THE ENGINEER'S YEAR-BOOK FOR 1931

THE UNIVERSITY
OF ILLINOIS
LIBRARY

620.8
En32
REMOTE STORAGE
v. 38

ALTGELD HALL STACKS

ENGINEERING

Will...

EN...

E... G

of ... ons

We do not ... We know fully that a... mall orders receive just as much attention as large indents. Urgent instructions have immediate consideration.

Wire, Write, or 'Phone for your every need—Willcox's will attend to it.

WE ARE AT YOUR SERVICE!

W. H. WILLCOX & CO., LTD.

GENERAL ENGINEERS' STORES,

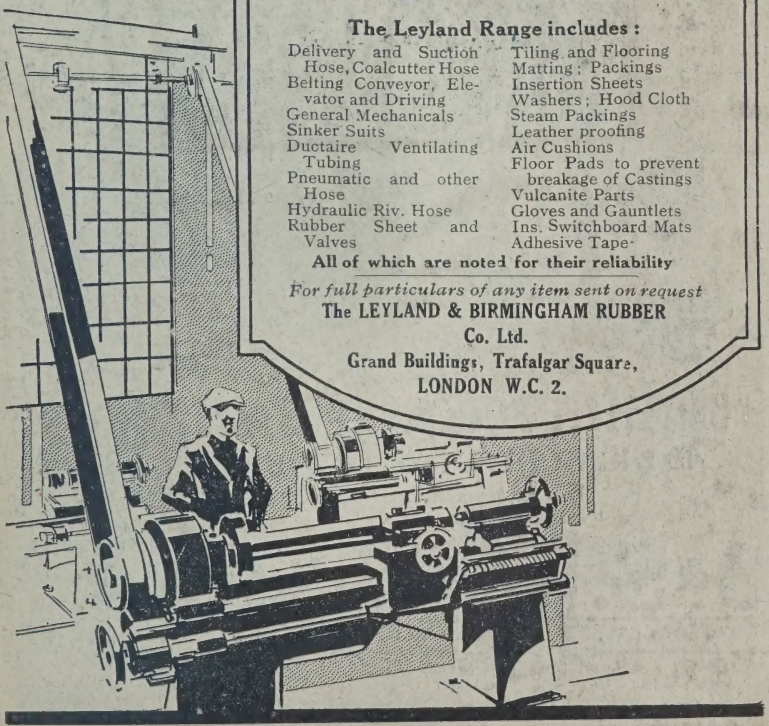
32-38, SOUTHWARK ST., LONDON, S.E.1.

'Phone: *Hop 3140 (7 lines).*

Wire: *"Enginsplys, Boroh, London."*

Use

LEYLAND Rubber Products FOR ALL ENGINEERING PURPOSES



The Leyland Range includes :

Delivery and Suction Hose, Coalcutter Hose	Tiling and Flooring Matting; Packings
Belting Conveyor, Elevator and Driving	Insertion Sheets
General Mechanicals	Washers; Hood Cloth
Sinker Suits	Steam Packings
Ductaire Ventilating Tubing	Leather proofing
Pneumatic and other Hose	Air Cushions
Hydraulic Riv. Hose	Floor Pads to prevent breakage of Castings
Rubber Sheet and Valves	Vulcanite Parts
	Gloves and Gauntlets
	Ins. Switchboard Mats
	Adhesive Tape

All of which are noted for their reliability

For full particulars of any item sent on request

The LEYLAND & BIRMINGHAM RUBBER Co. Ltd.

Grand Buildings, Trafalgar Square, LONDON W.C. 2.

RP 18-31. *Causton*

See pages 1132-1133

MASSEY HAMMERS & STAMPS

Equip your

SMITHY, FORGE & STAMPSHOP

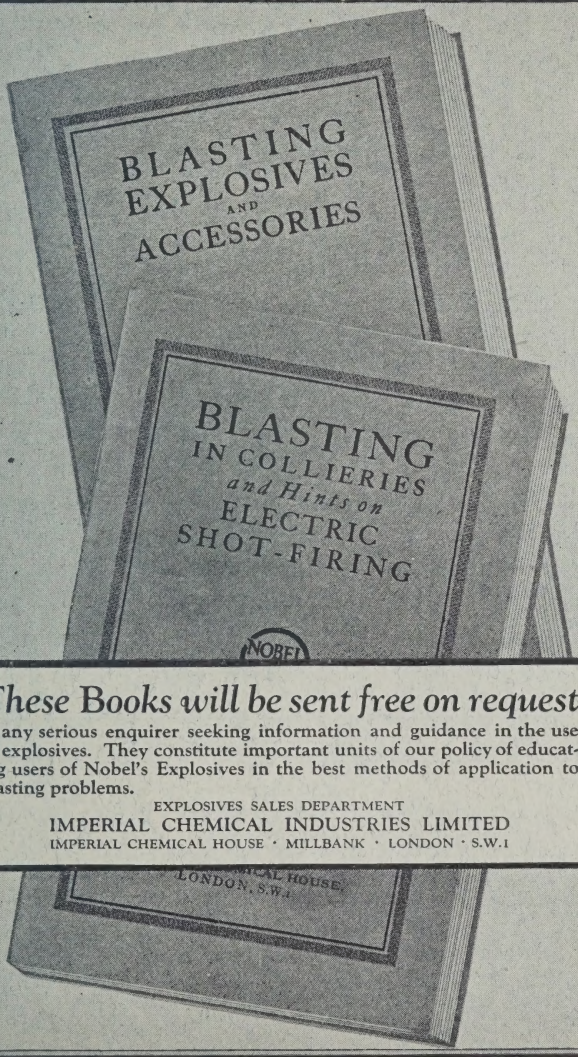
for **HIGH OUTPUT** with **LOW UPKEEP** costs.

Your enquiries will receive prompt attention.

B&S. MASSEY LTD.

OPENSHAW. MANCHESTER. ENGLAND.

NOBEL'S EXPLOSIVES



These Books will be sent free on request

to any serious enquirer seeking information and guidance in the use of explosives. They constitute important units of our policy of educating users of Nobel's Explosives in the best methods of application to blasting problems.

EXPLOSIVES SALES DEPARTMENT
IMPERIAL CHEMICAL INDUSTRIES LIMITED
IMPERIAL CHEMICAL HOUSE · MILLBANK · LONDON · S.W.1

THE

FRANÇOIS CEMENTATION PROCESS

is sure and safe, efficient and economical, and has the benefit of experience gained in many years of successful application.

SINKING SHAFTS; DRIVING DRIFTS or Tunnels; Building Dams; Sealing off Leakages in Shafts, Lodge Rooms, Tunnels or Roadways; Injecting Faulty Tubbing; Installing Ferro-Concrete Lining; Repairing and Re-Lining Old Furnace Shafts; Reclaiming Flooded Mines; Installing Shaft Plugs; Quenching Gob Fires; Diamond Drilling, Strengthening and Repairing Foundations, Piling, etc. etc.

Submit your Problems to—

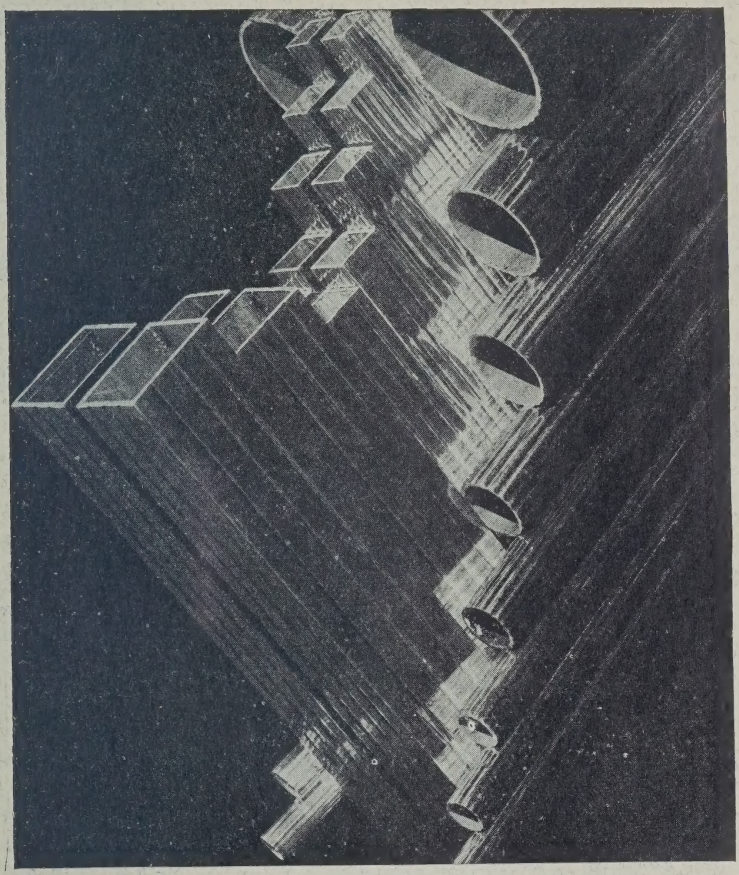
THE

FRANÇOIS CEMENTATION COMPANY, Ltd.

BENTLEY WORKS, BENTLEY, DONCASTER.

Telephone: 1177 & 1178 Doncaster. *Telegrams:* 'François, Doncaster.'

Directors: J. A. Agnew; Harry Clayton (*Chairman*); A. R. Neelands (*Managing Director*); L. Chevrillon (French), Paris Office; George Du Cros; John Pennington Harman; R. Scalliet (Belgian).



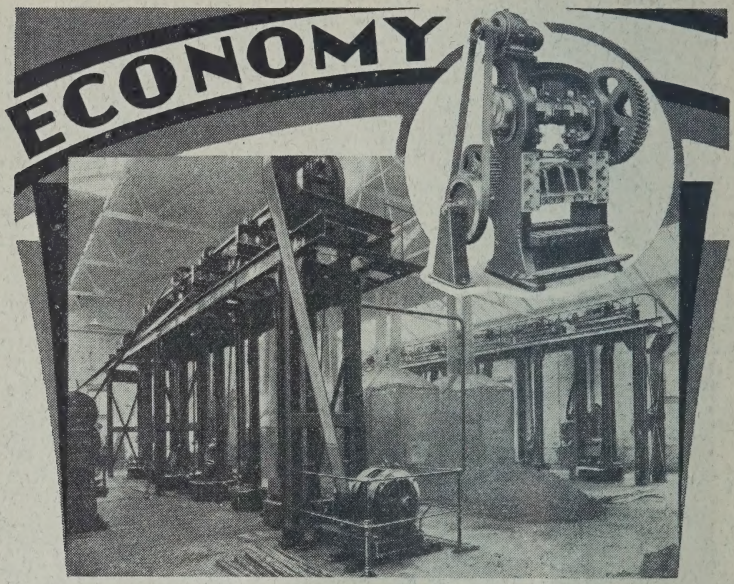
Aluminium

Aluminium developments are in step with Engineering progress.

THE BRITISH ALUMINIUM CO., LIMITED

ALUMINIUM PRODUCERS, ADELAIDE HOUSE, LONDON, E.C.4.

'Phone: Mansion House 5561 & 8074 (5 lines). *Telegrams:* Cryolite, Bilgate, London.



The above illustration shows an installation of friction driven drop hammers and inset a special type of clipping press for heavy work

BRETT

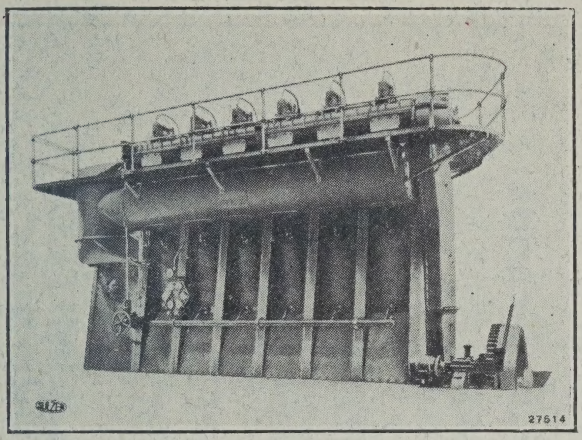
We specialise in all types of drop forging equipment and our 35 years experience in this branch of industry enables us to recommend and supply the most efficient and economical plant

DROP HAMMERS FOR STEAM, COMPRESSED AIR AND MOTOR DRIVE

BRETT'S PATENT LIFTER Co., LTD.

FOLESHILL WORKS COVENTRY

TELEGRAMS: LIFTERS, COVENTRY PHONE No.: 3658



LATEST TYPE OF

SULZER

DIESEL MARINE ENGINE

WITH CROSSHEAD AND SOLID INJECTION.

Although possessing the advantage of greatly reduced weight, this engine still embodies the well-known features of the Sulzer simple design and robust construction, ensuring extreme Reliability and Efficiency and making it ideal for ship propulsion where moderate powers are required. Built in standard sizes from 600 to 3000 BHP at speeds ranging between 200 and 150 revs. per min. respectively.

SPECIALITIES : Steam Engines & Boilers ; Electrode Boilers ; Land & Marine Diesel Engines ; Centrifugal Pumps & Fans ; Borehole Pumps ; Refrigerating Machinery ; Central Heating & Ventilating ; Coke Cooling Plant.

SULZER BROS. 31, BEDFORD SQUARE, LONDON, W.C.1

TRIUMPH–SUPERB | TRIUMPH–SUPERB 1000 | TRIUMPH–SUPERB XLD

TRIUMPH–SUPERB | TRIUMPH–SUPERB 1000

HIGH-SPEED STEELS

The Famous SAVILLE Brands

"TRIUMPH-SUPERB"

18 per cent. Tungsten Steel of Super quality, renowned for its regularity and uniformity. First made in 1909 and since consistently improved. Used all over the world.

"1000" TRIUMPH-SUPERB "1000"

Super Cobalt Vanadium Tungsten Steel possessing exceptional cutting power, easy to forge and harden, and exceptionally good for cutting hard heat-treated steels, close-grained cast-iron, manganese steel, etc.

"XLD" TRIUMPH-SUPERB "XLD"

The Multi-super Cobalt Vanadium Tungsten Steel possessing extraordinary cutting power on metals of all descriptions. WILL MACHINE THE HITHERTO "UNMACHINABLE."

TRADE MARK

J. J. SAVILLE & CO., LIMITED

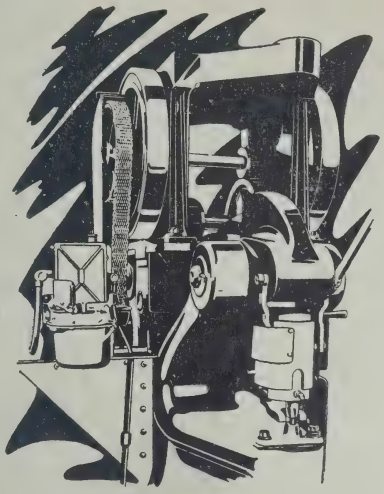
Triumph Steel Works, SHEFFIELD

ENGLAND

TRIUMPH–SUPERB | TRIUMPH–SUPERB 1000

TRIUMPH–SUPERB | TRIUMPH–SUPERB XLD

TRIUMPH–SUPERB XLD | TRIUMPH–SUPERB | TRIUMPH–SUPERB 1000



Coventry Inverted Tooth Chain Drive from Motor to Punching and Shearing Press.

Coventry Chains

for all classes of transmissions

STOCK DRIVES

We can supply from STOCK Roller Chain Drives to transmit 1—75 H.P. Booklet 1110

Reduction Gear Boxes. Chain Couplings. Tensioners.

The COVENTRY CHAIN Co. Ltd.

COVENTRY ENGLAND

J. CARTER, SONS & Co., Ltd.

Established 1820

IRWELL ENGINEERING WORKS

31 NEW BAILEY ST., SALFORD, MANCHESTER

Telephones: City 8352-4 *Telegrams: Carters, Engineers, Salford.*

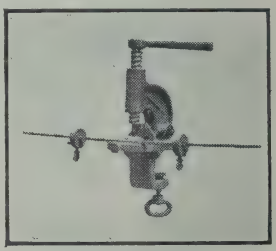
MANUFACTURERS OF

Admiralty Drilling Pillars, Machine Chasers, Square Thread Taps, Vise Benches, Sack Elevators, Jib Cranes, Crabs, Grinding Machines, Lifts, etc.

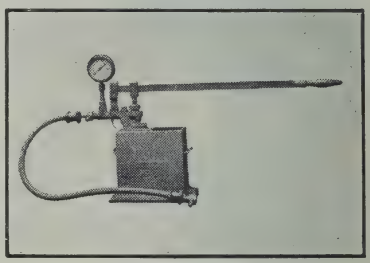
AND SPECIALISTS IN

FRICTION AND SELF-LANDING HOISTS.

We are specially equipped for the lines we manufacture and can offer attractive prices.



SLATE PUNCH.



TEST PUMP.

Also Merchants for all kinds of Machine Tools and Engineers Small Tools.

WE INVITE YOUR ENQUIRIES

Illustrated Catalogue on request.

EIGHT GOLD MEDALS.

WILLESDEN PAPER, WILLESDEN CANVAS,

TRADE MARK. WILLESDEN WATERPROOF WW UNIQUE

Registered 1880.

FOR ALL CLIMATES.

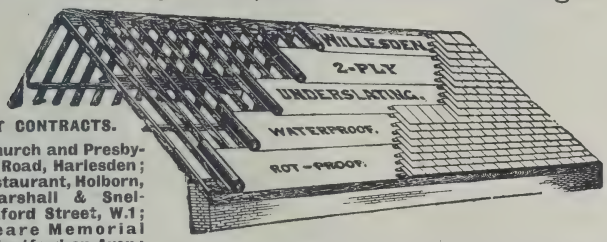
Waterproof, Rot-proof, Vermin-proof, Draught and Dust-proof, No Smell, Durable.

As used by H.M. Government, Admiralty, War Office, India Office, Colonial Office, General Post Office, Royal Engineers, Cape of Good Hope, West Australia, Queensland, New Zealand, and Foreign Governments, Home and Foreign Railways, Gasworks, Docks, Harbours, Municipalities, Estates, Contractors, leading Engineers and Architects, Noblemen, Private Users, and every Trade and Profession.

Proved to Resist the White Ant.

WILLESDEN 2-PLY UNDERSLATING

For Underlining Slates, Tiles and Iron Buildings.



RECENT CONTRACTS.

New R.C. Church and Presbytery, Nicoll Road, Harlesden; Holborn Restaurant, Holborn, W.C.1; Marshall & Snelgrove's, Oxford Street, W.1; Shakespeare Memorial Theatre, Stratford-on-Avon; Willesden General Hospital; Royal Air Force, Kidbrooke; Midland Railway, New Stores, Derby; New Merton Council Offices, Kingston Road, Merton; L.C.C. Housing Estate, Roehampton; Bush House, Aldwych, W.C.; Bank of England, Rebuilding, E.C.; Sheerness U.D.C.; Kempton Park Power Station; New Pier Pavilion, Colwyn Bay; Electricity Works, Burton-on-Trent; Carlton Club, Pall Mall, S.W.1; Old Oak Estate, Hammersmith, W.; New Offices, Metropolitan Water Board; New Garage for Daimler Hire, Herbrand St., Russell Square, W.C.1; New Tram Sheds, Rotherham; Boscombe Hospital, Bournemouth; Whiteley's Homes, Walton; Lloyd's New Buildings, Leadenhall Street, E.C.3; &c., &c.

WILLESDEN 4-PLY, FOR OUT-BUILDINGS, SIDES OF SHEDS, PARTITIONS, &c.

WILLESDEN 2-PLY LINING, FOR CORRUGATED IRON OR ZINC BUILDINGS, Preventing drip from condensation, and making metal buildings of even temperature.

WILLESDEN 2-PLY AND 4-PLY, FOR PARTITIONS, CEILINGS, &c. Preventing dust, &c.

WILLESDEN 2 AND 1-PLY, FOR PUGGING AND LINING BASEMENTS OR DAMP FLOORS. Used by eminent Building Firms. Also for Electrical Purposes.

WILLESDEN 1-PLY, FOR INSULATING CHILL ROOMS FOR COLD STORAGE, &c.: on land and board ship; for Dust and Draught-proof lining Mills, Factories, Houses, &c.: also for Damp Walls, underlaying Linoleum, Press Copy Sheets, &c.

DITTO, FOR STENCILS, PACKING, LABELS, UNDERLINING CARPETS, &c.

WILLESDEN PAPER & CANVAS WORKS Ld., Willesden Junction, NW.10.

Telegrams: 'Impermeable, Phone, London.'
Telephone No.: 1165 Willesden

ESTABLISHED 1870.

BRITISH ENGINEERS

require

BRITISH MICROMETERS

DEAD ACCURATE REPAIRS OF
MICROMETERS OF ANY MAKE
ALSO A SPECIALITY

✦ ✦ ✦

AMBROSE SHARDLOW & Co., LTD.

EALING WORKS, WASHFORD ROAD,

SHEFFIELD - - - ENGLAND

Telephone : 41247

TEN TYPES OF PLANT

TEN GRADES OF ELECTRODES

WILSON PLASTIC-ARC ELECTRIC WELDING PLANTS

and

"Colour-tipt"

WELDING METALS

are indispensable to the making of

A PERFECT WELD

Write for particulars

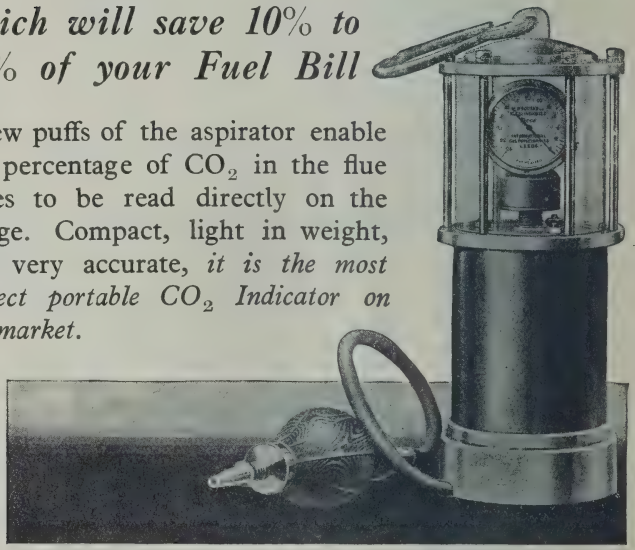
G. D. PETERS & Co. Ltd.,

3 CENTRAL BUILDINGS, WESTMINSTER, S.W.1.

A New INVENTION

which will save 10% to 15% of your Fuel Bill

A few puffs of the aspirator enable the percentage of CO_2 in the flue gases to be read directly on the gauge. Compact, light in weight, and very accurate, *it is the most perfect portable CO_2 Indicator on the market.*



Write now to:—

Sole Manufacturers and Patentees:

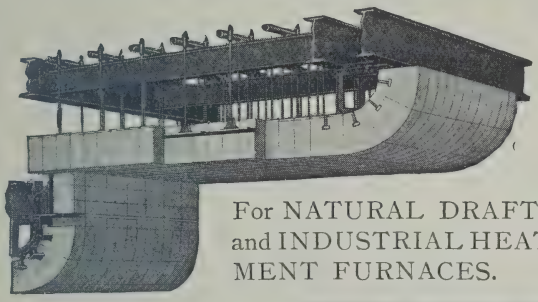
INTERNATIONAL GAS DETECTORS LTD.

(Inc. W. R. PATENTS, LTD.)

Great Wilson Street, LEEDS

Telephone 25268 *Telegrams "Dioxide, Leeds"*

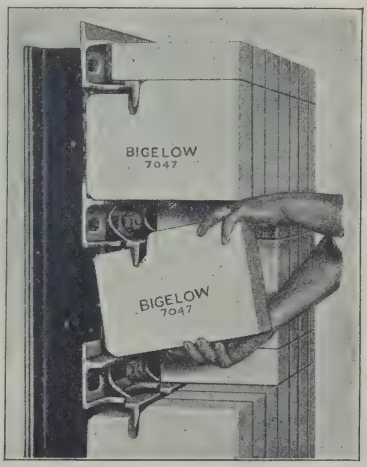
BIGELOW ARCHES and VENTILATED WALLS



For NATURAL DRAFT GRATES and INDUSTRIAL HEAT TREATMENT FURNACES.

BIGELOW Air-Cooled Suspended WALLS provide an inexpensive Construction for Furnace Walls, which can be relied on to give long service under conditions of high temperature erosion and slag.

A hot air supply is provided for use elsewhere.



Specify BIGELOW *for reduced maintenance costs*

LIPTAK FURNACE ARCHES Ltd.

38 Victoria Street, London, S.W.1

SWAN, HUNTER, & WIGHAM RICHARDSON

Limited

WALLSEND-ON-TYNE

BUILDERS AND REPAIRERS OF

all types of

STEAM AND MOTOR SHIPS

Our output of tonnage in 1930 was 213,229 gross tons ; top of the world's list.

FLOATING DOCK SPECIALISTS

Neptune Atlas-Polar, Wallsend-Sulzer,

Barclay Curle-Doxford

OIL ENGINES

Licensees of **BAUER-WACH** turbine installations

London Office: 21 Russell Square, W.C. 1

CONTRACTORS TO THE ADMIRALTY, WAR OFFICE, CROWN AGENTS FOR THE COLONIES, THE PRINCIPAL RAILWAY COMPANIES, ETC.

REPRESENTED IN ALL PARTS OF THE WORLD.

Electrodes and Welding Equipment

PATENTS NOS. 136016; 160957; 188855; 194395; 197196; 204686; 229780; 259365; 267612; 286762; 299575; 308475; 321279; 343052.

Representatives in:

ARGENTINA	CHINA	INDIA	SOUTH AFRICA
AUSTRALIA	CZECHO-SLOVAKIA	ITALY	SPAIN
BELGIUM	DENMARK	JAPAN	SWEDEN
BRAZIL	FINLAND	MEXICO	SWITZERLAND
BURMA	GERMANY	NEW ZEALAND	U.S.A.
CANADA	HOLLAND	NORWAY	VENEZUELA
	HUNGARY	PORTUGAL	

THE QUASI-ARC COMPANY, LIMITED

15 Grosvenor Gardens, Victoria, London, S.W.1

Telephone Nos.: **VICTORIA 6236-7.** *Telegrams:* **'QUAZARC, LONDON.'**

WALLSEND SLIPWAY AND ENGINEERING COMPANY LTD.

WALLSEND-ON-TYNE.

Builders of Marine Reciprocating, Turbine and Wallsend-Sulzer Diesel Oil Engines

Cylindrical and Water-Tube Boilers

Ship Repairers and Oil Burning Specialists

Outputs for Year 1930

NEW ENGINES	118,400 I.H.P.
BOILERS	NO. 25
OIL BURNING INSTALLATIONS	489,865 I.H.P.

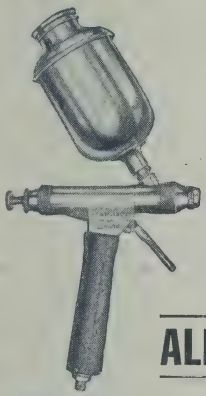
Head Office & Works: WALLSEND-ON-TYNE.

Telegrams: "Wall, Phone, Newcastle-on-Tyne."

London Office . 34 GREAT ST. HELENS, E.C. 3.

Liverpool Office CUNARD BUILDING.

WELLS' System of PAINT SPRAYING



THE NEW MODEL PISTOL TYPE No. 3

of all round service in connection with the Spraying of Oil Paints, Varnishes, Cellulose, etc.

Takes the lead for Efficiency with minimum of working parts. Portable and Stationary Equipment of every description supplied. State requirements and write for Catalogue H.

ALL BRITISH MANUFACTURE

WELLS' WASTE OIL FILTERS

(WITH SIGHT FEED SYPHONS)

THE ORIGINAL
AND WELL PROVED MACHINE
DIRTY LUBRICATING OILS OF
EVERY DESCRIPTION FILTERED
OVER AND OVER AGAIN
OVERHEAD CHARGES
REDUCED AND FIRST COST
QUICKLY RECOVERED

MANUFACTURERS OF
OIL CABINETS & STORAGE CISTERNS.
"UNBREAKABLE" LAMPS & OIL FEEDERS.
THE WELLS' LIGHT.
OIL HEATING & BRAZING PLANTS.

A. C. WELLS & CO., LTD.,
Carnarvon St., Cheetham, Manchester.
London Office: **4 Bradleys' Buildings, White Lion Street, Islington, N.1.**

VICKERS' DURALUMIN

"The Aeroplane," April 29, 1931, says:—
"The water is very saline, and everything corroaible corrodes very quickly. This has been the trouble with some foreign designed duralumin flying-boats, but the Supermarine Southampton boats seem to stand it well enough."

ALSO
"Y"
ALLOY
AND
WROUGHT
LIGHT
ALLOYS
TO ANY
SPECIFICATION

Telephone: East 1221 (Private Branch Exchange).
Telegrams: "Booth, Birmingham."
Cables: Liebers 5 Letter Code.
Bentley's Second Phrase Code, A.B.C. 5th and 6th Edition.

JAMES BOOTH & COMPANY (1915) LIMITED
ARGYLE ST. WORKS: NECHELLS.
BIRMINGHAM

EXTRUDED BRASS

SHAPES and SECTIONS. | SCREWING and FORGING RODS.

YELLOW METAL. RED METAL.
NAVAL BRASS. BRONZES.

Special alloys to Engineers' Specifications.

TUBES

ALUMINIUM BRONZE,
CUPRO-NICKEL, BRASS, COPPER, &c.
SOLID DRAWN, AND BRAZED.

METALS

BRASS, COPPER, BRONZE, NICKEL,
SHEETS, STRIPS, &c.

JAMES BOOTH

AND COMPANY (1915) LIMITED,

BIRMINGHAM.

On Admiralty, War Office, Air Board, Colonial & Foreign Government Lists.
Telegrams: 'BOOTH, BIRMINGHAM.'

Codes: Lieber's 5 Letter, ABC 5th & 6th Editions, Bentley's Second Phrase
Telephone: EAST 1221 (Private Branch Exchange)

Head Office: ARGYLE STREET WORKS, NECHELLS, BIRMINGHAM.

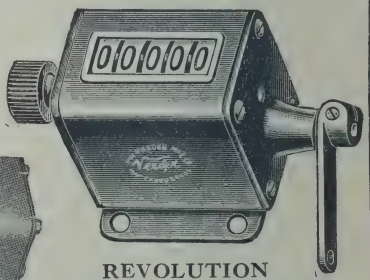
LONDON OFFICE:
VICKERS HOUSE, BROADWAY, WESTMINSTER, S W.1.

Veeder-Root

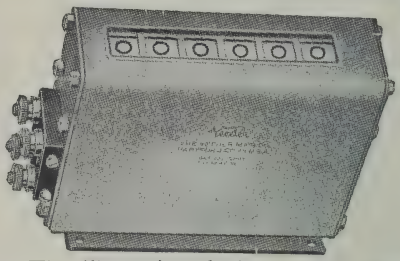
COUNTERS

FOR EVERY REQUIRED PURPOSE

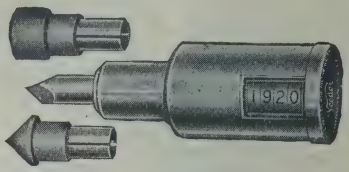
SMALL—COMPACT
DURABLE—STRONG



REVOLUTION
RATCHET COUNTER
with reset action enabling instrument to be reset to zero at any time.

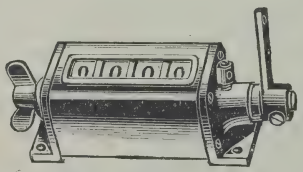


The illustration depicts one form of
VEEDER MAGNETIC COUNTER
which is obtainable for Direct or Alternating Current operating by the breaking and making of an electrical circuit.



CLUTCH SPEED COUNTER
for finding revolutions per minute made by a shaft or any other revolving part.

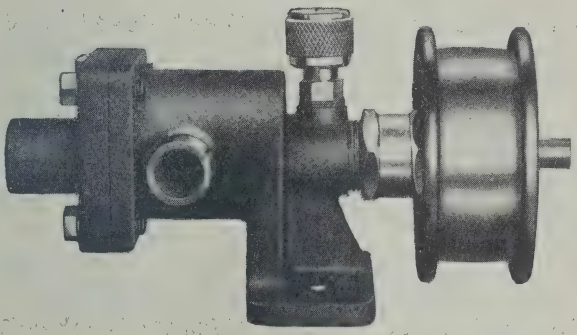
"SETBACK" & "LOCKED" COUNTERS
for Presses, Looms, Stamping Machines or any machine on which an accurate record of parts made is required.



LARGE SET BACK
REVOLUTION or ROTARY
RATCHET COUNTER

F. E. DICKINSON
TELEPHONE: CITY 8138
St. Andrew's House,
Holborn Circus, London, E.C.1

THE "ROTELLIP" PUMP



Embodies an entirely new principle in pumping, and combines efficiency of performance with simplicity of design.

SUITABLE FOR OIL, SUDS, PETROL AND PARAFFIN.

Light in weight, volumetric efficiency high, no valves employed, and flow is reversible according to direction of rotation.

ENQUIRIES TO

ROTHERHAM & SONS, LTD., COVENTRY.

'PHONE: 4154 'GRAMS: ROTHERHAMS, COVENTRY.

B.I.G

WELDING GASES

DISSOLVED ACETYLENE

99·5% PURE OXYGEN

in Special

FLAT BOTTOMED SAFETY CYLINDERS

which comply with all Home Office Regulations, and ensure absolute safety from Fire and Explosion.

No Cylinder stands—conserves storage space.

Depots at:

Glasgow, Newcastle, Hull, London, Southampton, Bristol, Newport, Swansea, Birmingham, Burnley, Warrington, Liverpool, Fleetwood, Leeds and Sheffield.

Send your enquiries to

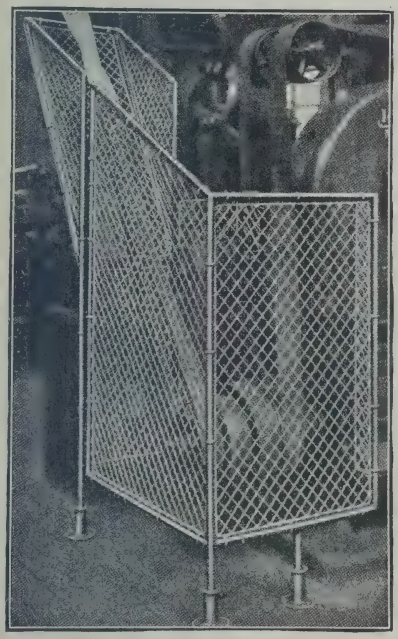
British Industrial Gases Ltd.

(CHARLES BINGHAM & CO.),

34 Victoria Street - London, S.W.1.

'Phone: Victoria 2356/8. 'Grams: "Binghamcy, Sowest, London."

Machinery Guards



Designed to meet the requirements of H.M. INSPECTORS of FACTORIES

Manufacturers of

WIREWORK
WOVEN WIRE
SHEET METAL
&
WROUGHT IRONWORK

Sole Makers of

"WING" VENTILATING FANS

For information regarding "Wing" Fans please refer to Editorial notice on page 2521

Illustrated Catalogues sent on request

F. W. POTTER & SOAR, LTD.

PHIPP STREET, LONDON, E.C. 2

Telephone Nos.: Bishopsgate 2177-8

Telegraphic Address: "Wirework-Finsquare, London"

Power Brakes

Manufacturers of Power Braking Apparatus and Fittings of every description, on either Vacuum or Air Systems, for all classes of Rolling Stock, including steam and electrically operated Railways, and for Tramways, Omnibuses and all types of Road Vehicles.

The Consolidated Brake & Engineering Co., Ltd.

3 Central Buildings, Westminster,
LONDON, S.W. 1.

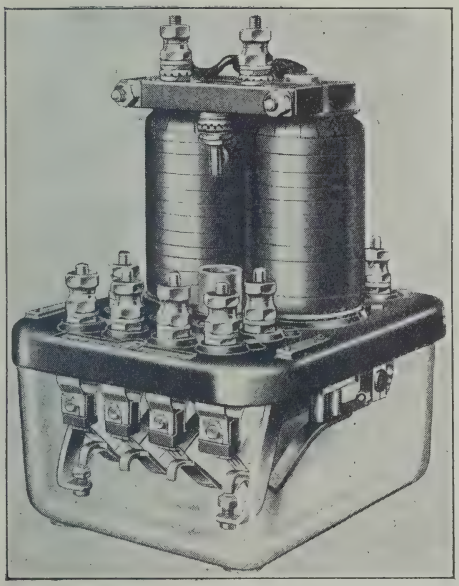
Windsor Works, SLOUGH, BUCKS.

The British Power Railway Signal

CO., LTD.

3 CENTRAL BUILDINGS,
WESTMINSTER, S.W. 1.

Railway Signal Engineers & Contractors



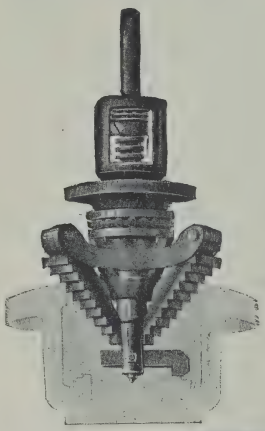
POWER
AND
MECHANICAL
SIGNALLING

Windsor Works,
SLOUGH, BUCKS

D.C. OR A.C. RELAYS
OF ALL DESCRIPTIONS
FOR
RAILWAY SIGNALLING

CROCKATT SIMPLEX

VALVE RESEATER



Reduces repair costs and saves time. Enables your engineering staff to keep Steam and Check Valves tight without recourse to a lathe.

This tool has been supplied to hundreds of ships, factories, repair shops, power stations, etc., and a large proportion of our orders are repeats.

MADE IN FOUR SIZES FOR VALVES FROM ⅜″ TO 12″

CROCKATT SIMPLEX TUBE BRUSHES

are made in a variety of designs for tubes from ⅜″ to 27″ diameter.

ALSO

ELECTRIC SALINOMETERS.
FEED WATER FILTERS.
MARBLE POLISHING MACHINES.
MARBLE SAWING MACHINES.
TERRAZZO FLOOR POLISHERS, etc.

COMPLETE CATALOGUE ON REQUEST.

W. CROCKATT & SONS, LTD.

62 Darnley Street, GLASGOW, S.1

2-3 Eldon St., London, E.C.2. **25 Water St., Liverpool.**

THE ELECTRICAL REVIEW

The Mirror of the Industry

FOUNDED 1872. PUBLISHED EVERY FRIDAY. PRICE 6D.

YEARLY SUBSCRIPTION RATES

United Kingdom £1-14-8 · Canada £1-12-6
Colonial & Foreign £2-1-6 · *Postage Free.*

The Editorial pages of the Review cover each week everything of interest and importance to the Electrical Industry—commercially, financially and technically. It is distributed all over the world, not only in the industry itself, but in all other industries and trades interested in the use and development of Electric Power. It has by far the largest circulation of any Electrical Journal in this country. Net Sales Certificate on Application.

THE ELECTRICAL REVIEW

4 LUDGATE HILL · LONDON, E.C.4.

Telephone: CENTRAL 5871 (5 LINES). *Telegrams:* 'AGEEKAY CENT, LONDON.'

ROBERT BOBY, LTD.

ENGINEERS

BURY-ST-EDMUNDS, ENG.

MANUFACTURERS OF FOLLOWING CLASSES OF MACHINERY

FOR BREWERIES, MALTINGS, AND DISTILLERIES

MECHANICAL HANDLING OF MATERIALS, STEEL STRUCTURES

ESTABLISHED 1856

GRAIN AND SEED CLEANING AND HANDLING

REFRIGERATION AND COLD STORAGE

TELEGRAMS: SCREENS.
'PHONES: 251-2-3
BURY ST. EDMUNDS.

FLAX SCUTCHING, FLAX PULLING, ETC.

CASTINGS, CAST IRON & BRASS GAS SYPHONS

DESIGNS AND ESTIMATES BY EXPERT ENGINEERS FREE.

GRAPHITE

PRODUCTS FOR ALL PURPOSES

PURE FLAKE GRAPHITE

GRAPHITE PIPE JOINT COMPOUND

SILICA-GRAPHITE & FINISHING PAINTS

GRAPHITE GREASES, ETC.

MADE IN ENGLAND

GRAPHITE PRODUCTS, LTD.

218-220 Queen's Road, Battersea

LONDON, S.W. 8

HENDERSON & GLASS,

Steel and Iron Merchants, LIVERPOOL.

Stockholders of All Descriptions of Iron and Steel for Engineers, Shipbuilders, Contractors, &c., &c.

Plates to Lloyds Tests of all sizes.
Joists in all Sections and Lengths. Always in Stock.

STRUCTURAL STEELWORK

We Design, Fabricate and Erect all types of Steelwork anywhere.

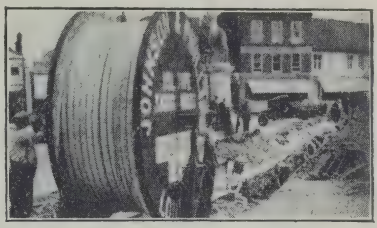
Telegraphic Address :	Codes :	Telephone :
Vulcan, Liverpool.	ABC 5 ABC 6	Liverpool. Bank 8865.

Offices and Warehouses :

VULCAN STREET, LIVERPOOL.

CABLES

Our manufactures include Paper Insulated, Lead Covered, Armoured, for pressures up to 33,000 volts. V.I.R. Insulated Wires, Cables and Flexibles, Semaphite Cables, Vulcanised Bitumen and Elastumen Bitumen Cables for Mines, etc. Telephone Cables, Tough Rubber-sheathed Cables, Ship Installation Cables, etc., to the standards of the Cable Makers' Association of which we are members.

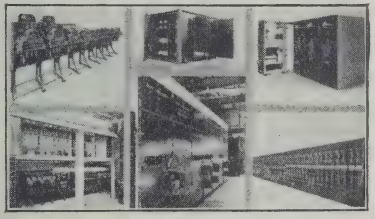


SWITCHGEAR

We manufacture all types of switchgear for A.C., E.H.T. and L.T.; also for continuous current. Draw-out Truck Type, Ironclad Type, Cubicle Type, Flat Back, etc., etc. Oil Switches, Knife Switches, etc., Outdoor Switchgear and Kiosks.

TRANSFORMERS

We were one of the first firms in this country to manufacture Static Transformers. To-day our service and reputation are in this direction world-renowned.



OVERHEAD LINES

Our Overhead Line contracts are increasing year by year, and our outside erecting staff is kept busy all the year round. The economy of overhead transmission is creating big demands, and we are always ready and willing to send a member of our engineering staff to consult and advise on any problem which such schemes of transmission involve.

JOHNSON & PHILLIPS, LTD.

CHARLTON, LONDON, S.E. 7

Branches in London (Aldwych), Birmingham, Bristol, Cardiff, Derby, Glasgow, Ipswich, Liverpool, Manchester, Newcastle, Portsmouth, Belfast and Australia, New Zealand, South Africa, East Africa, India, Ceylon, Egypt, Malaya, China, Argentine and Brazil.

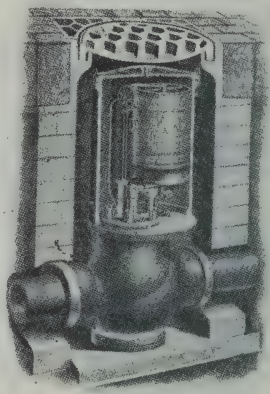
WASTE WATER
DETECTING METERS

(*See Editorial Notice pages* 1420 & 1436.)

Venturi Meters with Recording and Integrating Mechanism

Electrical and Mechanical Level Recorders

Recording Meters for Large Mains



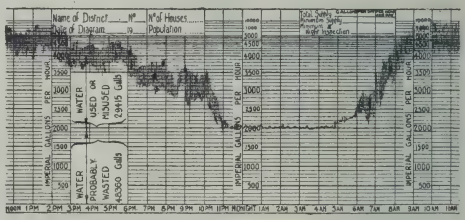
Trade Meters Rotary and Full Bore

Recorders for all types of Weirs and Flumes

Pressure Controlling Valves

Large size meters can be supplied with integrating mechanism

Deacon's waste water detecting meters



Used by all modern water authorities

THE FIRST 24-HOUR DIAGRAM AFTER FIXING A METER SHOWING THE CONDITIONS OF SUPPLY

PALATINE ENGINEERING CO. LTD.

Telegrams "WASTE" Code: "Bentley's"

BOOTLE, near Liverpool (ENGLAND)

Telephone BOOTLE 496 & 1235

Some of the productions of JOHN I.

THORNYCROFT

& CO., LIMITED

Designers and builders of all types of floating craft to 5,000 tons, Steam Turbines, Water Tube Boilers, Oil Fuel Gear, Marine Engines, Propellers, &c.

Full particulars from:

WOOLSTON WORKS, SOUTHAMPTON

Head Office:
Thornycroft House
WESTMINSTER
LONDON,
S.W.1

ENGINEERING TRAINING

LOUGHBOROUGH COLLEGE,

LEICESTERSHIRE, ENGLAND.

COURSES OF STUDY.

ENGINEERING :—

Civil, Mechanical, Electrical, Automobile, Chemical & Commercial

PURE AND APPLIED SCIENCE AND INDUSTRIAL ADMINISTRATION AND ECONOMICS.

The Engineering Works Departments accommodate
500 STUDENTS
and are exclusively engaged on
PRODUCTIVE OUTPUT for the COMMERCIAL MARKET
350 MODERN MACHINE TOOLS
are available for instructional purposes.

Students are prepared for the College Diploma and Associateship Examinations, the External Degrees of the University of London in Science, Engineering and Commerce, the Associateship of the Institute of Chemistry and for the Associate Membership Examinations of the Institutions of Civil, Mechanical, Electrical and Automobile, Engineering.

RESIDENTIAL HALLS ARE MAINTAINED FOR ALL STUDENTS.
THE COLLEGE ATHLETIC GROUNDS COVER 48 acres, and offer facilities for all branches of Sport.
THERE ARE EXTENSIVE FACILITIES FOR RESEARCH IN ALL DEPARTMENTS.

THE COLLEGE IS THE ONLY INSTITUTION IN THIS COUNTRY offering facilities for ACADEMIC STUDY OF DEGREE STANDARD concurrently with PRACTICAL INSTRUCTION ON A COMMERCIAL BASIS.

Appointments are found for all Students who satisfactorily complete the College Course.

JUNIOR COLLEGE for Boys 11-17—Residential.

The College Calendar, Senior and Junior (Illustrated—230 pages) will be forwarded upon application to :—

THE REGISTRAR, LOUGHBOROUGH COLLEGE.

LOUGHBOROUGH

ENGINEERING TRAINING

Heriot - Watt College,

EDINBURGH.

Principal—J. CAMERON SMAIL, O.B.E.

The College is conducted under the provisions of the Heriot-Watt College and George Heriot's Trust Order Confirmation Act, 1927, and is a Central Institution providing Technical, Trade, Commercial and General Education for both sexes for Edinburgh and the South East of Scotland.

DIPLOMA COURSES.

Full-time Day Courses of Instruction leading to Diplomas and Certificates are conducted in **Mechanical Engineering, Electrical Engineering, Mining Engineering** (Coal, Metalliferous, Oil), **Chemistry** (Analytical, Organic, Metallurgical, Bio-Chemical, Foods and Drugs), **Pharmacy, Building, Printing.**

The Diploma Courses extend over three years, but in Chemistry over four years.

RESEARCH AND POST-GRADUATE COURSES.

Provision can be made for Research Students who have already graduated, or hold practical qualifications, to undertake special lines of advanced work.

UNIVERSITY COURSES.

Arrangements exist with the University of Edinburgh by which certain courses for the Degrees in Mechanical Engineering, Electrical Engineering, Mining and Metallurgy, and Technical Chemistry are taken in the Heriot-Watt College. Full particulars are given in the University Calendar and the Heriot-Watt College Calendar.

EVENING CLASSES.

Advanced Classes in various Departments of Pure and Applied Science, Commerce, and Languages, which will prove of use to students who are studying for a Degree or for professional qualifications, are held in the College in the Evening.

FEES.—Day Classes. Composition Fees for Diploma Courses, £27 to £31 10*s*. per annum. **Evening Classes**, 10*s*. to 75*s*. Composition Fee for Approved Course, 42*s*.

All particulars regarding the College Classes, both Day and Evening, can be obtained from the PRINCIPAL at the College.

For the Scheme for Degrees in Civil, Electrical and Mechanical Engineering, Mining and Metallurgy, and Technical Chemistry arranged with the University of Edinburgh, see University Calendar, or the Calendar of Heriot-Watt College.

EDINBURGH

Manchester Municipal
COLLEGE OF TECHNOLOGY

Principal:
B. MOUAT JONES, D.S.O., M.A. (Oxon).

UNIVERSITY OF MANCHESTER
(FACULTY OF TECHNOLOGY)

DEGREE COURSES IN TECHNOLOGY

The Prospectus gives particulars of the courses leading to the Manchester University Degrees (B.Sc. Tech., M.Sc. Tech., and Ph.D.) and Certificates in the Faculty of Technology, in the following Departments:—

MECHANICAL ENGINEERING
(Prof. DEMPSTER SMITH, M.B.E., M.Sc. Tech., M.I.M.E.)

ELECTRICAL ENGINEERING
(Prof. MILES WALKER, M.A., D.Sc., M.I.E.E., F.R.S.)

MUNICIPAL ENGINEERING
(G. S. COLEMAN, D.Sc.Eng., A.M.Inst.C.E., A.M.Inst.M. & Cy.E., F.R.San.I.)

APPLIED CHEMISTRY, including General Chemical Technology, Metallurgy and Assaying, Fermentation Processes (including Brewing), Electro-Chemistry, Photography, Colouring Matters, Foodstuffs and Fuels.
(Prof. JAMES KENNER, D.Sc., Ph.D., F.R.S.)

TEXTILE CHEMISTRY (Bleaching, Dyeing, Printing and Finishing), Paper Manufacture.
(F. SCHOLEFIELD, M.Sc., F.I.C.)

TEXTILE TECHNOLOGY
(Prof. W. E. MORTON, M.Sc.Tech.)

BUILDING (W. B. McKAY, M.I.Struct.E., M.R.San.I.)

INDUSTRIAL ADMINISTRATION (Post-Graduate Certificate Course)
(J. A. BOWIE, M.A., D.Litt.)

Prospectus will be forwarded free on application to the Registrar, College of Technology, Manchester.

ENGINEERING TRAINING

Imperial College of Science and Technology.

VISITOR—H.M. THE KING.

CITY & GUILDS (ENGINEERING) COLLEGE,

Exhibition Road, South Kensington, London, S.W. 7.

The City and Guilds (Engineering) College, the Engineering Section of the Imperial College, provides complete Courses of Instruction leading up to, and including, the highest specialised instruction to fit Students to take eventually leading positions in the industrial world as Civil, Mechanical or Electrical Engineers, Scientific Advisers, Investigators or Experts in the Engineering industries. The College also provides opportunities for those already qualified who desire to take up more advanced courses of training and research in branches of Engineering in which they are specially interested.

The Departments of the College are as follows :—

ENGINEERING, MECHANICAL AND MOTIVE POWER—
Under the direction of Prof. Dalby, F.R.S., M.Inst.C.E.

ENGINEERING, CIVIL, AND SURVEYING—
Under the direction of Prof. Dixon, M.Inst.C.E.

ENGINEERING, ELECTRICAL—
Under the direction of Prof. Fortescue, M.I.E.E.

MATHEMATICS—
Under the direction of Prof. Chapman, F.R.S.

The following Diplomas are awarded to Students who satisfactorily complete a full Course of from two to four years :—

The Associateship of the City and Guilds of London Institute (A.C.G.I.).

For admission to the courses qualifying for this Diploma candidates are required to pass the Entrance or Matriculation Examination of the College held in September (application to be made not later than the first Monday in September); or to have passed the Matriculation of the University of London in certain subjects.

B.Sc. in Engineering of University of London.

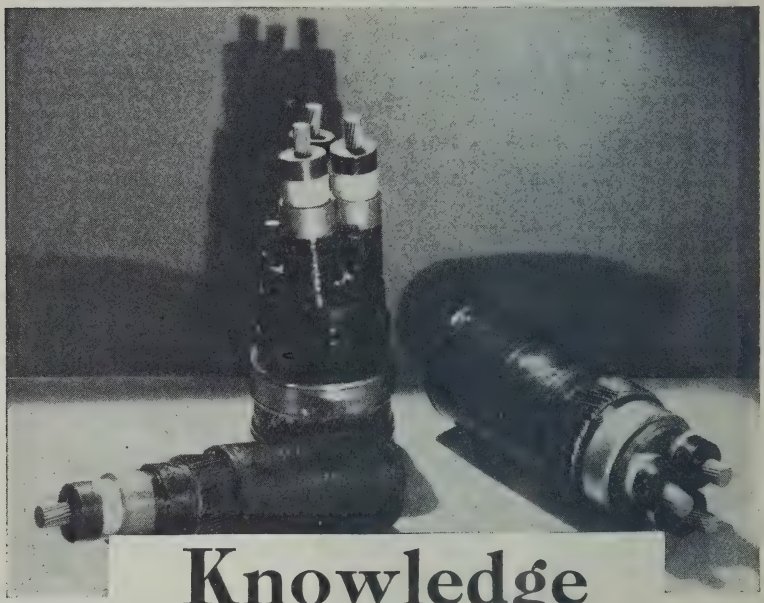
Students who enter by the Matriculation Examination of the University of London are eligible for the B.Sc. in Engineering of the University through the College Examinations alone.

The Diploma of the Imperial College (D.I.C.).

(For advanced specialised or fourth-year work.)

Further information and the Prospectus of the College may be obtained on application to the Registrar.

LONDON



Knowledge

Since the beginning of Electrical Transmission Henley's have been making Electric Cables and acquiring knowledge. To-day, that knowledge is embodied in every foot of cable manufactured.

The latest development in super tension cables is represented by the three types of cables shown. They are the cables for power. Power Cables and Cable Knowledge—both are at your service.

HENLEY
CABLES

823

W. T. HENLEY'S TELEGRAPH WORKS CO. LTD.
HOLBORN VIADUCT, LONDON, E.C.1.

Revised and Published Annually

Price 31s. 6d. net in Great Britain

THE ENGINEER'S YEAR-BOOK

OF

FORMULÆ, RULES, TABLES, DATA, & MEMORANDA

FOR 1931

A COMPENDIUM OF THE MODERN PRACTICE OF

CIVIL, MECHANICAL, ELECTRICAL, MARINE, GAS, AERO, MINE, & METALLURGICAL ENGINEERING

(*Originally compiled by H. R. KEMPE, M.Inst.C.E., M.I.Mech.E., and W. HANNEFORD SMITH, Assoc. Inst.C.E.*)

38th ANNUAL ISSUE

Revised under the direction
of
THE EDITOR of 'THE ENGINEER'

PUBLISHED BY
MORGAN BROS. (Publishers) LTD.
PROPRIETORS OF 'THE ENGINEER'
AT 28 ESSEX STREET, STRAND,
LONDON, W.C. 2

Publishing Trade Agents:
CROSBY LOCKWOOD AND SON
7 STATIONERS' HALL COURT, LUDGATE HILL,
LONDON, E.C. 4

Sold by the leading Booksellers in the United Kingdom, the Dominions Overseas, India, and Foreign Countries

1931

Printed in England

The Engineer.

Established 1856. Registered as a Newspaper

PROVIDES THE LATEST AND THE MOST RELIABLE INFORMATION FOR ENGINEERS

IN ALL BRANCHES OF THE PROFESSION.

The last issue in each month contains a Supplement, **The Metallurgist,** *a journal devoted to the science of metals, ferrous and non-ferrous.*

Subscriptions

			Per annum
INLAND, YEARLY			£3 5 0
CANADIAN SUBSCRIPTIONS.	Thin paper edition	...	£2 18 6
	Thick paper edition	...	£3 3 0
FOREIGN SUBSCRIPTIONS.	Thin paper edition	...	£3 3 0
	Thick paper edition	...	£3 7 6

'THE ENGINEER' DIRECTORY AND BUYER'S GUIDE

SENT FREE ON APPLICATION.

PREFACE TO FIRST EDITION (1894).

THE importance and value of such a work as THE ENGINEER'S YEAR-BOOK need hardly be insisted on. Perhaps there is no professional man who *needs* a YEAR-BOOK more than the Engineer of the present day, seeing that his duties are so multifarious and the requirements of his position so constantly varying. Yet, while other professions have their Year-Books, until now none has been provided for the Engineer; it is confidently believed that the present volume will meet a want that is constantly experienced by Practical Men in every branch of Engineering.

The aim of the compiler has been to produce a work which, being carefully brought up to date, shall take its place as the standard Book of Reference in the profession, and include only such Modern Formulæ, Rules, Tables, and Data as are required by the Engineer from day to day in the practical work of his calling, including Civil, Mechanical, Marine, Electrical, and Mine Engineering. The work is accordingly divided into Thirty-one Sections, each Section forming practically a complete treatise on its particular subject, and the whole embracing the present practice of Engineering.

The great advances made in Science, as applied to Engineering, during the last quarter of a century have fixed immutably many new principles, laws, and practical data. These, as well as the like particulars of older practice, have been carefully compiled from the best authorities, and find their place in THE ENGINEER'S YEAR-BOOK, whilst a large amount of original matter, presented in a handy and concise form, is also included in the volume.

In the work of compilation, most valuable assistance has been rendered by numerous practising Engineers, including those of the principal English Railway Companies, and of the Queensland Government, as well as by

771070

Girder Manufacturers, Boiler and Engine Makers, and many other persons of experience. Every available work has been searched through and consulted for information, but in making use of particulars so obtained the compiler has been at pains to trace the matter to its original source, so as to determine its authoritative value, and in every case where it appeared necessary he has redrawn existing figures and verified current formulæ, so that clearness and accuracy might be ensured.

Obviously a work of this nature, to fulfil its purpose of being kept up to date, will require periodical revision, and it is accordingly intended to REISSUE IT FROM YEAR TO YEAR. The facilities thus afforded for thorough revision throughout the volume will make it practicable to *insert every year new matter of importance in any Section*, and render the book at all times a safe and reliable DAILY REFERENCE BOOK for the PRACTICAL ENGINEER.

Kindly criticism and help towards making the work as complete and as useful as possible are earnestly invited, and any suggestions, addressed to the care of the Publishers, with this object, will be gladly received, and, as far as possible, acted upon.

I have to thank Mr. H. HARTNELL, Technical Officer of the Engineer-in-Chief's Office, for great assistance rendered in the work of compilation.

Finally, I think more than a word of thanks is due to the printers, Messrs. SPOTTISWOODE & Co., for the able manner in which they have executed the difficult task of concisely and clearly arranging the letter-press.

H. R. KEMPE.

ENGINEER-IN-CHIEF'S OFFICE,
GENERAL POST OFFICE, LONDON:
January 1894.

INTRODUCTION.

Amongst technical reference books ' Kempe ' holds an exceptional position, for whilst it devotes many of its pages to fundamental formulæ, tables, data, and theories, it gives also information concerning the practical application of those fundamentals, and describes proprietary commodities. There are some who hold that a handbook of the kind should rigidly reject anything that smacks of ' trade,' but experience has shown that the plan followed is of direct benefit to the reader. To take an example. It would, no doubt, be possible to give all the fundamental data about oil engines without describing a single particular make. But can anyone question seriously the advantage that is gained from the illustrations and descriptions of engines actually available ? Again, the properties desirable in insulating materials may be, and are, detailed, but the engineer who is called upon to use such materials is often in a hurry to know where they can be obtained. Or, once more, opening the volume at random, whilst there can be no doubt that the wiring rules for electricians are rightly included, is it not equally certain that it is a great convenience to have particulars of the materials, and the addresses of the makers of them, necessary for the proper observance of the rules ?

These are but a few examples taken as they came, but they indicate one of the characteristics which has made Kempe's Engineer's Year-Book popular with engineers in all parts of the world—and its encyclopædic nature makes it particularly valuable to the man on foreign service—for thirty-seven years.

In the present, the 38th edition, the principles which have guided the Editors from the beginning have been followed but with a slight modification which will appeal to all, and may prove of actual value to some, readers. Hitherto, it has been the custom to revise the text each year, with the result that complete repagination and re-indexing have been necessary. In the present volume the revision has only been made in the

text when it did not involve repagination ; in all other cases the latest information is given in addenda to the sections.

We believe this plan will be appreciated by all who desire to see at a glance the changes that the year has brought, for it will relieve them of the necessity of working through a whole section and picking out here and there a few lines, perhaps, that call for attention.

We submit the 38th Edition of the Year-Book, certain that it is as valuable as its predecessors and confident that it is worthy of the same popularity.

The Editor.

LIST OF ASSOCIATE-EDITORS AND CONTRIBUTORS.

In addition to the names included in this list, those of numerous other authorities who have furnished valuable information, or assisted in the revision of this work, are acknowledged in the text.

Contributor.	*Subject.*
A. H. Allen, M.I.E.E. . .	Electric Furnaces and data for other sections.
H. N. Allott, M.Inst.C.E. .	Arrangement and Construction of Engineering Workshops.
The Astronomer Royal . .	Astronomy.
S. W. Attwell, . . . National Physical Laboratory.	Measures.
W. Valentine Ball, M.A., O.B.E., Barrister-at-Law.	Law affecting Engineers.
Charles Bingham . .	Acetylene.
Ernest W. Birch, A.M.I.A.E.	Aero Engines; Automobiles; and Metallurgy.
C. Bride, M.A., B.Sc., A.I.C. .	Coke and Coking.
David Brownlie, A.M.I.Mech.E.	Fuels.
G. A. Burls, M.Inst.C.E . .	The Calculus and Curves.
Edward Butler, M.I.Mech.E. .	Wind and Wind Mills.
A. Vincent Clarke, . . . Assoc.M.Inst.C.E., M.I.Mech.E., A.M.I.E.E.	Internal Combustion Engines; Power Gas; and data for other sections.
A. G. Clausen, A.M.I.Mech.E.	Refrigeration.
H. F. Cornick, M.C., . . Assoc.M.Inst.C.E., late Major R.E.	Metal Bridges and Bridgework.
Brysson Cunningham, D.Sc., . F.R.S.E., M.Inst.C.E.	Harbour and Dock Engineering; and Navigable Waterways.
H. T. Davey, A.M.I.Mech.E. .	Gearing.
Alexander Drew . . .	Structural Engineering; and Reinforced Concrete Construction.
Prof. Owen W. Ellis, . . M.Sc. (Birm.), Assoc.M.Inst.C.E.	Metallurgy and Metallurgical Practice.
H. E. Fletcher, A.I.N.A., M.I.Mar.E.	Dredging for Alluvial Minerals.
John D. Frier, A.R.C.Sc., D.I.C., F.R.Aé.S.	Aircraft.

Contributor.	*Subject.*
Prof. JOHN GOODMAN, M.Inst.C.E.	Ball and Roller Bearings; Conveyors; Water Hammer in Pipes; and data for various other sections.
A. GROUNDS, B.Sc., A.I.C.	Briquette Fuel.
H. L. GUY, Assoc.M.Inst.C.E., M.I.Mech.E.	Steam Turbines.
Prof. B. P. HAIGH, M.B.E., D.Sc., M.Inst.C.E.	Fatigue in Metals.
R. T. HANCOCK, A.Inst.M.M.	Mining; Surveying; and Levelling.
LEONARD C. HARVEY	Pulverised Coal.
W. H. HATFIELD, D.Met.	Strength of Steel.
Prof. PERCY A. HILLHOUSE, D.Sc., M.I.N.A.	Naval Architecture and Marine Engineering.
R. B. HOBSON, A.M.C.T.	Fuels.
Prof. W. R. HODGKINSON, F.R.S.E., C.B.E.	Explosives.
J. L. HODGSON, B.Sc., Assoc.M.Inst.C.E., M.I.Mech.E.	Metering of Water, Air, Gas, Steam, and Coal; Dynamometers.
H. HOLCROFT, M.Inst.Loco.E.	Locomotives; and Railway Rolling Stock.
J. H. HOROBIN, Engineer, G.P.O., London.	Steam and Power Plants; Hydraulics; and data for other sections.
ALEXANDER L. HOWARD	Timber.
JOHN H. HUNTLEY, M.I.E.S.	Cranes and Lifting Appliances.
W. J. JONES, M.Sc., A.M.I.E.E.	Lighting.
C. NORMAN KEMP, B.Sc., F.R.S.E., A.I.C.	X-Rays in Industry.
COLIN LIVINGSTON	Blast-furnace Practice.
Professor HENRY LOUIS, M.A., D.Sc., M.Inst.C.E., F.G.S.	Coal and Metalliferous Mining; and Mineral Valuation.
J. F. MARYON, M.I.Mech.E.	Hydraulics.
ALWYNE MEADE, Assoc.M.Inst.C.E.	Gas and Gasworks Practice.
A. H. MIDDLETON	Coke and Coking.
R. FAIRFAX MIDDLETON, Assoc.M.Inst.C.E.	Waterworks Engineering.
H. A. MILES	Engineering Drawings; and Photo Copying.
HAROLD MOORE, M.Sc., F.C.S., M.I.Pet.Tech.	Fuels: Solid, Liquid, and Gaseous; and Lubricating Oils.
T. B. MORLEY, D.Sc., M.I.Mech.E.	Generation of Steam Power; Internal Combustion Turbines.
A. H. MUNDEY, Assoc.M.Inst.C.E., F.C.S., M.Inst.Met.	Die Casting.
E. W. L. NICOL, Assoc.M.Inst.C.E., M.I.M.E.	Coke as a Boiler Fuel.
HUGH S. POCOCK	Wireless Formulæ.
W. C. POPPLEWELL, M.Sc., Assoc.M.Inst.C.E.	Valve Gears.

Contributor.	*Subject.*
J. Roger Preston, M.I.Mech.E.	Heating by Hot Water, Steam, Gas, etc.
J. T. Rea	Cost data relating to Engineering Works.
D. Richardson, Wh. Ex., A.M.I.Mech.E., Capt. R.A.F.	Oxy-Acetylene and Electric Welding and Cutting.
L. S. Robson, A.M.I.Mech.E.	Steam Turbines.
H. F. Rutter, M.Inst.C.E.	Water Filtration.
Reginald Ryves, Assoc.M.Inst.C.E., F.R.Met.S.	Roads and Pavements; Irrigation; Rainfall; Bridges, Struts, Columns, Piles, etc.
Alfred B. Searle.	Mortars, Concretes, Cements, Plasters, and Glues; Clays and Allied Products; Grinding, Abrasives, and Polishing Appliances; and Depreciation of Plant and Machinery.
C. Seymour, District Engineer, Midland Great Western Railway of Ireland.	Railways; Piles, Pile-driving, Dams, Caissons; and data for other sections.
E. J. Silcock, M.Inst.C.E., F.G.S., F.S.I.	Sewerage, Sewage Disposal, and House Drainage.
Prof. Dempster Smith, M.B.E., M.Sc., M.I.Mech.E.	Machine Tools; Workshop Practice; Power Transmission; and Bearings.
Prof. S. Parker Smith, D.Sc., Assoc.M.Inst.C.E., M.I.E.E.	Electrical Engineering and Electric Traction.
Herbert Stone	Timber.
R. A. Sturgeon, Assoc.M.Inst.C.E.	Steel Helical Springs.
A. L. Summers.	Anthracite.
Arthur Surveyer, D.Eng., M.E.I.C.	Water Power Development.
James Thomson, M.I.N.A.	Marine Engineering.
F.W. Thorne, Assoc.M.Inst.C.E., A.M.I.Mech.E.	Fatigue in Metals.
Robert S. Vesey, Memb. Soc. of Chemical Industry.	Timber Creosoting.
C. F. Wade, A.M.I.Mech.E., A.M.I.E.E.	Commercial Selection of Coal; Waste Heat Recovery; and Boiler Draught.
G. Stanley Walpole, D.Sc., F.I.C.	Timber Seasoning.
Hal Williams, M.I.Mech.E., M.I.E.E.	Refrigeration.

STONE'S SPECIALITIES

Railway and Electrical Department

Bronze Locomotive Axle Boxes and Bearings, Connecting and Coupling Rod Bushes, etc. Locomotive Boosters.
Bronze Slide Valves, Bronze Carriage and Wagon Bearings.
Bronze Firebox Stays. Eitherside Wagon Brakes.
'Tanya' Suspension System for Tank Wagons. Roller Axle Boxes.
Train Lighting, Heating, Cooking, Refrigerating and Ventilating Installations.

Marine Department

Stone's High Tension Manganese Bronze Propellers and Propeller Blades.
Stone's Bronze for all Castings for High Tensile Work.
Stone's Whitemetals for lining Bearings.
The 'Stone' System for Hydraulically Controlling Bulkhead Doors.
Stone's Patent Hydraulic Underline Ash Expeller.
Ships' Windows and Sidelights.
Stone's Centrifugal Pumps for all duties. Ships' Pumps of all kinds.
Pumps for Fuel and Lubricating Oils and Industrial Viscous Fluids.
General Ships' Fittings. Copper Boat Nails and Rivets, etc.

Water and Drainage Ironwork Fittings Department

Valves and Fittings for Water Supply. Pumps for all Duties.
Internal Water Fittings for Buildings. Steam Fittings.
Ironwork for Sewerage and Sewage Disposal.

Copper Nail Department

Copper, Zinc, Aluminium, and Cast-Metal Nails.
Copper, Brass, and Aluminium Rivets and Washers.
Copper Boat Nails, Rose Burrs, Tacks, Brads, Spikes, etc.
Copper Furnace Pans. Coppersmiths' Work of all kinds.

Motor and Aero Department

Non-ferrous Alloys of all descriptions.
Motor and Aero Engine Castings in :—Manganese Bronze, Phosphor Bronze, Gunmetal, Brass, Aluminium, etc.
Stampings, Die Castings and Forgings in Non-Ferrous Alloys.
Cast and Machined Bushes and Parts of all descriptions.
Cored and Solid Phosphor Bronze Stick. Bearing Metals.
Copper and Aluminium Rivets and Washers.
Copper Nails, Roves, and Tacks.

H-R. Reduction Gears. Water Distilling Apparatus.
General Engineering in all its Branches.

We shall be pleased to send catalogues on application.

J. STONE & COMPANY LTD

Deptford, London—S.E. 14

London Office : 1a Cockspur Street, S.W. 1

CONTENTS.

SECTION		PAGE
I.	LENGTHS—SURFACES—WEIGHTS—CAPACITIES—FOREIGN MEASURES —WIRE GAUGES—SIZES OF PAPER, ETC.	1
II.	FORCES AND POWERS	23
III.	*Part I*: SURVEYING—ASTRONOMY IN FIELD WORK—*Part II*: DRAWING OFFICE PRACTICE	51
IV.	HEAT—SOUND—OPTICS	99
V.	*Part I*: WEIGHTS AND SPECIFIC GRAVITIES — *Part II*: TENSILE AND LATERAL STRENGTH OF MATERIALS	123
VI.	BEAMS—COLUMNS—STRUTS—RIVETING—STEEL ROOF FRAMING AND COVERING—BRITISH STANDARD SECTIONS	207
VII.	BRIDGES AND BRIDGEWORK (METAL)	247
VIII.	WOODEN BEAMS—STRUTTING AND PUGGING—ROOFING MATERIALS—FLOORS—PARTITIONS—STAIRS	311
IX.	*Part I*: BUILDING MATERIALS AND CONSTRUCTION—*Part II*: MORTARS—CONCRETES—CEMENT—PLASTERS—GLUES—*Part III*: PLASTERING—SLATING—PAINTING—PLUMBING	347
X.	CLAYS AND ALLIED PRODUCTS USED BY ENGINEERS	425
XI.	REINFORCED CONCRETE CONSTRUCTION	439
XII.	EARTHWORK — EXCAVATION — FOUNDATIONS — PILING — RETAINING WALLS — STAGINGS — SHORING — CHIMNEYS — ARCHES— PIERS—BRIDGES	481
XIII.	RAILWAY ENGINEERING—*Part I*: PERMANENT WAY—*Part II*: LOCOMOTIVES AND ROLLING STOCK—*Part III*: SIGNALS—*Part IV*: RAILWAY EXPENDITURE AND RECEIPTS	529
XIV.	ELECTRIC TRACTION	669
XV.	HIGHWAY ENGINEERING	711
XVI.	ENGINEERING WORKSHOPS—WORKSHOP PRACTICE	765
XVII.	FLOUR, COTTON, OIL, AND SUGAR MILLS	783
XVIII.	CRANES AND LIFTING APPLIANCES—AERIAL ROPEWAYS—CONVEYORS	801
XIX.	POWER TRANSMISSION — *Part I*: BELTING — ROPE DRIVING — CLUTCHES—*Part II*: CHAIN GEARING—*Part III*: GEARING—*Part IV*: LINK MOTIONS	855
XX.	*Part I*: BEARING PRESSURES — BALL AND ROLLER BEARINGS—*Part II*: LUBRICATING OILS—LUBRICATION AND LUBRICATORS—*Part III*: VISCOSITY AND VISCOUS FLOW OF OILS—*Part IV*: SHAFTING AND SHAFTS—*Part V*: SPRINGS	961
XXI.	*Part I*: MACHINE TOOLS—TOOL STEEL—TOOLS—*Part II*: GRINDING, ABRASIVES, AND POLISHING APPLIANCES—*Part III*: POWER HAMMERS	1019
XXII.	SCREW THREADS—SCREWS—BOLTS AND NUTS	1139
XXIII.	THE ATMOSPHERE—WIND—WINDMILLS—AIR-RESISTANCE—AIR COMPRESSION—AIR COMPRESSORS—PNEUMATIC DESPATCH TUBES	1181
XXIV.	REFRIGERATION—ICE MAKING—COLD STORAGE	1209
XXV.	*Part I*: HYDRAULICS—*Part II*: PUMPS—*Part III*: WATER POWER DEVELOPMENT—*Part IV*: TURBINES AND HYDRO-ELECTRIC PLANTS—*Part V*: HYDRAULIC MACHINERY	1239
XXVI.	*Part I*: SANITARY ENGINEERING—*Part II*: IRRIGATION—*Part III*: RAINFALL—WELLS—WATERWORKS—FLOW OF WATER IN PIPES AND CHANNELS	1359

SECTION		PAGE
XXVII.	Metering of Water, Air, Gas, Steam, and Coal	1435
XXVIII.	Harbours — Docks — Quay-Side Appliances — Graving and Floating Docks—Dredging—Subaqueous Rock Removal—Channel Demarcation—Buoys — Lightships—Lighthouses—Diving	1455
XXIX.	Navigable Waterways — River Canalisation — Canal Locks and Lifts—Traction and Haulage—Ship Canals	1505
XXX.	Steam Engineering.	
	Part I: Properties and Flow of Steam	1531
	Part II: Steam Generating Plant	1541
	Part III: Reciprocating Steam Engines	1705
	Part IV: Steam Turbines	1767
	Part V: Condensing Plant—Condensers—Air Pumps—Feed Pumps—Cooling Towers—Steam Accumulators	1825
XXXI.	Internal Combustion Engineering—*Part I*: Power Gas—*Part II*: Gas Engines—*Part III*: Oil Engines—Hot Air Engines	1861
XXXII.	Automobiles	1935
XXXIII.	Aero Engineering.	
	Part I: Aircraft—*Part II*: Aero Engines	1983
XXXIV.	Fuels: Solid, Liquid, and Gaseous	2077
XXXV.	Electrical Engineering	2203
XXXVI.	Naval Architecture and Marine Engineering	2353
XXXVII.	Gas—Gas Works—Acetylene Gas—Petrol Air Gas	2449
XXXVIII.	*Part I*: Heating Buildings by Hot Water, Steam, and Gas—*Part II*: Ventilation—*Part III*: Lighting—*Part IV*: Fire Extinction	2491
XXXIX.	Mine Engineering.	
	Part I: Mining and Mining Machinery—*Part II*: Mineral Valuation	2555
XL.	*Part I*: Metallurgy. *Part II*: Oxy-Acetylene and Electric Welding and Cutting	2659
XLI.	Explosives	2797
XLII.	Workshop Recipes and Processes — Painting — Varnishes—Metal Lacquers and Bronzes — Lubricants and Oils — Cleaning Compositions	2801
XLIII.	Radio Communication and Formulæ	2821
XLIV.	Patents, Trade Marks, and Designs	2833
XLV.	Cost Data relating to Engineering Works	2889
XLVI.	Depreciation of Plant and Machinery	2897
XLVII.	Legal Notes for Engineers	2903
INDEX		2913
Directory of Engineering Manufacturers, etc.		2989

For Classified Directory and Buyers' Guide, see pages 2999 to 3039.

SECTION I

LENGTHS — SURFACES — WEIGHTS — CAPACITIES — WIRE GAUGES — FOREIGN WEIGHTS, MEASURES, AND MONEY, WITH THEIR ENGLISH EQUIVALENTS—SIZES OF PAPER

(pp. 1–19)

(Revised by S. W. Attwell, of the National Physical Laboratory.)

SECTION I

LENGTHS—SURFACES—WEIGHTS—CAPACITIES—FOREIGN MEASURES—WIRE-GAUGES—SIZES OF PAPER, etc.

(Revised by S. W. Attwell, of the National Physical Laboratory.)

The accepted relations of English Imperial to International Metric Measures are thus given by the Standards Office of the Board of Trade :

1 inch = 25·399978 mm.; 1 metre = 39·370113 inches ; 1 gallon = 4·5459631 litres ; 1 litre = 1·75980 pint ; 1 pound = 0·45359243 kilogram ; 1 kilogram = 2·2046223 pounds.

In 1893 the United States of America recognised the metre as the fundamental standard of length, and adopted the conversion value of 1 metre = 39·37 inches.

10 penny diameters = 1 ft.; halfpenny = 1 inch ; 5-shilling piece = 1½ inch. Weight of 3 pennies, 1 five-shilling piece, or 5 shillings, each = 1 oz.

LENGTHS.

English.

The length of the standard yard of 36 inches is such that a pendulum vibrating seconds of mean time, temperature 62° F., at the level of the sea and latitude of Greenwich, has a length of 39·1393 inches.

Miles.	Furlongs.	Chains.	Rods.	Yards.	Feet.	Inches.
1	8	80	320	1,760	5,280	63,360
·125	1	10	40	220	660	7,920
·0125	·1	1	4	22	66	792
·003125	·025	·25	1	5·5	16·5	198
·00056818	·0045454	·045454	.181818	1	3	36
·00018939	·00151515	·01515151	·060606	·33333	1	12
·000015783	·000126262	·001262626	·0050505	·0277777	·083333	1

GUNTER'S SURVEYING CHAIN.

7·92 inches = 1 link. 100 links = 1 chain 4 rods, or 22 yards. 80 chains = 1 mile.

ROPES AND CABLES.

1 cable-length = 120 fathoms = 720 feet. 1 fathom = 6 feet.

GEOGRAPHICAL AND NAUTICAL MEASURES.

1 degree of a great circle of the earth round the equator = 69·174 statute miles = 365,239 feet. 1 nautical mile (knot) = 6,080 feet. 1 telegraph mile = 6,087 feet.

LOG-LINE.

The log-line should be about 150 fathoms long, and 10 fathoms from the log to the first knot on the line. If half-a-minute glass is used, it will be 50·66 feet between each succeeding knot. For 28-seconds glass it will be 47·29 feet = 7·88 fathoms per knot. This is the length of knot by calculation, but practically it is shortened to 7·5 fathoms per knot for 28-seconds glass.

Metric System.

Name.	Ins.	Feet.	Yards.	Miles.
Millimetre†	·03937	·003281		
Centimetre‡	·39370113	·032808		
Decimetre	3·9370113	·3280843	·10936	
Metre§	39·370113	3·280843	1·0936143	
Decametre	393·70113	32·80843	10·936143	
Hectometre	Road	328·0843	109·36143	·06213718
Kilometre	measures.	3280·843	1093·6143	·6213718
Myriametre		32808·43	10936·143	6·213718

† Nearly the $\frac{1}{25}$th part of an inch. ‡ Full $\frac{3}{8}$ inch.
§ Very nearly 3 ft. 3$\frac{3}{8}$ ins., which is too long by only 1 part in 8,056.

Conversion of English Inches into Centimetres.

Ins.	0	1	2	3	4	5	6	7	8	9
	Cms.	Cms.	Cms.	Cms.	Cms.	Cms.	Cms.	Cms.	Cms.	Cms.
0	·00	2·54	5·08	7·62	10·16	12·70	15·24	17·78	20·32	22·86
10	25·40	27·94	30·48	33·02	35·56	38·10	40·64	43·18	45·72	48·26
20	50·80	53·34	55·88	58·42	60·96	63·50	66·04	68·58	71·12	73·66
30	76·20	78·74	81·28	83·82	86·36	88·90	91·44	93·98	96·52	99·06
40	101·60	104·14	106·68	109·22	111·76	114·30	116·84	119·38	121·92	124·46
50	127·00	129·54	132·08	134·62	137·16	139·70	142·24	144·78	147·32	149·86
60	152·40	154·94	157·48	160·02	162·56	165·10	167·64	170·18	172·72	175·26
70	177·80	180·34	182·88	185·42	187·96	190·50	193·04	195·58	198·12	200·66
80	203·20	205·74	208·28	210·82	213·36	215·90	218·44	220·98	223·52	226·06
90	228·60	231·14	233·68	236·22	238·76	241·30	243·84	246·38	248·92	251·46
100	254·00	256·54	259·08	261·62	264·16	266·70	269·24	271·78	274·32	276·86

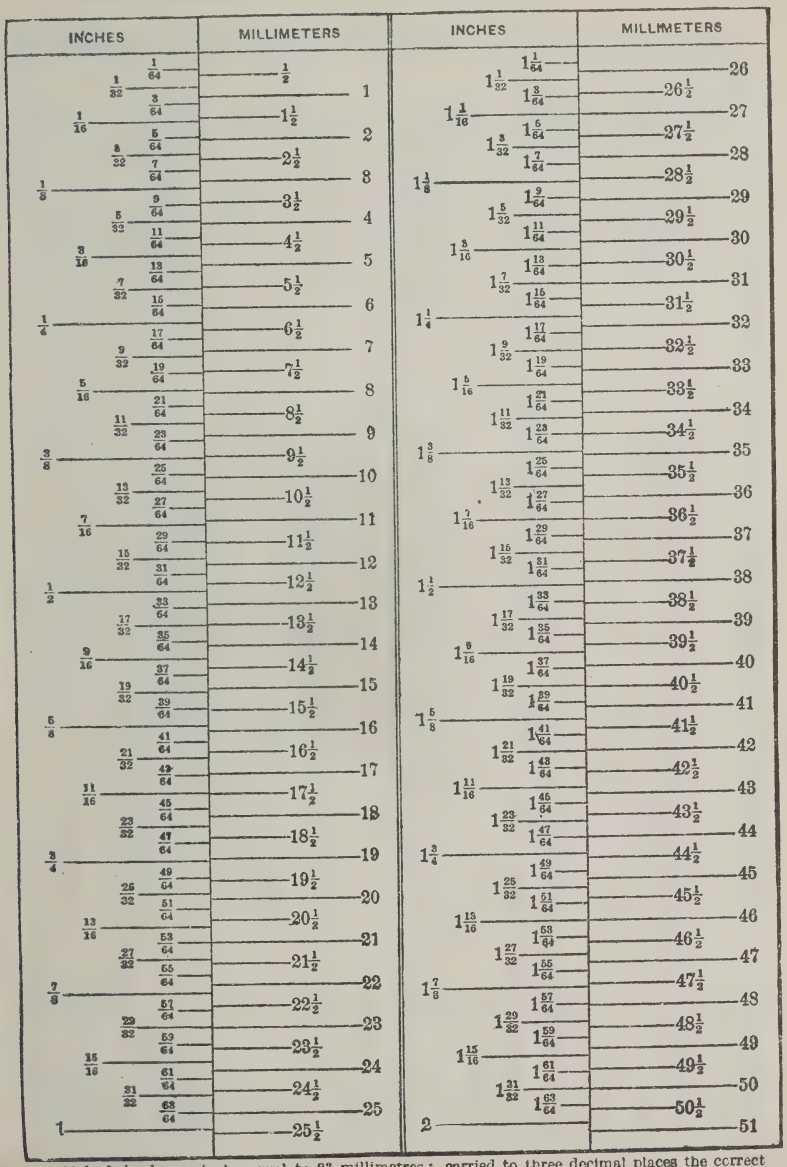
The above table is correct throughout to within 1 part in 1,000,000.

Conversion of Centimetres into English Inches.

Cms.	0	1	2	3	4	5	6	7	8	9
	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
0	·0000	·3937	·7874	1·1811	1·5748	1·9685	2·3622	2·7559	3·1496	3·5433
10	3·9370	4·3307	4·7244	5·1181	5·5118	5·9055	6·2992	6·6929	7·0866	7·4803
20	7·8740	8·2677	8·6614	9·0551	9·4488	9·8425	10·236	10·630	11·024	11·417
30	11·811	12·205	12·598	12·992	13·386	13·780	14·173	14·567	14·961	15·354
40	15·748	16·142	16·535	16·929	17·323	17·717	18·110	18·504	18·898	19·291
50	19·685	20·079	20·472	20·866	21·260	21·654	22·047	22·441	22·835	23·228
60	23·622	24·016	24·409	24·803	25·197	25·591	25·984	26·378	26·772	27·165
70	27·559	27·953	28·346	28·740	29·134	29·528	29·921	30·315	30·709	31·102
80	31·496	31·890	32·283	32·677	33·071	33·465	33·858	34·252	34·646	35·039
90	35·433	35·827	36·221	36·614	37·008	37·402	37·795	38·189	38·583	38·976
100	39·370	39·764	40·158	40·551	40·945	41·339	41·732	42·126	42·520	42·913

Conversion of English Feet into Metres.

Ft.	0	1	2	3	4	5	6	7	8	9
	Metres.	Metres.	Metres.	Metres.	Metres.	Metres.	Metres.	Metres.	Metres.	Metres.
0	·0000	·3048	·6096	·9144	1·2192	1·5240	1·8288	2·1336	2·4384	2·7432
10	3·0480	3·3528	3·6576	3·9624	4·2672	4·5720	4·8768	5·1816	5·4864	5·7912
20	6·0960	6·4008	6·7056	7·0104	7·3152	7·6200	7·9248	8·2296	8·5344	8·8392
30	9·1440	9·4488	9·7536	10·058	10·363	10·668	10·973	11·278	11·582	11·887
40	12·192	12·497	12·802	13·106	13·411	13·716	14·021	14·326	14·630	14·935
50	15·240	15·545	15·850	16·154	16·459	16·764	17·069	17·374	17·678	17·983
60	18·288	18·593	18·898	19·202	19·507	19·812	20·117	20·422	20·726	21·031
70	21·336	21·641	21·946	22·250	22·555	22·860	23·165	23·470	23·774	24·079
80	24·384	24·689	24·994	25·298	25·603	25·908	26·213	26·518	26·822	27·127
90	27·432	27·737	28·042	28·346	28·651	28·956	29·261	29·566	29·870	30·175
100	30·480	30·785	31·090	31·394	31·699	32·004	32·309	32·614	32·918	33·223



$\frac{29}{32}$ inch is shown to be equal to 23 millimetres; carried to three decimal places the correct value would be 23·019 millimetres.

(*C. H. W.*, '*Machinery*,' *June* 26, 1919.)

INCHES REDUCED TO DECIMALS OF A FOOT.

Ins.	Foot.	Ins.	Foot.	Ins.	Foot.	Ins.	Foot.	Ins.	Foot.	Ins.	Foot.
0	·0000	2	·1667	4	·3333	6	·5000	8	·6667	10	·8333
1/32	·0026	1/32	·1693	1/32	·3359	1/32	·5026	1/32	·6693	1/32	·8359
1/16	·0052	1/16	·1719	1/16	·3385	1/16	·5052	1/16	·6719	1/16	·8385
3/32	·0078	3/32	·1745	3/32	·3411	3/32	·5078	3/32	·6745	3/32	·8411
1/8	·0104	1/8	·1771	1/8	·3438	1/8	·5104	1/8	·6771	1/8	·8438
5/32	·0130	5/32	·1797	5/32	·3464	5/32	·5130	5/32	·6797	5/32	·8464
3/16	·0156	3/16	·1823	3/16	·3490	3/16	·5156	3/16	·6823	3/16	·8490
7/32	·0182	7/32	·1849	7/32	·3516	7/32	·5182	7/32	·6849	7/32	·8516
1/4	·0208	1/4	·1875	1/4	·3542	1/4	·5208	1/4	·6875	1/4	·8542
9/32	·0234	9/32	·1901	9/32	·3568	9/32	·5234	9/32	·6901	9/32	·8568
5/16	·0260	5/16	·1927	5/16	·3594	5/16	·5260	5/16	·6927	5/16	·8594
11/32	·0286	11/32	·1953	11/32	·3620	11/32	·5286	11/32	·6953	11/32	·8620
3/8	·0313	3/8	·1979	3/8	·3646	3/8	·5313	3/8	·6979	3/8	·8646
13/32	·0339	13/32	·2005	13/32	·3672	13/32	·5339	13/32	·7005	13/32	·8672
7/16	·0365	7/16	·2031	7/16	·3698	7/16	·5365	7/16	·7031	7/16	·8698
15/32	·0391	15/32	·2057	15/32	·3724	15/32	·5391	15/32	·7057	15/32	·8724
1/2	·0417	1/2	·2083	1/2	·3750	1/2	·5417	1/2	·7083	1/2	·8750
17/32	·0443	17/32	·2109	17/32	·3776	17/32	·5443	17/32	·7109	17/32	·8776
9/16	·0469	9/16	·2135	9/16	·3802	9/16	·5469	9/16	·7135	9/16	·8802
19/32	·0495	19/32	·2161	19/32	·3828	19/32	·5495	19/32	·7161	19/32	·8828
5/8	·0521	5/8	·2188	5/8	·3854	5/8	·5521	5/8	·7188	5/8	·8854
21/32	·0547	21/32	·2214	21/32	·3880	21/32	·5547	21/32	·7214	21/32	·8880
11/16	·0573	11/16	·2240	11/16	·3906	11/16	·5573	11/16	·7240	11/16	·8906
23/32	·0599	23/32	·2266	23/32	·3932	23/32	·5599	23/32	·7266	23/32	·8932
3/4	·0625	3/4	·2292	3/4	·3958	3/4	·5625	3/4	·7292	3/4	·8958
25/32	·0651	25/32	·2318	25/32	·3984	25/32	·5651	25/32	·7318	25/32	·8984
13/16	·0677	13/16	·2344	13/16	·4010	13/16	·5677	13/16	·7344	13/16	·9010
27/32	·0703	27/32	·2370	27/32	·4036	27/32	·5703	27/32	·7370	27/32	·9036
7/8	·0729	7/8	·2396	7/8	·4063	7/8	·5729	7/8	·7396	7/8	·9063
29/32	·0755	29/32	·2422	29/32	·4089	29/32	·5755	29/32	·7422	29/32	·9089
15/16	·0781	15/16	·2448	15/16	·4115	15/16	·5781	15/16	·7448	15/16	·9115
31/32	·0807	31/32	·2474	31/32	·4141	31/32	·5807	31/32	·7474	31/32	·9141
1	·0833	3	·2500	5	·4167	7	·5833	9	·7500	11	·9167
1/32	·0859	1/32	·2526	1/32	·4193	1/32	·5859	1/32	·7526	1/32	·9193
1/16	·0885	1/16	·2552	1/16	·4219	1/16	·5885	1/16	·7552	1/16	·9219
3/32	·0911	3/32	·2578	3/32	·4245	3/32	·5911	3/32	·7578	3/32	·9245
1/8	·0938	1/8	·2604	1/8	·4271	1/8	·5938	1/8	·7604	1/8	·9271
5/32	·0964	5/32	·2630	5/32	·4297	5/32	·5964	5/32	·7630	5/32	·9297
3/16	·0990	3/16	·2656	3/16	·4323	3/16	·5990	3/16	·7656	3/16	·9323
7/32	·1016	7/32	·2682	7/32	·4349	7/32	·6016	7/32	·7682	7/32	·9349
1/4	·1042	1/4	·2708	1/4	·4375	1/4	·6042	1/4	·7708	1/4	·9375
9/32	·1068	9/32	·2734	9/32	·4401	9/32	·6068	9/32	·7734	9/32	·9401
5/16	·1094	5/16	·2760	5/16	·4427	5/16	·6094	5/16	·7760	5/16	·9427
11/32	·1120	11/32	·2786	11/32	·4453	11/32	·6120	11/32	·7786	11/32	·9453
3/8	·1146	3/8	·2813	3/8	·4479	3/8	·6146	3/8	·7813	3/8	·9479
13/32	·1172	13/32	·2839	13/32	·4505	13/32	·6172	13/32	·7839	13/32	·9505
7/16	·1198	7/16	·2865	7/16	·4531	7/16	·6198	7/16	·7865	7/16	·9531
15/32	·1224	15/32	·2891	15/32	·4557	15/32	·6224	15/32	·7891	15/32	·9557
1/2	·1250	1/2	·2917	1/2	·4583	1/2	·6250	1/2	·7917	1/2	·9583
17/32	·1276	17/32	·2943	17/32	·4609	17/32	·6276	17/32	·7943	17/32	·9609
9/16	·1302	9/16	·2969	9/16	·4635	9/16	·6302	9/16	·7969	9/16	·9635
19/32	·1328	19/32	·2995	19/32	·4661	19/32	·6328	19/32	·7995	19/32	·9661
5/8	·1354	5/8	·3021	5/8	·4688	5/8	·6354	5/8	·8021	5/8	·9688
21/32	·1380	21/32	·3047	21/32	·4714	21/32	·6380	21/32	·8047	21/32	·9714
11/16	·1406	11/16	·3073	11/16	·4740	11/16	·6406	11/16	·8073	11/16	·9740
23/32	·1432	23/32	·3099	23/32	·4766	23/32	·6432	23/32	·8099	23/32	·9766
3/4	·1458	3/4	·3125	3/4	·4792	3/4	·6458	3/4	·8125	3/4	·9792
25/32	·1484	25/32	·3151	25/32	·4818	25/32	·6484	25/32	·8151	25/32	·9818
13/16	·1510	13/16	·3177	13/16	·4844	13/16	·6510	13/16	·8177	13/16	·9844
27/32	·1536	27/32	·3203	27/32	·4870	27/32	·6536	27/32	·8203	27/32	·9870
7/8	·1563	7/8	·3229	7/8	·4896	7/8	·6563	7/8	·8229	7/8	·9896
29/32	·1589	29/32	·3255	29/32	·4922	29/32	·6589	29/32	·8255	29/32	·9922
15/16	·1615	15/16	·3281	15/16	·4948	15/16	·6615	15/16	·8281	15/16	·9948
31/32	·1641	31/32	·3307	31/32	·4974	31/32	·6641	31/32	·8307	31/32	·9974

Conversion of Metres into English Feet.

Metres.	0	1	2	3	4	5	6	7	8	9
	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.
0	·0000	3·2808	6·5617	9·8425	13·123	16·404	19·685	22·966	26·247	29·528
10	32·808	36·089	39·370	42·651	45·932	49·213	52·493	55·774	59·055	62·336
20	65·617	68·898	72·179	75·459	78·740	82·021	85·302	88·583	91·864	95·144
30	98·425	101·71	104·99	108·27	111·55	114·83	118·11	121·39	124·67	127·95
40	131·23	134·51	137·80	141·08	144·36	147·64	150·92	154·20	157·48	160·76
50	164·04	167·32	170·60	173·88	177·17	180·45	183·73	187·01	190·29	193·57
60	196·85	200·13	203·41	206·69	209·97	213·25	216·54	219·82	223·10	226·38
70	229·66	232·94	236·22	239·50	242·78	246·06	249·34	252·62	255·91	259·19
80	262·47	265·75	269·03	272·31	275·59	278·87	282·15	285·43	288·71	292·00
90	295·28	298·56	301·84	305·12	308·40	311·68	314·96	318·24	321·52	324·80
100	328·08	331·37	334·65	337·93	341·21	344·49	347·77	351·05	354·33	337·61

Conversion of English Statute Miles into Kilometres.

Miles.	0	1	2	3	4	5	6	7	8	9
	Kiloms.	Kiloms.	Kiloms.	Kiloms.	Kiloms.	Kiloms.	Kiloms.	Kiloms.	Kilom.	Kilom.
0	·0000	1·6093	3·2187	4·8280	6·4374	8·0467	9·6561	11·265	12·875	14·484
10	16 093	17·703	19·312	20·921	22·531	24·140	25·749	27·359	28·968	30·578
20	32·187	33·796	35·406	37·015	38·624	40·234	41·843	43·452	45·062	46·671
30	48·280	49·890	51·499	53·108	54·718	56·327	57·936	59·546	61·155	62·764
40	64 374	65·983	67·592	69·202	70·811	72·420	74·030	75·639	77·248	78·858
50	80·467	82·076	83·686	85·295	86·905	88·514	90·123	91·733	93·342	94·951
60	96·561	98·170	99·779	101·39	103·00	104·61	106·22	107·83	109·44	111·04
70	112·65	114·26	115·87	117·48	119·09	120·70	122·31	123·92	125·53	127·14
80	128·75	130·36	131·97	133·58	135·18	136·79	138·40	140·01	141·62	143·23
90	144·84	146·45	148·06	149·67	151·28	152·89	154·50	156·11	157·72	159·32
100	160·93	162·54	164·15	165·76	167·37	168·98	170·59	172·20	173·81	175·42

Conversion of Kilometres into English Statute Miles.

Kiloms.	0	1	2	3	4	5	6	7	8	9
	Miles.	Miles.	Miles.	Miles.	Miles.	Miles.	Miles.	Miles.	Miles.	Miles
0	·0000	·6214	1·2427	1·8641	2·4855	3·1069	3·7282	4·3496	4·9710	5·5923
10	6·2137	6·8351	7·4565	8·0778	8·6992	9·3206	9·9419	10·563	11·185	11·806
20	12·427	13·049	13·670	14·292	14·913	15·534	16·156	16·777	17·398	18·020
30	18·641	19·263	19·884	20·505	21·127	21·748	22·369	22·991	23·612	24·234
40	24·855	25·476	26·098	26·719	27·340	27·962	28·583	29·204	29·826	30·447
50	31·069	31·690	32·311	32·933	33·554	34·175	34·797	35·418	36·040	36·661
60	37·282	37·904	38·525	39·146	39·768	40·389	41·011	41·632	42·253	42·875
70	43·496	44·117	44·739	45·360	45·982	46·603	47·224	47·846	48·467	49·088
80	49·710	50·331	50·952	51·574	52·195	52·817	53·438	54·059	54·681	55·302
90	55·923	56·545	57·166	57·788	58·409	59·030	59·652	60·273	60·894	61·516
100	62·137	62·759	63·380	64·001	64·623	65·244	65·865	66·487	67·108	67·730

SURFACES.

English.

Sq. Miles.	Acres.	Sq. Chains.	Sq. Rods.	Sq. Yards.	Sq. Feet.	Sq. Inches.
1	640	6,400	102,400	3,097,600	27,878,400	4,014,489,600
·001562	1	10	160	4,840	43,560	6,272,640
·0001562	·1	1	16	484	4,356	627,264
·000009766	.00625	·0625	1	30·25	272·25	39,204
·000000323	.0002066	·002066	·033	1	9	1,296
·0000000359	·00002296	·0002296	·00367	·1111111	1	144
·00000000025	·000000159	·00000159	·00002551	·0007716	·006945	1

Metrical.

Name.	Sq. Ins.	Sq. Ft.	Sq. Yards.	Acres.
Sq. Millimetre . .	·00155	·00001076	·0000012	
Sq. Centimetre. .	·1550	·0010764	·0001196	
Sq. Decimetre . .	15·50006	·1076393	·0119599	
Sq. Metre, or Centiare	1550·006	10·76393	1·195992	·000247
Sq. Decametre, or Are	1550006	1076·393	119·5992	·024711
Sq. Hectometre or Hectare . .	—	107639·3	11959·92	2·471058
Sq. Kilometre . .	·3861029 sq.	—	1195992	247·1058
Sq. Myriametre .	38·61029 miles	—	—	24710·58

WEIGHTS.

English.

One cubic inch of distilled water (weighed in air against brass weights) at temperature 62° F., the barometer being at 30 ins., weighs 252·326 grains, of which 7,000 go to the pound avoirdupois, and 5,760 to the pound troy.

AVOIRDUPOIS.

Ton.	Cwts.	Pounds.	Ounces.	Drams.
1	20	2240	35840	573440
·05	1	112	1792	28672
·00044643	·0089286	1	16	256
·0000279	·000558	·0625	1	16
·000001744	·00003488	·003906	·0625	1

TROY.

Pound.	Ounces.	Dwts.	Grains.	Pound Avoir.
1	12	240	5760	·822857
·083333	1	20	480	·068571
·004166	·05	1	24	·00342857
·0001736	·002083333	·0416666	1	·000142857
1·215278	14·58333	291·6666	7000	1

APOTHECARIES'.

Pound.	Ounces.	Drams.	Scruples.	Grains.
1	12	96	288	5760
·08333	1	8	24	480
·01041666	·125	1	3	60
·0034722	·0416666	·3333	1	20
·00017361	·0020833	·016666	·05	1

Metrical.

1 gramme = 15·432356 grains, and is the weight of a cubic centimetre of distilled water in vacuo at 4° Centigrade, or 39·2° Fahr.

Weight.	Grammes.	Avoirdupois Ounces.	Avoirdupois Lbs.	Cwts.	Ton.	Grains.
Milligramme .	·001	—	—	—	—	·015
Centigramme .	·01	—	—	—	—	·154
Decigramme .	·1	—	—	—	—	1·543
Gramme . .	1	·035	·0022	—	—	15·432356
Decagramme .	10	·35	·022			
Hectogramme	100	3·527	·22046			
Kilogramme .	1000	35·2739	2·204622	·019	·00098	
Myriagramme	10000	—	22·04622	·1968	·00984	
Quintal . .	100000	—	220·4622	1·9684	·0984	
Millier or Bar	1000000	—	2204·622	19·68413	·984206	

CONVERSION OF METRICAL TO ENGLISH WEIGHTS.

Grammes to Grains.		Kilogrammes to Ounces. (Avoir.)		Kilogrammes to Pounds (Avoir.)	
1=15·432356	6=92·594138	1=35·27396	6=211·64374	1=2·204622	6=13·227734
2=30·864713	7=108·02649	2=70·54791	7=246·91770	2=4·409245	7=15·432356
3=46·297069	8=123·45885	3=105·82187	8=282·19166	3=6·613867	8=17·636978
4=61·729426	9=138·89121	4=141·09583	9=317·46561	4=8·818489	9=19·841601
5=77·161782		5=176·36978		5=11·023111	

A milligramme is about $\frac{1}{65}$ grain.

CONVERSION OF ENGLISH TO METRICAL WEIGHTS.

Grains to Grammes.	Ounces (Avoir.) to Grammes.	Pounds (Avoir.) to Kilogrammes.	Hundredweights to Kilogrammes.
1=·0647989	1= 28·34953	1= ·4535924	1= 50·80235
2=·1295978	2= 56·69905	2= ·9071849	2=101·6047
3=·1943968	3= 85·04858	3=1·360777	3=152·4071
4=·2591957	4=113·3981	4=1·814370	4=203·2094
5=·3239946	5=141·7476	5=2·267962	5=254·0118
6=·3887935	6=170·0972	6=2·721555	6=304·8141
7=·4535924	7=198·4467	7=3·175147	7=355·6165
8=·5183914	8=226·7962	8=3·628739	8=406·4188
9=·5831903	9=255·1457	9=4·082332	9=457·2212

The Metric Carat.

By decision of the Standards Department of the Board of Trade, the metric carat of 200 milligrams, and its necessary multiples and sub-multiples, are standard weights in the United Kingdom.

CAPACITIES.

English.

Gallon.—The standard gallon (*International Bureau of Standards, Paris*, 1921) measures 277·420 cubic inches, and contains 10 lbs. (avoir.) weight of distilled water (weighed in air against brass weights) at the temperature of 62° F., the barometer being at 30 ins.

DRY MEASURE.

Cu. Yard.	Bushels.	Cu. Feet.	Pecks.	Gallons.	Cu. Ins.
1	21·0223	27	84·089	168·179	46656
·047568	1	1·28435	4	8	2219·35
·037037	·7786	1	3·1144	6·22884	1728
·011892	·25	·32109	1	2	554·838
	·125	·16054	·5	1	277·420
		·00579	·001802	·0036047	1

LIQUID MEASURE.

Gallon.	Quarts.	Pints.	Gills.	Cub. Ins.
1	4	8	32	277·420
·25	1	2	8	69·355
·125	·5	1	4	34·677
·03125	·125	·25	1	8·6693
·0036047	·014419	·028837	·11535	1

APOTHECARIES' MEASURES.

Gallon (C) = 8 pints (O); 1 pint = 20 fluid ounces (oz. weight of water).
Ounce (f℥) = 8 drachms (f℥) = 480 minims (♏) = 720 drops (gtt).
One wine-glass = 4 tablespoonfuls = 16 teaspoonfuls = 2 ounces.
Symbols.—f. or fl., fluid; ss., one half; aa., for each. Thus, f℥ss. means ½ a fluid ounce.

Conversion of Metrical to English Capacities.

Cub. Centimetres to Cub. Ins.	Litres to Fluid Ounces.	Litres to Pints.	Litres to Gallons.
1=·061024	1= 35·1961	1= 1·75980	1= ·219975
2=·122048	2= 70·3921	2= 3·51961	2= ·439951
3=·183072	3=105·5882	3= 5·27941	3= ·659926
4=·244096	4=140·7842	4= 7·03921	4= ·879902
5=·305120	5=175·9803	5= 8·79902	5=1·099877
6=·366143	6=211·1764	6=10·55882	6=1·319852
7=·427167	7=246·3724	7=12·31862	7=1·539828
8=·488191	8=281·5685	8=14·07842	8=1·759803
9=·549215	9=316·7646	9=15·83823	9=1·979779

The capacity of a litre is approximately that of a cube each side of which is a decimetre. One litre equals 1000·027 cubic centimetres.

Conversion of English to Metrical Capacities.

Cub. Ins. to Cub. Centimetres.	Fluid Ounces to Cubic Centimetres.	Pints to Litres.	Gallons to Litres.
1= 16·3870	1= 28·413	1= ·568245	1= 4·54596
2= 32·7740	2= 56·826	2=1·136491	2= 9·09193
3= 49·1611	3= 85·239	3=1·704736	3=13·63789
4= 65·5481	4=113·652	4=2·272982	4=18·18385
5= 81·9351	5=142·065	5=2·841227	5=22·72982
6= 98·3221	6=170·478	6=3·409472	6=27·27578
7=114·7091	7=198·891	7=3·977718	7=31·82174
8=131·0962	8=227·304	8=4·545963	8=36·36770
9=147·4832	9=255·717	9=5·114209	9=40·91367

Angle Measurement by the Opening of a Two-foot Rule.

Opening Rule.	Fractions of an Inch.															
	0		1/8		1/4		3/8		1/2		5/8		3/4		7/8	
Ins.	°	′	°	′	°	′	°	′	°	′	°	′	°	′	°	′
0		0		36	1	12	1	47	2	23	2	59	3	35	4	11
1	4	47	5	22	5	58	6	34	7	10	7	46	8	22	8	58
2	9	34	10	10	10	46	11	22	11	57	12	34	13	10	13	46
3	14	22	14	58	15	34	16	10	16	46	17	22	17	59	18	35
4	19	11	19	48	20	24	21	0	21	37	22	13	22	50	23	26
5	24	3	24	39	25	16	25	53	26	30	27	7	27	43	28	20
6	28	57	29	34	30	11	30	48	31	26	32	3	32	40	33	18
7	33	55	34	32	35	10	35	48	36	25	37	3	37	41	38	19
8	38	57	39	35	40	13	40	51	41	29	42	7	42	46	43	24
9	44	3	44	42	45	20	45	59	46	38	47	17	47	56	48	36
10	49	15	49	54	50	34	51	14	51	53	52	33	53	13	53	53
11	54	34	55	14	55	54	56	35	57	16	57	57	58	38	59	19
12	60	0	60	41	61	23	62	5	62	47	63	29	64	11	64	53
13	65	35	66	18	67	1	67	44	68	27	69	11	69	54	70	38
14	71	22	72	6	72	51	73	35	74	20	75	5	75	51	76	36
15	77	22	78	8	78	54	79	41	80	27	81	14	82	2	82	49
16	83	37	84	25	85	14	86	3	86	52	87	41	88	31	89	21
17	90	12	91	3	91	54	92	46	93	38	94	31	95	24	96	17
18	97	11	98	5	99	0	99	56	100	51	101	48	102	45	103	43
19	104	41	105	40	106	40	107	40	108	41	109	43	110	45	111	49
20	112	53	113	58	115	5	116	12	117	20	118	30	119	40	120	52

PRESSURES

Conversion of Kilogrammes per Square Centimetre to Pounds per Square Inch.

Kilos. per Sq. Cm.	0	·1	·2	·3	·4	·5	·6	·7	·8	·9
	Pounds per Square Inch.									
0	0·000	1·4223	2·8447	4·2670	5·6893	7·1117	8·5340	9·9563	11·379	12·801
1	14·223	15·646	17·068	18·490	19·913	21·335	22·757	24·180	25·602	27·024
2	28·447	29·869	31·291	32·714	34·136	35·558	36·981	38·403	39·825	41·248
3	42·670	44·092	45·515	46·937	48·359	49·782	51·204	52·626	54·049	55·471
4	56·893	58·316	59·738	61·160	62·583	64·005	65·427	66·850	68·272	69·694
5	71·117	72·539	73·961	75·384	76·806	78·228	79 651	81·073	82·495	83·918
6	85·340	86·762	88·185	89·607	91·029	92·452	93·874	95·296	96·719	98·141
7	99·563	100·99	102·41	103·83	105·25	106 67	108·10	109·52	110·94	112·36
8	113·79	115·21	116·63	118·05	119·48	120·90	122·32	123·74	125·17	126·59
9	128·01	129·43	130·85	132·28	133·70	135·12	136·54	137·97	139·39	140·81
10	142·23	143·66	145·08	146·50	147·92	149·34	150·77	152·19	153·61	155·03

Conversion of Pounds per Square Inch to Kilogrammes per Square Centimetre.

Lbs. per Sq. In.	0	1	2	3	4	5	6	7	8	9
	Kilogrammes per Square Centimetre.									
0	0·0000	0·07031	0·14061	0·21092	0·28123	0·35154	0·42184	0·49215	0·56246	0·63276
10	0·70307	0·77338	0·84369	0·91399	0·98430	1·0546	1·1249	1·1952	1·2655	1·3358
20	1·4061	1·4764	1·5468	1·6171	1·6874	1·7577	1·8280	1·8983	1·9686	2·0389
30	2·1092	2·1795	2·2498	2·3201	2·3904	2·4607	2·5311	2·6014	2·6717	2·7420
40	2·8123	2·8826	2·9529	3·0232	3·0935	3·1638	3·2341	3·3044	3·3747	3·4450
50	3·5154	3·5857	3·6560	3·7263	3·7966	3·8669	3·9372	4·0075	4·0778	4·1481
60	4·2184	4·2887	4·3590	4·4293	4·4997	4·5700	4·6403	4·7106	4·7809	4·8512
70	4·9215	4·9918	5·0621	5·1324	5·2027	5·2730	5·3433	5·4136	5·4840	5·5543
80	5·6246	5·6949	5·7652	5 8355	5·9058	5·9761	6·0464	6·1167	6·1870	6·2573
90	6·3276	6·3979	6·4683	6·5386	6·6089	6·6792	6·7495	6·8198	6·8901	6·9604
100	7·0307	7·1010	7·1713	7·2416	7·3119	7·3822	7·4526	7·5229	7·5932	7·6635

1 kilogramme per sq. centimetre = 0·91 tons per sq. ft.
1 ton per sq. ft. = 1·10 kilogrammes per sq. centimetre.
1 ton per sq. ft. = 15·6 lbs. per sq. in., nearly.

Conversion of Pounds to Tons.

Tons = Pounds × 0·00044643.

Pounds.	Tons.	Pounds.	Tons.	Pounds.	Tons.	Pounds.	Tons.
1,000	0·446	24,000	10·714	47,000	20·982	88,000	39·286
2,000	0·893	25,000	11·161	48,000	21·429	90,000	40·179
3,000	1·339	26,000	11·607	49,000	21·875	92,000	41·072
4,000	1·786	27,000	12·053	50,000	22·321	94,000	41·964
5,000	2·232	28,000	12·500	51,000	22·768	96,000	42·857
6,000	2·679	29,000	12·946	52,000	23·214	98,000	43·750
7,000	3·125	30,000	13·393	54,000	24·107	100,000	44·643
8,000	3·571	31,000	13·839	56,000	25·000	110,000	49·107
9,000	4·018	32,000	14·286	58,000	25·893	120,000	53·572
10,000	4·464	33,000	14·732	60,000	26·786	130,000	58·036
11,000	4·911	34,000	15·179	62,000	27·679	140,000	62·500
12,000	5·357	35,000	15·625	64,000	28·572	150,000	66·965
13,000	5·804	36,000	16·071	66,000	29·464	160,000	71·429
14,000	6·250	37,000	16·518	68,000	30·357	170,000	75·893
15,000	6·696	38,000	16·964	70,000	31·250	180,000	80·357
16,000	7·143	39,000	17·411	72,000	32·143	190,000	84·822
17,000	7·589	40,000	17·857	74,000	33·036	200,000	89·286
18,000	8·036	41,000	18·304	76,000	33·929	210,000	93·750
19,000	8·482	42,000	18·750	78,000	34·822	220,000	98·215
20,000	8·929	43,000	19·196	80,000	35·714	230,000	102·680
21,000	9·375	44,000	19·643	82,000	36·607	240,000	107·140
22,000	9·821	45,000	20·089	84,000	37·500	250,000	111·610
23,000	10·268	46,000	20·536	86,000	38·393	260,000	116·072

Corresponding Heights of Barometer in Millimetres and English Inches.

Milli-metres.	English Inches.	Milli-metres.	English Inches.	Milli-metres.	English Inches.
720	28·347	739	29·095	758	29·843
721	28·386	740	29·134	759	29·882
722	28·425	741	29·174	760	29·922
723	28·465	742	29·213	761	29·961
724	28·504	743	29·252	762	30·000
725	28·543	744	29·292	763	30·039
726	28·583	745	29·331	764	30·079
727	28·622	746	29·370	765	30·118
728	28·662	747	29·410	766	30·158
729	28·701	748	29·449	767	30·197
730	28·740	749	29·488	768	30·236
731	28·780	750	29·528	769	30·276
732	28·819	751	29·567	770	30·315
733	28·858	752	29·606	771	30·355
734	28·898	753	29·645	772	30·394
735	28·937	754	29·685	773	30·433
736	28·976	755	29·724	774	30·473
737	29·016	756	29·764	775	30·512
738	29·055	757	29·803		

COMPARISON OF VARIOUS FOREIGN AND ENGLISH MEASURES.

Country.	Measure.	English Value.	Country.	Measure.	English Value.
America .	Bushel . .	2150·4 cu. ins.	Germany .	Morgen .	27,484 sq. feet
,, .	Gallon (dry)	268·80 cu. ins.	,, .	Pfund . .	1·01 to 1·23 lbs.
,, .	,, (fluid)	231 cu. ins.	India . .	Seer . .	32·914 oz.
,, .	Peck . .	537·605 cu. ins.	,, . .	Tola . .	180 grains
,, .	Pint . .	28·875 cu. ins.	Netherlands	Myle . .	3,281 feet
,, .	Quart . .	57·750 cu. ins.	Norway .	Centner .	110·231 lbs.
Austria .	Joch . .	1·422 acres	,, .	Mile . .	4·6803 miles
,, .	Klafter .	6·2213 feet	,, .	Morgen .	27,484 sq. feet
Bengal .	Maund . .	74·667 lbs.	Russia . .	Archine .	28 inches
Canada .	Mino . .	2,339 cu. ins.	,, . .	Dessatine .	2·69977 acres
China . .	Catty or Gin	1·3333 lbs.	,, . .	Dola . .	·68573 grains
,, . .	Li . .	608·5 yds.	,, . .	Pud or Pood	36·112 lbs.
,, . .	Picul . .	133·33 lbs.	,, . .	Sagene . .	7 feet
,, . .	Tael . .	583 grains	,, . .	Verst . .	3,500 feet
Denmark .	Centner .	110·231 lbs.	Spain .	Arroba .	25·35 lbs.
,, .	Mile . .	4·6803 miles	,, . .	,, .	2·77 gallons
,, .	Morgen .	27,484 sq. feet	Sweden .	Mile . .	6·64 miles
France .	Aune . .	47·245 inches	,, .	Tunnland .	1·2198 acres
,, .	Brasse . .	5·328 feet	Turkey .	Archin . .	26·9 inches
,, .	Lieu . .	6,077 yds.	,, .	Oke . .	2·829 lbs.
Germany .	Centner .	110·231 lbs.	,, .	Pie . .	25·69 inches
,, .	Mile . .	4·6803 miles	Vienna .	Centner .	123·46 lbs.

WIRE GAUGES.

BIRMINGHAM WIRE-GAUGE.

Number.	Dimensions.	Number.	Dimensions.	Number.	Dimensions.	Number.	Dimensions.
	Mils.		Mils.		Mils.		Mils.
0000	454	7	180	17	58	27	16
000	425	8	165	18	49	28	14
00	380	9	148	19	42	29	13
0	340	10	134	20	35	30	12
1	300	11	120	21	32	31	10
2	284	12	109	22	28	32	9
3	259	13	95	23	25	33	8
4	238	14	83	24	22	34	7
5	220	15	72	25	20	35	5
6	203	16	65	26	18	36	4

BIRMINGHAM WIRE-GAUGE FOR SILVER AND GOLD.

No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.
1	·004	6	·013	11	·029	16	·051	21	·072	25	·095	29	·120	33	·145
2	·005	7	·015	12	·034	17	·057	22	·074	26	·103	30	·126	34	·148
3	·008	8	·016	13	·036	18	·061	23	·077	27	·113	31	·133	35	·158
4	·010	9	·019	14	·041	19	·064	24	·082	28	·120	32	·143	36	·167
5	·013	10	·024	15	·047	20	·067								

SHEET ZINC TRADE GAUGE.

No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.
1	·00395	5	·0097	9	·0177	12	·0260	15	·0375	18	·0526	21	·0699	24	·0915
2	·00554	6	·0114	10	·0196	13	·0292	16	·0426	19	·0577	22	·0768	25	·0980
3	·0067	7	·0132	11	·0228	14	·0323	17	·0478	20	·0632	23	·0843	26	·1052
4	·0082	8	·0149												

(See also page 138.)

Standard Wire-Gauge.*

No.	Diameters. Mils.†	Differences.	Millimètres.	No.	Diameters. Mils.	Differences.	Millimètres.	No.	Diameters. Mils.	Differences.	Millimètres.
0,000,000	500	..	12·70	13	92	12	2·34	32	10·8	·8	·274
000,000	464	36	11·78	14	80	12	2·03	33	10·0	·8	·254
00,000	432	32	10·97	15	72	8	1·83	34	9·2	·8	·234
0,000	400	32	10·16	16	64	8	1·63	35	8·4	·8	·213
000	372	28	9·45	17	56	8	1·42	36	7·6	·8	·193
00	348	24	8·84	18	48	8	1·22	37	6·8	·8	·173
0	324	24	8·23	19	40	8	1·016	38	6·0	·8	·152
1	300	24	7·62	20	36	4	·914	39	5·2	·8	·132
2	276	24	7·01	21	32	4	·813	40	4·8	·4	·122
3	252	24	6·40	22	28	4	·711	41	4·4	·4	·112
4	232	20	5·89	23	24	4	·610	42	4·0	·4	·102
5	212	20	5·38	24	22	2	·559	43	3·6	·4	·0914
6	192	20	4·88	25	20	2	·508	44	3·2	·4	·0813
7	176	16	4·47	26	18	2	·457	45	2·8	·4	·0711
8	160	16	4·06	27	16·4	1·6	·417	46	2·4	·4	·0610
9	144	16	3·66	28	14·8	1·6	·376	47	2·0	·4	·0508
10	128	16	3·25	29	13·6	1·2	·345	48	1·6	·4	·0406
11	116	12	2·95	30	12·4	1·2	·315	49	1·2	·4	·0305
12	104	12	2·64	31	11·6	·8	·295	50	1·0	·2	·0254

* This gauge is the only legal standard wire-gauge for the United Kingdom.
† 1 Mil. = $\frac{1}{1000}$th of an inch.

Warrington Wire-Gauge.

Mark. No.	Size. Inch.	Mark. No.	Size. Inch.	Mark. No.	Size. Inch.	Mark. No.	Size. Inch.
7/0	$\frac{1}{2}$	2	·274	10	·133	16	·0625, or $\frac{1}{16}$
6/0	$\frac{15}{32}$	3	·25, or $\frac{1}{4}$	10½	·125, or $\frac{1}{8}$	17	·053
5/0	$\frac{7}{16}$	4	·229	11	·117	18	·047
4/0	$\frac{13}{32}$	5	·209	12	·10, or $\frac{1}{10}$	19	·041
3/0	$\frac{3}{8}$	6	·191	13	·090	20	·036
2/0	$\frac{11}{32}$	7	·174	14	·079	21	·0315, or $\frac{1}{32}$
0	·326	8	·159	15	·069	22	·028
1	·300	9	·146				

Whitworth's Wire-Gauge, in Decimals of an Inch.

Mark.	Size.	Mark.	Size.	Mark.	Size.	Mark.	Size.	Mark.	Size.
1	·001	14	·014	34	·034	85	·085	240	·240
2	·002	15	·015	36	·036	90	·090	260	·260
3	·003	16	·016	38	·038	95	·095	280	·280
4	·004	17	·017	40	·048	100	·100	300	·300
5	·005	18	·018	45	·045	110	·110	325	·325
6	·006	19	·019	50	·050	120	·120	350	·350
7	·007	20	·020	55	·055	135	·135	375	·375
8	·008	22	·022	60	·060	150	·150	400	·400
9	·009	24	·024	65	·065	165	·165	425	·425
10	·910	26	·026	70	·070	180	·180	450	·450
11	·011	28	·028	75	·075	200	·200	475	·475
12	·012	30	·030	80	·080	220	·220	500	·500
13	·013	32	·032						

BIRMINGHAM GAUGE ('B.G.') FOR SHEETS AND HOOPS.

(Legalised by the Board of Trade Nov. 1, 1914.)

Descriptive Number.	Equivalents in parts of an Inch.	Descriptive Number.	Equivalents in parts of an Inch.
No.	Inch.	No.	Inch.
15/0 B.G.	1·000	20 B.G.	·0392
14/0 B.G.	·9583	21 B.G.	·0349
13/0 B.G.	·9167	22 B.G.	·03125
12/0 B.G.	·8750	23 B.G.	·02782
11/0 B.G.	·8333	24 B.G.	·02476
10/0 B.G.	·7917	25 B.G.	·02204
9/0 B.G.	·750	26 B.G.	·01961
8/0 B.G.	·7083	27 B.G.	·01745
7/0 B.G.	·6666	28 B.G.	·015625
6/0 B.G.	·625	29 B.G.	·0139
5/0 B.G.	·5883	30 B.G.	·0123
4/0 B.G.	·5416	31 B.G.	·0110
3/0 B.G.	·500	32 B.G.	·0098
2/0 B.G.	·4452	33 B.G.	·0087
1/0 B.G.	·3964	34 B.G.	·0077
1 B.G.	·3532	35 B.G.	·0069
2 B.G.	·3147	36 B.G.	·0061
3 B.G.	·2804	37 B.G.	·0054
4 B.G.	·250	38 B.G.	·0048
5 B.G.	·2225	39 B.G.	·0043
6 B.G.	·1981	40 B.G.	·00386
7 B.G.	·1764	41 B.G.	·00343
8 B.G.	·1570	42 B.G.	·00306
9 B.G.	·1398	43 B.G.	·00272
10 B.G.	·1250	44 B.G.	·00242
11 B.G	·1113	45 B.G.	·00215
12 B.G.	·0991	46 B.G.	·00192
13 B.G.	·0882	47 B.G.	·00170
14 B.G.	·0785	48 B.G.	·00152
15 B.G.	·0699	49 B.G.	·00135
16 B.G.	·0625	50 B.G.	·00120
17 B.G.	·0556	51 B.G.	·00107
18 B.G.	·0495	52 B.G.	·00095
19 B.G.	·0440		

Stubs' Steel Wire Gauge (Dimensions in Mils.).

No.	Mils.	No.	Mils.	No.	Mils.	No.	Mils.	No.	Mils.	No.	Mils.	No.	Mils.	No.	Mils.	No.	Mils.	No.	Mils.	No.	Mils.	No.	Mils.
Z	413	Q	332	H	266	1	227	9	194	17	172	25	148	33	112	41	95	49	72	57	42	65	33	73	23
Y	404	P	323	G	261	2	219	10	191	18	168	26	146	34	110	42	92	50	69	58	41	66	32	74	22
X	397	O	316	F	257	3	212	11	188	19	164	27	143	35	108	43	88	51	66	59	40	67	31	75	20
W	386	N	302	E	250	4	207	12	185	20	161	28	139	36	106	44	85	52	63	60	39	68	30	76	18
V	377	M	295	D	246	5	204	13	182	21	157	29	134	37	103	45	81	53	58	61	38	69	29	77	16
U	368	L	290	C	242	6	201	14	180	22	155	30	127	38	101	46	79	54	55	62	37	70	27	78	15
T	358	K	281	B	238	7	199	15	178	23	153	31	120	39	99	47	77	55	50	63	36	71	26	79	14
S	348	J	277	A	234	8	197	16	175	24	151	32	115	40	97	48	75	56	45	64	35	72	24	80	13
R	339	I	272																						

Table showing Relative Values of Standard, Birmingham, and American (Brown and Sharpe) Wire Gauges.

S.W.G.	B.W.G.	A.W.G.	Equivalent in Mils.	Equivalent in Mms.	S.W.G.	B.W.G.	A.W.G.	Equivalent in Mils.	Equivalent in Mms.	S.W.G.	B.W.G.	A.W.G.	Equivalent in Mils.	Equivalent in Mms.	S.W.G.	B.W.G.	A.W.G.	Equivalent in Mils.	Equivalent in Mms.
7/0			500	12·699	5			212	5·384	15	15	13	072	1·828	28	28	27	014	3555
6/0			464	11·785			4	204	5·181		16		065	1·650	29	29		013	3300
		0000	460	11·683		6		203	5·156	16		14	064	1·625			28	0122	3100
	0000		454	11·531	6			192	4·876		17		058	1·472	30	30		012	3046
5/0			432	10·972			5	182	4·622			15	057	1·447	31		29	011	2800
	000		425	10·794		7		180	4·571	17			056	1·421	32			0108	2743
		000	409	10·388	7			176	4·470			16	050	1·270	33	31	30	010	2539
0000			400	10·159		8		165	4·191		18		049	1·244	34	32	31	009	2300
	00		380	9·651			6	162	4·114	18			048	1·218	35	33	32	008	2031
000			372	9·448	8			160	4·064			17	045	1·142	36	34	33	007	1777
		00	365	9·271		9		148	3·759		19		042	1·066	37			0068	1727
00			348	8·839	9		7	144	3·657	19		18	040	1·016	38		34	006	1523
	0		340	8·635		10		134	3·403	20		19	036	9140			35	0056	1422
		0	325	8·254	10		8	128	3·251		20		035	8886	39	35	36	005	1269
0			324	8·229		11		120	3·047	21	21	20	032	8124	40			0048	1219
1	1		300	7·620	11			116	2·946		22		030	7617	41		37	0044	1118
		1	289	7·340			9	114	2·895			21	0284	7213	42	36	38	004	1015
	2		284	7·213		12		109	2·768	22			028	7109	43		39	0036	0914
2			276	7·010	12			104	2·641			22	0253	6126	44		40	0032	0813
	3		259	6·578			10	102	2·590		23		025	6347	45			0028	0713
		2	257	6·527		13		095	2·412	23			024	6093	46			0024	0610
3			252	6·400	13			092	2·336	24	24	23	022	5585	47			002	0507
	4		238	6·045			11	090	2·286	25	25	24	020	5078	48			0016	0406
4			232	5·892		14		083	2·108	26	26	25	018	4570	49			0012	0305
		3	229	5·816	14		12	080	2·032	27	27	26	016	4062	50			001	0253
	5		220	5·588															

WIRE GAUGES USED IN THE UNITED STATES.

Dimensions of Sizes in Decimal Parts of an Inch.

Number of Wire Gauge.	American or Brown and Sharpe.	Birmingham or Stubs' Wire.	Washburn and Moen Mfg. Co., Worcester, Mass.	Imperial Wire Gauge.	Stubs' Steel Wire.	U.S. Standard for Plate.	Number of Wire Gauge.
000000	—	—	—	·464	—	·46875	000000
00000	—	—	—	·432	—	·4375	00000
0000	·46	·454	·3938	·400	—	·40625	0000
000	·40964	·425	·3625	·372	—	·375	000
00	·3648	·38	·3310	·348	—	·34375	00
0	·32486	·34	·3065	·324	—	·3125	0
1	·2893	·3	·2830	·300	·227	·28125	1
2	·25763	·284	·2625	·276	·219	·265625	2
3	·22942	·259	·2437	·252	·212	·25	3
4	·20431	·238	·2253	·232	·207	·234375	4
5	·18194	·22	·2070	·212	·204	·21875	5
6	·16202	·203	·1920	·192	·201	·203125	6
7	·14428	·18	·1770	·176	·199	·1875	7
8	·12849	·165	·1620	·160	·197	·171875	8
9	·11443	·148	·1483	·144	·194	·15625	9
10	·10189	·134	·1350	·128	·191	·140625	10
11	·090742	·12	·1205	·116	·188	·125	11
12	·080808	·109	·1055	·104	·185	·109375	12
13	·071961	·095	·0915	·092	·182	·09375	13
14	·064084	·083	·0800	·080	·180	·078125	14
15	·057068	·072	·0720	·072	·178	·0703125	15
16	·05082	·065	·0625	·064	·175	·0625	16
17	·045257	·058	·0540	·056	·172	·05625	17
18	·040303	·049	·0475	·048	·168	·05	18
19	·03589	·042	·0410	·040	·164	·04375	19
20	·031961	·035	·0348	·036	·161	·0375	20
21	·028462	·032	·03175	·032	·157	·034375	21
22	·025347	·028	·0286	·028	·155	·03125	22
23	·022571	·025	·0258	·024	·153	·028125	23
24	·0201	·022	·0230	·022	·151	·025	24
25	·0179	·02	·0204	·020	·148	·021875	25
26	·01594	·018	·0181	·018	·146	·01875	26
27	·014195	·016	·0173	·0164	·143	·0171875	27
28	·012641	·014	·0162	·0149	·139	·015625	28
29	·011257	·013	·0150	·0136	·134	·0140625	29
30	·010025	·012	·0140	·0124	·127	·0125	30
31	·008928	·01	·0132	·0116	·120	·0109375	31
32	·00795	·009	·0128	·0108	·115	·01015625	32
33	·00708	·008	·0118	·0100	·112	·009375	33
34	·006304	·007	·0104	·0092	·110	·00859375	34
35	·005614	·005	·0095	·0084	·108	·0078125	35
36	·005	·004	·0090	·0076	·106	·00703125	36
37	·004453	—	—	·0068	·103	·006640625	37
38	·003965	—	—	·0060	·101	·00625	38
39	·003531	—	—	·0052	·099	—	39
40	·003144	—	—	·0048	·097	—	40

Care should be taken to distinguish the difference between Stubs' Iron Wire Gauge and Stubs' Steel Wire Gauge. The former is commonly known as the English Standard Wire or Birmingham Wire Gauge, and designates the Stubs' *Soft* wire sizes.

The Stubs' Steel Wire Gauge is the one employed in measuring drawn steel wire or drill rod, of Stubs' make, and is also used by some makers of American drill rods.

(*Brown and Sharpe.*)

VARIOUS FOREIGN WEIGHTS AND MEASURES WITH THEIR APPROXIMATE BRITISH EQUIVALENTS.

(Compiled by the Board of Trade (July 1923) from information published by the respective Governments.)

[NOTE.—In the case of fractional equivalents the last figure shown is uncertain.]

Country.	Length.	Area.	Weight.	Capacity.	Approximate British Equivalent.
1. United States of America	—	—	Barrel (flour)	—	196 lbs. avoir.
	—	—	Short Ton	—	2,000 lbs. avoir.
	—	—	Long „	—	2,240 „ „
	—	—	—	Bushel (Winchester)—dry	0·96897 imperial bushel.
	—	—	—	Gallon (Old English)—wet	0·83268 „ gallon.
	Metre	—	—	—	1·09361 yards.
	Kilometre	—	—	—	0·62137 mile.
	—	Sq. Kilometre = 100 Hectares	—	—	0·38610 square mile.
	—	Are	—	—	0·024711 acre.
	—	Hectare	—	—	2·4711 acres.
2. Austria	—	—	Kilogramme	—	2·20462 lbs. avoir.
	—	—	Quintal métrique (Centner) = 100 kilogs	—	220·462 lbs. avoir.
	—	—	Tonne	—	2204·62 lbs. avoir.
	—	—	—	Cubic metre	1·30795 cubic yards.
	—	—	—	Hectolitre (liquid)	21·9975 imperial gallons.
	—	—	—	„ (cereals)	2·74969 „ bushels.
3. Belgium	—	As No. 2	—	—	—
4. Bulgaria	—	„ „	—	—	—
5. China	Ch'ih (= 10 Ts'un)	—	—	—	*14·1 inches.
	Chang (= 10 Ch'ih)	—	—	—	*11·75 feet.
	—	—	Tael (Liang)	—	*1⅓ ozs. avoir.
	—	—	Catty	—	*1⅓ lbs. avoir.
	—	—	Picul	—	*133⅓ lbs. avoir.
6. Czechoslovakia	—	As No. 2	—	—	—
7. Denmark	Weights and measures as No. 2 have been compulsory since April 1922.				
	Pic for textiles (0·58 metres) (Diraa Baladi)	—	—	—	22·83466 inches.
	—	Feddan (land)	—	—	1·038050 acres.
8. Egypt	—	—	Oke (400 Dirhems) = 1·248 kilogs.	—	2·751367 lbs. avoir.
	—	—	Kantar (36 Oke) = 44·928 kilogs.	—	99·049272 „
	—	—	—	Ardeb (198 litres)—dry	5·144391 bushels.
9. Esthonia	—	As No. 2	Russian weights and measures as No. 24 also in use.		
10. Finland	—	„ „	—	—	—
11. France	—	„ „	—	—	—
12. Germany	—	„ „	—	—	—
13. Greece	Metric system (as No. 2) compulsory since 1922	—	Ocque = 1·280 kilogs.	—	2·82 lbs. avoir.
			Slater (Quintal = 56·320 kilogs.	—	124·16 „

13. Greece—*cont.*	—	—	Livre (Great Venetian)	—	1·06 lb. avoir.
	—	—	Drachme (Dram)	—	$\frac{1}{9}$ oz. avoir.
14. Holland	—	As No. 2	—	—	—
15. Hungary	—	„ „	—	—	—
16. Italy	—	„ „	—	—	—
	Ri	—	—	—	2·4403 miles.
	Cho	—	—	—	5·423 chains.
	Ken	—	—	—	1·9834 yards.
	—	Square Ri	—	—	5·95505 square miles.
	—	Cho (land)	—	—	2·4507 acres.
	—	Tsubo	—	—	3·9538 square yards.
17. Japan	—	—	Kwan	—	8·26733 lbs. avoir.
	—	—	Kin	—	1·32277 „
	—	—	—	Koku (wet)	39·7033 gallons.
	—	—	—	„ (dry)	4·9629 bushels.
	—	—	—	„ (capacity of vessel)	$\frac{1}{10}$ of a measurement ton.
	—	—	—	Sho (wet)	1·5881 quarts.
	—	—	—	„ (dry)	0·1985 peck.
18. Latvia	As No. 2.				
19. Lithuania	Russian system as No. 24, but at same time metric system is in use. Arrangements are being made for making latter compulsory.				
20. Norway	—	As No. 2	—	—	—
	Lokieć	—	—	—	0·62992 yard.
	—	Square lokieć (lokieć kwadretowe)	—	—	0·39680 square yard.
	—	Morg	—	—	1·3835 acres.
21. Poland	—	—	Funt	—	0·893983 lb. avoir.
	—	—	Center = 100 funt	—	89·3983 lbs.
	—	—	—	Cubic lokieć (lokieć szescienne)	0·24995 cubic yard.
	—	—	—	Kwarta (wet and dry) = 1 litre	0·21998 gallon.
22. Portugal	—	As No. 2	—	—	—
23. Roumania	—	„ „	—	—	—
	Verst	—	—	—	0·6629 mile.
	—	Square verst	—	—	0·4394 square mile.
	—	Dessiatine	—	—	2·6997 acres.
24. Russia	—	—	Pood	—	36·113 lbs. avoir.
	—	—	Berkovets (=10 pood)	—	361·13 „
	—	—	—	Tchetvert (dry)	5·7717 imperial bushels.
	—	—	—	Vedro (wet)	2·705 „ gallons.
25. Serb-Croat-Slovene State	As No. 2	—	—	—	—
26. Spain	„ „	—	—	—	—
27. Sweden	„ „	—	—	—	—
28. Switzerland	„ „	—	—	—	—

* As fixed for foreign trade by treaty between Great Britain and China in 1902. Customary weights and measures of the same name vary in every province and trade in China.

Various Foreign Standard Monetary Units.

Country.	Unit of Currency.	Country.	Unit of Currency.
1. United States of America	Dollar	15. Hungary	Pengő
2. Austria	Schillinge	16. Italy	Lire
3. Belgium	Belgas	17. Japan	Yen
4. Bulgaria	Lev	18. Latvia	Lat
5. China	Haikwan Tael	19. Lithuania	Litas
6. Czechoslovakia	Koruna C.S. Kronen	20. Norway	Krone
7. Denmark	Kroner	21. Poland	Zlotych
8. Egypt	£ Egyptian	22. Portugal	Escudo
9. Esthonia	Kroon	23. Roumania	Lei
10. Finland	Markaa	24. Russia	Rouble
11. France	Franc	25. Serb-Croat-Slovene State	Dinar
12. Germany	Reichsmark	26. Spain	Pesetas
13. Greece	Drachma	27. Sweden	Kronor
14. Holland	Florin	28. Switzerland	Franc

Foreign Measures into English Equivalents.

Country.	Foreign Measure.	British Measure.
China	1 Ts'un	1·41 in.
	1 Ch'ih	14·1 in.
	1 Chang	11·75 ft.
	1 Li	586 yds. 2 ft.
Egypt	1 Feddan	1·08808 acres
Greece	1 Pike	27 in.
Japan	1 Ken	5·965 ft.
	1 Cho	117 yds. 1 ft.
	1 Ri	2·44 miles.
Russia	1 Vershok	1·75 in.
	1 Arshin	28 in.
	1 Sajene	7 ft.
	1 Verst	0·663 mile.
	1 Sq. Verst	0·4394 sq. mile
	1 Dessiatine	2·70 acres.
Turkey	1 Donum	1096·76 sq. yds.

Country.	Foreign Weight.	British Weight.
China	1 Liang or Tael	1·3 oz.
	1 Chin or Catty	1·3 lb.
	1 Picul	133 lb.
Egypt	1 Okis	1·3207 oz.
	1 Roti	·99049 lb.
	1 Oke	2·7513 lb.
	1 Quantar	99·049 lb.
Greece	1 Livre	1·05 lb.
	1 Oke	2·80 lb.
	1 Stator	123·20 lb.
Russia	1 Pound	·9 lb.
	1 Pood	26 lb.
Turkey	1 Oke	2·83 lb.
	1 Cantar	1·25 lb.

PRINTING

ENGINEERING FIRMS requiring high-class brochures, catalogues, calendars, and other forms of creative print to illustrate their products and achievements are invited to communicate with us.

WELBECSON PRESS
LIMITED

137 BLENHEIM CRESCENT
LONDON W.11

WELBECSON

A first-rate printing service for Engineering Firms

WELBECSON PRESS
Limited
137 Blenheim Crescent
London W.11

Telephone: PARK 9293
(Private Branch Exchange)

A Welbecson Imprint
is synonymous
with merit

SIAMESE WEIGHTS AND MEASURES.

$$1 \text{ rai} = \frac{4}{25} = 0{\cdot}16 \text{ hectare} = \frac{79}{200} = 0{\cdot}395 \text{ acre}$$

1 hectare = 6·25 rai; 1 acre = 2·43 rai

$$1 \text{ picul} = \frac{10}{168} \text{ ton} = \frac{100}{84} \text{ cwt.} = 133\tfrac{1}{3} \text{ lbs.}$$

1 ton = 16·8 piculs; 1 cwt. = 0·84 piculs

1 picul per rai = 0·1506 ton per acre = 3·012 cwt. per acre

Sizes of Paper and Books.

SIZES OF PAPER USED FOR PRINTING BOOKS, ETC.

	Ins. Ins.		Ins. Ins.		Ins. Ins.
Foolscap . .	$13\frac{1}{2} \times 17$	Medium . .	18×23	Super royal .	20×27
Crown . . .	15×20	Royal . . .	$20 \times 25\frac{1}{2}$	Imperial . . .	$22\frac{1}{2} \times 31$
Demy . . .	$17\frac{1}{2} \times 22\frac{1}{2}$				

Most of these papers are also made double size.

A ream of paper contains 20 quires of 24 sheets each. A bale is 10 reams. A roll of parchment is 60 skins.

SIZES OF BOOKS.

Broadside, folio, quarto (4to), octavo (8vo), 12mo, 16mo, 18mo, 32mo, 48mo.
These names indicate the number of pages in the half-sheet; for instance, an octavo has 8 pages in the half-sheet. A crown octavo is about 5 ins. × 8 ins.

Sizes and Descriptions of Type.

The following descriptions of type have been used in the printing of this book :—

LONG PRIMER SMALL CAPITALS.

BOURGEOIS CAPITALS.

Bourgeois Roman.

Bourgeois Italic.

MINION CAPITALS.

MINION SMALL CAPITALS.

NONPAREIL ANTIQUE CAPS.

Nonpareil Antique.

NONPAREIL CAPITALS.

NONPAREIL SMALL CAPITALS.

Nonpareil Roman.

Nonpareil Italic.

PEARL ANTIQUE CAPITALS.

Pearl Roman.

Pearl Italic.

Diamond Italic.

SECTION II

POWERS

STATICS—CENTRE OF GRAVITY—LEVERS—PULLEYS—PARALLELOGRAM OF FORCES—INCLINED PLANE—DYNAMICS—MOVING BODIES—ACCELERATION—WORK—PENDULUM—OSCILLATION AND PERCUSSION—GYRATION—INERTIA—BALLISTIC PENDULUM—POWER BRAKES—ANIMAL POWER (pp. 23–48)

SECTION II

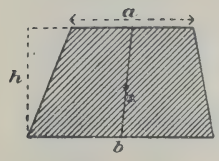
POWERS

STATICS — CENTRE OF GRAVITY — LEVERS — PULLEYS — PARALLELOGRAM OF FORCES — INCLINED PLANE — DYNAMICS — MOVING BODIES — ACCELERATION — WORK — PENDULUM — OSCILLATION AND PERCUSSION — GYRATION — INERTIA — BALLISTIC PENDULUM — POWER BRAKES — ANIMAL POWER.

STATICS.

CENTRE OF GRAVITY.

Plane Figures.



Quadrangle.
a and b parallel.

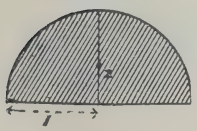
$$z = \frac{h}{2} - \frac{h}{6}\left(\frac{b-a}{b+a}\right).$$

Fig. 1.

Triangle.

$$z = \frac{h}{3}.$$

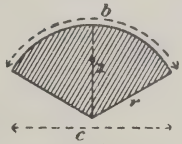
Fig. 2.



Half a Circle Plane or Elliptic Plane.

$$z = 0{\cdot}424r.$$

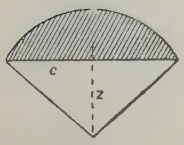
Fig. 3.



Circle Sector.

$$z = \frac{2cr}{3b}.$$

Fig. 4.

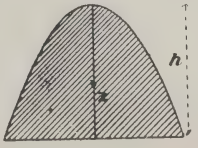


Circle Segment.

$$z = \frac{c^3}{12a}.$$

a = area.

Fig. 5



Parabola.

$$z = \frac{2h}{5}.$$

Fig. 6.

Shell or Convex Surface, and Solid Figures.

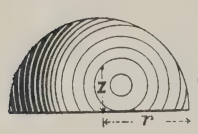


Fig. 7.

Half Sphere.

Convex surface, $z = \frac{1}{2}r$.
Solid, $z = \frac{3}{8}r$.

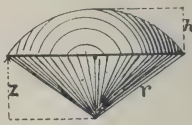


Fig. 8.

Spherical Sector.

Solid, $z = \frac{1}{4}\left(r - \frac{h}{2}\right)$.

Fig. 9.

Spherical Segment.

Convex surface, $z = \frac{h}{2}$.

Solid, $z = \frac{h}{2} \cdot \left[\frac{2r^2 + h^2}{3r^2 + h^2}\right]$.

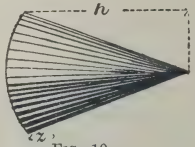


Fig. 10.

Cone.

Shell, $z = \frac{h}{3}$.

Solid, $z = \frac{h}{4}$.

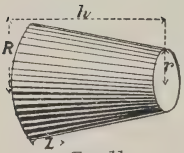


Fig. 11.

Conic Frustum.

Convex surface, $z = \frac{h}{2} - \frac{h}{6}\left[\frac{R - r}{R + r}\right]$. Solid, $z = \frac{h}{4} \cdot \left[\frac{R^2 + r(2R + 3r)}{R^2 + r(R + r)}\right]$.

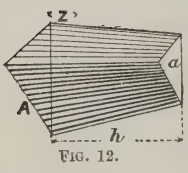


Fig. 12.

Pyramidal Frustum.

Solid, $z = \frac{h}{4}\left[\frac{A + 3a + 2\sqrt{Aa}}{A + a + \sqrt{Aa}}\right]$.

A and a = areas of the two bases.

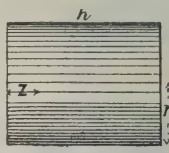


Fig. 13.

Cylindric Surface with a Bottom in one End.

$z = \frac{h}{r + 2h}$.

Circumferential Figures.

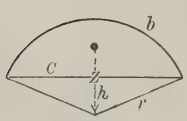


Fig. 14.

Circle Arc or Elliptic Arc.

$z = \frac{cr}{b} = \frac{c\sqrt{4h^2 + c^2}}{4b}$.

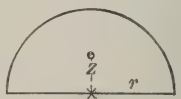


Fig. 15.

Half a circumference of a Circle or Ellipse.

$z = 0{\cdot}6367r$.

LEVERS.

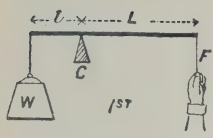


FIG. 16.

$F : W = l : L,$

$FL = Wl.$

$F = \frac{Wl}{L},$

$W = \frac{FL}{l}.$

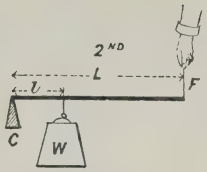


FIG. 17.

$F : W = l : L,$

$FL = Wl.$

$F = \frac{Wl}{L},$

$W = \frac{FL}{l}.$

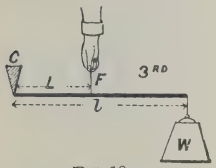


FIG. 18.

$F : W = l : L,$

$FL = Wl.$

$F = \frac{Wl}{L},$

$W = \frac{FL}{l}.$

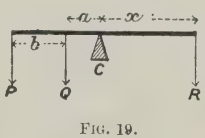


FIG. 19.

To find the fulcrum, C, when three forces act on one lever.

$Rx = Qa + P(b + a),$

$x = \frac{Qa + P(b+a)}{R}$

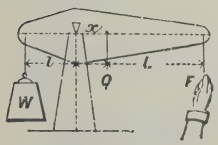


FIG. 20.

Q = weight of the lever.
x = distance from the centre of gravity of the lever to the fulcrum.

$F = \frac{Wl - Qx}{L},$

$W = \frac{FL + Qx}{l}.$

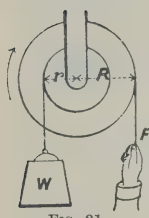


FIG. 21.

$F : W = r : R,$

$FR = Wr.$

$F = \frac{Wr}{R},\ R = \frac{Wr}{F},$

$W = \frac{RF}{r},\ r = \frac{RF}{W}.$

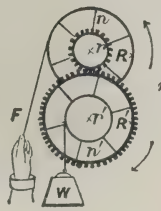


FIG. 22.

$F = \frac{Wrr'}{RR'},\qquad W = \frac{FRR'}{rr'}.$

n, n' = numbers of revolutions of the wheels.

$n : n' = R' : r,\quad v : v' = RR' : rr'.$

v = velocity of F,

v' = velocity of W.

PULLEYS.

$\frac{v'}{v}$ = ratio of distances moved by W and F.

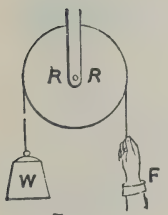


FIG. 23.

A single fixed Pulley.

$F : W = R : R,$

or $F = W.$

$\frac{v'}{v} = 1.$

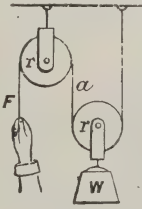


FIG. 24.

A single movable Pulley.

$F : W = r : 2r,$

or $F = \frac{1}{2}W.$

If the force F be applied at *a* and act upwards, the result will be the same.

$\frac{v'}{v} = 2.$

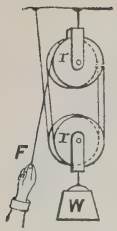


FIG. 25.

A double movable Pulley.

$F : W = r : 4r,$

or $F = \frac{1}{4}W.$

$\frac{v'}{v} = 4.$

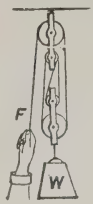


FIG. 26.

A double movable Pulley.

$F = \frac{1}{4}W.$

$\frac{v'}{v} = 4.$

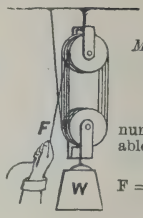


FIG. 27.

Multiple movable Pulley.

Let n = any number of movable pulleys, then

$F = \frac{W}{2n}, \quad \frac{v'}{v} = 2n.$

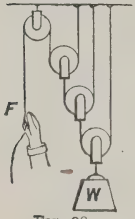


FIG. 28.

Compound Pulleys.

n = number of movable pulleys.

$F = \frac{W}{2^n}, \quad W = 2^n F,$

$\frac{v'}{v} = 2^n.$

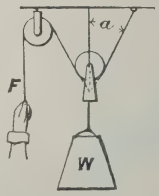


FIG. 29.

An oblique fixed Pulley.

$F : W = \sec \alpha : 2.$

$W = \frac{2F}{\sec \alpha}, \quad F = \frac{W \sec \alpha}{2}.$

PARALLELOGRAM OF FORCES.

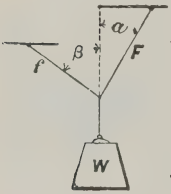


FIG. 30.

$W : F = \sin(\alpha+\beta) : \sin \beta,$

$W : f = \sin(\alpha+\beta) : \sin \alpha.$

$F = \frac{W \sin \beta}{\sin(\alpha+\beta)},$

$f = \frac{W \sin \alpha}{\sin(\alpha+\beta)},$

$W^2 = F^2 + f^2 + 2Ff \cos(\alpha+\beta).$

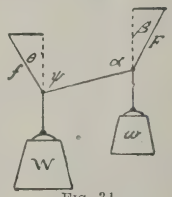


FIG. 31.

$F = \frac{w \sin \alpha}{\sin(\alpha+\beta)},$

$f = \frac{W \sin \psi}{\sin(\psi+\theta)}.$

INCLINED PLANE.

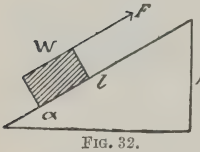


FIG. 32.

No Friction.

$F = \frac{Wh}{l} = W \sin \alpha,$

$W = \frac{Fl}{h} = \frac{F}{\sin \alpha}.$

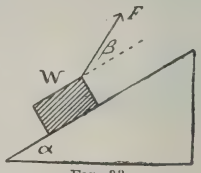


FIG. 33.

No Friction.

$F = W \frac{\sin \alpha}{\cos \beta},$

$W = F \frac{\cos \beta}{\sin \alpha}.$

Friction.

$F = W \frac{\mu \cos \alpha + \sin \alpha}{\mu \sin \beta + \cos \beta},$

where μ = coefficient of friction.

$F = W \frac{\sin(\phi + \alpha)}{\cos(\beta - \phi)},$

where ϕ = limiting angle of resistance.

$\mu = \tan \phi.$

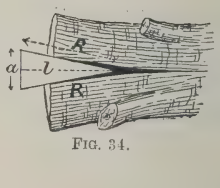


FIG. 34.

Wedge.

$F = \frac{Ra}{l},$

$R = \frac{Fl}{a}.$

F = force required to drive the wedge.

Force Acting at an Angle

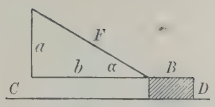


Fig. 35.

Let the line F represent the magnitude and direction of a force acting at an angle a to move the body B on the line CD; then the line a represents a part of F which presses the body B against CD, and the line b represents the magnitude of the force which actually moves the body B.

$$b = \sqrt{F^2 - a^2}, \quad b = F \cos a.$$

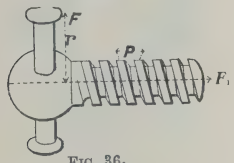


Fig. 36.

Force by a Screw.

P = pitch of the screw.
r = radius on which the force F acts.
π = 3·1416.

$$F_1 : W :: P : 2\pi r.$$

$$F_1 = \frac{WP}{2\pi r}. \qquad W = \frac{F_1 2\pi r}{P}.$$

DYNAMICS.

Uniform Motion.

s = space passed through in time t. v = velocity.

$$s = vt.$$

If s is in ft. and t in secs., then v = ft. per sec.

FALLING BODIES.

Body Falling Freely.

s = space passed through in time t. v = velocity at end of time t.
g = coefficient of terrestrial acceleration = 32·2 (where s is in ft. and t in secs.).

$$s = \tfrac{1}{2}gt^2. \qquad v = gt. \qquad v^2 = 2gs.$$

Body Thrown Downward.

If body is *thrown downward* with a velocity V,

$$v = V + gt. \qquad s = Vt + \tfrac{1}{2}gt^2. \qquad v^2 = V^2 + 2gs.$$

Body Thrown Upward.

If body is *thrown upward* with a velocity V,

$$v = V - gt. \qquad s = Vt - \tfrac{1}{2}gt^2. \qquad v^2 = V^2 - 2gs.$$

IMPACT OF BODIES.

The Bodies inelastic.

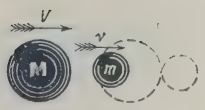


Fig. 37.

The Bodies move in the same direction.

$$v'(M+m) = MV + mv,$$

$$v' = \frac{MV + mv}{M+m}.$$

Fig. 38.

The Bodies move in opposite directions.

$$v'(M+m) = MV - mv,$$

$$v' = \frac{MV - mv}{M+m}.$$

Fig. 39.

Only one Body in motion.

$$v'(M+m) = MV,$$

$$v' = \frac{MV}{M+m}.$$

where M and m are either mass or weight.

IMPACT OF BODIES.

The Bodies elastic.

{ K = coefficient of elasticity of M
  k = " " " m }

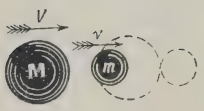


FIG. 40.

The Bodies move in the same direction.

$$V' = \frac{V(M - Km) + vm(1 + K)}{M + m},$$

$$v' = \frac{MV(1 + k) + (m - kM)}{M + m}.$$

FIG. 41.

The Bodies move in opposite directions.

$$V' = \frac{V(M - Km) - vm(1 + K)}{M + m},$$

$$v' = \frac{MV(1 + k) - v(m - kM)}{M + m}.$$

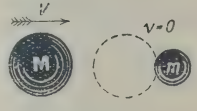


FIG. 42.

Only one Body in motion.

$$V' = \frac{V(M - Km)}{M + m},$$

$$v' = \frac{VM(1 + k)}{M + m}.$$

where M and m are either mass or weight.

ACCELERATION.

ACCELERATION PRODUCED BY A GIVEN PRESSURE.

If body (W lbs.) be on a perfectly smooth horizontal plane, and be acted upon by a pull of P lbs., then acceleration, f, is

$$f = \frac{P}{W} g ;$$

Where g = coefficient of terrestrial acceleration = 32·2.
If coefficient of friction on plane = μ,

$$f = \frac{P - \mu W}{W} g.$$

INCLINED PLANE.

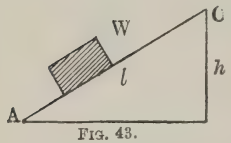


FIG. 43.

WORK ON AN INCLINED PLANE.

W = weight of body in pounds. μ = coefficient of friction.

Work in moving body from A to C = W ($\mu l + h$).
" " " C to A = W ($\mu l - h$).

TABLE OF COEFFICIENTS OF FRICTION.

Sliding Surface.	Coefficient (μ)
Wood on Wood:—	
Ash on Oak, fibres parallel to motion	·40
Beech,, ,, ,, ,, ,, ,,	·36
Elm ,, ,, ,, ,, ,, ,,	·43
,, ,, ,, moving fibres at right angles to the others, and to motion	·45
Fir ,, ,, fibres parallel to motion	·36
Oak ,, ,, ,, ,, ,, ,,	·48
,, ,, ,, moving fibres at right angles to the others, and to motion	·32
,, ,, ,, fibres at right angles to motion	·34
,, ,, ,, moving fibres on end, resting fibres parallel to motion	·19
White Pine on White Pine, fibres parallel to motion	·40
Wood on Wood, soaped	·2 to ·04

TABLE OF COEFFICIENTS OF FRICTION—*continued.*

Sliding Surface.	Coefficient (μ)*
Metal on Wood—	
Wrought Iron on Elm, fibres parallel to motion	·25
" " " Oak, " " "	·62
Cast Iron on Elm, " "	·20
" " " Oak, " "	·49
Copper " " " "	·62
Brass " Pine, " "	·19
" " Mahogany " "	·24
Steel " Pine, " "	·16
" " Mahogany, " "	·18
Metal on Metal:—	
Wrought Iron on Wrought Iron, fibres parallel to motion	·14
" " " Cast " " " "	·19
" " " Brass, " "	·17
Cast " " Cast Iron	·15
" " " Brass	·15
Steel on Steel (*Trautwine*)	·14
" " Cast Iron	·20
" " Brass	·15
Copper on Cast Iron	·19
Metal on Metal (average), dry	·17
" " " " wet and clean	·36
" " " " damp and slimy	·14
Leather on Metal:—	
Leather on Metal, dry	·56
" " " wet	·36
" " " greasy	·23
" " " oily	·15
Wood on Stone:—	
Oak, across the grain, on hard or soft Limestone	·38
Metal on Stone:—	
Wrought Iron on soft Limestone, well dressed	·49
" " " hard " " "	·24
" " " " " " wet	·30
" " " " Marble, sawn (*Trautwine*)	·17
Stone on Stone, or Brickwork:—	
Polished Marble on polished Marble (*Trautwine*)	·16
" " " common Brick	·44
Soft Limestone, well dressed, on soft Limestone, well dressed	·64
Hard " " " " hard " "	·38
" " " " " soft " "	·67
Common Brick on soft Limestone, well dressed	·65
" " " hard " "	·60
" " " common Brick	·64
Masonry on dry Clay	·51
" " wet "	·33
Miscellaneous:—	
Leather on Oak	·27 to ·38
Hemp " " dry	·53
" " " wet	·33
Perodo fibre dry, heat limit 230° Fahr.	·45 to ·65
" bonded asbestos dry, heat limit 520° Fahr.	·3 to ·4
Surfaces continually lubricated	·8 to ·13

* μ varies with unit pressure.

WORK.

ACCUMULATED WORK.

W = weight of body. V = velocity.

$$\text{Work accumulated} = \frac{WV^2}{2g};$$

Where g = coefficient of terrestrial acceleration = 32·2.

If body changes its velocity from V to V_1,

$$\text{Work gained or lost} = \pm \frac{W(V^2 - V_1^2)}{2g}.$$

Accumulated Work.

(Continued.)

If a shot of weight w is fired into a mass of weight W lbs. with a velocity of V ft. per sec., then

$$\text{Work accumulated in shot} = \frac{1}{2} \cdot \frac{W}{g} \cdot V^2.$$

$$\text{Work lost} = \frac{1}{2} \cdot \frac{W}{g} \cdot \frac{W}{W+w} \cdot V^2. \qquad \text{Work remaining} = \frac{1}{2} \cdot \frac{W}{g} \cdot \frac{W}{W+w} \cdot V^2.$$

$g = 32{\cdot}2.$

Gravity.

g = acceleration in centimetres per second due to gravity. l = length of seconds pendulum in centimetres.

Values of g and l in various localities.

Place.	Latitude.	g.	l.
Equator	°	978·1	99·103
Greenwich	51·29	981·17 *	99·413 †
Paris	48·50	980·94	99·39
Berlin	52·30	981·25	99·422
Dublin	53·21	981·32	99·429
Glasgow	55° 52′ 42·″8	981·44	99·442
Edinburgh	55·57	981·54	99·451
N. Pole	90°	983·11	99·61

* Equals 32·1902, in ft. per sec.
† Equals 3·2615 ft.

THE PENDULUM.

Simple Pendulum.

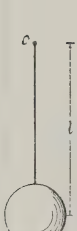


Fig. 44.

l = length of pendulum in inches. t = time in secs. of one oscillation (the time of moving from the highest point on one side to the highest point on the other). g = coefficient of terrestrial acceleration at equator = 32·097. ft. per sec. = 385·163 ins. per sec. $\pi = 3{\cdot}1416$.

$$t = \pi \sqrt{\frac{l}{g}} = 3{\cdot}1416 \sqrt{\frac{l}{385{\cdot}163}} = \frac{\sqrt{l}}{6{\cdot}247}.$$

Length of seconds pendulum at equator = 39·0265 ins.

If L = latitude,

$$l = 39{\cdot}0265 + {\cdot}1608 \sin^2 \mathrm{L}, \text{ for a 1-second pendulum.}$$

Compound Pendulum.

Fig. 45.

o = centre of suspension. l = equivalent length of simple pendulum to give the same time of oscillation. W = weight of lower bob. w = weight of upper bob.

Fig. 45, $l = \dfrac{a^2\mathrm{W} + b^2 w}{a\mathrm{W} - bw}$; Fig. 46, $l = \dfrac{a^2\mathrm{W} + b^2 w}{a\mathrm{W} + bw}$.

The length is actually the distance of the *centre of suspension* from the *centre of oscillation* (this is the same as the *centre of percussion*), the *centre of oscillation* being that point at which, if all the matter were collected, the time of oscillation would remain unaltered.

In case fig. 46, if w = W, then rate of oscillation will be greatest if w is put in such a position that $b = (\sqrt{2} - 1)a$, in which case $l = 2(\sqrt{2} - 1)a$.—*H.R.K.*

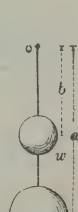


Fig. 46.

Conical Pendulum.

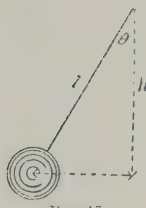


Fig. 47.

t = time of one revolution in secs., g = coefficient of terrestrial acceleration = 32·2, π = 3·1416.

$$t = 2\pi\sqrt{\frac{h}{g}}. \qquad (A)$$

This formula is correct irrespective of the length, l, of the rod (fig. 47).

Governor of Steam Engine.

Hence from (A), if

θ = angle (fig. 47) corresponding to a given period of rotation, t
g = coefficient of terrestrial acceleration = 32·2, π = 3·1416.

$$l = \frac{t^2 g}{4\pi^2 \cos\theta}.$$

CENTRE OF OSCILLATION OR PERCUSSION

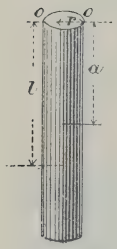


Fig. 48.

The *Centre of Oscillation* is that point in a vibrating body where, if all the matter were collected, the time of vibration would be unchanged.

The *Centre of Percussion* is that point in a revolving body at which, upon the point striking against an immovable obstacle, the whole of the motion of the body is destroyed. The centres of oscillation and percussion are the same.

o = centre of suspension. r = radius of cylinder. a = half length.

$$l = \frac{16a^2 + 3r^2}{12a} = \frac{4a}{3} + \frac{r^2}{4a}.$$

If r is small,

$$l = \frac{4a}{3}$$

i.e. in a bar the centre of oscillation or percussion is at a point distant from the centre of suspension two-thirds of the *total* length of the bar.

CENTRIFUGAL FORCE.

F = *Centrifugal Force*, in lbs.
W = *Weight* of revolving body in lbs.
v = *Velocity* of revolving body, in ft. per sec.
R = *Radius* of circle in which body revolves, in ft.
n = *Revolutions* per min.
g = Coefficient of terrestrial acceleration = 32·2. π = 3·1416.

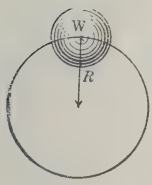


Fig. 49.

$$F = \frac{Wv^2}{gR} = \frac{Wv^2}{32{\cdot}16R}. \qquad F = \frac{4WR\pi^2 n}{60^2 g} = \frac{WRn^2}{2933}.$$

$$W = \frac{FgR}{v^2} = \frac{2933F}{Rn^2}. \qquad R = \frac{Wv^2}{Fg} = \frac{2933F}{Wn^2}.$$

$$n = \sqrt{\frac{2933F}{WR}}. \qquad v = \sqrt{\frac{FRg}{W}}.$$

Fig. 50.

Centrifugal force of a ring.

$$F = \frac{Wn^2\sqrt{R^2 + r^2}}{4150}.$$

Fig. 51.

Centrifugal force of a grindstone, circle-plane, cylinder, rotating round its centre.

$$F = \frac{WRn^2}{4150}.$$

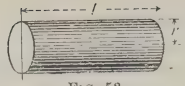


FIG. 52.

Centrifugal force of a cylinder rotating round the diameter of its base.

$$F = \frac{W n^2 \sqrt{4l^2 + 3r}}{10260}.$$

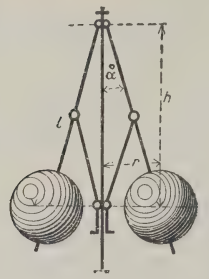


FIG. 53

Governor.

$$n = \frac{60}{2\pi}\sqrt{\frac{g}{h}} = \frac{54{\cdot}16}{\sqrt{h}} = \frac{54{\cdot}16}{\sqrt{l\cos\alpha}},$$

$$h = \frac{2933}{n^2}, \quad l = \frac{2933}{n^2\cos\alpha} = \frac{h}{\cos\alpha},$$

$$\cos\alpha = \frac{2933}{n^2 l} = \frac{h}{l}, \quad r = \sqrt{l^2 - h^2}$$

RADIUS OF GYRATION.

The *Centre of Gyration* is that point at which, if all the revolving matter were collected, it would obtain equal angular velocity from, and sustain equal resistance to, the force that gives the rotary motion. The *Radius of Gyration* (x) is the distance of the centre of gyration from the axis, A.

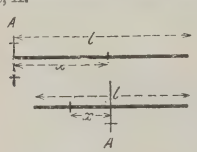


FIG. 54.

A Line or Bar.

$x = 0{\cdot}5775l,$

$x = 0{\cdot}2887l.$

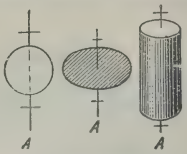


FIG. 55.

A Circumference round its Diameter.

A Circle-plane round its Centre.

A Cylinder round its Axis.

$x = 0{\cdot}7071r.$

FIG. 56.

A Circle-plane round its Diameter.

$x = 0{\cdot}5r.$

FIG. 57.

A Sphere round its Diameter.

Convex surface.
$x = 0{\cdot}8165r.$

Solid.
$x = 0{\cdot}6324r.$

r = radius.

FIG. 59.

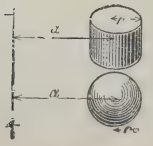


FIG. 58.

Cylinder.

$x = \sqrt{a^2 + \frac{1}{2}r^2},$

Sphere.

$x = \sqrt{a^2 + \frac{2}{5}r^2}.$

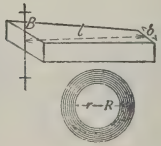


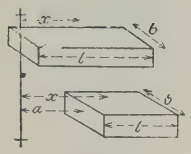
FIG. 60.

Wedge.

$x = 0{\cdot}204\sqrt{12l^2 + B^2 + \delta^2}.$

Ring.

$x = \sqrt{\frac{R^2 + r^2}{2}}.$

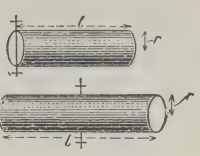


Parallelopiped.

$$x = \sqrt{\frac{4l^2 + b^2}{12}}.$$

$$x = \sqrt{\frac{4l^2 + b^2}{12} + a^2 + al}.$$

FIG. 61.

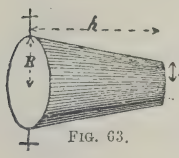


Cylinder.

$$x = \sqrt{\frac{4l^2 + 3r^2}{12}}.$$

$$x = \sqrt{\frac{l^2 + 3r^2}{12}}.$$

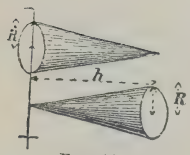
FIG. 62.



Conic Frustum.

$$x = \sqrt{\frac{h}{10}\left(\frac{R^2 + 3Rr + Rr^2}{R^2 + Rr + r^2}\right) + \frac{3}{20}\left(\frac{R^5 - r^5}{R - r^3}\right)}.$$

FIG. 63.

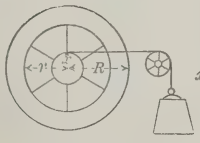


Cone.

$$x = \sqrt{\frac{2h^2 + 3R^2}{20}},$$

$$x = \sqrt{\frac{12h^2 + 3R^2}{20}}.$$

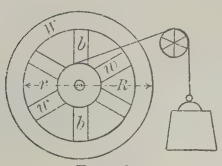
FIG. 64.



Flywheel.

$$x = \sqrt{\frac{R^2 + r^2}{2}},$$

FIG. 65.



Flywheel with Arms.

$$x^2(W + w) = W\frac{R^2 + r^2}{2} + w\frac{4r^2 + b^2}{12},$$

$$= \sqrt{\frac{6W(R^2 + r^2) + w(4r^2 + b^2)}{12(W + w)}}.$$

FIG. 66.

MOMENTS OF INERTIA.

If each element of the mass of any solid body be supposed to be multiplied by the square of its perpendicular distance from a given line or axis, the sum of all these products is called the moment of inertia of the body with respect to that axis. The following Table gives the most useful cases :

M = mass of the body. I = moment of inertia.

Uniform thin rod, axis at end, length = l . . . . . $I = M\frac{l^2}{3}$.

„ „ „ through middle, length = l . . . . . $I = M\frac{l^2}{12}$.

Rectangular lamina, axis through centre and parallel to one of sides, a and b lengths of sides, a the side bisected . . . . . $I = M\frac{a^2}{12}$.

Rectangular lamina, axis through centre and parallel to one of sides, a and b lengths of sides, b the side bisected . . . . . $I = M\frac{b^2}{12}$,

Rectangular lamina, axis through centre and perpendicular to the plane, a and b lengths of sides . . . . . $I = M\frac{a^2 + b^2}{12}$

Rectangular parallelopiped, axis through centre and perpendicular to a side, a, b, and c lengths of sides:

1. Axis perpendicular to side contained by a and b . . . $I = M\frac{a^2 + b^2}{12}$.
2. ,, ,, ,, ,, a ,, c . . . $I = M\frac{a^2 + c}{12}$.
3. ,, ,, ,, ,, b ,, c . . . $I = M\frac{b^2 + c^2}{12}$.

Circular plate, axis through centre perpendicular to the plate, radius $= r$ $I = M\frac{r^2}{2}$.

Circular plate, axis any diameter, radius $= r$ . . . . $I = M\frac{r^2}{4}$.

Circular ring of round material, axis through centre perpendicular to plane of ring, outer radius $= R$, inner radius $= r$ . . . . $I = M\frac{7R^2 + 2Rr + 7r^2}{16}$

Circular ring of round material, axis any diameter . . . . . $I = M\frac{9R^2 - 2Rr + 9r^2}{32}$

Circular ring of flat material (washer), axis through centre perpendicular to plate of ring, outer radius $= R$, inner radius $= r$ . . . $I = M\frac{R^2 + r^2}{2}$.

Right cylinder, axis the axis of figure, r = radius of section . . . $I = M\frac{r^2}{2}$.

Right cylinder, axis through centre perpendicular to axis of cylinder of length, l . . . . . $I = M\left(\frac{l^2}{12} + \frac{r^2}{4}\right)$.

Hollow cylinder, axis the axis of figure, outer radius $= R$, inner radius $= r$ $I = M\frac{R^2 + r^2}{2}$.

Right cone, axis the axis of figure, r = radius of base . . $I = \frac{3}{10}Mr^2$.

Sphere, axis any diameter, r = radius . . . . $I = \frac{2}{5}Mr^2$.

ENERGY OF A ROTATING BODY.

The energy of a rotating body $= \frac{1}{2}\omega^2 I$, ω being the angular velocity, and I the moment of inertia.

BALLISTIC PENDULUM.

W = weight of pendulum. P = weight of a shot fired into W. V = velocity of shot. θ = angle through which pendulum moves. n = number of vibrations per minute pendulum makes. a = distance from centre of suspension of point at which shot enters block. h = distance of centre of gravity of pendulum weight with shot in it from centre of suspension. g = coefficient of terrestrial acceleration = 32·2. $\pi = 3{\cdot}1416$.

$$V = \frac{W + P}{P} \cdot \frac{120g \sin \frac{\theta^\circ}{2}}{\pi n} \cdot \frac{h}{a}$$

BRAKES FOR POWER MEASUREMENT.

Hirn's Rope Brake.

A convenient form of brake for power measurement is one consisting of a band of rope enveloping the fly-wheel of the engine being measured.

Fig. 67 (p. 35) represents the rope brake as arranged by Mr. B. H. Thwaite for the purpose. It consists of a band of rope carrying five wood blocks, with side flanges to hold the brake on the wheel. Each end of the rope is attached to a spring balance.

The net pull upon the wheel rim is the difference of the two spring balance readings, and the power developed by the engine is

$$(W - w) \times D\ \pi \times N \div 33{,}000 = B.H.P.,$$

where,

W = reading of the large balance or the pull in lbs.; w = reading of the small balance or the pull in lbs.; D = diameter in ft. of the rim on which the brake lies; N = revolutions per minute; B.H.P. = brake horse-power; $\pi = 3{\cdot}1416$.

The results of the formula are slightly below the actual power, owing to the brake belt not being absolutely flexible. French engineers measure D at the centre of the rope thickness, an error perhaps slightly in the other direction.

For the large spring balance a weight may be substituted.

A rope brake never sticks, nor does it ever suddenly lose its grip.

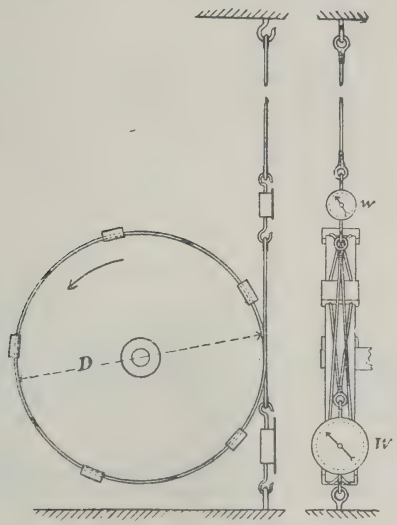


FIG. 67.

The Prony Brake.

This form of brake is useful for small power measurements. In its simple form, fig. 68, it consists of a couple of wooden blocks clipped upon the pulley or shaft transmitting the power

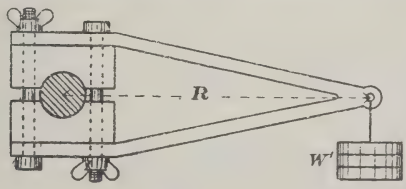


FIG. 68.

to be indicated. Attached to the blocks is a lever, to which is hung a scale pan or a spring balance. The diameter D in the formula for Thwaite's brake is, in the case of the Prony brake, double the radius R measured to the point of suspension of the pan, and $W - w$ is the weight W'.

The difficulty with the Prony Brake is the constant vibration of the weighted arm. The difficulty has been reduced at Purdue University by the addition of a pendulum lever weight, B, fig. 69 (p. 36), the quadrant of which may be divided by actual weight determination.

Fig. 70 shows a form of stabiliser* which, it is stated, will damp down the vibrations to a very marked extent. Between the balance and the brake arm are placed in series a heavy weight A and a resilient coil spring B. The pointer of the balance is then set back to offset the

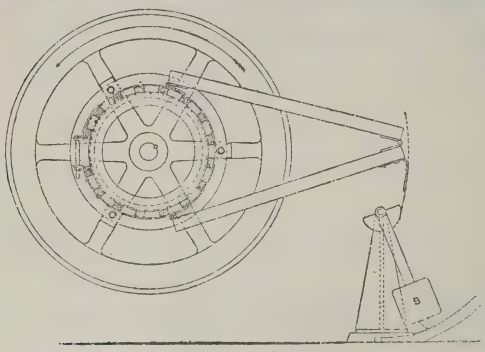


FIG. 69

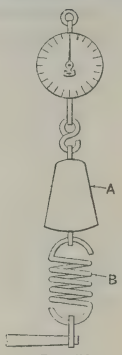


FIG. 70.

initial force, or tare, exerted by the weight. When the motor being tested is in motion, the average torque is transmitted through the spring and suspended weight, and is accurately indicated on the dial of the balance. The momentary vibrations, however, are not of sufficient duration to move the weight, and their motion is dissipated by the auxiliary spring. By employing this type of stabiliser, an oil dashpot may be dispensed with entirely, and readings of a high degree of accuracy may be obtained.

Ringelmann's Brake.

In the brake of M. Ringelmann (fig. 71) the brake band is a strap of iron, on which, by a hook, *C*, is suspended a weight, *Q*. If the wheel runs faster and lifts the weight, the hook travels

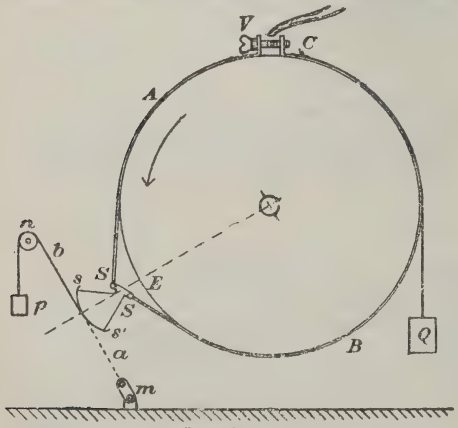


FIG. 71.

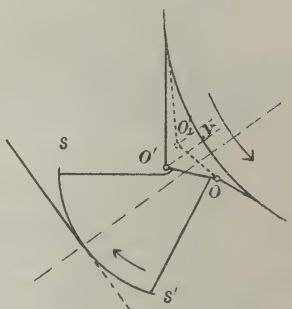


FIG. 72.

in the direction of the arrow, and *vice versa*. The brake is in two halves, *A* and *B*, joined by a tightening screw at *V*, and by a peculiar joint at *E*, shown enlarged in fig. 72. The sector, *S S*, which carries the ends of the brake straps, is held by a pair of cords, *mas* and

* J. W. R., *Machinery*, June 7, 1923.

s'bnp, the first fixed to a regulating roller *m*, attached to the floor; the second to a small weight, *p*, over a pulley, *n*. The mean position of the sector is as shown, the cords *b* and *a* being in one straight line. If in the course of a test the brake is pulled round the point *O* (fig. 72) is lowered, the sector rolls on the cord, and the point *O'* comes to *O'*$_1$, lengthening the brake strap by the amount *y*. Soap and water is kept flowing on at the point *V*. The tension of the brake is thus maintained to a constant power, and the lubrication is also constant, so that the motor is compelled to exercise a constant power. This brake was used for the testing of small motors at Meaux by M. Ringelmann in 1894.

Water-pipe Absorption Dynamometer.

In place of ropes the encircling band may be made of flexible water-pipes, the rim being of wood. Water is circulated through the pipes to keep them cool. Each band of $\frac{3}{4}$ in. iron pipe on

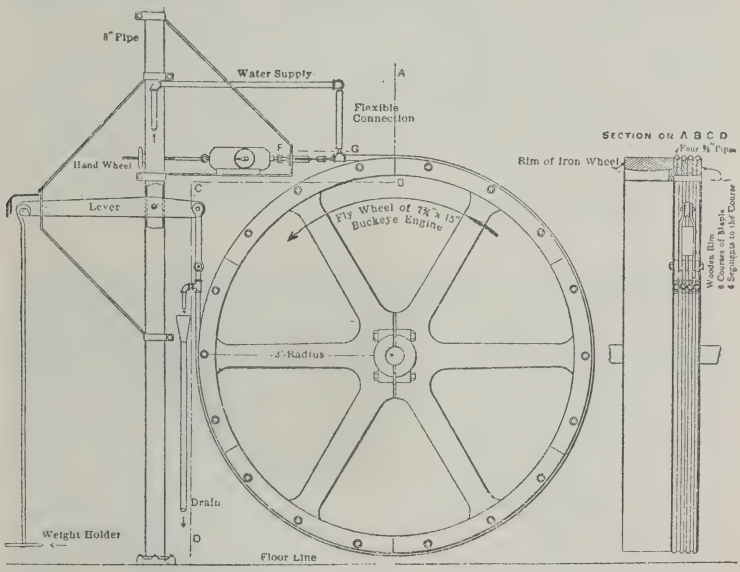


FIG. 73.

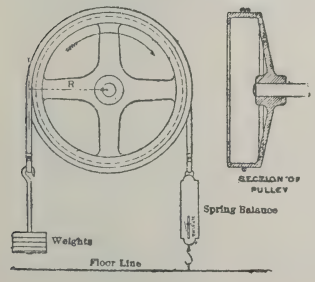


FIG. 74.

a 6 ft. wooden rim will absorb 10 horse-power at 150 revolutions. Fig. 73, as used at Purdue University, shows a brake of this description.

Cooling Brake Rims.

In many cases the inside of a brake rim is of trough form (fig. 74), which will hold water when revolving, by virtue of the centrifugal action. The trough being filled with water is kept to (at most) 212° F. by the evaporation of the water, which is kept constantly running into the rim in a small stream.

Another means of cooling is to encircle a sheet-steel brake band with iron wire netting, which holds by capilliary action water poured on it; this evaporates and cools the strap. Surplus water runs off at the bottom of the band.

The Alden Brake.

This brake consists, for 200 horse-power, of a 56-inch disc of cast iron revolving between two plates of copper, containing cylinder oil, which is circulated from the circumference towards the centre. Outside the copper plates water circulates at about 15 lbs. pressure, and serves both to cool the plates and maintain contact pressure on the moving disc. By circulating more water the brake resistance may be sensibly increased through the diminished viscosity of the oil, and *vice versâ*. This brake is said to run for hours at great constancy, and to be nearly perfect in its action.

The Brotherhood Fluid Friction Dynamometer.

This arrangement (*Peter Brotherhood, Ltd., Peterborough*) is specially suitable for high and moderate speed testing. All power given to the machine is recorded accurately: no losses in shaft bearings, etc., have to be allowed for, as the friction of bearings is measured by the dynamometer and included in the readings; no calibration is required. Calculations involve simple multiplication only from the direct readings of a spring balance or weighing machine; no clutch is required. Motors under test may be started up with the brake set for full loads, the torque at the lowest speeds being extremely small and rising rapidly as the normal speed is reached. The resistance rises very rapidly with increase of speed and also diminishes rapidly with decrease in speed, so that the brake is self-governing, and any sudden increase in friction is impossible. The machine is extremely accurate, the only power not recorded being the infinitesimal friction between the casing and its supporting roller bearings. The casing rests freely on the rollers, and motion at the supports only occurs when the load is changing, and is then very small, so that no measurable energy is absorbed. The constants may be verified by use of a rule and standard weights only, the only factors being the distance from the shaft centre to the point of application of the spring balance, and the readings of the spring balance which may be directly tested by deadweights. This feature enables the user to check the readings immediately before and after each test if desired.

The principle employed is that of the frictional resistance between fluids and rotating plates. Several perforated annular plates are secured to the casing of the brake, and between each pair a disc rotates with the shaft. The rotating discs run near the fixed plates, but do not touch them. On the introduction of water or other fluid into the casing, friction is set up between the fluid and the plates. The moment of the frictional resistance between the fluid and the fixed plates is measured by means of a spring balance acting at a convenient radius, the casing being free to rotate within limits on trunnions and anti-friction rollers. A platform weighing machine may be substituted for the spring balance if preferred. The energy delivered to the dynamometer is transformed by fluid friction into heat which is absorbed by the fluid; as no solid friction occurs, unequal heating and seizing of surface cannot arise. The quantity of water flowing through the dynamometer need be only sufficient to absorb the power and keep the temperature down to the desired limits. For example, when absorbing 100 b.h.p., assuming that the temperature of the water supply is 60° F., and that the water is raised to 160° F. before leaving the casing, about 5 gallons per minute would be sufficient. This water enters and leaves the casing quietly—no water is splashed about. The load on the motor at any desired speed can be quickly and smoothly adjusted by varying the amount in the casing; any desired quantity of water may be retained, and therefore any load may be steadily maintained, although water is flowing through continuously. For short tests sufficient water may be placed in the casing, and no continuous flow provided. This feature is of great use in comparing and calibrating motors, engines, etc. With these dynamometers a constant and definite torque resistance will always be obtained at any given speed and with a definite quantity of water in the casing. It is thus easy to adjust the machine to give a predetermined load curve. This feature has been found of great value in testing engines for marine propulsion, the dynamometer resistance varying with speed proportionally in precisely the same way as the speed and power of a marine propeller varies.

The Froude Water Dynamometer.

This dynamometer (*Heenan & Froude, Ltd., Worcester*), is made in the following types, viz.:

1. *Type D.P.X.*, in small and medium sizes for dealing with high-speed prime movers, including internal combustion engines for motor cars and aeroplanes.

2. *Types S. and F.*, in medium and large sizes, for dealing with slow-speed prime movers, such as land and marine Diesel, oil, gas, and steam engines, also with medium and slow-speed and hydraulic turbines.

Fig. 75 shows a cross-sectional drawing through the D.P.X. type 'Froude' dynamometer.

The engine is directly coupled to the main shaft, transmitting the power to a rotor revolving inside a casing, through which water is circulated to provide the hydraulic resistance and simultaneously to carry away the heat developed by destruction of power.

In each face of the rotor are formed pockets of semi-elliptical cross-section divided one from another by means of oblique vanes. The internal faces of the casing are also pocketed in the same way.

When in action, the rotor discharges water at high speed from its periphery into the casing pockets, by which it is then returned at diminished speed into the rotor pockets at a point near the shaft. Thus, the pockets in rotor and casing together form elliptical receptacles round which the water courses at high speed, creating vortices which destroy the power of the engine as quickly as it is developed.

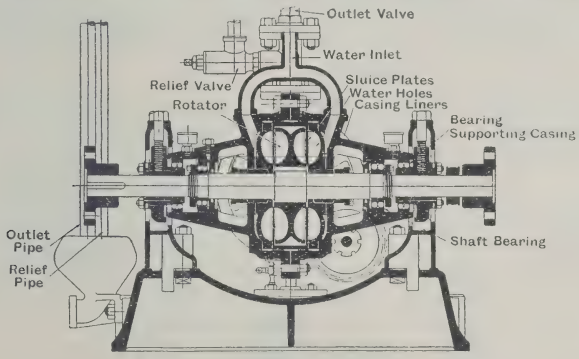
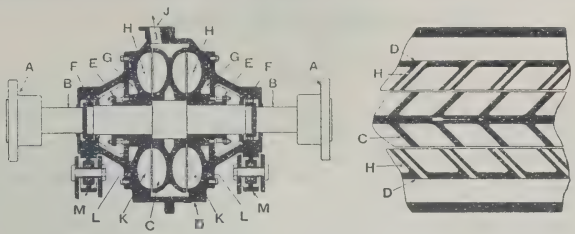


FIG. 75.

Between the faces of the rotor and the casing are interposed thin metal plates or sluices capable of sliding radially towards or away from the shaft. By rotating an external handwheel these sluices can be caused to mask or uncover more or less of the power-absorbing cups, and in this way to regulate the load exerted by the dynamometer.

In this type of dynamometer under running conditions the pockets should remain always full of water, which enters through holes in the casing to a point near the centre of each vortex where



A...Couplings	E...Gland	J....Water Outlet
B...Shaft	F....Bearing	K....Air Holes
C...Rotor	G...Water Inlet Channel	L.... Air Channel
D...Casing	H...Water Holes	M...Casing Trunnions

FIG. 76.—Sectional Elevation through Casing of Single Rotor Non-Reversible Dynamometer showing Section through Vanes.

the pressure is low. After being heated by the destruction of power, the water passes to waste through an outlet valve, which in practice is regulated to give to the water a temperature of about 140° F.

All ' Froude ' dynamometers have casings supported upon anti-friction trunnions and free to swivel about the axis of the shaft. The shaft is mounted upon bearings fitted into the casing and not in independent supports. The piping conveying water to and away from the casing is flexible, and precautions are taken to prevent external forces from resisting the free swivelling motion of the casing. When a turning moment is applied to the shaft, any resistance created by the dynamometer, whether by hydraulic or solid friction, must of necessity react upon the casing, which tends to swivel upon its supports. In the ' Froude ' dynamometer this tendency is resisted solely

by a weighing machine, which correctly indicates the magnitude of the torque. *This arrangement ensures scientific accuracy.*

The form of weighing apparatus varies somewhat. In the D.P.X. type dynamometer it consists of an arm directly attached to the casing, and connected through pin joints to a spring balance upon which dead weights are suspended so as to put the springs of the balance into tension. When the dynamometer is at rest, uncoupled from the engine, and the arm is in a horizontal position, the pointer indicates zero. The dial is graduated so as to give a direct reading of the lifting force exerted by the end of the lever arm when at work.

Fig. 76 shows a cross-sectional drawing through the 'Froude' dynamometer types S. and F.

The power-absorbing pockets and vanes are generally similar in shape to those incorporated in the D.P.X. type, but the vanes in the casing are drilled with air holes in addition to water

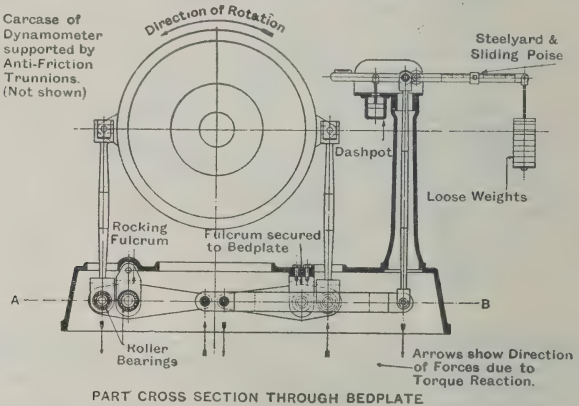


FIG. 77.—Arrangement of Torque Meter for Froude Dynamometer.

holes. No sluice plates are interposed between rotor and casing, but the load is regulated by varying the volume of water contained by the power-absorbing pockets.

The shape of the pockets is such as to create the formation of eddies in exactly the same manner as previously described, but in the case of the S. and F. type dynamometers the pockets do not always remain full of water, and the vortices which they contain are free to cavitate under the action of centrifugal forces.

When working full of water they exert maximum resistance to rotation; when nearly empty, each pocket contains a vortex, consisting merely of a thin film of water coursing around the periphery of the pocket, and resistance to rotation is small. The partial closing of the outlet valve fixed upon the casing increases both the volume of water in the pockets and the load, and *vice versâ*.

The dynamometer carcase carries a special centrifugal pump (not shown in the above figure) positively driven from dynamometer shaft, controlling the rate of flow of water to the power absorbing cups. This gives the dynamometer automatic self-governing properties, such that should the speed of the dynamometer (and therefore of the pump) become slightly higher, the dynamometer load will increase in a higher proportion, owing to the augmented rate of water supply which increases the volume of water in the cups. The contrary effect takes place upon a decrease in speed, and this patented feature is of great assistance in keeping steady the speed of the engine upon test and minimising 'hunting,' often caused previously by the usual slight fluctuations of engine output.

The pump is fitted with a fine-adjustment valve which is used in conjunction with the outlet valve for control of load and speed. The whole of the power absorbed by pump and drive is correctly indicated by the weighing apparatus, so that no errors in B.H.P. readings result on this account.

In the new 'S' type of dynamometer, the reaction is balanced by coupling up the measuring apparatus to two diametrically opposed points on the periphery of the casing, as seen in fig. 77. This shows a diagrammatic arrangement of Heenan & Froude's new Torque Meter. The casing is coupled up at the two points through stout rods having articulated joints, to a system of levers

fixed in the underlying baseplate. When the direction of rotation is counter-clockwise, as shown, the right-hand connecting rod transmits a vertically upward force upon the short arm of main lever B; the left-hand connecting rod pushes downwardly upon main lever A; and the long arms of this pair of levers are connected together by the short arm of a third lever, the long end of which pulls down upon a steelyard type of weighing machine. The direction of the forces due to torque is shown by the arrows. This system of levers is equivalent to a lever arm fixed on the dynamometer casing, and having an effective length 600 times as great as the radius from axis of dynamometer to one of the points where the connecting rods are attached to the casing. The torque can thus be measured by a number of very small weights and a steelyard, instead of by enormously heavier weights.

Dynamometers for aircraft engine testing.

The D.P.X. type Froude dynamometer is usually employed for aircraft engine testing, a motor-driven centrifugal fan and wind tunnel being added when required for cooling air-cooled engines. In this form the plant is manufactured complete in all details, and is known as the 'Froude' wind tunnel testing plant. It is usually found that a single plant of this type possesses sufficient range of power and ventilating capacity to cover all modern aircraft engines, whether direct drive or geared drive, air cooled, or water cooled.

For testing air-cooled aircraft engines, all of substantially similar powers and speeds (e.g., for manufacturers' production testing), the same makers have developed the 'Heenan-Fell' patent air brake dynamometer, in which the engine power is absorbed by a centrifugal air impeller in a casing fitted with patented shutter for regulating the load. The range of power and speed is less extensive than that of the 'Froude' dynamometer, but the advantage is obtained that the engine provides its own cooling air blast. B.H.P. readings are strictly accurate under all conditions of temperature, barometric pressure, etc., and the dynamometer is silent running.

The Fan Dynamometer.

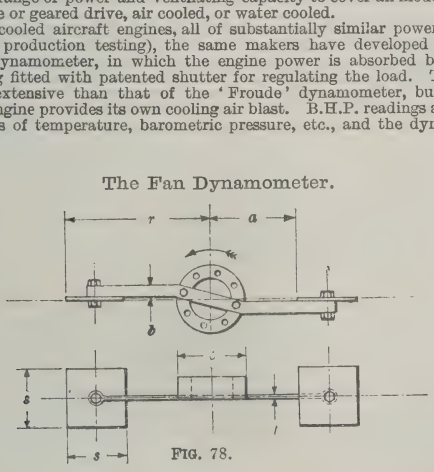


FIG. 78.

If

w = torque driving the fan in inch-lbs.; w_a = resisting torque due to the arms alone;
w_b = resisting torque due to the blades alone; N = revs. per min. of the fan;

s = length of side of square blades or diameter of round blades in inches;
r, a, b, t, and c are other dimensions in inches;

W = density of the fluid in which the fan rotates, in lbs. per cubic foot,

$$= 62{\cdot}33 \text{ for water at } 50^\circ \text{ F.} = \frac{1{\cdot}317 \times \text{barometric pressure in in., for air}}{(460 + \text{temp. in } ^\circ \text{F.})};$$

H.P. = horse-power absorbed by fan;

K = a coefficient depending upon the value of r/s and upon the shape of the blade; see values tabulated below for square and circular blades having square edges.

Values of r/s . . .	2·5	3·0	3·5	4·0	4·5	5·0	5·5	6·0	6·5	9·0	12·0
$K \times 10^{-5}$ for circular blades	1·69	1·69	1·66	1·63	1·60	1·59	1·58	1·575	1·575	1·575	1·575
$K \times 10^{-5}$ for square blades	2·25	2·28	2·24	2·18	2·16	2·12	2·10	2·08	2·04	2·04	2·04

σ = a coefficient depending upon the ratio b/t.

Values of b/t . .	2	3	6
Values of σ . .	1·3	0·94	0·62

then,

$$wa = 0{\cdot}35 \times 10^{-6}\sigma N^2 a^4 \ell W\,; \quad wb = KN^2 r^3 s^2 W\,; \quad w = wa + wb^*\,; \quad \text{H.P.} = wN/63{,}000.$$

Notes on the Fan Dynamometer.

Care should be taken that the arms and fastenings are amply strong enough to withstand the centrifugal stresses. For reasonably accurate work, with fans rotating in air, one side of the fan should be completely free from obstruction for a distance of at least '3 r' inches from the plane of rotation, and on the other side 'O' should not exceed 'a.' Also the floor should be distant at least 'r' inches. The linear velocity of the centre of the blades must not exceed, say, 350 ft. per second. In order to compute W with accuracy, the barometric pressure, and the temperature of the air at the periphery of the fan should be taken. Fans rotating in water must have at least as much clearance around them as is specified above, and must not be rotated at such a speed as to produce cavitation. To facilitate work in the test-shop the 'H.P., Rev.' curves for each blade used should be plotted on logarithmic paper for some standard value of W. Since the H.P. varies approximately as N^3, N should be observed with great accuracy—a speedometer is usually not sufficiently accurate.

See also Hodgson, *Proc. Inst. Automobile Engineers*, March 1916.

(*J. L. Hodgson.*)

The Torsion Meter.

(*Hopkinson-Thring Patent.*)

The Hopkinson-Thring Torsion Meter has been designed for determining the power transmitted by turbine and other shafts. It requires only a very short length of shaft, is extremely compact and strong, and it gives a direct reading of the torsion on the shaft. The zero point of the apparatus can readily be ascertained by barring the shaft. The instrument offers also a simple means of determining the friction of the shafts themselves.

The principle of the apparatus consists in the observation of the twist between two adjacent points on the shaft by means of two beams of light projected on to a scale from a fixed and a movable mirror.

A diagrammatic arrangement of the complete apparatus is shown in end elevation and plan in fig. 79. A collar R, clamped to the shaft of which the torque has to be measured, is provided with a flange projecting at right angles to the shaft and an extension. A sleeve T, provided with a similar flange and extension at one end, is clamped at its further end on to the shaft in such a manner that its flange is close to that on the collar R, whilst its extension overlaps that of the collar R, on which it is supported to keep it concentric. Both the collar and sleeve are quite rigid, so that when the shaft is twisted by the transmission of power, the flange on the sleeve T will move relatively to that on the collar R, the movement being equal to that between the two parts of the shaft on which these fittings are clamped. This movement is made visible by one or more systems of torque mirrors mounted between the two flanges, which reflect a beam of light, projected from a lamp, on to a scale divided in a suitable manner on ground glass.

Each system of torque mirrors consists of a mounting, pivoted top and bottom on one or other of the flanges, in which two mirrors are arranged back to back. This mounting is provided with an arm, the end of which is connected by a flat spring to an adjustable stop on the other flange. Any relative movement of the two flanges will turn the torque mirror and thereby cause the beam of light to move on the scale, the deflection produced being directly proportional to the torque applied to the shaft. Hence, if the rigidity of the material and the number of revolutions per minute are known, the h.p. transmitted can be arrived at immediately.

With the arrangement described, a reflection will be received from each mirror at every half revolution of the shaft; but where the torque varies during a revolution (as with reciprocating engines), a second system of mirrors may be arranged at right angles to the first system, so that four readings can be taken during one revolution; or, if two scales are used, eight readings can be taken.

Fig. 80 shows how the beam of light reflected by the mirror when in its highest position passes through the upper part of the scale; while the second reflection will occur when the mirror is in the position occupied by the zero mirror, the beam of light passing through the lower part of the scale. The position of the torque mirror in fig. 80 is such that if the reflected beam strikes the scale to the right of the zero line, when the shaft has made a further half revolution, the reflected beam from the other mirror will strike the scale to the left of the zero line.

The fixed mirror is attached to one of the flanges (in fig. 79 to the flange of the sleeve T). This must be adjusted so that the beam of light reflected from it is received at the same point on the scale as those from the movable mirrors when there is no torque on the shaft. To facilitate

* For this relation to hold, t must be less than $\frac{s}{7}$.

HOPKINSON-THRING TORSION METER.

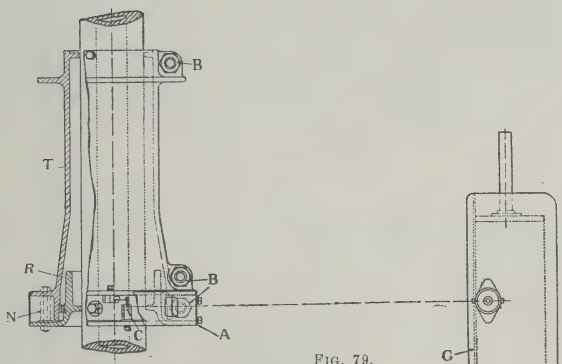


FIG. 79.

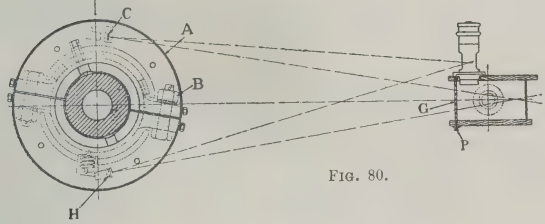
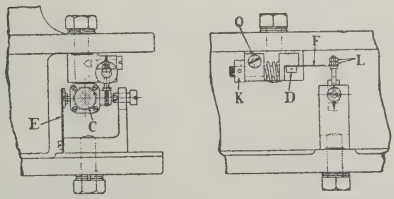


FIG. 80.



the erection and adjustment of the apparatus, the box containing the scale and carrying the lamp is fitted with trunnions, so that it can be inclined as required.

If the position of the apparatus on the shaft becomes altered relatively to the scale, owing to the warming up of the shaft or from other causes, this is indicated immediately to the observer by an alteration in the position of the zero as reflected by the fixed mirror. Hence the scale zero may be adjusted, if desired, by moving the scale so that its zero coincides with the reflection from the fixed mirror, but this is not necessary to obtain a correct result, since the mean of the two readings will be the same. It will readily be understood that, as a movement of the torque mirrors can only occur through a relative movement of the two flanges, the vibration of the shaft or of the ship will not influence the readings in any way.

The apparatus is made in seven standard sizes, to fit shafts from 1 up to 21 inches in diameter, the overall length of the meters being 13½ to 58½ inches. All sizes have been in practical operation for a considerable time, with entirely satisfactory results, in warships and other turbine-driven steamers, running at speeds up to 1,500 revolutions per minute. The smaller sizes of the instrument are designed to run up to 3,000 revolutions per minute.

ANIMAL POWER.

MEN.

The mean effect of the power of a man, unaided by a machine, working to the best practicable advantage, is the raising of 70 lbs., 1 foot in a second, for 10 hours in a day.

Two men, working at a windlass with the handles at right angles to each other, can raise 70 lbs. more easily than one man can 30 lbs.

A man of ordinary strength exerts a force of 30 lbs., for 10 hours in a day, with a velocity of 2½ feet per second = 4,500 lbs. raised 1 foot in a minute = ·2 of the work of a horse.

A man can travel, without a load, on level ground, during 8½ hours a day, at the rate of 3·7 miles an hour, or 31½ miles a day. He can carry 111 lbs. 11 miles in a day.

A porter going short distances, and returning unloaded, can carry 135 lbs. 7 miles a day. He can transport, in a wheelbarrow, 150 lbs. 10 miles in a day.

WEIGHT OF CROWDS.

Experiments carried out by Professor L. J. Johnson, of Harvard University, show that it is possible for this to reach the high figure of 181·3 lbs. per square foot of floor surface. If the crowd jumps, the apparent load may be 50 per cent. more than the steady load, and running four abreast will show a similar result. A weight of 140 lbs. per square foot is quite possible where there are throngs of people; while a load of 80 lbs. per square foot is quite common in buildings and private houses where social gatherings are frequent, and a maximum of 40 lbs. or 45 lbs. is far short of what actually does prevail.

Some experiments carried out in America show that a crowd of men pushing against a balustrade may exert a horizontal pressure of 168 lbs. per foot run when they are three deep.

HORSES.

A horse can travel 400 yards, at a walk, in 4½ minutes; at a trot, in 2 minutes; and at a gallop, in 1 minute. A draught horse can draw 1,600 lbs. 23 miles a day, weight of carriage included. The ordinary work of a horse may be stated at 22,500 lbs., raised 1 foot in a minute, for 8 hours a day. In a horse-mill a horse moves at the rate of 3 feet in a second; the diameter of the track should not be less than 25 feet. The strength of a horse is equivalent to that of 5 men.

TABLE OF THE AMOUNT OF LABOUR A HORSE OF AVERAGE STRENGTH IS CAPABLE OF PERFORMING, AT DIFFERENT VELOCITIES, ON CANALS, RAILROADS, AND ROADS.

Velocity per Hour.	Duration of Work.	Useful Effect for 1 Day, drawn 1 Mile.			Velocity per Hour.	Duration of Work.	Useful Effect for 1 Day, drawn 1 Mile.		
		On a Canal.	On a Railroad.	On a Road.			On a Canal.	On a Railroad.	On a Road.
Miles.	Hours.	Tons.	Tons.	Tons.	Miles.	Hours.	Tons.	Tons.	Tons.
2½	11·5	520	115	14·0	6	2·000	30·0	48·0	6·0
3	8·0	243	92	12·0	7	1·500	19·0	41·0	5·1
4	4·5	102	72	9·0	8	1·125	12·8	36·0	4·5
5	2·9	52	57	7·2	10	0·750	6·6	28·8	3·6

A horse in a mill can produce an effect of 106 lbs., at a velocity of 3 feet in a second, for 8 hours in a day. A mule can produce, under a like velocity and time, an effect of 71 lbs.; and an ass, 37 lbs.

An ox, walking at a velocity of 2 feet in a second (1·34 miles per hour), will exert a pull of 154 lbs. for 8 hours in a day.

The draught of man and animals by traces is as follows:

Man, 150 lbs. | Horse, 600 lbs. | Mule, 500 lbs. | Ass, 360 lbs.

A man rowing a boat 1 mile in 7 minutes performs the labour, while rowing, of 6 fully-worked labourers at ordinary occupations of 10 hours.

The force of traction on a railroad or road is constant, but the mechanical power necessary to move the carriage increases as the velocity.

Labour upon Embankments.

Single Horse and Cart.—A horse with a loaded dirt-cart, employed in excavation and embankment, will make 100 lineal feet of trip, or 200 feet in distance per minute, while moving. The time lost in loading, dumping, awaiting, etc. = 4 minutes per load.

A medium labourer will load with a cart in 10 hours, the following earths, measured in the bank:

Gravelly Earth, 10; *Loam*, 12; *Sandy Earth*, 14 cubic yards.

Carts are loaded as follows: *Descending Hauling*, $\frac{1}{2}$ of a cubic yard in bank; *Level Hauling*, $\frac{2}{3}$ of a cubic yard in bank; *Ascending Hauling*, $\frac{1}{3}$ of a cubic yard in bank.

Loosening, etc.—In *Loam*, a three-horse plough will loosen from 250 to 800 cubic yards per day of 10 hours.

WORK DONE BY MEN AND ANIMALS.

Nature of Labour.	Daily Duration of Work in Hours.	No. of Units of Work per Day.	No. of Units of Work per Min.	Weight raised or Mean Pressure.	Velocity.	
					Feet per Min.	Miles per Hour.
(1) *Raising weights vertically.*						
A man mounting a gentle incline or ladder without burden, i.e. raising his own weight . .	8·0	2,032,000	4,230	145	29	0·33
Labourer raising weights with rope and pulley, the rope returning without load . .	6·0	563,000	1,560	40	39	0·44
Labourer lifting weights by hand	6·0	531,000	1,480	44	34	0·38
Labourer carrying weights on his back up a gentle incline or up a ladder and returning unladen	6·0	406,000	1,130	145	8	0·09
Labourer wheeling materials in a barrow up an incline of 1 in 12 and returning with the empty barrow	10·0	313,000	520	130	4	0·045
Labourer lifting earth with a spade to a mean height of 5½ ft.	10·0	281,000	470	6	78	0·9
(2) *Action on Machines.*						
Labourer walking and pushing or pulling horizontally . .	8·0	1,500,000	3,130	27	116	1·32
Labourer turning a winch .	8·0	1,250,000	2,600	18	144	1·64
Labourer pulling and pushing alternately in a vertical direction	8·0	1,146,000	2,390	11	216	2·70
Horse yoked to a cart and walking	10·0	15,688,000	26,150	150	175	2·00
Do. to a whim gin . . .	8·0	8,440,000	17,600	100	175	2·00
Do. do. trotting . . .	4·5	7,036,000	26,060	66⅔	391	4·44
Ox yoked to a whim gin and walking	8·0	8,127,000	16,930	145	117	1·33
Mule do. do.	8·0	5,627,000	11,720	66⅔	176	2·00
Ass do. do.	8·0	2,417,000	5,030	30	168	1·95

The following table gives the useful effect of men and animals employed in the horizontal transport of burdens. The second and third columns give the useful effect, viz. the product of the weight in lbs. and the distance in feet. This must not be mistaken for the units of work done by the agent, the agent being employed *not* in raising the weight, but in overcoming the passive resistances, friction, etc., which depend on the weight indeed, but are only a fraction of it.

USEFUL EFFECT OF AGENTS EMPLOYED IN THE HORIZONTAL TRANSPORT OF BURDENS.

Agent.	Daily Duration of Work in Hours.	Useful Effect Daily.	Useful Effect per Minute.	Weight transported.*	Velocity.	
					Feet per Min.	Miles per Hr.
Man walking on a horizontal road without burden, i.e. transporting his own weight . .	10·0	25,398,000	42,330	145	292	3·32
Labourer transporting materials in a truck on two wheels, returning with it empty for a new load	10·0	13,025,000	21,710	220	99	1·12
Do. in a wheelbarrow . .	10·0	7,815,000	13,030	130	160	1·14
Labourer walking with a weight on his back	7·0	5,470,000	13,030	90	145	1·64
Labourer transporting materials on his back and returning unburdened for a new load .	6·0	5,087,000	14,110	145	97	1·10
Do. on a handbarrow . .	10·0	4,298,000	7,160	110	65	0·74
Horse transporting materials in a cart, walking, always laden .	10·0	200,582,000	334,300	1,500	223	2·53
Do. trotting	4·5	90,262,000	334,300	750	44	5·06
Do. transporting materials in a cart returning with the cart empty for a new load . .	10·0	109,408,000	182,350	1,500	121	1·38
Horse walking with a weight on his back	10·0	34,385,000	57,310	270	212	2·41
Do. trotting	7·0	32,092,000	76,410	180	424	4·82

* Exclusive of the weight of the barrow, truck, cart, etc.

In advocating the employment of mechanical tractors on the farm, *Modern Farming* states that the horse eats annually the product of five acres. According to Edison, the efficiency of the horse is about 2 per cent.

NOTES ON MAN POWER.

(*R. T. Hancock.*)

The amount of foot-lbs. of energy which a man engaged on heavy labour can exert during a working day is entirely bound up with fatigue and opportunity for recovery from fatigue, and is a question of physiology rather than of dynamics. The number of hours constituting a working day and the opportunities given during the working day for at least partial replacement of the waste of the muscle tissues are the governing factors. All forms of human energy, physical or mental, are similarly conditioned. All unnecessary motions or distractions produce fatigue just as much as if they formed part of the useful work done, and correspondingly reduce the useful work possible. The basis of the subject is fully treated by Taylor in his 'Principles of Scientific Management.' Instances are given of the great increase in work possible by careful attention to length of working day, provision of short periods for rest, fixing the actual percentage of the working day during which the labourer can remain under load, the optimum load, and the planning of the work to enable it to be done with the minimum of motions. The performance of labourers loading pig-iron from the ground into railroad

cars an average distance of 36 ft. away was raised from 12½ to 47½ tons per man per day; of labourers shovelling iron ore from cars, from 16 to 59 tons per day; and the number of bricks laid by a bricklayer, from 120 to 350 per hour.

The average labourer is not only incapable of planning his work so as to effect it with the least waste of energy, but many of the conditions affecting its amount are not under his control, but that of the management. The intelligence of the labourer of course plays no small part, but it is the function of scientific management, according to Taylor, to train each individual to follow the routine which the management has evolved as tending to maximum production for a given expenditure of energy. The expenditure of energy by the labourer, or the amount of fatigue involved, being much the same as where no system is followed, the increased production benefits chiefly the employer who has made this state of things possible, and the extra pay which the labourer may receive bears only a small proportion to the extra useful work he accomplishes. Even in highly skilled and specialised trades the improvement in output following on the adoption of scientific systems of work-planning is very great, but its principal field would appear to lie in the improvement of the performance of the lowest and most unintelligent class of labour. As instancing the differences which already exist due solely to the class of labour employed, Hoover ('Principles of Mining') considers that in the simplest forms of mining work, such as shovelling or tramming, one white labourer equals two to three members of the coloured races, and in the more highly skilled branches such as mechanics, the proportion rises to 1 to 7, or even 1 to 11. As under the principles laid down by Taylor the labourer eventually ceases to use any initiative of his own but becomes practically an automaton, it would appear possible to train the lowest type of labour so that they could eventually replace on an equal footing men whose higher attributes fit them for a better class of work.

COSTS AND CAPACITIES OF VARIOUS METHODS OF TRANSPORT.

Transport Unit.	Load.	Total Daily Mileage Capacity.	Total Daily Mileage Laden.	Capacity in Ton-Miles a Day.	Charges per Ton-Mile.	Gross Daily Takings.
Porters . .	60 lb.	15	Assumed always laden	0·4	5*s.* to 1*s.* 6*d.*	2*s.* to 7*d.*
Pack animals .	200 to 400 lb.	24	12 to 18	1 to 3	5*s.* to 2*s.*	5*s.* to 6*s.*
Animal-drawn vehicles	1 to 5 tons	16	8 to 12	8 to 60	1*s.* 6*d.* to 9*d.*	12*s.* to 2*l.* 5*s.*
Mechanical pneumatic-tyred wheeled vehicles	½ to 1 ton	80	40 to 60	20 to 60	2*s.* 6*d.* to 1*s.* 6*d.*	2*l.* 10*s* to 4*l.* 10*s.*
Roadless vehicles	1 to 2 tons	70	35 to 50	35 to 100	1*s.* 6*d.* to 1*s.*	2*l.* 12*s.* 6*d.* to 5*l.*
Roadless vehicles	2 to 5 tons	60	30 to 45	60 to 225	1*s.* to 8*d.*	3*l.* to 7*l.* 10*s.*
Roadless vehicles	5 to 20 tons	60 to 40	30	150 to 600	8*d.* to 4*d.*	5*l.* to 10*l.*
Roadless vehicles	20 to 200 tons	40 to 32	20 to 24	400 to 4,800	4*d.* to 2*d.*	6*l.* 13*s.* 4*d.* to 40*l.*
Arterial railway goods trains	200 to 500 tons	150 to 300	75 to 225	15,000 to 112,500	2*d.* to 1*d.*	125*l.* to 468*l.* 15*s.*

NOTE.—The column of total daily mileage is based on no-return loads as a minimum up to half-return loads as a maximum.

(Philip Johnson; paper read at a Joint Meeting of Section G (Engineering) and Section F (Economics) at the Southampton Meeting of the British Association, August 31, 1925.

Power Required for Ploughing.

$$\text{Horse-power} = \frac{rwdv}{375};$$

where,

r = soil resistance in lbs. per sq. in.; w = total width in ins. of all furrows ploughed at once; d = depth of furrows in ins.; v = speed of travel in miles per hour.

The quantity rwd is the draught of the plough, and at steady speeds it equals the pull exerted at the draw-bar of the tractor of the haulage rope.

The soil resistance r varies according to the type and condition of the soil. It is measured

in lbs. per sq. in. of furrow cross-section, and may vary as much as 40 per cent. in one field.

Values of r are as follows :—

Type of Soil.	Sandy.	Sandy Loam.	Light.	Dry Clay Loam.	Heath Land.	Heavy Clay Sod.	Heavy Loam.	Firm Clay.
r	3	3–6	5–7	7–8	8·8	10–11	12	16

The ' angle of hitch ' is most important. When the hitch is not correct vertically, either the share digs into the ground too deeply or does not penetrate at all.

A re-pointed and re-sharpened share will show a less draught than a dull one, while a new one may reduce the draught by 36 per cent. over the re-sharpened one.

Experiments carried out at Rothamsted indicated that the speed of ploughing could be considerably increased without proportionately increasing the draught. In fact, an increase of speed of from 1 to $2\frac{1}{2}$ m.p.h., or 150 per cent., increased the draught by 14·5 per cent.

It was found that the draught in heavy clay could be reduced some 15 per cent. by chalking the soil.

Cost per acre of steam ploughing, 15*s.* to 20*s.*; horse ploughing, 20*s.* to 35*s.*; electric ploughing, 5*s.* to 6*s.* See also ' Electric Ploughing,' by Borlase Matthews, *Journal of the Institution of Electrical Engineers*, No. 383, Nov. 1928.

(*R. Borlase Matthews.*)

See also ' Electricity in Agriculture ' Section XXXV.

SECTION III

PART I

SURVEYING—SURVEYING INSTRUMENTS—ASTRONOMY—ASTRONOMY IN FIELDWORK (pp. 51–86)

(With contributions by R. T. Hancock, Assoc.Inst.M.M., Reginald Ryves, Assoc.M.Inst.C.E., Fred. Simpson, A.M.Inst.C.E., and others.)

PART II

DRAWING (pp. 87–96)

H. A. Miles

SECTION III

PART I

SURVEYING.

Chaining.

Chains.—The surveying chain is 22 yards = 66 feet long. It consists of 100 links, and each link = 7·92 inches. This chain is known as 'Gunter's' chain.

Another chain is known as the '100-foot' chain, and consists of 100 links, each 1 foot long. It is used almost exclusively for work done abroad.

A third form is the 'Decimetre' chain; this is divided into 100 links, each of which measures a decimetre = ·328 foot.

Chain Measurement.—The most careful measurement that can be made with the common surveying chain is liable to an error of $\frac{1}{3000}$ of the distance measured, and an ordinary measurement across fields, &c., to an error of $\frac{1}{500}$.

Chain measurement alone will not suffice for a greater extent than a square mile on a scale of 6 inches to a mile; or than $\frac{1}{16}$ square mile on a scale of 20 inches to a mile.

Arrows.—With each chain 10 arrows or pins of strong wire about $1\frac{1}{2}$ feet long, and preferably having a piece of red cloth tied to the ring of each, are carried.

Ranging.

Poles and Ranging Rods.—The former are poles of red pine about 15 feet high, 2 inches thick at the bottom and $\frac{3}{4}$ inch at the top; each should be shod with an iron shoe. The ranging rods are about 6 feet high, and have a white and red flag about 9 inches square at the top.

Offset Staff.—A narrow slip of deal about $1\frac{1}{2}$ inches wide, 1 inch thick, and 10 links long, divided into links. It is used for the purpose of measuring small distances from the line to hedges, &c.

The statute acre in England consists of 10 square chains, that is, of 10 square blocks of land whose length and breadth are 1 chain each; or, taking 10 blocks as one whole block, then an acre is equal to the area of a rectangle whose base is 10 chains and perpendicular 1 chain; or to any other rectangle, the product of whose base and perpendicular also equals 10 chains: thus, 4 chains base with $2\frac{1}{2}$ chains perpendicular, and 2 chains base with 5 chains perpendicular, are each equal to 1 acre.

With any quantity less than 1 chain, to bring feet into links add $\frac{1}{2}$ more; to bring links into feet take $\frac{1}{3}$ less.

MENSURATION.

To find the Area of a Triangle when the Base and Perpendicular Height are given.—Multiply the base by half the height, or *vice versâ*.

To find the Area of a Triangle when the Three Sides are given.—Take half the sum of the sides, subtract each side severally from this sum, then multiply this and the three remainders together, and take the square root for the area.

To find the Area of a Rectangular Figure.—Multiply the length by the breadth; the product will be the area.

To find the Area of a Trapezoid.—Multiply half the sum of the two parallel sides by the distance between them.

To find the Area of a Parallelogram whose Angles are not Right Angles.—Multiply the length of any one of the sides by the perpendicular let fall on it from the angle opposite that side.

To find the Area of a Trapezium.—Divide the trapezium into two triangles, and find the areas of the latter by the first rule.

To find the Area of an Irregular Polygon.—Divide the polygon into trapeziums and triangles, and find the sum of the areas of each.

To find the Area of an Irregular Figure.—Draw the figure on fine cardboard or thin sheet-metal, cut the same carefully out, and weigh it with an accurate balance. Then this weight, compared with the weight of a piece of the cardboard or metal of a definite size—say 1 inch square—gives at once the area required.

Calculation of Areas from Co-Ordinates.

If necessary adjust the traverse so that the co-ordinates correctly close the figure.

Consider each line round the figure, not the stations.

Take the total, or reduced, departures of the stations at each end of one line, add them algebraically, and divide by two. This gives the 'middle departure' of that line.

Repeat for all the lines of the figure.

It makes no difference from what initial point the departures are reckoned, whether from one of the stations on the figure, or from another point either inside or outside the figure.

Taking each line round the figure in order—it makes no difference which way round—take the difference between the reduced latitudes of the stations at each end of the line. The result is the same as the actual amount of Northing or Southing made by that particular line.

Multiply these results for each line by the middle departure of the same line, paying attention to the algebraic sign of each factor, and add all the results together algebraically.

The result is that the area of the figure in squares of the units employed in measuring the length of the sides.

If these are feet, then

divide by 27,878,400 (Log 7·445 2678) for sq. miles: divide by 43,560 (Log 4·639 0878) for acres.

In the logarithmic computation of middle departures by latitudes, it is convenient to use the arithmetical complements of the logs given in preceding paragraph, adding all three logs, and thus avoidlng taking out differences.

The terms latitude and departure are mutally interchangeable throughout the above.

TRIANGULATION.

For large surveys a base line is accurately measured and triangles built upon it, the sides being for 6 inches scale from $\frac{3}{4}$ to $1\frac{1}{2}$ mile. Each point should, if possible, be fixed by three distances.

Example.—AB (fig. 1) is the base line; C, D, E, F are convenient points to be fixed by triangulation.

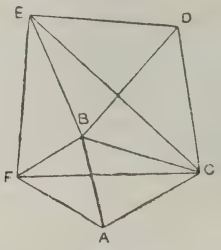


Fig. 1.

AC, AF, BF and BC being obtained, two values of FC can then be found from FBC and FAC, and then two values of FE from FEB and FEC, and so on. In plotting the triangulation the long lines EF, FC, EC should be first laid down.

AB = 5,230 feet measured by chain.
BAC = 55° 30′ 15″
CBA = 64° 21′ 25″ } measured by instrument.
BCA = 60° 8′ 20″

$$BC = \frac{AB \sin BAC}{\sin BCA}.$$

Log AB = 3·7185017
Log sin BAC = 9·9130154
13·6345171
Log sin BCA = 9·9381368
Log BC = 3·6963803
BC = 4970

The triangles are then broken up into smaller ones and measured by the chain.

Simson's Rule for Area Computation.

FIG. 2.

Divide base into any even number of equal parts, and measure length of perpendiculars or ordinates through each division.

If S = sum of lengths of first (f) and last (l) ordinates; E = sum of lengths of intermediate even ordinates; N = sum of lengths of intermediate odd ordinates; d = common distance between ordinates; then

$$\text{Area} = \frac{S + 4E + 2N}{3} \times d.$$

Rate of Surveying.

The average daily progress of a good surveyor in England, surveying with one chainman, for the 6-inch-to-a-mile scale, is, according to Frome—

Close large village	about	5 acres.
Villages and surrounding fields, etc.	„	14 acres.
Close country, gentlemen's houses, etc.	„	20 acres.
Medium country, ordinary fields, and scattered farms	„	30 to 32 acres.
Open moorland, with roads, streams, boundaries, cart tracks, etc. (no fields)	„	55 acres.

A survey of a piece of country of about 6 square miles, on a scale of 6 inches to a mile (which would comprise measuring and levelling base line, triangulating, finding variation of the compass, traversing the principal roads, filling in detail by prismatic compass, contouring to every 25 feet, and completing fair plan), would take a good surveyor thirty working days.

A greater area takes less time in proportion, for the base is the same, as also variation of compass and part of the triangulation.

TO ASCERTAIN THE DISTANCE OF AN INACCESSIBLE OBJECT.

To measure a Base Line across a River.

Let DA (fig. 3) be the direction of the line which has been measured up to the river. It is required to ascertain the distance BA at once upon the ground, so as to continue the measurement of the line.

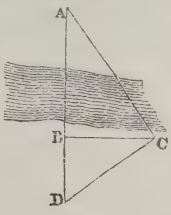


FIG. 3.

On the line DA take any point, B, whence a perpendicular, BC, can be taken, which will be free from obstruction, so that BC can be accurately measured; carry the range on across the river, and at A set up a flag; on BC take any point, C, whence A can be seen; and at C erect a perpendicular to AC, intersecting the line AD on D.

Measure DB; then

$$AB = \frac{BC^2}{BD}.$$

These angles can either be taken by a cross-staff or by the chain, with the distances of 30 links, 40 links, and 50 links; 50 being the hypotenuse of a right-angled triangle when the base and perpendicular are respectively 30 and 40.

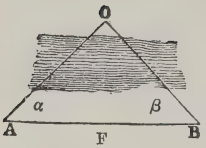


FIG. 4.

To measure the Width of a River with the Base Line alongside of it.

Take at the ends of the base line, AB (fig. 4), with a theodolite, the angles α and β, made between the base line and a flag, O, placed at the edge, on the other side of the river; then width OF is

$$OF = AB \frac{\tan \alpha \tan \beta}{\tan \alpha + \tan \beta}.$$

To find the Distance of one Object from another, where a River divides them, without using the Theodolite.

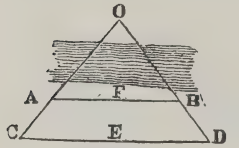


FIG. 5.

Let O and A (fig. 5) be the given objects, and AO the distance required. From A draw AB at *any* angle to AO, and produce OA to C, measuring AC about one-third length of AB; from C measure CD, parallel to AB, OBD being in one straight line. To make CD parallel to AB, at A and B erect equal perpendiculars, which can be done either by a cross-staff or by means of the unit proportion of the sides of a right-angled triangle, viz. 3, 4, and 5, or any multiples of them whatever. Then

$$AO = \frac{AB \cdot CA}{CD - AB}.$$

If the perpendicular distance OF be required, then this can be obtained by the formula—

$$OF = FE \frac{OA}{AO}.$$

Or we can so select the point A as to have OAC at right angles to the river, and make AB and CD at right angles to OAC, when AO, given by the foregoing formula, gives, of course, the required perpendicular distance.

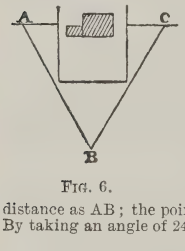


FIG. 6.

To continue the Measurement and Direction of a given Line when any Obstacle stands in the way, which cannot be crossed but can be avoided by going to the right or left.

At any point, A (fig. 6), on the given line, AC, take an angle with the given line, of 120° if you would turn to the left, or 240° if to the right, as in the present case, and proceed measuring to B till an angle of 60° made with this line towards the first line, AC, will carry you clear of the obstacle. Take this angle, ABC, 60°, and measure BC the same distance as AB; the point C will then be on the given line, and AC will be equal to AB or BC. By taking an angle of 240° with the line BC, the range of the line can be continued.

General Remarks.

1. Avoid taking bearings and distances along a circuitous boundary line like *abc* (fig. 7), but run the straight line *ac*, and at right angles to it measure offsets to the crooked line. 2. If it is desired to survey a straight line from *a* to *c*, but the instrument cannot be directed precisely towards *c*, on account of intervening obstacles, first run a trial line, *am*, as nearly in the proper direction as can be guessed. Measure *mc*, and calculate a distance, *y*, say, so that *am* is to *mc* as is 100 ft. to *y*. Lay off *ao* equal to 100 ft., and *os* equal to *y*, and run the final line *asc*. Or, if *mc* is quite small, calculate offsets like *os* for every 100 ft. along *am*. 3. When *c* is visible from *a*, but the intervening ground difficult to measure along on account of marshes, &c., extend the side *ya* to good ground at *t*; then, making the angle *ytd* equal to *yac*, run the line *tn* to that point, *d*, at which the angle *ndc* is found by trial to be equal to the angle *atd*. It will rarely be necessary to make more than one trial for this point *d*.

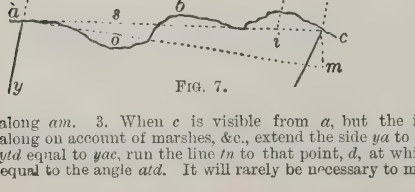


FIG. 7.

for, suppose it to be made at x, see where it strikes ac at i; measure ic, and continue from x, making $xd = ic$. 4. In the case of a very irregular piece of land, or a lake (fig. 8), surround it by straight lines. Survey these, and at right angles to them measure offsets to the crooked boundary. 5. When between two objects, m and n (fig. 9), and it is wished to place oneself in range with them, lay a straight rod, cb, on the ground, and point it to one of the objects, m; then go to the end c, and see if it points to the other object; if it does not, then, by successive trials, find the position, ed, in which the rod points to both objects, and consequently is in range with them. If no rod is at hand, two stones will answer, or two chain-pins. A plumb-line (a pebble tied to a piece of thread) will add to the accuracy of ranging the rod, or stones, &c.

FIG. 8.

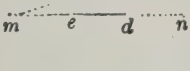


FIG. 9.

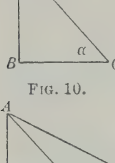


FIG. 10.

TO MEASURE THE HEIGHT OF AN OBJECT.

(1) When the Base is Accessible.

Measure any distance from the base of the object as nearly equal as possible to the height, and take the angle of elevation by the theodolite.

Let BA (fig. 10) be the object; measure BC, and take the angle a; then

$$BA = BC \tan a.$$

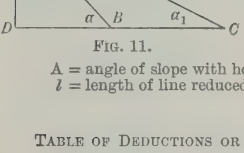


FIG. 11.

(2) When the Object is Inaccessible.

If the object be inaccessible, so that DB *cannot be measured,* range DB (fig. 11) on to C, and taking BC nearly equal to DB, measure BC, and at B and C take the angles of elevation, α and α_1; then

$$AD = BC \tan \alpha_1 \left(\frac{\tan \alpha}{\tan \alpha - \tan \alpha_1} \right).$$

Chaining on Slopes.

A = angle of slope with horizon. L = length of line chained on slope.

l = length of line reduced to the horizontal.

$$l = L \cos A.$$

TABLE OF DEDUCTIONS OR ADDITIONS TO BE MADE PER 100 FEET IN CHAINING OVER SLOPING GROUND

In order to reduce the inclined measurements to horizontal ones.

Slope in Deg.	Deduct Feet.	Rise in 100 ft. Hor.	Slope in Deg.	Deduct Feet.	Rise in 100 ft. Hor.	Slope in Deg.	Deduct Feet.	Rise in 100 ft. Hor.	Slope in Deg.	Deduct Feet.	Rise in 100 ft. Hor.
¼	·001	·436	5½	·420	9·189	10½	1·596	18·08	15½	3·521	27·26
½	·004	·873	⅝	·460	9·629	⅝	1·675	18·53	⅝	3·637	27·73
¾	·009	1·309	¾	·503	10·07	¾	1·755	18·99	¾	3·754	28·20
1	·015	1·746	6	·548	10·51	11	1·837	19·44	16	3·874	28·67
¼	·024	2·182	¼	·594	10·95	¼	1·921	19·89	¼	3·995	29·15
½	·034	2·619	½	·643	11·39	½	2·008	20·35	½	4·118	29·62
¾	·047	3·055	⅝	·693	11·84	¾	2·095	20·80	¾	4·243	30·10
2	·061	3·492	7	·745	12·28	12	2·185	21·26	17	4·370	30·57
¼	·077	3·929	¼	·800	12·72	¼	2·277	21·71	¼	4·498	31·05
½	·095	4·366	½	·856	13·17	½	2·370	22·17	½	4·628	31·53
¾	·115	4·803	¾	·913	13·61	¾	2·466	22·63	¾	4·760	32·01
3	·137	5·241	8	·973	14·05	13	2·563	23·09	18	4·894	32·49
¼	·161	5·678	¼	1·035	14·50	¼	2·662	23·55	¼	5·030	32·98
½	·187	6·116	½	1·098	14·95	½	2·763	24·01	½	5·168	33·46
¾	·214	6·554	⅝	1·164	15·39	¾	2·866	24·47	¾	5·307	33·95
4	·244	6·993	9	1·231	15·84	14	2·970	24·93	19	5·448	34·43
¼	·275	7·431	¼	1·300	16·29	¼	3·077	25·40	¼	5·591	34·92
½	·308	7·870	½	1·371	16·73	½	3·185	25·86	½	5·736	35·41
¾	·343	8·309	⅝	1·444	17·18	¾	3·295	26·33	¾	5·882	35·90
5	·381	8·749	10	1·519	17·63	15	3·407	26·79	20	6·031	36·40

The foregoing table gives deductions or additions to be made every 100 ft. as actually chained along sloping ground in order to reduce the sloping measurements to horizontal ones. Even when it is so nearly level that the eye cannot detect the slope, an over-measurement of an inch or two in 100 ft. may readily occur. It is plain that if we measure all the undulations of the ground we shall get greater totals than if we measure horizontally, as is *supposed* always to be done; but since few surveyors pretend to measure horizontally until the slope becomes apparent to the eye, their lines are usually too long by from 1 to 2 inches in 100 ft. To counteract this to some extent chains are frequently made from 1 to 2 inches longer than 100 ft.; and for ordinary purposes the precaution is a good one. When greater accuracy is required the chainmen should be attended by a third person, with a rod and slope-level, for taking the inclinations of the ground. These deductions being made, the remainder will be the actual horizontal distance.

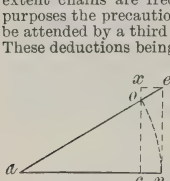


FIG. 12.

For example, in fig. 12 each 100 ft. *ao* measured up or down the slope *ae* plainly corresponds to the shorter horizontal distance, *ac*, the difference or deduction being *cn*. Taking *ao* as radius, then *ac* is the cosine, and *cn* the versed sine of the angle *ean* of the slope. Therefore *ao* multiplied by the nat. cosine of the angle *ean* gives the reduced horizontal distance, *ac*, which, taken from *ao*, gives the deduction *cn* of our table.

But if, while chaining along the slope *ae*, we wish to drive stakes that shall correspond with horizontal distances, *an*, of 100 ft., it is evident that we must add *cn* to each 100 ft. *ao*, as shown at *xe*, and the stake must be driven at *e* instead of at *o*. Observe that $xe = cn$ must be measured horizontally.

When the ground is very sloping all this calculation may be avoided, where great accuracy is most required, by actually holding the chain horizontal, as nearly as can be judged by eye, and finding, by means of a plumb-line, where its raised end would strike the ground. A whole chain at a time cannot be measured in this way, but shorter distances must be taken, as the ground requires—at times, on very steep ground, not more than 5 or 10 ft.

The following short rules may be used for finding the approximate correction to be made :—

(1.)

Square the number of degrees in the slope, multiply by $1\frac{1}{2}$, and obtain correction in hundredths of a foot.

Example.—10° slope. $10^2 \times 1\frac{1}{2} = 150$, or 1·50 ft. correction.

(2.)

Square the rise in feet per 100 ft. horizontal, and divide by 2; this gives correction in hundredths of a foot.

Example.—10 ft. rise in 100. $10^2 \div 2 = 50$, or 0·50 ft. correction.

SCALES FOR SURVEYS.

Ordnance Survey Scales.

VARIOUS SCALES OF MAPS OF GREAT BRITAIN AND IRELAND.

Natural Scales.	Inches to one Statute Mile.	Statute Miles to one Inch.	Chains to one Inch.	REMARKS.
$\frac{1}{500}$	126·720	0·0079	0·63$\dot{1}$	Town maps.
$\frac{1}{528}$	120·000	0·008$\dot{3}$	0·6$\dot{6}$	
$\frac{1}{1056}$	60·000	0·01$\dot{6}$	1·$\dot{3}$	
$\frac{1}{2500}$	25·344	0·0395	3·15$\dot{6}$	Cadastral, or parish, maps of the United Kingdom.
$\frac{1}{2640}$	24·000	0·041$\dot{6}$	3·$\dot{3}$	Map of Alnwick.
$\frac{1}{5280}$	12·000	0·08$\dot{3}$	6·$\dot{6}$	Special maps of districts around London and Plymouth.
$\frac{1}{10560}$	6·000	0·1$\dot{6}$	13·$\dot{3}$	County maps of United Kingdom, and special maps.
$\frac{1}{15840}$	4·000	0·25	20·0	Special index.
$\frac{1}{63360}$	1·000	1·0	80·0	General map of United Kingdom, and special maps.

Small-Scale Maps of Great Britain and Ireland.

Fifteen miles to one inch: British Isles.

Ten miles to one inch: Great Britain: (*a*) in outline, with water coloured blue; (*b*) hills shaded, water blue, roads sienna. Ireland: (*a*) outline engraved; (*b*) hills shaded, water blue, roads sienna.

Four miles to one inch: England: (*a*) engraved in outline; (*b*) conspicuously coloured. Also county maps with main roads coloured. Ireland and Scotland: (*a*) outline, water blue; (*b*) coloured. County maps of Scotland.

Two miles to one inch: England and Scotland; coloured.

One mile to one inch: England: (*a*) in outline with *contours*; (*b*) hills shaded in brown; (*c*) hills shaded in black; (*d*) coloured, with *contours* in red. Ireland: (*a*) in outline; latest editions with *Contours*; (*b*) hills shaded in black (unrevised); (*c*) coloured; (*d*) in outline, with *boundaries* of counties and urban and rural districts in red. Scotland: (*a*) in outline, with contours; (*b*) hills shaded in brown, zincographed; (*c*) hills shaded in black (unrevised); (*d*) coloured.

Maps of the India Survey Department.

One mile to one inch: Topographical maps; about one-eighth of the total number has been published (1914).

Four miles to one inch: Prepared in sheets of one degree by one degree. Publication only recently begun.

Thirty-two miles to one inch: The outline engraving of this new map of India and adjacent countries is completed. The hills are shown in the 'layer' system. (*a*) A new edition of the railway, canal, and road map of India. (*b*) A new map of India and adjacent countries, hills shown on the 'layer' system. (*c*) A new edition of the map showing railway stations.

Dip of Horizon.

d = dip of horizon, in seconds. h = height of observer's eye, in feet.

s = distance of horizon, in statute miles. n = distance of horizon, in nautical miles.

$d = 57{\cdot}4\sqrt{h}$; approximate, varying with temperature. $h = {\cdot}666s^2 = {\cdot}86n^2$.

Table of Dip and Distance of Horizon at Various Heights.

h.	s.	n.	d.	h.	s.	n.	d.
Feet.	Miles.	Miles.	′ ″	Feet.	Miles.	Miles.	′ ″
5	2·739	2·411	2 8·35	80	10·959	9·644	8 33·4
10	3·874	3·409	3 1·51	90	11·624	10·229	9 4·54
15	4·745	4·176	3 42·31	100	12·253	10·783	9 34
20	5·480	4·822	4 16·7	150	15·007	13·206	11 43
25	6·126	5·391	4 47	200	17·329	15·249	13 31·76
30	6·711	5·906	5 14·39	300	21·223	18·676	16 34·2
35	7·249	6·379	5 39·58	400	24·507	21·566	19 8
40	7·749	6·819	6 3·03	500	27·399	24·111	21 23·5
45	8·219	7·233	6 25·05	1,000	38·749	34·099	30 15·1
50	8·664	7·624	6 45·88	2,000	54·799	48·223	42 47
60	9·491	8·352	7 24·62	3,000	67·115	59·061	51 23·9
70	10·252	9·021	8 0·24	4,000	77·498	68·198	60 30·3

To reduce a Measured Base to the Level of the Sea.

Let r represent the radius of the earth corresponding to the base b at the level of the sea, and $r + a$ the radius referred to the level of the measured base, B; then

$$r + a : r :: B : b, \quad \text{or,} \quad b = B \times \frac{r}{r+a}, \quad \text{and,} \quad B - b = B - B\frac{r}{r+a} = B\left(1 - \frac{r}{r+a}\right)$$

$$= B\left(\frac{a}{r+a}\right) = B\left(\frac{\frac{a}{r}}{1+\frac{a}{r}}\right) = \frac{Ba}{r}\left(\frac{1}{1+\frac{a}{r}}\right) = \frac{Ba}{r}\left(1 - \frac{a}{r} + \frac{a^2}{r^2} - \text{\&c.}\right) = B\left(\frac{a}{r} - \frac{a^2}{r^2} + \frac{a^3}{b^3} - \text{\&c.}\right).$$

But the radius of the earth being very great in comparison with the difference of level, a, we have the correction, $B - b$, or d, say, sufficiently accurate by retaining only the first term. Hence

$$d = \frac{Ba}{r} \text{ is the value to be deducted from the measured base.}$$

Spherical Excess.

$$\text{E, in seconds} = \frac{S}{r^2 \sin 1''} = \frac{ab \sin C}{2r^2 \sin 1''}$$

S = the area of the triangle. r = the radius of the earth.

$$S = \frac{ab \sin C}{2} = \sqrt{s(s-a)(s-b)(s-c)}, \quad s \text{ being} = \frac{a+b+c}{2}.$$

Between latitudes 45° and 25° the spherical excess amounts to about 1′ for an area of 75·5 square miles.

Hence, if the area in square miles be known, a close approximation to the spherical excess will be had by dividing the area by 75·5. Log mean radius of the earth in yards = 6·8427917. If the three angles of a triangle are assumed to have been equally well determined, the previous determination of the spherical excess is not necessary for the calculation of the sides, though it will be required for estimating the relative value of the observations. For the sides of a spherical triangle may be computed as if they were rectilineal if one-third the excess of the sum of the three angles above 180° is deducted from each of the three observed angles, in which case

$$\text{side } b = \text{side } a \sin (B - \tfrac{1}{3}E) \div \sin (A - \tfrac{1}{3}E).$$

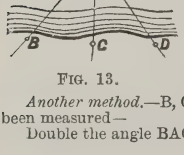


FIG. 13.

Surveying Soundings.

To determine the position of any sounding, A (fig. 13), when afloat, take simultaneously the bearings of three known objects on shore, B, C, D, plot the angles on tracing-paper, and move them on plan until the lines cut the points B, C, D.

Another method.—B, C, and D being known points on shore, and the angles BAC, CAD having been measured—

Double the angle BAC, and deduct this doubled angle from 180°. Halve the remainder, and set off from B and C angles, each equal to the half remainder—that is, BCE, CBE; then BE = EC. With E as a centre and radius EC, describe the circle BCA. Then the point A will be in the circumference of this circle.

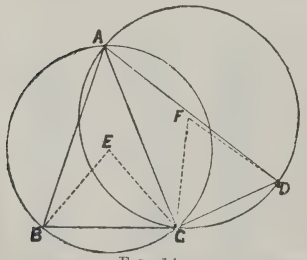


FIG. 14.

In like manner, double the angle CAD and proceed as before; the point A will be in the circumference of the circle CAD. Being in the circumference of the two circles, the point A will therefore be at the intersection of the two.

Suppose the angle BAC to have been 40°, CAD 30°, the double of BAC is 80°; then 180°−80°=100°, which divided by 2 gives 50°. Set off from B and C angles equal to 50°. In like manner, 2CAD=60°, and 180°−60°=120°; half 120°=60°; from C and D set off 60°. With E and F as centres (fig. 14), and radii EB, FC, describe the two circles, whose intersection determines the position, A, of the observer.

To reduce a Sounding to Low Water.

h = vertical rise of tide, in feet. h' = height of tide above low water, in feet, at the time t.
t = time between high and low water. t' = time from time of sounding to low water, in hours.

$$\frac{h}{2}\left(1 \mp \cos\frac{180t'}{t}\right) = h'; \quad - \cos \text{ when } \frac{180t'}{t} < 90°, \text{ and } + \cos \text{ when } > 90°.$$

EXAMPLE.—Low water occurring at 3.45, and high water at 10.15 P.M., a sounding taken at 5.30 P.M. was 18·25 feet; what was the depth at low water, the vertical rise being 10 feet?

$$h = 10 \textit{ feet}; \; t = 5h.\ 30m. - 3h.\ 45m. = 1h.\ 45m. = 1{\cdot}75 \textit{ hours}.$$
$$t = 10h.\ 15m. - 3h.\ 45m. = 6h.\ 30m. = 6{\cdot}5 \textit{ hours}.$$

Then

$$\frac{10}{2}\left(1 \mp \cos\frac{180 \times 1{\cdot}75}{6{\cdot}5}\right) = 5(1 - 48°\ 27') = 5 \times (1 - {\cdot}662) = 1{\cdot}69 \textit{ feet}.$$

Sounding	18·25 *feet*.
Reduction	1·69 ,,
	16·56 ,,

SURVEYING INSTRUMENTS.

(W. F. Stanley.)

The Plain Theodolite.

In this instrument (fig. 15) there is a horizontal circle reading to minutes by two verniers, and a vertical half circle reading also to minutes by a single vernier—both readings being provided with clamps to fix the telescope nearly to the reading required, and tangent screws to make the final adjustment. Below the horizontal divided limb of this theodolite a separate axis is formed, so that every reading may be set to start from zero. The telescope has two turned collars which rest in a cradle formed of Y-form supports. This form of cradle permits the telescope to be revolved upon its axis to set it in collimation with the object glass, by adjustment of the spider webs in front of the eye-piece until the revolution of the telescope does not appear to displace any small object observed. A magnetic needle of about three inches in length is placed in a compass box over the axis of the instrument. In old instruments this is very difficult to read, from interference of vision by the standards of the telescope. In modern instruments (fig. 15) an aluminium ring compass is provided, which is read from one side by a microscope. This theodolite is set up upon a tripod stand and adjusted to level, before commencing a survey, by four adjusting screws (parallel plate screws) which are used oppositely to tip the instrument one way or the other until the bubbles of the level tubes on the limb are brought to the centre of their run. As the pressure of these parallel plate screws must be more or less according to the force used, the axis is slightly distorted by them; dull bubble tubes have therefore to be used. Where greater exactness is required, the support, as of every scientific instrument, is placed upon three points only, an arrangement which is termed the tribrach system, and which admits of quick adjustment.

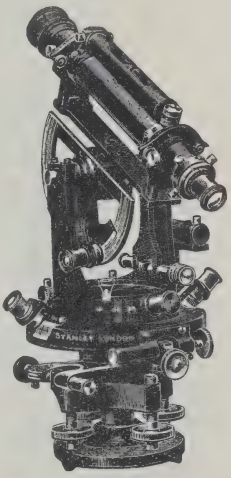


FIG. 15.

To set up a Tribrach.—Place the level linable with either pair of screws B.C. (fig. 16, p. 60), and bring the bubble to the centre of its run. Then place the level at right angles to this, so that the screw A comes opposite the centre of the telescope. Bring the bubble to the centre of its run. Now return to first position to see that this is right. If not exact, repeat the operation.

With tribrach support a modern well-ground bubble will read sensibly to five seconds of arc, or, if required in high-class work, even to a single second or less.

Adjustment of the Theodolite.

(R. T. Hancock.)

The first and most important adjustment consists in correctly levelling the instrument with its vertical axis pointing truly to the zenith, every time it is set up. Other errors in adjustment can be corrected or compensated by special methods of observation, but correct results are impossible if the theodolite is out of level. This adjustment must precede all others.

Set the instrument up firmly with its legs well spread and bring it approximately level by means of the base adjusting screws, with the aid of the bubble tubes on the vernier plate. Place the telescope as nearly horizontal as possible, with the vertical arc clamped to zero if desired, and bring it over one pair of the base levelling screws, or parallel to a pair in the case of a tribrach base, and observe the position of one end of the bubble of the telescope level in relation to the gradations on the tube. Select and use one end throughout, say, the end of the bubble nearest the eyepiece. Revolve the upper part of the theodolite through 180°, bring the bubble back half way, if it has moved, by means of the base levelling screws. Check by turning the instrument back to its original position. If the bubble remains in its mean position, turn the upper part of the instrument through 90°, so that telescope lies over the other pair of levelling screws, or the remaining one of the tribrach, and bring the bubble to the same position by these. Finally check by turning the instrument slowly through 360°, and see that the bubble remains unmoved. If not, repeat adjustment from the beginning. The bubble may not finally occupy the centre of its tube : this is unimportant. What is important is that it should not move when the instrument is revolved about its vertical axis. If the bubbles of the levels on the vernier plate are not well centered when the instrument is thus levelled, it will be convenient to make them so now by means of the turret screws at the ends of the cases, so that a close approximation to true levels may be obtained when setting up on future occasions. It should make no difference in levelling up whether the two plates revolve together or whether the lower is clamped and the upper free. If it does, it shows that the cones require regrinding.

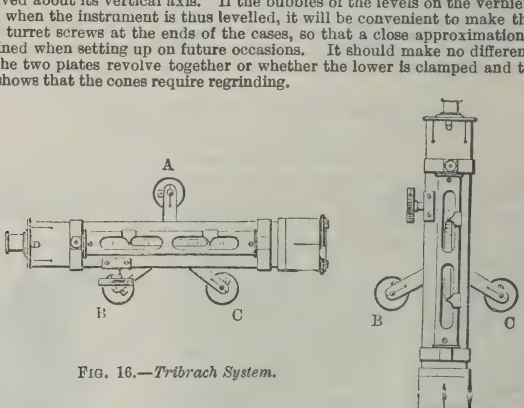


FIG. 16.—*Tribrach System.*

Adjustment for parallax consists in moving the eyepiece in or out, altering the focus of the object glass to correspond, until the crosshairs of the diaphragm do not appear to move relative to the point viewed when the eye is moved slightly from side to side of the lens opening. Care should be taken not to revolve the eyepiece tube in its casing, as this affects the optical centering of the system of lenses. For this reason the lens cell and its casing are usually provided with corresponding marks by the maker.

The transit axis upon which the telescope swings should be accurately at right angles with the vertical axis of the instrument. The adjustment of this is effected by sliding bearings at the top of the standards, but in cheaper instruments no provision for adjustment in this respect is made, the makers supposedly securing it once and for all by correctly proportioning the height of the standards. Direct the telescope to some high object, such as the finial of a spire. Depress the telescope and mark on the ground the point where the intersection of the hairs falls. Transit the telescope, sight up again, and mark where the intersection falls on the ground the second time. Raise or lower one bearing so that the intersection falls one-quarter of the distance between the marks from the second mark. Check, and repeat if necessary.

The collimation line may be defined as a line joining the optical centre of the object glass and the intersection of the crosshairs. To make it at right angles to the axis on which the telescope swings, set up, level, and sight to a well-defined mark about 100 feet away ; transit the telescope, and mark where the intersection falls in the opposite direction, at about the same distance away and on the same level as near as may be. Revolve the plates through 180° about, and sight to the original mark. Again transit, and mark the new point of intersection, if altered, near the former one. Measure a point one-fourth of the distance between the two from the latter, and move the diaphragm carrying the crosshairs by means of the adjusting screws holding it in the telescope until the intersection covers the measured mark. It will be necessary to loosen all four screws to do this, but care should be taken not to move the diaphragm up or down more than can be helped, and to keep the crosshairs vertical and horizontal. Check by transiting back again to the original mark, which will now not fall under the crosshairs ; bring it under the intersection by the slow motion tangent screw, and transit back again. The intersection should now fall half way between the two original marks.

Having performed the foregoing adjustments we now have the vertical axis of the instrument pointing truly to the zenith, the horizontal axis on which the telescope swings truly at right angles

to the vertical axis, and the line of collimation truly at right angles to the horizontal axis. The collimation line then lies wholly in a vertical plane cutting the horizontal axis at right angles, but does not necessarily lie parallel to the axis of the level on the telescope in that plane. It can be made to do so by altering the lie of one or the other, and in order to avoid disturbing the adjustment just completed, it is better to bring the axis of the telescope-level parallel to the collimation line as it now lies. This is why it was recommended not to disturb the up-and-down position of the collimation line more than can be helped, so as to avoid the necessity for much alteration in the telescope level. This adjustment is not really required for the theodolite, unless it is intended to use it occasionally as a level, and the surveyor who desires to effect it can proceed as described below for the level, except that after setting up beyond the levelled pegs, the telescope is lowered or raised by the slow-motion tangent screw of the vertical arc till the crosshairs read the same on both pegs; the bubble of the telescope level is then brought to the centre of its run by altering the capstan-headed screws at each end of its casing.

Everest's Theodolite.

This instrument was much used in the survey of India, and is a modification of the plain theodolite, in which the cradle fitting, somewhat liable to derangement from its light loose parts, is replaced by a telescope fixed in a solid collar. The segments of arc of this theodolite are used upon each side of the axis, so that double vernier reading may be employed to vertical angles. It was the first English-made theodolite in which the tribrach system of adjustment was employed, and is still considered to be an excellent instrument for survey upon fairly level country.

Lister's Inclinometer.

The construction (fig. 17) is similar to that of a plain theodolite, but with the addition of an extra vertical axis to the telescope, carried through the horizontal axis at right angles. By this means the telescope can be moved upon this supplementary vertical axis in a plane of any inclination desired, so that if the horizontal axis is set to the slope of a railway cutting a large number of pegs may be set out continuously with the same setting by direct observation through

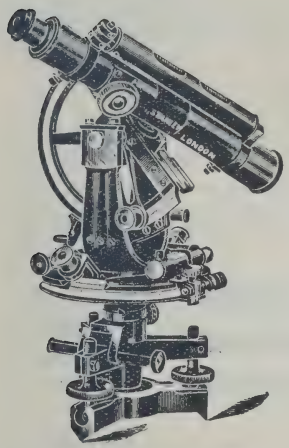


Fig. 17.

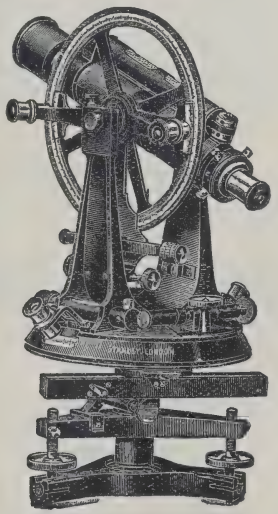


Fig. 18.

the telescope across any irregularity or inclination of the land surface. The slopes for curves also are readily pegged by means of the special horizontal angle repeater with which the instrument is fitted, so that one angle only need be set and that can be repeated round a curve until complete, when the angle repeater is simply released to return to a straight line.

The instrument is readily made into an ordinary surveying theodolite by fixing the telescope by means of a catch, which places it firmly true to the reading of the arc and limb.

Transit Theodolite.

In modern practice this instrument (fig. 18), in some form, is generally used. Its great merit is that the telescope being placed high enough above the limb to revolve over half a revolution on its axis, a direct fore-and-back sight can be taken in line with its centre, and, as this may also be done by moving the reading through 180°, it is a good test for the division and the true centering of the instrument. Except for the transitting telescope the general principle of structure of this theodolite is the same as that of the plain theodolite above described. The height necessary for the telescope is generally somewhat reduced by replacing the circular magnetic compass by a magnetic needle placed in a trough under the limb. The needle can move only 20° of arc, but, as it can be brought co-incident with the magnetic north by the lower adjustment of the axis of the theodolite, this arc is quite sufficient. It possesses the advantage of having a longer needle placed in clear daylight for observation. With the bad object-glasses formerly used the cradle adjustment in Y's was necessary for true collimation. In modern instruments the object glasses are truly centered so that the webs or other indices may be placed exactly in the axis of the telescope. On this point, in the old pieced-up instruments every severe shake or journey threw the instrument out of adjustment, and it was necessary that the experienced surveyor should have every facility for restoring this. In modern good work with refined apparatus for adjustment, a permanent index, or diaphragm, as it is termed, may be fixed to position by the maker, which will remain in perfect adjustment with careful usage the whole life of the surveyor. As this index matter is a subject not too well known, it will be discussed further on.

NOTES ON DIAPHRAGMS.

Spider's Webs (fig. 20)—the oldest form of telescope index—are clear and good, but very delicate, sagging with damp and breaking by tension if they remain so long in the instrument as to become very dry. Unless the user is specially skilled, the re-webbing and re-adjustment must be done by an optician.

Lines on Glass (fig. 21).—These are accurate and permanent. The presence of the glass diaphragm obstructs a little light at all times, but the great defect of this diaphragm is that it becomes bedewed by changes of temperature and requires cleaning. Where glass is fitted it should be easily removable with means of returning it exactly to its former adjustment.

Point Diaphragms (fig. 22) should be formed of very sharp fine needle points made of platinum-iridium alloy, which is as hard as finely-tempered steel and is non-corrosive.

This index gives the most exact reading and is permanent, so that an instrument with points will remain in adjustment for many years. In case of dust on the points this can be removed gently with a camel hair brush. The only objection made to this diaphragm is that it requires great care in bringing an object exact to reading, but certainly not greater care than is always desirable. Its precision may be demonstrated most simply. Upon first impression the

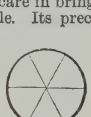


Fig. 20.

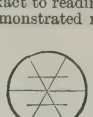


Fig. 21.

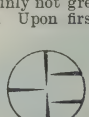


Fig. 22.

point may not appear so fine as a web. Practically it is more exact, as it does not cover the reading, as will be seen by the magnified appearance of the readings (fig. 23).

Both these readings are of ·4 upwards of the black band, but this quantity is less clear with the web W W than with the point P. This principle is most important with subtense measuring as will be shown presently.

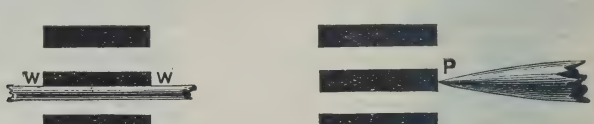


Fig. 23.

Some objections have been made by railway engineers to the large size of the transit theodolite, as they do not require the vertical circle except occasionally in mountainous districts. To meet this difficulty Messrs. W. F. Stanley & Co Ltd. have made a transit theodolite (fig. 24) as low and as portable as the plain theodolite. This transits over the eye end only, which is loaded by a heavy casting to balance the object end, which is made in aluminium for lightness.

Tunnelling Transit Theodolite.

This is a most useful form (fig. 25) for all underground work. It is built very low and the telescope transits. The vertical axis is made hollow, so that any angle to about 10° may be read

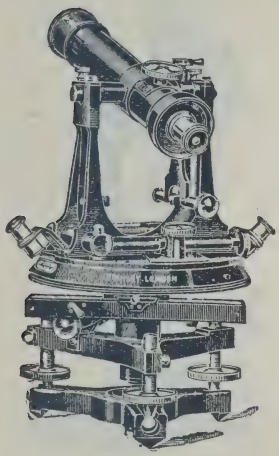


FIG. 24.

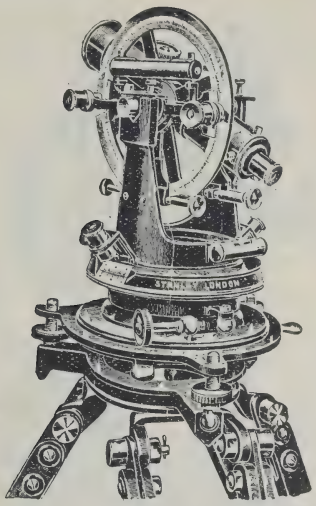


FIG. 25.

directly below the vertical. This will be found very useful in transferring lines underground from the surface, by sighting down a shaft, and also for plumbing. The hollow centre is supported upon a sliding fitting so that it may be displaced 1½ in. about the centre of the tripod and clamped to its position. It is a form of theodolite most convenient for the mining engineer, in which a long diagonal eye-piece permits a surface line to be observed from the bottom of a deep shaft.

DUNBAR SCOTT'S AUXILIARY TOP AND SIDE TELESCOPE.

Is fitted to theodolites for reading steeper horizontal angles than can be read with the ordinary telescope, also precipitous vertical angles. The zenith-nadir telescope is placed on top of telescope for the former and upon the outside of the standards to project beyond the limb for the latter. This telescope can also be removed so that the instrument then becomes an ordinary transit theodolite.

MICROMETRIC THEODOLITES.

In modern practice for superior triangulation, a vernier reading does not give the necessary accuracy, so the more exact system of reading by a microscope provided with a fine scale in the focus of the eye-piece is employed; over this scale an adjustable web is moved by a fine screw with divided head. With this appliance (fig. 26, page 64) much finer divisions may be made upon the instrument, and the micrometer scale will subdivide these divisions to read to any exactness desired. Micrometers were formerly employed upon large theodolites only. In modern practice they are used for as small as five-inch circles. The precision of the micrometric system may be shown in that a vernier on a six-inch theodolite can only be made to read satisfactorily 20 seconds of arc. The micrometer on the same instrument will read clearly five seconds. As a rule six-inch instruments are made to read to five seconds; five-inch, to ten seconds; eight-inch, to two seconds; ten-inch, to single seconds. It is necessary with micrometric instruments, even five-inch, that they should be figured at every degree; large instruments to half degrees. The micrometric theodolite is made the same in general structure as the ordinary transit, except for the micrometer.

THE OMNIMETER.

The omnimeter (fig. 27, page 64) is an instrument in which the tangent to a radius reads direct from the axis of the telescope to a rule or stadia of constant length, so that the subtense angle varies with the distance. The instrument consists of a theodolite to which the subtense tangential system is added. In the construction of the instrument there are two sights, one, a telescope to sight the object, and the other, a powerful compound microscope to read divisions upon a tangential scale. The telescope and microscope are united together at 90° to each other with their axes exactly crossing the transverse axis of the theodolite, so as to move together through the

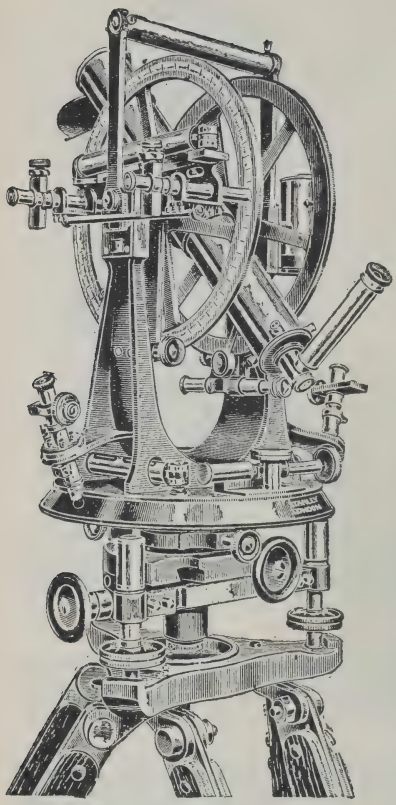


FIG. 26.

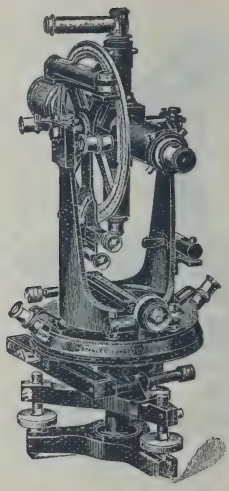


FIG. 27.

same amount of angle by the motion of the telescope in traversing the azimuth. A delicate level is placed upon the telescope, and when the bubble is in the centre of its run the scale is at right angles to the axis of the microscope. The scale, which lies flat on the limb at a central distance of six inches, is four inches long, placed in a sliding fitting to adjust longitudinally to its central position by means of a micrometer screw. It is generally divided into 100 parts for numerical calculation. The divisions are subdivided by shorter lines, making the total division 200. The micrometer screw has fifty threads to the inch, and moves over one of the divisions of the scale only. The micrometer head is divided into 100, numbered at the tens ; a vernier placed against the head subdivides each of these divisions into five, making the total micrometer 500 for one complete revolution. The total division of the four-inch scale therefore becomes : 200 (divisions of scale) × 500 (micrometer) = 100,000 in four inches. The scale is placed centrally to the instrument, so that when the telescope is level the microscope is vertical, and reads 50,000 when in perfect adjustment.

The mathematical theory of the instrument is very simple :

In fig. 28, O is the centre point of the axis of rotation, directly over the divided scale, AB ; O*b*, the base line of the instrument, which may be found by experiment or by calculation ; *mn*, a staff of known length, say 10 feet ; *b*, the departure-point of the scale when the telescope is parallel to AB ; in this position, from the nature of the construction, the microscope points in the direction of the line O*b*, which is perpendicular to AB.

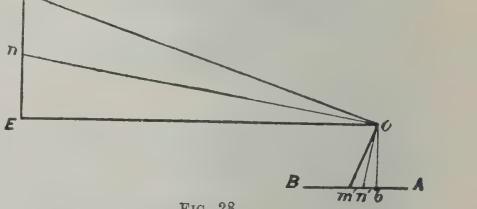


FIG. 28.

Then if the telescope be pointed to *m*, the microscope will be directed to *m'*, the line O*m'* being

perpendicular to the line Om. Again, if the telescope be directed to the point n, the microscope will be in the direction of On' and perpendicular to On. Omn and $Om'n'$ being similar triangles, we have $m'n' : Ob :: mn : OE$; the perpendicular distance from the line mnE passing through the staff mn, $\therefore$ OE $= \frac{mn \times Ob}{m'n'}$; $m'n' : bn' :: mn : n$E, consequently the altitude nE $= \frac{mn \times bn'}{m'n'}$, or $= \frac{bn' \times OE}{Ob}$.

The perpendicular Ob, which is termed the base line of the instrument, when the temperature is not very variable, may be found very accurately by taking OE about 100 feet, and placing the 10-foot staff, mn, as shown in fig. 28 ; for we have $Ob = \frac{m'n' \times OE}{mn}$.

Let $m'n'$ be = 47,735 — 46,015 = 1,720, as in the example we have taken ; and suppose the staff mn = 10 feet between the upper and lower marks, Ob = 6 inches (by construction), therefore Ob = 150,000 equal parts, when 4 inches contain 100,000 such parts.

$$\therefore \text{OE} = \frac{mn \times Ob}{m'n'} = \frac{10 \times 150{,}000}{1{,}720} = 872{\cdot}09 \text{ feet.}$$

Hence, for calculating distances we have the following easy rule : Multiply the length of the staff in feet by the length of the base of the instrument in micrometric equal parts, and divide the product by the difference of the two micrometer readings (of the staff) ; the quotient will be the required distance.

Subtense Measurements. Tacheometers.

It is quite common at present to place a pair of webs—or much better, a pair of points—in the diaphragm of a theodolite or level, these being at such a distance from each other that they will by vision correspond to a measurement (upon a distant levelling staff), which is proportional to the base of the subtense angle, so as to show thereby, in the space between the lines or points, the true distance of the instrument from the staff. Fig. 29 shows the form these points appear in focus of the eye piece. By the employment of this system chain measurements are entirely dispensed with, and the system has the merit of a simple sight measurement, as upon a rule, without any calculation. Theoretically it is scarcely as exact as the work of the Omnimeter, as there is great difficulty of exact adjustment of the points without the use of complicated optical means, but practically, as testified in some recent work by the Surveyor-General of Lagos and Nigeria, a good instrument in correct adjustment may be read within less than ·001

Fig. 29.

Fig. 30.

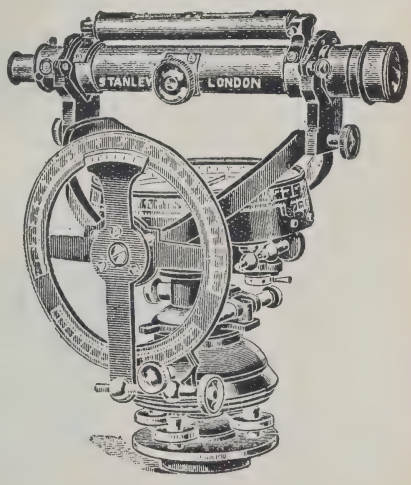


Fig. 31.

of total distance, which is nearer than is possible by chaining over uneven ground. One merit over the Omnimeter is that the instrument is not touched by the hand during the observation of the points, cut by vision upon the staff, which give the distance.

Mining Surveying Instruments.

Where the mining theodolite cannot be conveniently used, in consequence of the absence of height of gallery, constant short turns or otherwise, as also for economy, some form of mining dial is used. One of the most general is Hedley's (fig. 30, page 65). This being sighted only, it permits the eye to take observation from nearly the top of a gallery or cutting. The sights rock on an axis to any desired inclination, which may be read off on a small divided arc on the side. The compass reads by an edge bar needle. Dials of various patterns are made into instruments nearly resembling theodolites, the important points being observed that the instrument must be low and light, and read as nearly as possible at its highest part. The tripod of a dial is made with the legs to separate about the centre so that the instrument may be brought low, or what is better with sliding legs to adjust to any height required.

Stanley's Cranked Gimbal Dial.

This instrument (fig. 31, page 65) is fitted with sliding stand with solid head, and an improved form of ball head to give the necessary amount of tip to the instrument, when the tripod is upon sloping ground, to bring it level. It has clamp and tangent movement to the lower axis, also to the limb, which is outside reading by two verniers to single minutes. The gimbal is cranked so that the telescope will read 90° both ways, thus enabling a bearing to be taken of any object upon the surface from the top of the shaft, and a line being sighted at the bottom of the shaft in the same vertical plane with it, without changing the horizontal adjustment of the instrument. It is also useful for taking vertical sights up a shaft. The vertical arc has a clamp and tangent, and is divided to read single minutes. The telescope is fitted with stadia points for reading distances *without the use of a chain*. It is also fitted with folding sights for use without the telescope and a needle pin to lock compass at zero. The entire instrument is constructed as low as possible for use in shallow workings.

The Dumpy Level.

This is a perfect knock-about instrument. If stations are taken at equal distances apart, say one chain, this form of level will act correctly although out of adjustment. Thus with the level

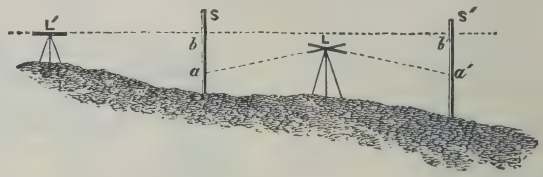


Fig. 32.

at L (fig. 32), place a staff vertically at one chain distance from the instrument at S. The inaccurate instrument will read the staff at a. Now place the staff vertically at S′. It will read at a'. The distances and verticality being equal, a and a' will be level. If we now place

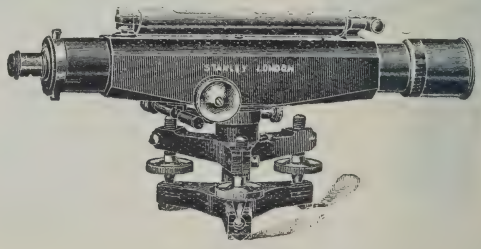


Fig. 33.

the level at some distant position nearly in line L′ and adjust the level by the bubble screw to read equal distances above *a* and *a′* as *b* and *b′*, the level must be true.

The stability of the level, without increase of weight, has always been a very desirable consideration with all engineers. This is best attained by shaping the level as nearly as possible out of solid castings. Fig. 33 (page 66) shows a level made in this way, and it is the strongest and most compact level made.

Adjustment of the Level.

(*R. T. Hancock.*)

Select a fairly level piece of ground, set up the instrument, carefully levelled up as described for the theodolite (p. 59), and drive a peg at a distance of 100 feet or so away. Hold the levelling staff on the peg and note the exact reading. On the opposite side of the instrument, and exactly the same distance away, set another peg, driving it down till the staff held on it reads exactly the same as on the first peg. The tops of the pegs are then on the same level, independent of instrument error. Set up on the prolongation of the line beyond one of the pegs, level up carefully, and make the crosshairs read the same on both pegs by raising or lowering the diaphragm cell by means of the screws holding it in the telescope tube. The collimation line is now horizontal.

It is not necessary to level the pegs if the difference in level is noted, and the crosshairs are altered to read the same difference. To facilitate the adjustment in this case, set up beyond the pegs at a distance which bears some simple relation to the distance between the pegs, make a sketch of the position of instrument and staffs, insert readings and distances, and calculate the amount it is necessary to alter the reading on either staff (by moving the crosshairs), on the principle of similar triangles.

The telescope bubble may not finally occupy the centre of its tube when the collimation has been put horizontal as described. This is unimportant, as the instrument can always be correctly levelled up if the procedure as described for the theodolite is followed, in fact the central position of the bubble should never be relied on as a test for levelling up, even if it has once been secured by adjustment. It may be brought to the centre of its run by adjusting the screws holding the level to the telescope, care being necessary not to disturb the instrument.

The Gradiometer.

This instrument (fig. 34) (*Stanley's patent*) is designed for taking vertical inclines at fixed small angles for railways, irrigation, drainage works, &c., &c., by means of an open extended scale, which may be conveniently and distinctly read without the use of a vernier.

It is also the most rapid levelling instrument yet made, as the telescope can be put level instantly at any sight by moving the drum, without any setting up whatever by the levelling screws. When the drum is at zero it forms an ordinary level of the most approved form. The gradiometric arrangement is effected by the telescope being mounted in trunnions, one being adjustable vertically; the amount of elevation or depression being indicated by a drum graduated to read rise or fall from 1 in 12 to 1 in 1,200.

The additional parts do not increase the size, and add little to the weight.

By its use a great saving of work is effected; for instance, a town drainage in which it is desired to work out an inclination which the levels indicate to have a fall of, say, 10 feet between the extreme points. If the line which the drainage is intended to take is measured, then however angular or zigzag it may be, the length of that line will be that into which the 10 feet is to be divided: thus if the line of streets measure 5,000 feet, this gives 1 foot in 500. Therefore if the drum is set to that proportion, then all the pipes may be laid directly without further setting.

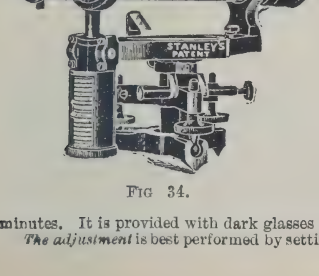


Fig. 34.

For levelling on steep inclines the instrument also saves a great number of settings up, as instead of levelling for say every 14 feet rise or fall, the gradient of the total distance can be taken, and also the distance, by stadia reading, and by reference to a gradient table the difference of level can at once be seen. Telemetrical readings at great distance may also be taken with accuracy by this instrument.

This gradiometer has been adopted by the Government for producing contour maps.

Box Sextant.

This is the most convenient instrument for filling in details of work performed by the theodolite, or for small surveys. It is constructed upon the principle of the nautical sextant, but the arrangement is made very portable by being placed in a circular box. The radius arm reads to single minutes. It is provided with dark glasses for adjustment upon the sun.

The adjustment is best performed by setting the sextant so that the sun appears a perfectly cir-

cular disc in the telescope with the index at zero. In defect of the sun some distant object may be taken, such as one edge of a chimney, house, or flagstaff, this object being looked at through the eye-hole and horizon glass. Turn the index arm, by means of the large mill-headed screw, until the reflected image of the selected object coincides in the horizon glass with the object seen direct. Having effected this, the vernier should then be examined, to note whether the zero-point of the vernier coincides with the zero-point of the arc. If this coincidence occurs, and the observed object be distant *fully half a mile*, then the instrument is in horizontal adjustment, but if the two zero-points do not agree, the index arm is moved until they do, and the key placed in the key-holes by the side of the sextant to move the horizon glass by which the reflected image of the distant object is made to coincide with the object seen direct through the lower part of the horizon glass. The key is inserted in the keyhole on the *top* of the instrument when it is required to move the image up or down for vertical adjustment. In all cases it is better to see that a perfect disc of the sun appears in the telescope when the index is at zero.

The Plane Table.

This instrument is now generally used in new countries and in our colonies, for filling in details of work with the theodolite. It has the great merit that an intelligent native without technical education can be instructed to do valuable detail work with it. In its simplest form

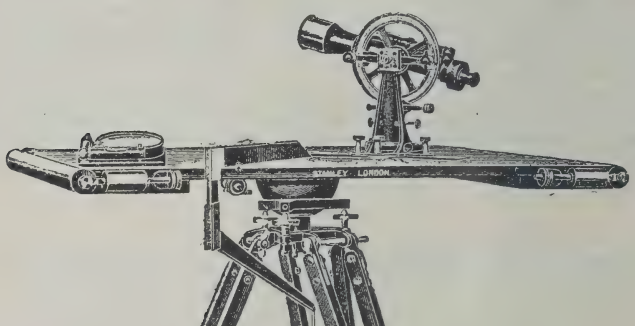


Fig. 35.

it consists of a sighted rule placed on a small board supported by a light tripod, the work being performed on the surface of paper. Many civil engineers find its work extremely convenient, because this work being made most definite by sketches upon the paper used, from these sketches they have a certainty of position in making maps from the same; so that they employ plane tables with all the refinement of means of observation possessed by the theodolite. One of these instruments is shown by fig. 35.

Prismatic Compass.

Details of small surveys or filling-in work are made by this compass, which is sighted for observation of direction, so that the eye may also see the compass ring by reflection through a prism at the instant of sighting. A reflector is commonly fixed to the forward sight so that distant objects may be seen up or down hill. The instrument is generally held in the hand for work, but where greater refinement is desired it is supported on a tripod stand. 2½ in. or 3 in. instruments are used by hand; 3½ in., 4 in., and 4½ in. are each set on a stand.

The Abney Reflecting Level.

A small level to hold in the hand. For flying work and approximate measurements from bench marks on an ordnance map the instrument is very useful. It may be used to level uphill from or to a bench mark by setting to zero and taking successive horizontal sights to points on the ground such as a stone or a daisy. The observer walks to the daisy and takes another forward sight, and so on up to the point whose level is required. Then the difference of level is the height of the observer's eye × number of sights taken. Or if the site is visible from the bench mark it can be sighted at once, the bubble being brought to the line of sight and the angle of the vernier arm read. The distance horizontally can be measured on the map, and the difference of elevation is this length multiplied by the tangent of the angle.

(See also 'Surveying and Levelling Instruments,' Stanley: published by E. and F. N. Spon.)

The Steward Surveying Telemeter.

(J. H. Steward, Ltd.)

This Pocket Telemeter (fig. 36) is especially useful when the survey of a tract of country, a preliminary survey for a railway, or some topographical reconnaissance, has to be made in a short space of time, and when the transport of cumbersome instruments is inconvenient. The size of the telemeter is 4½ × 1½ in., and its weight 10 oz.

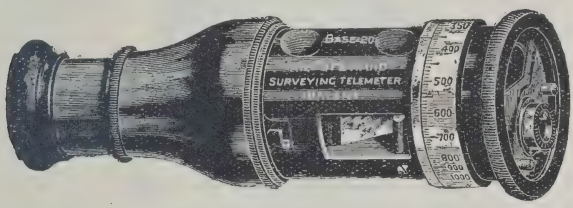


FIG. 36.

The arrangement consists of two plane mirrors mounted upon pivots and serving as index and horizon mirrors. The mirrors are mounted upon a base enclosed in a metal tube, to one end of which is attached a small sighting telescope. To the other end is adapted a movable collar, connected to the index mirror by means of a metal arm. By rotating this collar the index mirror is moved in azimuth, and the angular displacement is measured by reference to a graduated scale upon the exterior of the collar. When the scale is set to zero the mirrors form an optical square and reflect a right angle. A small toothed wheel attached to a screw acts on the horizon mirror so that the angle of double reflection can be varied about 5 degrees out of the right angle, in order to facilitate the determination of the direction of the base.

The scale of distances is divided upon the German silver rotating collar, each division representing 10 units of distance. Any unit of measurement may be employed with the Telemeter, the graduations showing the range in terms of units of the base.

The principle of the Telemeter is to measure the two angles at the extremities of the base, forming a triangle with the object of which the distance is required. The distance or range of the object is read directly from a scale graduated on the instrument, and is expressed in terms of units of the base, so that any scale of measurement (yards, feet, metres, etc.) may be employed. The scale of distances is computed for a base of 20 units, and as the unit of measurement employed may be of any length, it follows that distances of any length can be determined, the limit of distance being governed by the length of base that can be employed and the visibility of the object. The instrument is also adapted for use with a base which is of constant ratio to the distance of the object.

The accuracy of the indications is assured by empirically graduating the scale. The results obtained with the Telemeter have been most satisfactory, the mean error working out at less than 1 per cent.

The Steward Hypsometric Aneroid, with Swing Thermometer.

(J. H. Steward, Ltd.)

This aneroid (fig. 37, page 70) is constructed so as to eliminate the many sources of error present in aneroids of ordinary construction and to facilitate operations by doing away with the necessity of calculations.

The altitude scale is regularly divided, to the left for 'ascents' and to the right for 'descents,' and forms a complete circle of equal divisions closed at the two extremities. The zero of the altitude scale can be displaced relatively to the barometric scale and set opposite the index hand at any point of the barometric scale without introducing an error; consequently it is only necessary to set the zero to the index hand by means of the rack and pinion at one station, and the index hand will indicate the difference in altitude at the next station. The divisions are so arranged that with an altitude scale extending up to 10,000 feet, differences as small as five feet or less can be read without the application of a vernier; and by a special arrangement of the reading lens there is no error of parallax. The movement being compensated, changes of

temperature do not affect the readings. The aneroid is read and set without removing it from its case, and the altitude scale is automatically locked so that it cannot shift during transit.

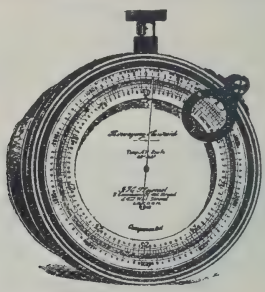


FIG. 37.—The Steward Hypsometric Aneroid.

The diameter of the dial is 3¼ in. A small swing thermometer, for ascertaining the air temperature, to make the correction for difference in the weight and volume of the atmosphere, is carried in a pocket at the back of the case.

REVERSING LEVELS.

(*Carl Zeiss (London), Ltd., Winsley House, Wells Street, Oxford Street, London, W.*1.)

The reversing level is a type of instrument very suitable for all accurate work in surveying. The telescope can be rotated about its optical axis through 180° and this automatically eliminates errors of collimation by providing two settings, the mean of which gives the true level. As is customary with all Zeiss levelling instruments, the vertical axis is made of hardened steel and is accurately ground cylindrical. The thrust due to the weight of the telescope is taken by a ball bearing; this relieves the central axis of all strain and thus ensures permanent accuracy of adjustment.

The telescope is provided with an internal focussing lens which is so arranged that the magnification is constant for all object distances, and the stadia lines in the eyepiece therefore give the

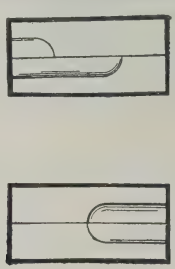


FIG. 38.

object distance from the staff reading by merely multiplying by 100 without the addition of a constant.

The bubble is viewed through a prism system which enables the setting to be made by bringing into coincidence the images of the two half ends of the bubble as shown in fig. 38. This method eliminates errors due to parallax, and at the same time doubles the accuracy of setting as compared with the traditional method of setting with a graduation.

The firm of Zeiss also manufactures a large reversing level which can be fitted with an optical micrometer for interpolating between two lines on the staff graduation. The device consists of a plane parallel glass plate which can be tilted relative to the optical axis of the telescope by means of a drum bearing a scale divided into 1/1,000ths of a foot. The readings thus obtained apply equally well at all working distances, as the effect of the tilted glass plate is independent of the distance. This instrument is specially suitable for work requiring the highest precision.

The Zeiss Theodolite.

This instrument, fig. 39, differs from the usual theodolite in that the use of verniers and screw micrometers is dispensed with. The horizontal and vertical circles are both read simultaneously in the eyepiece of the reading microscope. The method of reading is with the aid of a special prism system, fig. 40, which presents the images of opposite sides of the circles side by side in the eyepiece as shown in the illustration. Generally the graduations showing the angle to be measured and those showing 360° minus this angle will not coincide. By turning the micrometer head the graduations can be brought to coincidence, then the angle can be read to the nearest 10 minutes by estimation from the index line. The figured lines give whole degrees and the lines in between

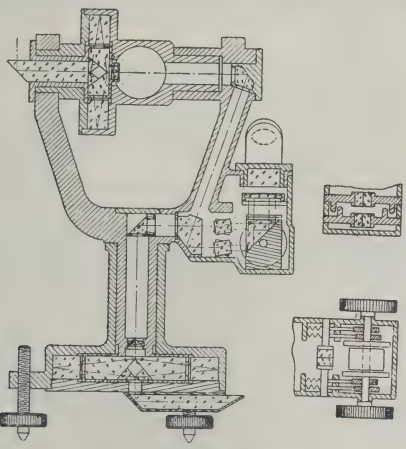


Fig. 39. Section of Zeiss Theodolite.

give intervals of 20 minutes. The actual minutes and seconds will now be read in the same eyepiece on the micrometer scale below. The lower figures give minutes, and the higher row of figures give tens of seconds. Single seconds are read direct and fractions of seconds by estimation. To take readings on the other circle, all that has to be done is to bring the corresponding graduations into coincidence by turning the same drum and to note the readings on the micrometer scale as before. It should be mentioned that the micrometer is purely an optical device and it does not affect the sighting of the telescope.

The divided circles in the Zeiss theodolite are totally enclosed and thus protected from dust and moisture; they are made of glass, and the graduations are permanent. The circle readings

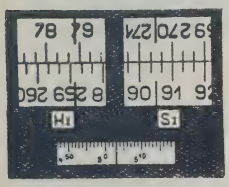


Fig. 40.

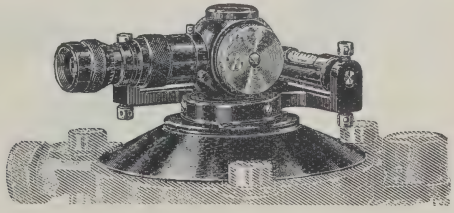


Fig. 41.

are taken in the micrometer eyepiece next to the eyepiece of the telescope in whatever direction the instrument is sighted, thus obviating the necessity for the observer to walk round the tripod to take the readings. The horizontal circle can be moved independently and set to any reading desired. This takes the place of the old type repetition arrangement. A protecting cap is

provided for the milled head to prevent its being turned inadvertently. For night work, and mining and tunnelling work, the instrument can be provided with electric illumination. Two types of optical plummet, fig. 41, can be provided, one for downward sighting and the other for sighting both downward and upward.

The Zeiss Self-reducing Tacheometer.

This instrument is designed for accurate stadiometry in mountainous or closely built over districts, also in the laying out of railways. In its general outline the arrangement is a repeating theodolite, but it serves at the same time as a double image rangefinder. When the telescope is sighted on the centre of the horizontal staff, the observer sees (fig. 42), in the top half of the field

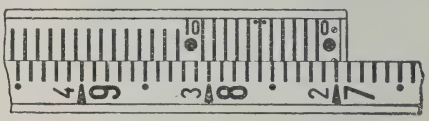


Fig. 42. Tacheometer Staff as seen in the Eyepiece of the Instrument.

an image of the top edge of the staff displaced sideways with respect to the image of the bottom edge of the staff, which appears in the lower half of the field in its natural position ; this displacement is proportional to the distance of the staff. The reading is taken in the following manner : the first line of the vernier serves as an index line and the other lines to the right give the decimals of the main graduations. The accuracy of reading is up to 1 in 10,000.

The Zeiss Sectioning Alidade.

This instrument, fig. 43, enables the engineer to obtain accurate cross-sections of structures, such as railway embankments, cuttings, canals, etc., in the field, without subsequent calculations. The telescope is coupled mechanically with a graduated ruler which follows the elevation and depression of the telescope, thus transferring the angle of elevation to the drawing on the plane

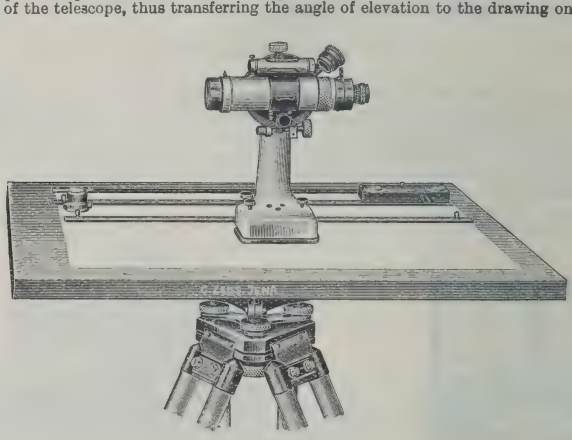


Fig. 43.

table. The distance of the section point is found by means of stadia lines and marked on the paper at the corresponding graduations on the ruler. As many as five sections can be completed in one hour's work. The instrument can be placed on any plane table and also can be used as an ordinary telescopic alidade. A special staff is provided for stadiometric distance readings, fitted with a sighting device for setting the staff at right angles to the line of sight.

To Find the Variation of the Compass.

1. By the Pole Star (Polaris).

The pole star is on the meridian, either above or below the pole, when a vertical plane passes through it, and the bright star at the root of the tail of the Great Bear ε Ursæ Majoris (Ahoth). Thus, during the twenty-four hours, two opportunities will be presented of finding the pole star on the meridian. Set up a fine plumb-line in a convenient place, free from currents of air, and the observer, placing himself behind the thread, waits until the moment when the two stars are hidden by the thread. If the bearing of the pole star be then read, it will show the variation of the compass. But as it is not very easy to read a compass at night, it is best to fix a stick with a light on it in the line of the pole star, and as far off as possible, the line joining the observer's position, and the light is the meridian line, and its bearing may be read in the morning.

2. By a Single Altitude of the Sun.

1. Compare watch with chronometer to get correct mean time.

2. Note readings of barometer and thermometer.

3. Adjust theodolite and note index error.

4. Set up instrument at . . . . . Δ station; make the temporary adjustments, and set instrument at 360° on . . . . . Δ station.

5. About three hours from 12 o'clock, morning or afternoon, unclamp upper horizontal plate and vertical arc. See that the dark eye-piece is down, and move upper part of instrument till the sun is in the field of object glass. Clamp upper horizontal plate and vertical arc, and with the two tangent screws follow (morning observation) the sun's movement till the sun's upper limb just touches the intersection of the cross-hairs. Note the time by watch, and read on horizontal arc the azimuth, and on vertical arc the altitude of ⊃, sun's upper limb.

6. Unclamp horizontal plate, and either with a slight motion, or, if plate be clamped, wholly with tangent screw, follow the sun's motion in azimuth till the sun's lower limb (⌓) becomes tangent to the intersection of cross-hairs, the vertical arc being untouched. Note the time and the reading on the horizontal arc.

7. The mean of the times and the mean of the azimuths will give the time and azimuth at which the vertical reading of the arc denotes the apparent altitude of the sun's centre ⊙.

8. Correct apparent altitude, ⊙.

Correction 1. For index error of instrument. Correction 2. For refraction, modified by barometric and thermometric readings: corrections to be found in tables in Frome's 'Surveying,' Drayson, or Chambers. Correction 3. For parallax. To be found in tables in 'Nautical Almanac,' and modified to noted altitude by formula:—Sin par. in alt. = sin hor. par. × cos. alt.

9. Apparent altitude thus corrected gives true alt. ⊙.

10. The true alt. ⊙ subtracted from 90° gives the zenith distance ZS.

11. As the altitude of the elevated pole is equal to the latitude of the place, 90° minus latitude = zenith distance of the pole, viz. PZ.

12. The declination of the sun is given in the 'Nautical Almanac,' page 1, of each month, for each day at apparent noon. The variation or declination for each hour is also given, and the direction of declination, N. or S. Hence the declination at the time of observation is obtained. If the time of observation be 9 A.M., and declination increasing, the correction for three hours' variation must be subtracted from the declination at apparent noon, *vice versâ* if the declination be decreasing.

13. Then we have the three sides of the astronomical triangle to ascertain the angle Z, which is the measure of the angular deviation of the sun from the true meridian.

14. Compute the angle Z by the formula

$$\cos \frac{Z}{2} = \sqrt{\frac{\sin S \sin (S - PS)}{\sin PZ \sin ZS}}$$

where $S = \frac{1}{2}$ sum of the three sides of the triangle.

15. The magnetic bearing of the Δ station from which the sun's azimuth was taken should be noted two or three times with great care. If possible a stand to be used for the compass.

16. Then, knowing the sun's azimuth from a station and the direction of that station as regards the magnetic north, we get the deviation of the sun from the magnetic north. The computed angle, Z, is the deviation of the sun from the true north, and the difference of the two deviations is the variation of the compass.

17. Note that for afternoon observations the computed angle, Z, s to be subtracted from 360° before comparison with sun's magnetic bearing.

Annual Compass Variation.

The mean declination at the observing station (in connection with the Royal Observatory, Greenwich) on the northern slope of Leith Hill, Surrey, was, for 1927, 12° 58·4′ W. The value of the declination at Greenwich may be inferred by subtracting the station difference, viz. 12·8′.

Horizontal and Vertical Force of Earth's Magnetism.

At Leith Hill the horizontal force was 0·18575, and the vertical force 0·42932, c.g.s. units.

Magnetic Dip.

The mean magnetic dip for 1927 at Leith Hill was 66° 36·2′.

To Determine the South by Means of a Watch.

Point the hour hand of the watch at the sun, then half-way round (clockwise) between the hour hand and 'XII' on the dial points 'south.' Thus, suppose it to be 8 P.M., point the hour hand at the sun, then the 'IIII' of the dial points to south. In southern latitudes, to find 'north,' point 'XII' at the sun, then half-way round between 'XII' and the hour is due 'north.'

To Determine a True East and West.

Set up a *vertical* rod (any size) on a flat floor. Describe a circle round the base about twice (this varies with the latitude) the height of the rod. Some time in the forenoon mark the point where the shadow of the rod just touches the circle, and repeat this in the afternoon on the other side of the circle; mark this point also. A line through these points will be true east and west.

As a guide to the true north, the following notes as to the whereabouts of the pole star at convenient hours in the evening or night will be useful, and correct enough for the purpose for the next hundred years.

The times given (Greeenwich mean time) are for the first day of each month, and become about 4 minutes (3 mins. 59·34 secs.) *less* every day.

Month.	1° W.	True N.	1° E.	Month.	1° W.	True N.	1° E.
January .	9·45	6·30	..	July . .	..	..	9·45
February .	7·45	..	..	August .	..	..	7·45
March . .	11·15	..	..	September .	..	..	11·15
April . .	9·15	12·30	..	October .	..	12·30	9·15
May . .	7·15	10·30	..	November .	..	10·30	7·15
June . .	..	8·30	11·45	December .	..	8·30	5·15

Equation of Time.

TABLE SHOWING TIME TO BE ADDED TO OR SUBTRACTED FROM SUNDIAL TIME IN ORDER TO OBTAIN TRUE OR CLOCK TIME.

Date.	Minutes (approximate).	Date.	Minutes (approximate).	Date.	Minutes (approximate).	Date.	Minutes (approximate).
Jan. 1	+ 4	Mar. 28	+ 5	Aug. 9	+ 5	Oct. 27	−16
„ 3	+ 5	April 1	+ 4	„ 15	+ 4	Nov. 15	−15
„ 5	+ 6	„ 4	+ 3	„ 20	+ 3	„ 20	−14
„ 7	+ 7	„ 7	+ 2	„ 24	+ 2	„ 24	−13
„ 9	+ 8	„ 11	+ 1	„ 28	+ 1	„ 27	−12
„ 12	+ 9	„ 15	0	„ 31	0	„ 30	−11
„ 15	+10	„ 19	− 1	Sept. 3	− 1	Dec. 2	−10
„ 18	+11	„ 24	− 2	„ 6	− 2	„ 5	− 9
„ 21	+12	„ 30	− 3	„ 9	− 3	„ 7	− 8
„ 25	+13	May 13	− 4	„ 12	− 4	„ 9	− 7
„ 31	+14	„ 29	− 3	„ 15	− 5	„ 11	− 6
Feb. 10	+15	June 5	− 2	„ 18	− 6	„ 13	− 5
„ 21	+14	„ 10	− 1	„ 21	− 7	„ 16	− 4
„ 27	+13	„ 15	0	„ 24	− 8	„ 18	− 3
Mar. 4	+12	„ 20	+ 1	„ 27	− 9	„ 20	− 2
„ 8	+11	„ 25	+ 2	„ 30	−10	„ 22	− 1
„ 12	+10	„ 29	+ 3	Oct. 3	−11	„ 24	0
„ 15	+ 9	July 5	+ 4	„ 6	−12	„ 26	+ 1
„ 19	+ 8	„ 11	+ 5	„ 10	−13	„ 28	+ 2
„ 22	+ 7	„ 28	+ 6	„ 14	−14	„ 30	+ 3
„ 25	+ 6			„ 19	−15		

LENGTH OF A DEGREE OF LONGITUDE AND LATITUDE.

Lat.	Degree of Longitude.		Degree of Latitude.		Lat.	Degree of Longitude.		Degree of Latitude.	
$\phi°$	Stat. Miles.	Naut. Miles.	Stat. Miles.	Naut. Miles.	$\phi°$	Stat. Miles.	Naut. Miles.	Stat. Miles.	Naut. Miles.
0	69·174	60·000	68·711	59·598	45	49·002	42·503	69·053	59·895
1	69·162	59·991	68·711	59·598	46	48·136	41·752	69·077	59·916
2	69·132	59·964	68·712	59·599	47	47·261	40·993	69·089	59·926
3	69·080	59·918	68·713	59·600	48	46·372	40·222	69·101	59·937
4	69·006	59·854	68·715	59·602	49	45·469	39·439	69·113	59·947
5	68·911	59·774	68·718	59·604	50	44·552	38·643	69·125	59·958
6	68·798	59·674	68·720	59·606	51	43·621	37·836	69·137	59·968
7	68·661	59·557	68·723	59·609	52	42·677	37·017	69·149	59·978
8	68·505	59·420	68·725	59·611	53	41·719	36·186	69·160	59·988
9	68·327	59·265	68·730	59·615	54	40·749	35·345	69·172	59·998
10	68·130	59·094	68·734	59·618	55	39·766	34·492	69·183	60·008
11	67·912	58·905	68·739	59·622	56	38·771	33·629	69·195	60·018
12	67·673	58·698	68·744	59·627	57	37·765	32·777	69·206	60·028
13	67·413	58·473	68·749	59·631	58	36·746	31·896	69·216	60·037
14	67·133	58·230	68·755	59·637	59	35·716	30·979	69·227	60·046
15	66·832	57·969	68·761	59·642	60	34·675	30·076	69·238	60·056
16	66·511	57·690	68·767	59·647	61	33·623	29·163	69·248	60·064
17	66·172	57·396	68·774	59·653	62	32·561	28·242	69·258	60·073
18	65·810	57·080	68·780	59·658	63	31·488	27·312	69·268	60·082
19	65·428	56·751	68·789	59·666	64	30·406	26·374	69·277	60·089
20	65·028	56·404	68·796	59·672	65	29·315	25·427	69·287	60·098
21	64·608	56·040	68·804	59·679	66	28·215	24·479	69·296	60·106
22	64·167	55·654	68·813	59·687	67	27·106	23·511	69·303	60·112
23	63·709	55·260	68·822	59·695	68	25·988	22·594	69·313	60·121
24	63·228	54·843	68·830	59·702	69	24·862	21·565	69·321	60·128
25	62·731	54·411	68·840	59·710	70	23·729	20·582	69·329	60·135
26	62·212	53·961	68·849	59·718	71	22·589	19·598	69·336	60·140
27	61·676	53·496	68·859	59·727	72	21·441	18·597	69·343	60·147
28	61·119	53·013	68·869	59·736	73	20·287	17·596	69·350	60·153
29	60·550	52·520	68·879	59·744	74	19·126	16·590	69·357	60·158
30	59·957	52·005	68·890	59·754	75	17·960	15·562	69·363	60·164
31	59·347	51·476	68·901	59·763	76	16·788	14·561	69·368	60·168
32	58·718	50·933	68·911	59·772	77	15·611	13·540	69·374	60·174
33	58·071	50·372	68·923	59·782	78	14·429	12·515	69·379	60·177
34	57·409	49·795	68·935	59·792	79	13·242	11·486	69·383	60·181
35	56·726	49·203	68·946	59·802	80	12·051	10·453	69·387	60·185
36	56·027	48·597	68·957	59·812	81	10·857	9·417	69·391	60·188
37	55·312	47·976	68·968	59·822	82	9·659	8·378	69·394	60·191
38	54·579	47·232	68·980	59·832	83	8·458	7·336	69·397	60·193
39	53·829	46·583	68·992	59·842	84	7·254	6·292	69·400	60·196
40	53·064	46·027	69·004	59·853	85	6·049	5·247	69·402	60·198
41	52·282	45·348	69·016	59·863	86	4·842	4·199	69·404	60·200
42	51·483	44·655	69·028	59·874	87	3·632	3·151	69·405	60·200
43	50·670	43·950	69·041	59·885	88	2·422	2·102	69·406	60·201
44	49·840	43·230	69·053	59·895	89	1·211	1·051	69·406	60·201
45	48·996	42·503			90	0·000	0·000		

Length of a degree of longitude at $\phi°$ latitude = cos $\phi_1°$ × 69·174 statute miles; where tan $\phi_1° = b \div a$ tan $\phi°$; a = equatorial radius (3963·36 miles), and b = polar radius (3950·06 miles); $b \div a = 3950{\cdot}04 \div 3963{\cdot}36 = 0{\cdot}99664$.

Example: What is the length of a degree of longitude at 50° latitude?

tan 50° = 1·1917536, therefore,

tan $\phi_1°$ = ·99664 × 1·1917536 = 1·1877493 = tan 49° 54′ 18″; cos 49° 54′ 18″ = ·6440569, therefore length of degree = ·6440569 × 69·174 = 44·552 statute miles.

Length of a degree of latitude at $\phi°$ latitude ($\phi°$ being at the middle of the arc) = 69·174 × (·99330 + ·01005 sin² $\phi°$) = 68·711 + ·6952 sin² $\phi°$ statute miles.

Example: What is the length of a degree of latitude between 30° and 31°?

$\phi° = 30°\ 30'$, sin 30° 30′ = ·5075384, therefore length of degree = 68·711 + (·6952 × ·5075384 × ·5075384) = 68·711 + ·179 = 68·890 statute miles. (*H. R. Kempe.*)

APPROXIMATE DISTANCE OF OBJECTS WHEN FIRST SEEN AT SEA, THE OBSERVER'S EYE BEING SUPPOSED AT SEA LEVEL.

Height. Feet.	Distance. Naut. Miles.	Height. Feet.	Distance. Naut. Miles.	Height. Feet.	Distance. Naut. Miles.	Height. Feet.	Distance. Naut. Miles.	Height. Feet.	Distance. Naut. Miles.	Height. Feet.	Distance. Naut. Miles.
1	1·1	22	5·4	43	7·5	120	12·6	300	19·9	520	26·2
2	1·6	23	5·5	44	7·6	125	12·9	310	20·2	540	26·7
3	2·0	24	5·6	45	7·7	130	13·1	320	20·6	560	27·2
4	2·3	25	5·7	46	7·8	135	13·4	330	20·9	580	27·7
5	2·6	26	5·9	47	7·9	140	13·6	340	21·2	600	28·1
6	2·8	27	6·0	48	8·0	145	13·8	350	21·5	620	28·6
7	3·0	28	6·1	49	8·1	150	14·1	360	21·8	640	29·1
8	3·2	29	6·2	50	8·2	160	14·5	370	22·1	660	29·5
9	3·4	30	6·3	55	8·5	170	15·0	380	22·4	680	29·9
10	3 6	31	6·4	60	8·9	180	15·4	390	22·7	700	30·4
11	3·8	32	6·5	65	9·3	190	15·8	400	23·0	720	30·8
12	4·0	33	6·6	70	9·6	200	16·3	410	23·2	740	31·2
13	4·1	34	6·7	75	9·9	210	16·7	420	23·6	760	31·7
14	4·3	35	6·8	80	10·3	220	17·0	430	23·8	780	32·1
15	4·4	36	6·9	85	10·6	230	17·4	440	24·1	800	32·5
16	4·6	37	7·0	90	10·9	240	17·8	450	24·4	820	32·9
17	4·7	38	7·1	95	11·2	250	18·1	460	24·7	840	33·3
18	4·9	39	7·2	100	11·5	260	18·5	470	24·9	860	33·7
19	5·0	40	7·3	105	11·8	270	18·9	480	25·2	880	34·1
20	5·1	41	7·4	110	12·1	280	19·2	490	25·5	900	34·5
21	5·3	42	7·5	115	12·3	290	19·6	500	25·7	920	34·9

Example : From what distance will a hill 300 feet high be visible to an observer 20 feet above the sea ?

To an observer on the summit of the hill the horizon would be 19·9 miles distant, and to the observer in the example it would be 5·1 miles distant, hence two such observers facing one another would just see each other at a distance of 19·9 + 5·1 = 25 miles, which is therefore the distance the hill will be visible.

ASTRONOMY.

The Earth and Sun.

Equatorial radius of the earth = 6,378,200 metres (3963·30 miles) (*Helmert*) = 6,378,388 metres (3963·42 miles) (*Hayford*); mean value, 6,378,294 metres (3963·36 miles). Polar radius of the earth = 6,356,818 metres (3950·01 miles) (*Helmert*) = 6,356,909 metres (3950·06 miles) (*Hayford*); mean value, 6,356,864 metres (3950·04 miles).

Equatorial horizontal parallax of sun (angle under which the equatorial radius of the earth would appear at the sun's centre) = 8″·80. Mean distance of sun from earth = $\frac{3963{\cdot}23}{\sin 8''{\cdot}80}$ = $\frac{3963{\cdot}23}{{\cdot}000042664}$ = 92,890,000 miles. Maximum distance (1st July) = 93,950,000 miles. Minimum distance (1st Jan.) = 90,950,000 miles.

Angle under which the radius of the sun would appear at the earth's centre, at the earth's mean distance from the sun = 16′ 1″·18 (*Auwers*).

Radius of sun = $3963{\cdot}23 \frac{\sin 16'1''{\cdot}18}{\sin 8''{\cdot}8} = 3963{\cdot}23 \frac{{\cdot}0045699}{{\cdot}000042664}$ = 424,500 miles.

None of the foregoing numerical values can be accepted as absolute, as authorities are not agreed on the subject.

Identification of Stars.

In stellar observations for time, latitude, or azimuth, the star selected for observation may be conveniently found from the following notes, which describe the relative positions of the larger stars in the constellations. The angular distance between two stars can be closely estimated by comparison with half the distance between the observer's zenith and the horizon—i.e. 45°.

(*a*) In the constellation *Ursa Major* the two well-known stars α and β (THE POINTERS) point directly to POLARIS, the Pole star. The latter also forms the end or tail of *Ursa Minor*.

(*b*) If the curve of the tail of *Ursa Major* be continued, about 30° along it will be found ARCTURUS (*α Boötes*).

(*c*) A line from POLARIS at right angles to the line from THE POINTERS gives 50° from the former, CAPELLA (*α Auriga*).

(*d*) A line from POLARIS through the last star but one of the tail of *Ursa Major* passes through SPICA, 30° beyond ARCTURUS.

(*e*) The POINTERS' line continued through POLARIS for 60° indicates THE GREAT SQUARE OF PEGASUS. The star in the corner opposite to the Pole is MARKAB.

(*f*) Midway between the SQUARE and POLARIS is the constellation *Cassiopeia* (five stars grouped like W).

(*g*) A diagonal through the SQUARE S.E. to N.W. produced 40° gives DENEB (*α Cygni*).

(*h*) 25° further and 10° to the right the same line gives VEGA (*α Lyræ*), a large white star.

(*j*) 35° south of DENEB and VEGA is ALTAIR (*α Aquilae*). This star is between two companions, and forms the apex of an isosceles triangle with DENEB and VEGA.

(*k*) A line from POLARIS through CAPELLA will touch RIGEL, 65° from CAPELLA. This line passes between ALDEBARAN (*α Tauri*), 10° west and 30° from CAPELLA, and BETELGEUSE, 10° east and 40° from CAPELLA. RIGEL and BETELGEUSE are diagonal stars in *Orion*.

(*l*) A line from ALDEBARAN through *Orion's Belt* passes near SIRIUS (*α Canis Majoris*) at an equal distance from ALDEBARAN. This is the brightest star in the heavens.

(*m*) 50° east of BETELGEUSE, forming nearly an equilateral triangle with BETELGEUSE and SIRIUS, is PROCYON.

(*n*) 30° north of PROCYON is CASTOR, and 5° S.E. of CASTOR is POLLUX.

(*o*) The apex of an isosceles triangle with CASTOR and PROCYON, 45° east, is REGULUS (*α Leonis*).

(*p*) Eastward from PROCYON to REGULUS 30° is DENEBOLA. DENEBOLA forms an equilateral triangle with ARCTURUS and SPICA.

ASTRONOMY IN FIELD WORK.

(*Fred. Simpson, Assoc.M.Inst.C.E.*)

Choice of Instrument.

Choice of Instrument.—The most suitable instrument is the Transit Theodolite provided with a delicate level attached to the vernier arms of the vertical circle, and a sensitive striding level for use on the pivots of the horizontal axis. See figs. 25, 26, pp. 63, 64. The adjustments must be made as perfectly as possible, and the residual index error carefully ascertained. The value of a division of each of the sensitive levels should be determined, and the position of the bubbles should be noted at the time of observing, and an allowance made for any variation in their readings. The most suitable form of diaphragm for astronomical work is that with simple vertical and horizontal cross hairs. For night work on stars it is necessary to illuminate the cross hairs, if the instrument is not provided with an illuminated axis; a lamp held a little distance in front of, and a little on one side of the object glass, may be used for this purpose.

With the ordinary eyepiece, altitudes exceeding 60° cannot be conveniently taken; for greater altitudes some form of diagonal attachment to the eyepiece is required. A makeshift arrangement can be improvised with a small piece of mirror held in the hand, or provisionally attached in front of the eyepiece at an angle of 45° to the axis of the telescope.

In observing the sun, a very convenient method, less trying to the eyes, is to cast the image of the sun and cross hairs on to a piece of white paper or card (to act as a screen) held about

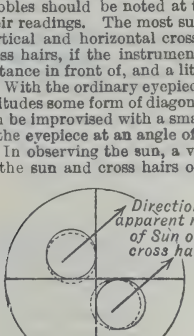


FIG. 44.

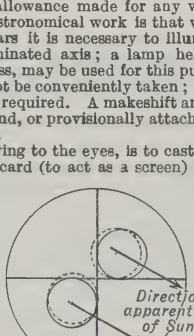


FIG. 45.

three or four inches from the eyepiece; the telescope must be focused for long-distance vision, with the cross hairs in focus; the telescope is then directed to the sun, and the eyepiece drawn out until perfect images of the sun and cross hairs are projected on the screen, the diameter of the image of the sun being smaller as the eyepiece is withdrawn. If the telescope inverts with direct vision, the image on the screen will be erect.

The limbs of the sun are observed when tangential to the respective cross hairs; the image of the sun should be allowed to leave one cross hair clamped a little in advance and followed with the other cross hair, using the tangent motion, so that the moving hair is brought tangential to the other limb as the first limb is just leaving the fixed hair; in this way only one hand is

occupied with the tangent motion. The correction for semi-diameter must be applied to the readings in altitude and azimuth. In the case of the altitudes, the correction for semi-diameter is applied as taken from the Almanac; in that of the azimuths, the correction to be applied to the horizontal circle reading is approximately the semi-diameter, taken from the Almanac, multiplied by the secant of the angle of altitude. These corrections apply to observations taken on any one limb of the sun to reduce the reading to the centre of the sun; but by taking the means of pairs of observations, on the upper and lower limbs for altitude, and on the right and left limbs for azimuth, the necessity for correction for semi-diameter is eliminated.

In a double observation on upper and lower and right and left limbs, a pair of observations are made, one in which the horizontal hair is fixed in advance, the vertical hair being moved by the tangent motion of the horizontal circle; the other observation is then made with the vertical hair fixed in advance, the tangent motion of the vertical circle being then used to follow the lower or upper limb with the horizontal cross hair, up to the time of the vertical hair, with the right or left limb which is being observed.

The cross hair which is being moved should always be kept slightly on the disc of the sun, up to the time of contact with the other hair, as it becomes indistinct when off the disc.

The means of the circle readings of the altitudes and azimuths taken in this way will be thus reduced to the centre, and will correspond to the mean of the times of observations, and no correction for semi-diameter is required.

Altitudes require correction for index error (+ or −), for refraction corresponding to altitude (always −), parallax (in case of sun or planets) corresponding to altitude (always +); semi-diameter in case of sun or planets, + if lower limb is observed, − if the upper; this correction is eliminated if observations on upper and lower limbs are combined.

Refraction.—The mean refraction values are near enough for ordinary purposes when the altitude observed is over 20°, but for lower altitudes a correction may be necessary, corresponding to the altitude, temperature, and barometric pressure; refraction decreasing with higher temperature and increasing with higher barometric pressure.

Parallax.—The horizontal parallax of the sun is approximately 9″, and for any given altitude = 9 × cosine of altitude, in seconds.

Declination as taken from the Almanacs must be corrected to correspond to the time of observation referred to time at the principal meridian adopted in the Almanac; generally it is sufficient to know his time to within ten minutes in spring or autumn, or within thirty minutes in summer or winter.

To Find Latitude.

Latitude may be deduced from a single altitude of a star, or the upper or lower limb of the sun, when crossing the meridian. This altitude is corrected for index error and refraction in the case of a star; in that of the sun a further correction is required for parallax and semi-diameter to reduce the altitude to the sun's centre. The declination is corrected to the time of observation.

In the case of the sun, and of stars with declination less than the latitude if of the same name:—
Latitude = 90° − (altitude + declination), if declination is of the opposite name to latitude.
Latitude = 90° − (altitude − declination), if declination is of the same name as latitude.
In case of stars with declination of the same name, but greater than latitude:—
Latitude = (altitude + declination) − 90° at the upper transit.
Latitude = altitude + (90° − declination) at lower transit of circumpolar stars.

More exact determination of the meridian altitude can be obtained from a series of ex-meridian altitudes taken about the time of the star or sun's meridian passage, provided three altitudes are taken at equal intervals of time, or equal intervals of azimuth (it is immaterial what intervals are taken, but they must be equal and not extend more than about twenty minutes on either side of the meridian passage).

The corrected altitude at the meridian may be obtained by the following reduction:—

Subtract the lowest altitude from the other two and let these differences, reduced to seconds, be denoted by d_1 and d_2; then the meridian altitude will be = lowest altitude + $\frac{(4d_1 - d_2)^2}{8(2d_1 - d_2)}$ seconds.

EXAMPLE.—Three altitudes of the upper limb of the sun were taken at intervals of 6 minutes as follows:

	° ′ ″	° ′ ″	° ′ ″
	38 10 35	38 12 40	38 12 30
+ correction	2 17	− 38 10 35	× 38 10 35
= altitude on meridian, upper limb	38 12 52	2 05 = 125′	1 55 = 115′

$$\frac{(4 \times 125 - 115)^2}{8(2 \times 125 - 115)} = 137'' = 2' 17''.$$

To Find Time and Azimuth.

Time and Azimuth from sun or star altitudes.

Let a = altitude observed (corrected for under error and refraction in case of star, as well as for parallax and semi-diameter in case of sun);

l = latitude

p = polar distance = 90° − declination, if latitude and declination are of the same name, = 90° + declination, if latitude and declination are of different names.

$$s = \frac{a + l + p}{2};$$

then,

$$\log \sin \tfrac{1}{2} \text{ Hour Angle} = \frac{\log \sec l + \log \operatorname{cosec} p + \log \cos s + \log \sin (s - a)}{2}$$

$$\log \cos \tfrac{1}{2} \text{ Azimuth} = \frac{\log \sec a + \log \sec l + \log \cos s + \log \cos (p - s)}{2}$$

The Hour Angle in arc is reduced to time; if from a P.M. observation of the sun this will be Apparent Time P.M., but if from an A.M. observation this Hour Angle in time should be deducted from 12, the result being Apparent Time A.M. To arrive at Mean Time, the Equation of Time must be added or subtracted as directed in the Almanac, this Equation of Time being first corrected to correspond to the approximate time of the observation at the principal meridian to which the Almanac refers.

In star observations for time, the Hour Angle found and reduced to time is deducted from or added to the Right Ascension of the star (deducted from if before the meridian transit, but added to if after), this gives the Sidereal Time of observation. Take the Right Ascension of the Mean Sun for the previous noon from the Almanac, and correct this to the meridian of the place of observation; this deducted from the Sidereal Time of observation, and the difference reduced to Solar Time, will give the Mean Time of the observation.

	h. m. s.	h. m. s.
E.g., suppose Hour Angle found after transit of meridian		4 30 24
Right Ascension of star		8 10 43
Sidereal Time at observation		12 41 07
R.A. Mean Sun	1 35 28	
Correction for longitude 3 h. 54 m. West	+ 38	
		1 36 06
Interval in Sidereal Time from Mean Noon		11 05 01
Correction of interval, Sidereal to Mean Time		− 1 49
Mean Time of observation		11 03 12

The horizontal circle reading of the azimuth found as above should be corrected for the inclination of the horizontal axis of the vertical circle as denoted by the reading of the cross striding level; the difference of level in seconds of arc multiplied by the tangent of the altitude taken will give the correction to the horizontal circle reading this being added if the left-hand pivot is highest, and subtracted if the right-hand pivot is highest. The circle reading of some distant well-defined object should be used as a Reference Point to which the azimuth observed will then be connected.

In a series of observations where the sun has been utilised in the daytime, and it is desired to connect up a series of observations on stars, the adjustment of the instrument should be carefully gone over after sundown, and when the instrument has attained an even temperature, the Reference Point being utilised to check the results of the day observations with those of the night.

The degree of accuracy of the results will depend upon the care taken to eliminate slight errors in the data assumed or instrumental defects. In the case of observations for time, the taking of the mean of a combination of A.M. and P.M. observations will to a great extent eliminate the slight errors that may be due to instrumental defects, incorrect latitude assumed, or almanac corrections to date. The same remarks apply also to Azimuth except that the cross level error can only be eliminated by observations taken with reversed positions of vertical axis. In observations for latitude the errors of the instrument can be to a great extent corrected by taking the mean of meridian observations on stars to the north and to the south of the zenith.

EXAMPLE (in time and azimuth; P.M. observation). Data: latitude (l) 53° 29′ 05″ N.; corrected declination, 0° 35′ 13″ S.; corrected altitude of the sun's centre (a) 20° 37′ 18″; polar distance (p) 90° + 0° 35′ 13″ = 90° 35′ 18″.

Time.

	h. m. s.	
a =	20 37 18	
l =	53 29 05	log sec 0·22546
p =	90 35 13	log cosec 0·00002
	2)164 41 86	
s =	82 20 48	log cos 9·12444
$s-a$ =	61 43 30	log sin 9·94482
		2)19·29474
½-hour angle	26 21 30	log sin 9·64787
	2	
hour angle	52 43 00 in arc	
	4	
hour angle	3 30 52 in time	

Azimuth.

	° ′ ″	
a =	20 37 18	log sec 0·02876
l =	53 29 05	log sec 0·22546
p =	90 35 13	
	2)164 41 36	
	82 20 48	log cos 9·12444
$(p-s)$ =	8 14 25	log cos 9·99549
		2)19·37415
	60 53 30	log cos 9·68707
	2	

Azimuth N. 121 47 00 W.

To Find Azimuth from Greatest Elongation of Circumpolar Stars.

At instant of greatest elongation :—

$$\sin \text{Azimuth} = \cos d \sec l;$$
$$\cos \text{Hour Angle} = \tan l \cot d;$$
$$\sin \text{Altitude} = \sin d \operatorname{cosec} l.$$

The Hour Angle, reduced to time, is added to the Right Ascension of the star if the Greatest Elongation is to the west of the meridian, or subtracted if to the east; this will give the Sidereal time of the Greatest Elongation. The sidereal time of the mean sun's passage of meridian for the succeeding noon subtracted from this will give the sidereal interval of time from noon, which, reduced to mean time, will give the Mean Time of Greatest Elongation. The altitude at Greatest Elongation is useful in setting the instrument for the observation.

Without knowledge of the latitude the Azimuth may be obtained from two observations on stars, one to the east, the other to the west of the meridian, and from the horizontal angle between them, which will be the sum of their Azimuths. Let this angle be denoted by ($A + A_1$), and let d and d' be the respective declinations of the stars used, then,

$$\tan\left(\frac{A - A_1}{2}\right) = \tan\frac{d + d_1}{2} \times \tan\frac{d - d_1}{2} \times \tan\left(\frac{A + A_1}{2}\right),$$

giving the half difference of Azimuths, which added to and subtracted from the half sum of Azimuths will give A and A_1, respectively. The latitude can then be computed from one of the Azimuths and found by formula

$$\cos l = \cos d \operatorname{cosec} A.$$

EXAMPLE.—Taking stars β Crucis (d_1 = 59° 06′ 30″S.); γ Hydri (d = 74° 33 59′ S.); angle between their Greatest Elongations 57° 53′ 33″ (= A + A_1).

$\frac{d + d_1}{2}$ =	66° 50′ 14½″	l tan 0·36874
$\frac{d - d_1}{2}$ =	7 43 44½	l tan 9·13264
$\frac{A + A_1}{2}$ =	28 46 46½	l tan 9·73980
$\frac{A - A_1}{2}$ =	9 53 05	l tan 9·24118
	± 28 46 46½	
Greatest Elongation γ Hydri	18 53 41½	
,, ,, β Crucis	38 39 51	

The latitude may then be calculated as follows, from γ Hydri :—

Azimuth	18° 53′ 41½″	l cosec 0·48968
d	74 33 59	l cos 9·42508
		l cos 9·91476

latitude = 34 43 55 S.

To reduce Mean Solar Time intervals to Sidereal Time intervals (approximate).

Reduce interval into hours and decimals of an hour, multiply by 10 and diminish product by $\frac{1}{16}$; the result will be correction in seconds to be *added* to Mean Time interval.

	h. m. s.	
EXAMPLE.	16 30 18 M.T.	16·505 × 10 = 165·05
Correction +	2 42·7	− $\frac{1}{16}$ 2·36
	16 33 00·7 Sidereal T.	162·69 sec.
		= 2 m. 42·69 sec.

To reduce Sidereal to Mean Time intervals.

Reduce intervals into hours and decimals of an hour, multiply by 10 and diminish product by $\frac{1}{16}$; the result will be correction to be *subtracted* from Sidereal Time interval.

	h. m. s.	
EXAMPLE.	16 33 00·7 Sid. T.	16·5502 × 10 = 165·50
Correction −	2 42·7	− $\frac{1}{16}$ 2·76
	16 30 18 M.T.	162·74 sec.
		= 2 m. 42·74 sec.

The error in using these approximations will be less than $\frac{1}{10}$ second in a 24-hour interval.

To Find Longitude by Moon Culminating Stars.

This is the most convenient method for determining the longitude when the Mean Time at Greenwich is not accurately known, the only requirements being a well-adjusted transit theodolite, a watch the rate of which can be depended upon for a short interval of time but the actual error of which need not be known, and a Nautical Almanac.

The moon varies comparatively rapidly in Right Ascension (R.A.), but its R.A. at the instant of its transit across the plane of the meridian can be accurately determined by comparing its time of transit with that of a star or stars which pass the meridian about the same time.

The theodolite is adjusted with its line of collimation in the plane of the meridian by the aid of one of the methods before described for obtaining azimuths ; the time of transit, over the centre thread, of the selected stars of reference and the moon's bright limb are carefully noted.

The intervals (reduced to Sidereal Time) between the times of transit of the respective stars and the time of transit of the moon's bright limb, added to the R.A. of the respective stars (in the case of stars which transit before the moon), or subtracted from the R.A. of any stars which transit after the moon, will give the R.A. of the moon's bright limb at the instant of its transit over the meridian.

We have, under the heading of 'Moon Culminating Stars,' the R.A. of the moon's bright limb at its transit at Greenwich, and also the variation of the moon's R.A. in one hour of longitude expressed in seconds ; the difference between the R.A. of the moon's bright limb when passing the meridian of Greenwich and the R.A. as found when passing the meridian at the place of observation gives the change of R.A., and this difference (reduced to seconds), being divided by the variation corresponding to one hour difference of longitude, will give the difference of longitude between Greenwich and the place of observation expressed in hours and fractions of an hour.

Under the same heading of 'Moon Culminating Stars' will be found convenient pairs of stars which transit within half an hour or so of the time of transit of the moon, and which have a declination nearly that of the moon, so that a slight movement of the telescope of the instrument

in altitude will bring them within its field. As these stars are generally of small magnitude and not easily distinguishable it is advisable to (approximately) calculate the altitude at which they cross the meridian ; the instrument may then be set in advance so that the star will cross approximately in the centre of the field. Any of the stars catalogued in the Nautical Almanac of greater magnitude, and easily distinguishable, may be used, but their declination and R.A. should not differ much from those of the moon.

When the longitude of the place of observation differs considerably from that of Greenwich the ' variation for one hour ' given in the tables should be corrected by interpolation to correspond to the approximate middle interval of time between the transit at Greenwich and that of the observed transit.

Let

t_1 t_2 be the observed times of transit of the star and the moon's bright limb ;

RA_1 RA_2 be the Right Ascensions of the star and the moon's bright limb taken from the catalogue ;

c the hourly variation (corrected, if necessary, to the middle time) ;

then the longitude in time $= \dfrac{(t_1 - t_2) - (RA_1 - RA_2)}{c}$,

the interval $(t_1 - t_2)$ if in Mean Time must be reduced to Sidereal Time,

$(RA_1 - RA_2)$ must be expressed in seconds,

the longitude will be expressed in hours and fractions of an hour, and to the west if positive and to the east if negative.

Timing Observations.

When working with an assistant the time may be noted at the instant a signal is given by the observer, but when observing alone it is difficult to note the exact time of contact or transit on watch or chronometer when the hands are employed in manipulating the instrument and the eye engaged in taking the sight.

In this case a convenient method is to use a watch whose time of beat is known (in modern watches usually $\frac{1}{5}$ second) ; this watch should be in such a position that the beats are distinctly heard, for example, suspended over the ear from the cap, in a small bag or watch pocket. The beats are then counted from the instant of contact or transit and continued until the seconds finger of the standard timepiece can be noted on some convenient second ; this time noted, less the number of beats of the auxiliary watch, reduced to seconds and fractions, will be the actual time of the observation.

As probably some little time may elapse after the observation before the standard timepiece can be read, especially at night where a lamp has to be used, this extended counting of beats of say $\frac{1}{5}$ second becomes difficult when arriving at the double syllable numbers ; but by adding a ' dummy ' syllable to the first twelve counted and then continuing with ' thir-teen,' ' four-teen,' and so on, giving one syllable to each beat, the counting of $\frac{1}{5}$ seconds becomes quite easy, even, if carried to 100 beats and over, e.g. taking the dummy syllable as ' and,' and starting the instant of observation with ' nought,' the counting would continue :—' nought-and,' ' one-and, ' two-and,' ' three-and,' . . . ' 'leven-and,' ' twelve-and,' ' thir-teen,' ' four-teen,' . . . ' twen-ty, ' twent'-one,' and so on. Then taking the first syllable as units and the second syllable as half, the counted beats multiplied by 4 and divided by 10 will give the required intervals in seconds and tenths, in the case of the auxiliary watch beating fifths of seconds.

Taking examples, say the beats counted between the instant of observation and the time noted on the timepiece was on the first syllable to thirty-seven, the time at that beat being 3 hrs. 47 mins. 15 secs. ; then the interval to be deducted would be $\dfrac{37 \times 4}{10} = 14{\cdot}8$ seconds.

and the corrected time of the observation would then be 3 hrs. 47 mins. 15 secs. $-$ 14·8 secs. $=$ 3 hrs. 47 mins. 0·2 secs. In another case, say the second syllable of ' nine-and ' corresponded to 6 hrs. 22 mins. 55 secs. on the timepiece, the correction would then be $\dfrac{9\frac{1}{2} \times 4}{10} = 3{\cdot}8$ seconds,

and the corrected time of observation 6 hrs. 23 mins. 55 secs $-$ 3·8 secs. $=$ 6 hrs. 23 mins. 51·2 secs.

This method of measuring short intervals of time is useful in other instances ; for example in estimating distances by the velocity of sound, etc.

TABLE OF MEAN REFRACTIONS.

Corresponding to altitudes, A; at 50° F. and Barometer 29½ ins.

A	0′	5′	10′	15′	20′	25′	30′	35′	40′	45′	50′	55′
°	′ ″	′ ″	′ ″	′ ″	′ ″	′ ″	′ ″	′ ″	′ ″	′ ″	′ ″	′ ″
0	34 54	33 51	32 49	31 50	30 52	29 57	29 3	28 12	27 23	26 35	25 50	25 6
1	24 25	23 45	23 7	22 30	21 56	21 23	20 51	20 21	19 52	19 24	18 58	18 33
2	18 7	17 45	17 23	17 1	16 41	16 20	16 1	15 42	15 23	15 5	14 48	14 31
3	14 15	13 59	13 44	13 29	13 15	13 1	12 48	12 36	12 24	12 12	12 1	11 50
4	11 39	11 28	11 18	11 8	10 59	10 49	10 40	10 30	10 21	10 12	10 3	9 55
5	9 47	9 39	9 31	9 23	9 16	9 9	9 2	8 55	8 48	8 54	8 36	8 29
6	8 23	8 17	8 12	8 6	8 0	7 55	7 49	7 44	7 39	7 34	7 29	7 24
7	7 20	7 15	7 11	7 6	7 2	6 57	6 53	6 49	6 45	6 42	6 37	6 33
8	6 30	6 26	6 22	6 18	6 15	6 12	6 8	6 5	6 2	5 59	5 55	5 52
9	5 49	5 46	5 43	5 40	5 38	5 35	5 32	5 29	5 26	5 24	5 21	5 19
10	5 16	5 14	5 11	5 9	5 6	5 4	5 2	4 59	4 57	4 55	4 53	4 51
11	4 49	4 46	4 44	4 42	4 40	4 38	4 36	4 34	4 32	4 30	4 29	4 27
12	4 25	4 23	4 21	4 20	4 18	4 16	4 15	4 13	4 11	4 10	4 8	4 7
13	4 5	4 3	4 2	4 0	3 59	3 57	3 56	3 54	3 53	3 52	3 50	3 49
14	3 47	3 46	3 45	3 43	3 42	3 41	3 39	3 38	3 37	3 36	3 35	3 33
15	3 32	3 31	3 30	3 29	3 27	3 26	3 25	3 24	3 23	3 22	3 21	3 20
16	3 19	3 18	3 16	3 15	3 14	3 13	3 12	3 11	3 10	3 9	3 8	3 7
17	3 7	3 6	3 5	3 4	3 3	3 2	3 1	3 0	2 59	2 58	2 57	2 57
18	2 56	2 55	2 54	2 53	2 52	2 52	2 51	2 50	2 49	2 48	2 48	2 47
19	2 46	2 45	2 45	2 44	2 43	2 42	2 42	2 41	2 40	2 39	2 39	2 38
20	2 37	2 37	2 36	2 35	2 35	2 34	2 33	2 33	2 32	2 31	2 31	2 30
21	2 29	2 29	2 28	2 27	2 27	2 26	2 26	2 25	2 24	2 24	2 23	2 22
22	2 22	2 21	2 21	2 20	2 20	2 19	2 18	2 18	2 17	2 17	2 16	2 16
23	2 15	2 15	2 14	2 14	2 13	2 13	2 12	2 11	2 11	2 10	2 10	2 9
24	2 9	2 8	2 8	2 7	2 7	2 7	2 6	2 6	2 5	2 5	2 4	2 4

A	0′	10′	20′	30′	40′	50′
°	′ ″	′ ″	′ ″	′ ″	′ ″	′ ″
25	2 3	2 2	2 1	2 0	2 0	1 59
26	1 58	1 57	1 56	1 55	1 54	1 54
27	1 53	1 52	1 51	1 50	1 50	1 49
28	1 48	1 47	1 47	1 46	1 45	1 45
29	1 44	1 43	1 42	1 42	1 41	1 40
30	1 40	1 39	1 38	1 38	1 37	1 36
31	1 36	1 35	1 35	1 34	1 33	1 33
32	1 32	1 32	1 31	1 30	1 30	1 29
33	1 29	1 28	1 27	1 27	1 26	1 26
34	1 25	1 25	1 24	1 24	1 23	1 23
35	1 22	1 22	1 21	1 21	1 20	1 20
36	1 19	1 19	1 18	1 18	1 17	1 17
37	1 16	1 16	1 16	1 15	1 15	1 14
38	1 14	1 13	1 13	1 12	1 12	1 12
39	1 11	1 11	1 10	1 10	1 10	1 9

A	0′	10′	20′	30′	40′	50′
°	′ ″	′ ″	′ ″	′ ″	′ ″	′ ″
40	1 9	1 8	1 8	1 8	1 7	1 7
41	1 6	1 6	1 6	1 5	1 5	1 4
42	1 4	1 4	1 3	1 3	1 3	1 2
43	1 2	1 1	1 1	1 1	1 0	1 0
44	1 0	59	59	59	58	58
45	58	57	57	57	56	56
46	56	55	55	55	54	54
47	54	54	54	53	53	52
48	52	52	51	51	51	50
49	50	50	50	49	49	49
50	48	48	48	48	47	47
51	47	46	46	46	46	45
52	45	45	45	44	44	44
53	43	43	43	43	42	42
54	42	42	41	41	41	41

A	0°	1°	2°	3°	4°	5°	6°	7°	8°	9°
°	″	″	″	″	″	″	″	″	″	″
50	48	47	45	43	42	40	39	37	36	35
60	33	32	31	29	28	27	26	25	23	22
70	21	20	19	18	17	15	14	13	12	11
80	10	9	8	7	6	5	4	3	2	1

SUN'S PARALLAX IN ALTITUDE.

A	0°	10	20°	30°	40°	50°	60°	70°	80°	90°
Parallax	9″	9″	8″	8″	7″	6″	5″	3″	2″	0″

Graphic Method of finding Time and Azimuth from Altitude of Sun or Star.

Describe a circle, fig. 46, of any convenient radius; through the centre O draw diameter MOM_1. From O draw PO making angle POM equal to the latitude; draw OQ at right angles to PO. Through O draw OA, making AOM equal to the corrected altitude, and OD, making DOQ equal to the corrected declination, and on the side of OQ towards P if the declination and latitude are of the same name, or on the opposite side if of different names.

Through A draw AA_1 parallel to MM_1 and through D draw DD_1 parallel to QO, intersecting AA_1 in s.

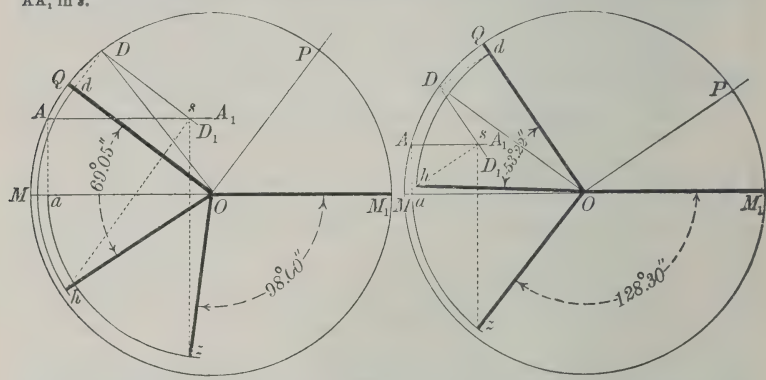


FIG. 46.

Data.	Result.	Data.	Result.
Lat. 53° 30′ N.	Azimuth N. 98° 00′ W.	Lat. 34° 30′ S.	Azimuth S. 128° 30′ W.
Dec. 14° 50′ N.	Hour Angle 69° 05′	Dec. 19° 50′ N.	Hour Angle 53° 22′
Alt. 24° 20′ P.M.		Alt. 15° 40′ P.M.	

To find the Azimuth.

Drop a perpendicular from A to OM, intersecting this latter in a; from centre O and radius Oa draw arc az, and from s draw sz perpendicular to MOM_1, intersecting arc az in z; through z draw Oz. Then the angle M_1Oz will represent the azimuth from the elevated pole east or west of the meridian according as the time (A.M. or P.M.) at which the observation was made.

To find the Hour Angle.

Draw Dd at right angles to QO, and with centre O and radius Od draw arc dh; from s draw line sh perpendicular to QO, intersecting arc dh in h; draw line Oh. Then QOh will be the hour angle at the time of observation before or after apparent noon.

This diagrammatic method may be utilised for approximate determination of other problems, such as azimuth and time of sun or stars at instant of rising or setting; altitude and azimuth at a given time; or for altitude and time of passing prime vertical.

Geodetic Formulæ.

From measurements made of various meridians and parallels it has been found that the earth has the form of an ellipsoid of revolution with its greatest diameter at the equator and its least between the poles; and that the radius of curvature of the meridian varies with the latitude, being least at the equator and greatest at the poles, so that the length of a degree of latitude is more at the poles than at the equator. The length of a degree of longitude varies also with the latitude, being greatest at the equator and zero at the poles.

In treating with measurements of an extensive character over the surface of the earth, it is necessary to take into account its ellipsoidal shape, and it is convenient to employ certain approximations, so that spherical trigonometry may be used in the solution of the problems involved. The following formulæ enable the radius of a spherical surface to be found that approximates most nearly to the ellipsoidal surface embracing the area under consideration.

In these formulæ

a = the equatorial radius of the earth; e = the eccentricity; ϕ = the mean latitude; A = the azimuth.

Radius of curvature of meridian = $Q = \frac{a(1-e^2)}{\sqrt{(1-e^2\sin^2\phi)^3}}$.

Radius of curvature of normal section perpendicular to meridian = $N = \frac{a}{\sqrt{(1-e^2\sin^2\phi.)}}$

Radius of curvature of a normal section which cuts meridian at an angle A

$$= \frac{QN}{Q \sin^2 A + N \cos^2 A}.$$

Radius of curvature of an area of the earth's surface, or mean curvature $= \sqrt{QN}$.

Radius of a parallel of latitude $= N \cos \phi = \frac{a \cos \phi}{\sqrt{1 - e^2 \sin^2 \phi}}$

The logarithms of the most probable values of the constants used above are as follows (taking a in statute miles):—

$$\log. a = 8{\cdot}59806. \qquad \log. e^2 = 7{\cdot}83047. \qquad \log. (1 - e^2) = 9{\cdot}99853.$$

Adopting the radius which most nearly corresponds to the surface considered at the mean latitude ϕ, any measured length l on this surface may be reduced to arc by one of the following formulæ, ρ being the radius adopted reduced to the same unit as l:—

$\frac{l \times 57{\cdot}2958}{\rho}$ in degrees of arc; $\frac{l \times 3437{\cdot}747}{\rho}$ in minutes of arc; $\frac{l \times 206265}{\rho}$ in seconds of arc.

The table on page 75, 'Length of a Degree of Longitude and Latitude,' is calculated from the accepted dimensions of the earth. Values for intermediate latitudes may be obtained from this table, by interpolating, with sufficient exactitude for most purposes. The length of a degree on a great circle at right angles to the meridian at a given latitude may be obtained by multiplying the length of a degree of longitude (as taken from the table) by the secant of the latitude.

Convergence of Meridians.

In traverses and route surveys, a survey line is referred to a meridian at a certain point crossing the meridian with an azimuth A, this line, if continued in the same direction, having increased or diminished azimuths. The differences of azimuth can be approximately determined by calculating the convergence of the meridian corresponding to a given unit departure at the mean of the latitudes, and multiplying this by the departure as calculated from the traverse. To obtain the azimuth on crossing the second meridian, the convergence obtained in the manner just explained is added to or subtracted from the azimuth crossing the first meridian, being added where the longitude is increased and subtracted where the longitude is diminished.

The convergence, corresponding to various unit lengths of departure, may be calculated by the following formulæ, with sufficient approximation for use in ordinary traverse surveying. The length of a degree of longitude corresponding to the mean latitude is taken in miles from table, page 75, interpolating if necessary; then:—

Log. of convergence in seconds per mile of departure
$= \log. \sin \phi + 3{\cdot}55630 - \log.$ miles per 1° longitude.

Log. of convergence in seconds per 1,000 feet of departure
$= \log. \sin \phi + 2{\cdot}83367 - \log.$ miles per 1° longitude.

Log. of convergence in seconds per 1,000 metres of departure
$= \log. \sin \phi + 3{\cdot}34966 - \log.$ miles per 1° longitude.

To range a parallel of latitude by setting out chords of a given length:—

The chords are ranged making an angle to the meridian of 90° — half the angle of convergence corresponding to the length of the chord at the proposed latitude. The succeeding chords will have an angle of deflection from the preceding chord produced, equal to the convergence corresponding to the length of chord used.

To range a parallel of latitude by means of offsets:—

A line is run, perpendicular to the meridian, then the lengths of offsets are equal to the distances along this perpendicular line multiplied by the sine of half the angle of convergence corresponding to the lengths.

If the offsets are equidistant the length of the second offset will be four times the length of the first; the third, nine times the first, and so on; the nth offset will be n^2 times the length of the first.

Horizontal Sun Dial.

On a line NS, fig. 47, mark off a convenient length NO; from O set off a line towards L, making an angle NOL equal to the latitude of the position (taken in diagram 53° 30′), and from N draw a perpendicular to OL, cutting the latter in point L. From N and with length NL mark off a point C in the line NS, and with C as centre and the same length NL as radius describe the quadrant NQ and join NQ with a straight line.

Divide the quadrant NQ into six equal parts (each corresponding to one hour) and from these divisions on the arc draw lines towards the centre C and mark the points at which these radial lines cross the line NQ.

From O draw a line perpendicular to NS and mark off OE equal to CQ and join NE.

From each division on line NQ draw a line parallel to NS, marking where this line crosses the line NE in each case.

Then lines drawn from O through each of these points, marked off in NE, will be the dial lines corresponding to the hour divisions on the quadrant from which they were found respectively.

Having completed the divisions of one quadrant, these angles may be transferred to the other quadrants but in reverse order (VI–VII equals VI–V, VI–VIII equals VI–IIII, and so on).

An allowance must be made for the thickness of the style between the AM and PM portions of the dial, the AM dial being exactly similar to the PM dial but drawn with the XII lines the thickness of the style apart.

The style is constructed with the angle of inclination of the inclined face equal to the latitude of the site, its lower point at O and its upper towards the elevated pole, so that its

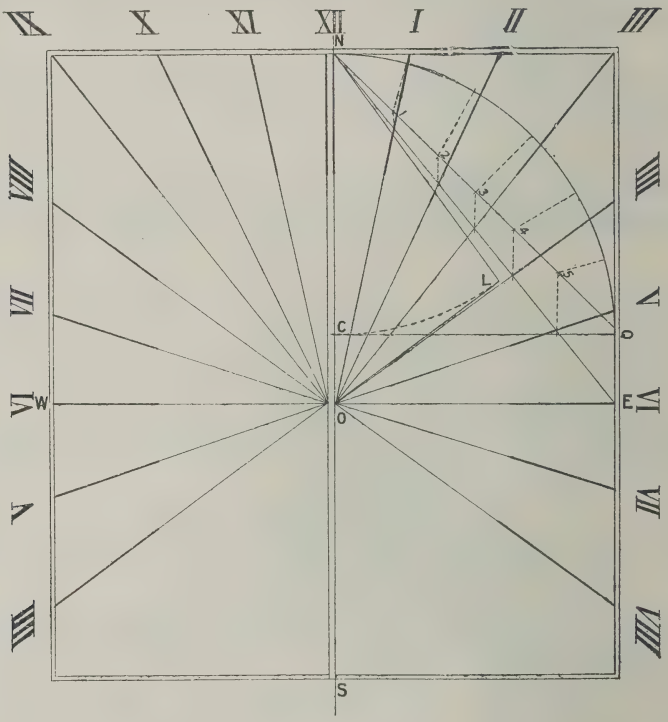


FIG. 47.

plane is in that of the meridian, the XII (noon) lines pointing towards the north in northern latitudes or towards the south in southern latitudes, and the VI–VI lines pointing east and west, so that the upper face of the style will be parallel to the Polar axis. The plane of the dial should be perfectly level.

(Methods of fixing the direction of the true north and south line will be found on page 74.)

The time indicated by the dial is sun time, and to get mean time the table of Equation of Time, page 74, should be applied as there directed. To arrive at standard time a constant correction must be applied according to the difference of longitude between the dial and the standard meridian (Greenwich in Great Britain) adopted in the country (subtracted if to the east or added if to the west).

The dial angles may be calculated if preferred by formula :

$$\tan t = \sin l \tan h ;$$

where

t = angle at O from NO to the hour line on dial corresponding to h ; l = latitude of position ; h = hour angle.

SECTION III.

PART II.

DRAWING.

(Contributed by H. A. Miles.)

Engineering drawings are usually required for one of two purposes, viz. as illustrations or for workshop use. The former, which are not necessarily to scale, or dimensioned, may be either 'line' or 'wash' drawings—the first for preference—or prepared from photographs in the following manner. A bromide print is outlined and worked up with waterproof ink, after which the photographic image is bleached out. This is useful in complicated perspective views, or where the skill of the draughtsman is not equal to elaborate freehand work.

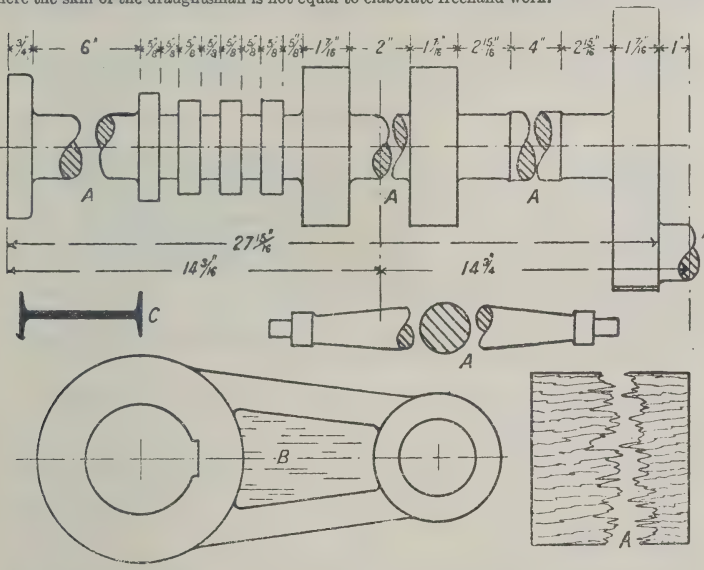


FIG. 1.

For reproduction purposes, drawings should always be prepared to an enlarged scale—twice full size is usual. If a scale is necessary it should be *drawn* (not denoted in words), so as to be reduced proportionately with the illustration to which it refers. Bold and striking effects should be aimed at, simplicity rather than elaboration being most effective. Shade lines and dotted hidden details are both useful, and surface lining, B, fig. 1, is sometimes necessary, but care must be taken to avoid overcrowding. A good rule is never to introduce a superfluous line or to omit one which is necessary.

The best material is 'Turkey Mill' paper or 'Bristol Board,' but it is sometimes advantageous to use tracing cloth, and (provided it is not yellow) good results will be secured with this. The blackest of black ink should be employed. All artists' colourmen now stock the liquid carbon or waterproof ink, and process black, either of which will be found suitable.

For workshop purposes a drawing should be looked upon as a book explaining the designer's views—it should be easily 'read' by the mechanic, and must contain all information necessary for the construction of the item represented. To this end there are certain conventions generally employed, and in the same shop these always mean the same thing, but unfortunately, owing to the absence of a recognised standard, each firm is at liberty to please itself, and consequently there is no uniformity.

The conventions which follow are those most frequently employed :—

Construction and visible boundaries . . .

Ditto hidden from view .

Centre or pitch lines . .

Projection lines . . .

Dimension lines . . . 1'. 3"

One view of an article is seldom sufficient for a proper understanding of its construction, and plans, elevations and sections are usually given, minor details being generally drawn separately and frequently to an enlarged scale.

All such drawings should be fully dimensioned. Where a number of small dimensions combine to make up a total, the overall should be given, not only to save time, but to act as a check upon the others.

Where coloured drawings are desired to indicate sections, the following will meet requirements :—

Cast iron	Neutral tint or Payne's grey.
Wrought iron	Prussian blue.
Steel	Violet carmine.
Copper	Crimson lake, little gamboge.
Brass	Gamboge, little burnt umber.
Lead	Indian ink and Prussian blue.
Brickwork	Light red, little crimson lake.
Wood	Yellow ochre and burnt sienna.

In each case the section is coloured a considerably deeper shade of the same tint than the other parts of the drawing.

In building plans (otherwise uncoloured) prepared to show alterations or additions to existing premises, it is a common practice to colour new work red.

Different methods of lining are sometimes employed to represent different materials, but if followed to any extent this is likely to lead to mistakes unless supplemented by a key or written explanation, in which case it can scarcely be considered necessary.

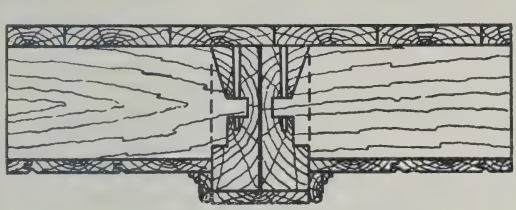
A very general practice is to adopt C (fig. 2) for all metals; to line up woodwork in imitation of the grain A (fig. 2); and to use as few others as possible—the material being specified in words thus :—'Brass'—'Copper 75 mils'—'Spring Steel 22 S.W.G.'—etc.

It is obviously impossible to fix any definite spacing between the lines. Small details must necessarily be treated differently from larger surfaces, and very large areas need only be sectioned around their boundaries.

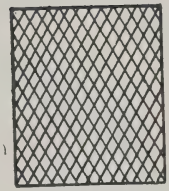
Typical specimens of various section conventions are shown in fig. 2. Where two pieces of material butt against each other, the section lining should run in opposite directions: if three meet, the lines running in the same direction should break joint. (D, fig. 2.)

Small or thin sections such as boiler plates, engine beds, girders, joists, etc., may be blacked in solid. (C, fig. 1.)

Long details such as columns, shafting, pipes, etc., need not necessarily be shown full length, and in such cases break lines, A, fig. 1, are employed. Should the diameter vary, one or more sections may be shown, as in the firebar example.

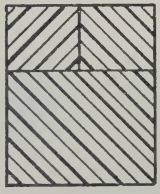


A. Wood, with and across grain.

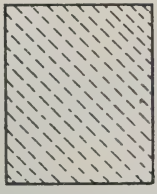


B. Lead.

C. Cast Iron.



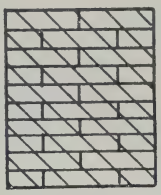
D Wrought Iron.



E. Steel or Stone.

F. Cast Steel.

G. Brass or Brick.

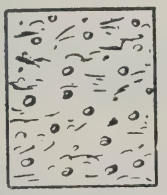


H. Brickwork.

I. Stonework.

K. Insulators.

L. Earth.



M. Concrete.

N. Water

O. Glass.

FIG. 2.

Patent Specification Drawings.

All necessary information will be found in a pamphlet entitled ' Instructions to Applicants for Patents,' published by and obtainable free on application at the Patent Office, 25 Southampton Buildings, Chancery Lane, London, W.C., or it will be sent free on receipt of 1*d.* for postage.

One point to remember in this class of work is that the 13-inch side of each sheet is bound up with the specification, and should be looked upon as the left-hand side. All lettering, etc., should therefore be upright, viz., not facing different ways.

For whatever purpose drawings are prepared, they will inevitably indicate to the expert eye the character of the draughtsman. One man scarcely varies the thickness of his lines, muddles his lettering, and does not space his work out to advantage. Another has the knack of ' displaying ' his work, as a printer would say : his construction lines are bold and the others thin, his lettering varies according to whether it is of major or minor importance—heavy block, small capitals, or small italics being used as considered advisable ; all his details are nicely spaced, and the whole effect is one of neatness, clearness, and completeness.

Drawing Appliances.

The Harling Pantagraph.

(*W. H. Harling*, 117, *Moorgate*, *London*, *E.C.* 2.)

This Pantagraph (fig. 3), which has been recently designed, is used for copying plans or parts of plans to a reduced scale.

The instrument is made of boxwood with brass fittings, and has a heavy iron weight, or fulcrum, which is further secured to the board at option, by two loose-fitting brass pins. When opened for use the pantagraph forms a square the sides of which measure 17 ins.

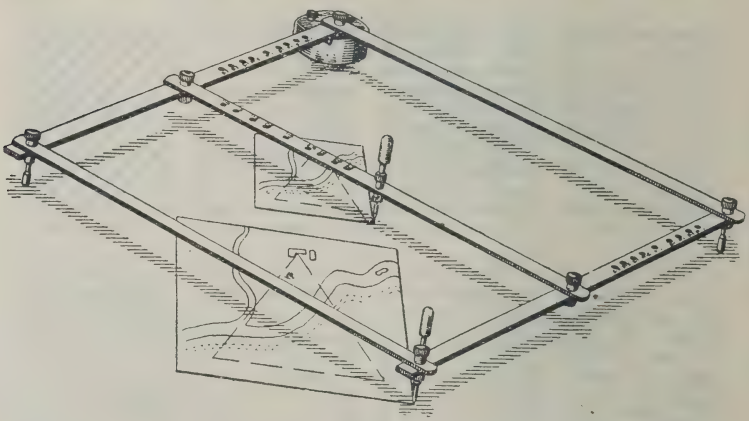


FIG. 3.

There are sixteen different ratios at which the instrument may be used.

The instrument will work quite smoothly and accurately, there being no perceptible friction or shake.

The price, complete with lead holder and tracing point, and fitted in strong cardboard box, is 25*s.*

Stanley's Patent Drawing Pen.

(*W. F. Stanley & Co., Ld.*, 286 *High Holborn, London, W.C.* 1.)

This pen (fig. 4) is made with one nib to swivel round, so that it can be opened out for

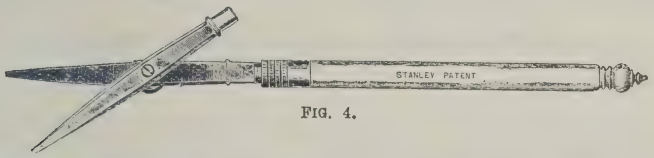


Fig. 4.

clearing, the arrangement being such that it locks back again in exact position, and the adjustment for the thickness of line is not interfered with.

A Compass Pencil-Leg Kink for Draughtsmen.

Draughtsmen who find that the jaws of the lead-holding clamp of a compass pencil-leg have become so smooth that it is difficult to tighten them sufficiently to hold the lead, can easily overcome this difficulty in the following way. First remove the clamping screw and insert a tap in place of the lead; then press the jaws together gently with the fingers and run the tap backwards and forwards five or six times; this will roughen the jaws so that they will grip the lead firmly.

(*O. T. R.*, '*Machinery.*')

A Steel Drawing-board Edge.

One of the difficulties a draughtsman quite frequently bumps up against is that of trying to draw long lines parallel to each other. When drawn, he often finds that, instead of being parallel throughout, they are from $\frac{1}{16}$ to $\frac{1}{8}$ in. farther apart at one end than the other. This condition is almost surely to be found in steam-heated offices, where the cross-grained end of the board is depended upon to make a straight edge upon which the head of the T-square is expected to slide.

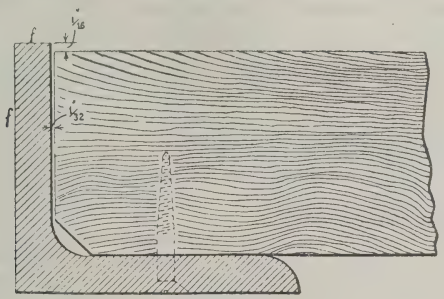


Fig. 5.

Fig. 5 shows an arrangement fitted to a board to overcome this difficulty, and found to answer the purpose admirably. The straight edge is made from a piece of angle iron 2 by 2 by $\frac{1}{4}$-in. ground or otherwise finished on the top and outside edges to ensure its being perfectly straight. The top edges project about $\frac{1}{16}$ in. above the surface of the drawing-board to give the T-square clearance above the drawing-pins when sliding up and down, and small particles of dirt are thus prevented from being ground into the paper. The angle iron is fastened to the drawing-board by two wood screws about $1\frac{1}{4}$ in. long, screwed in about 5 ins. from each end of the angle iron. The screw hole in one end should be slightly slotted to take care of swelling and shrinking of the wood.

(*W. F.*, '*Machinery.*')

SIZES OF DRAWING-PAPER.

	Ins. Ins.		Ins. Ins.		Ins. Ins.
Antiquarian .	. 52 × 31	Columbia .	. 34 × 23	Super royal .	. 27 × 19
„ extra	. 56 × 40	Elephant .	. 27½ × 22½	Royal . .	. 24 × 19
Double elephant	. 40 × 26	Double crown	. 42 × 22	Medium . .	. 22 × 17
Atlas . . .	. 33 × 26	Imperial .	. 30 × 21	Demy . .	. 20 × 15

To Trisect an Angle by Means of a Set-Square.

(*H. R. Kempe.*)

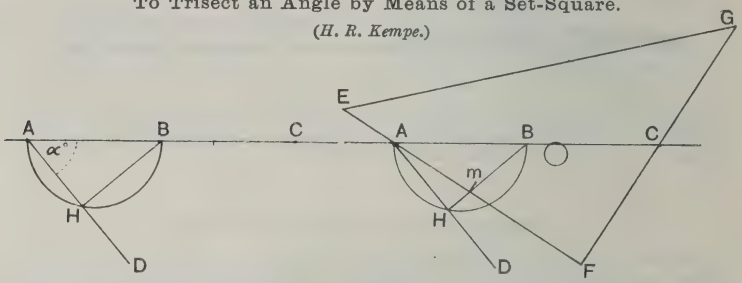


FIG. 6.

On the line AC, fig. 6, AB is a semicircle, the diameter AB being equal to BC. At A and C are inserted pins of as small a gauge as possible (fine needles are best).

Let DAB be the angle ($\alpha°$) to be trisected. Join B and H.

Drop the set-square between the pins A and C and move it about until the mark (which is in such a position on the edge EF of the square that $Fm = AB$) comes on the line HB, then the edge EF of the set-square trisects the angle.

BRITISH STANDARD ENGINEERING DRAWING OFFICE PRACTICE.*

(No. 308—1927.) (*Abstract.*)

(*Recommendations.*)

1. The following to be the standard overall sheet sizes for drawings and tracings:

72 in. × * 40 in.		40 in. × 15 in.	
60 in. × * 40 in.	* The width of rolls from which these sizes are cut.	30 in. × 20 in.	These sizes are subdivisions of the 40 × 30 in. sheets.
53 in. × * 30 in.		20 in. × 15 in.	
* 40 in. × * 30 in.		15 in. × 10 in.	
* 40 in. × 27 in.		10 in. × 7½ in.	

2. The drawing number, date, and firm's name to be placed in the right-hand bottom corner of the drawing and repeated, the same way up, in the top left-hand corner to meet the requirements of both horizontal and vertical filing.

3. The scale or scales of all drawings to be clearly indicated.

4. The standard method of projection to be first angle projection—namely, that in which each view is so placed that it represents the side of the object remote from it in the adjacent view.

6. The use of colours should be avoided as far as possible. When colours are used, centre lines to be brown, and dimension lines blue with black arrowheads and figures.

Note.—To facilitate reproduction by photo-printing brown should be used instead of red, and a mixture of Prussian blue and Chinese white should be used for the blue.

COPYING PROCESSES.

GLYCERINE-JELLY PROCESS.

(1.)

8 ozs. glycerine, 2 ozs. best glue; to the glue add 8 ozs. of water, and when dissolved stir in glycerine. Pour the mixture into a zinc dish, and as soon as it sets it is ready for use. Use aniline ink.

* By permission of the British Engineering Standards Association.

(2.)

16 ozs. glycerine, 8 ozs. golden syrup, 3 ozs. flake gelatine, 16 ozs. water. Soak gelatine in water for 10 hours; heat gently in water bath; add glycerine and golden syrup. Pour into dish. Use aniline ink.

(3.)

Gelatine, 1 lb.; water, 2 lbs.; glycerine, 5 lbs. Melt together and pour in shell dish. For the ink: aniline violet, 1 oz.; water, 5 ozs.; methylated spirit, 1 oz.

(4.)

Whiting to fill dish; glycerine in sufficient quantity to mix the whiting to the consistency of glaziers' putty. Press into dish and smooth with knife, and finally with small indiarubber squeegee. For ink use violet aniline, ½ oz.; sugar, ½ drachm; boiling water, 2 ozs.; when cold add 1 drachm each of alcohol and glycerine and 2 drops carbolic acid.

Write or draw original on dry smooth paper with a pen well filled with the aniline ink. Directly it is quite dry place it face down for 1 to 5 minutes, according to number of copies required, on composition, the surface of which must previously be moistened with water and dried with paper (newspaper will do); gently smooth down with hand to remove air bubbles. On peeling it off a negative is left on the composition. To take copies, apply to the negative sheet after sheet of any paper, smoothing them down with slight pressure, and removing them at first as quickly as possible; but as the copies become faint the paper may be left down longer on the negative, or it may be damped before putting down. The copies having been taken, sponge off the negative immediately with cold water, or tepid water if it does not come off readily. When the surface is worn and uneven, re-melt composition; a little glycerine may have to be added at each re-melting.

(5.)

The following formula for making hectograph pads is given by Mr. H. E. Smith in the 'American Engineer and Railroad Journal': Clear hide glue, 1 lb.; water, 1½ pint; glycerine, 2½ pints. The glue should be of a good quality, and the kind that comes in transparent, light-brown sheets, as the white or brown opaque glue does not give as good results. Break the glue into small pieces and soak it in the water over-night in a covered vessel. Then melt it in a water-bath, and add the glycerine, which should previously be heated to the same temperature as the melted glue. Stir only as much as is necessary to mix the glue and the glycerine, as too much stirring introduces air bubbles, which are difficult to remove. Pour the hot mixture through a cheese-cloth bag into the pans. When the pans are filled and the jelly is still quite fluid, sweep off air bubbles or impurities from the surface with the edge of a card. Let the pans stand 48 hours before using. This formula calls for much less water than usually required by other formulas, since it is preferred to secure the requisite softness by means of the glycerine, which does not evaporate and allow the pads to dry out and harden, as does the water.

In writing the original copy always use hard, glazed paper, and write with hectograph ink. In making the negative moisten the surface of the pad with a cold, wet sponge, wiping off the excess of moisture. Dry off with a newspaper, and let it stand for two minutes. Place the original face down on the pad for from one to three minutes, rubbing down to a perfect contact, and then carefully remove. In printing apply a clean sheet of any kind of paper so that it touches the hectograph at all points, rubbing as little as possible. Never use hot water for removing the negative; as soon as the copies are all made, wash off with a sponge and cold water and dry off well with a newspaper. Never let the pad stand with ink in it after the copies are made, and always keep it closed when not in use.

Blakeley Process.

The following process, devised by Mr. R. W. Blakeley, of Pendleton, Manchester, is a modification of the glycerine-jelly process, but dispenses with the glycerine-jelly.

Take a pad of 10 to 12 sheets of tissue paper, damp well and screw up for a minute between damping sheets in the ordinary letter-copying press to distribute the moisture evenly. The pad should be thoroughly damp—if too dry it will not take up the ink from the original and if too wet the ink will run.

The original is drawn on any hard-surfaced paper with aniline ink, such as violet blue copying ink, and then placed face down on the damp tissue paper pad and screwed up in the press for five minutes; this gives the negative on the damp pad, from which copies are taken by screwing up blank sheets with the negative in the press. For the first copy 20 seconds in the press is enough. For the fifth or sixth, 45 or 50 seconds is required.

As the negative pad loses moisture to every copy this damping cloth under the negative is redamped every half-dozen copies or so.

AUTOCOPYIST.

Tough paper is coated by means of a sponge, at a temperature of 120° F., with a gelatine (flake) 5 ozs.; glycerine, 5 drachms; water, 20 ozs.; and then allowed to dry.

The best paper to use is either prepared vegetable tracing-paper, or else vegetable-parchment paper.

The prepared paper is stretched tightly over a felted wooden block or board. The original is written or drawn on smooth paper with an ink composed of a saturated solution of alum tinged with aniline black, and then applied to the prepared paper as in the glycerine process. An inked roller (lithographic ink) is then passed over the surface, a sheet of paper laid on the latter, and a felted roller passed briskly over two or three times, and the paper taken off. One hundred to two hundred sharp, clear copies can thus be obtained. The prepared paper cannot be cleaned and used a second time.

CYCLOSTYLE.

A stencil is formed by writing or drawing on prepared paper with a pen fitted with a revolving perforator. Copies are taken by passing over the stencil an india-rubber scraper charged with lithographic ink thinned with lard-oil.

BLUE PHOTOGRAPHIC COPYING-PROCESS.

Draw design on thin paper. Brush paper over with a solution of (1), and dry in the dark.

(1.)

Double citrate of iron and ammonia, 44 grs.; water, 1 oz.

Expose under plan, which is pressed out flat by a sheet of plate-glass, and develop with solution (2).

(2.)

Ferrocyanide of potassium, 50 grs.; water, 1 oz.

Wash with water slightly acidulated with hydrochloric acid, and then in water alone, and dry.

This gives white lines on a blue ground, or a negative of the drawing.

The following gives blue lines on a white ground, or a positive print:—

Sensitising solution—

(1.)

Ferric chloride, 50 grs.; oxalic acid, 25 grs.; water, 1 oz.

Develop with

(2.)

Ferrocyanide of potassium, 80 grs.; water, 1 oz.

All the steps being the same as in the other process.

BLACK PHOTOGRAPHIC COPYING-PROCESS.—TO COPY DRAWINGS IN BLACK LINES ON WHITE GROUND.

(1.)

The following process has been patented by A. Colas, of Neuilly, France. Paper is prepared with the following solution:—$\frac{1}{3}$ oz. each of sulphate of iron, sesquichloride of iron, gelatine, and tartaric acid, in $10\frac{1}{2}$ ozs. of distilled water. The paper, kept flat, is stored in a dark closet. When exposed to the light beneath a transparent drawing, the parts influenced by the light lose their greenish-yellow colour. It is afterwards dipped into a bath made of 7 ozs. of gallic acid, $\frac{3}{4}$ fluid oz. of alcohol, and 1 quart of water. The greenish-yellow lines turn black at once, and the sheet requires only to be rinsed in water.

(2.)

The following is known as the 'Polygraphic' method:—A solution of $\frac{1}{3}$ oz. of aniline colour and 8 drops of glacial acetic acid in $3\frac{1}{2}$ fluid ozs. of water is used as an ink in this process. Japanese paper, a copying press, and a smooth, polished zinc plate are used for producing copies. The paper, after the corners have been moistened, is spread out upon the zinc plate. On the top of the paper is placed a linen cloth, first soaked in water and then wrung out, and on the top of this a sheet of waterproof paper. A cushion formed of ordinary paper by folding it several times together is pressed upon the zinc plate thus prepared. The side of the sheet written on is placed upon the sheet fastened on the zinc plate, and on the top of this a sheet of paper of the same size. The sheets of paper which are to receive the copy must be damped. It is now pressed: in a few minutes the impression will be ready, which can be used at once for 20 to 25 copies. After the impression is drawn off the Japanese paper is removed and the zinc plate washed and cleansed. If 2 pages of a sheet have to be copied, the pages must be enveloped in sheets of paper, which will prevent the ink from one page depositing itself on the other.

The Ferrocyanide Process.

This process, by which it is possible to make direct positive prints, was the first step taken towards important improvements in the reproduction of drawings. The ferrocyanide paper is

prepared with a sensitising solution, the nature of which varies according to the class of work to be done. The following are two formulas:

	Grammes.		Grammes.
1. Perchloride of iron	80	2. Tartaric acid	90
Gum	120	Persulphate of iron	110
Citric acid	30	Perchloride of iron	120
Water	1,000	Gum solution	1,000

This paper when prepared is of a yellow colour; when exposed to light beneath the tracing important changes take place in the sensitised surface which is not protected by the lines of the drawing. The salts of iron are reduced, the exposed surface becomes white, and as soon as the drawing appears in yellow lines, which can be ascertained by raising the back of the frame, the paper is removed and put in a ferrocyanide or potassium bath, which immediately forms Prussian blue by combining with the iron salts that are not reduced—that is to say, in those parts of the sheet that have been protected from the light. In this way the drawing is developed in a bright blue tint. The sheet is afterwards washed, and then placed in a bleaching bath made up in the proportion of 40 grammes of sulphuric acid and 1,000 grammes of water; finally, the sheet is subjected to a liberal washing, and any spots which do not appear to have been completely bleached are treated with a soft brush.

Photo-Copying.

The simplest outfit for photo-copying consists of a flat frame, a developing bath and rubber squeegee, cutting and trimming table and knife, drying racks, and some metal tubes to hold the sensitised papers. The frame (of ash or oak) has a plate-glass front, a thick sheet of felt, back boards saw cut and battened, pressure bars and springs. In each frame there should be an equal number of boards and bars, and all bars and boards should be interchangeable.

The developing bath consists of an oblong shallow tray; the larger sizes are fitted with outfall socket and plug. These baths are usually stocked (in zinc) up to 56 ins. by 36 ins.; for larger sizes, wooden baths lead-lined, or simply made water-tight and provided with a glass or slate slab at the bottom, are generally used. The bath should be fixed with a fall of about ½ in. per ft., so that the water used in developing will flow off readily.

FIG. 7.

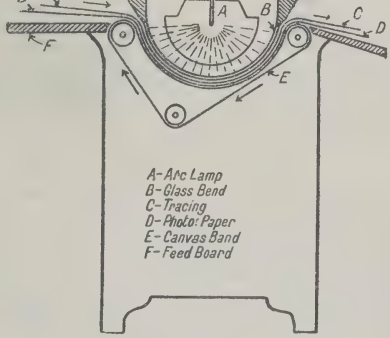


FIG. 8.

The cutting table is simply a board fitted with a strip of zinc about 6 ins. wide on the upper side, running the full length, and close to the edge. A piece of felt should be tacked to the opposite edge of the table sufficiently large to cover the whole table. This can be drawn over undeveloped prints or sensitised papers. Such a board may be very conveniently supported on trestles. It is often the practice to substitute for the trestles a cabinet of drawers, in which tracings and finished prints can be stored.

The metal tubes for holding the papers should be of tin or zinc, with sliding lids and at least twice the diameter of the roll of paper before unwrapping, otherwise, when unwrapped, it will expand and be difficult to remove. The simplest form of rack for natural drying consists of two brackets attached to a wall, about 3 ft. 6 ins. apart; clips are attached to the prints when in the bath, a rod is slipped through and the whole hung on the brackets.

A flat squeegee is frequently used for removing the superfluous moisture from the prints whilst lying in the bath. It consists of a thin strip of rubber about 10 ins. long secured in a wooden handle.

The outfit above described is suitable for offices having good daylight and no great pressure of copying work. For large and rapid outputs an electrical outfit is necessary.

Such a machine, capable of taking two tracings, is shown in fig. 7. It is made in several sizes varying from 40 ins. by 27 ins. to 60 ins. by 40 ins. The tracings are clamped closely around a glass cylinder, inside which an arc lamp is alternately raised and lowered until the prints have been fully exposed.

An improved type of copier for a considerably larger output is shown in fig. 9. It enables prints from tracings of any length up to that of a complete roll of paper (30 or 32 feet) to be produced in one piece. The paper A, fed with tracings B, travels slowly downwards between the

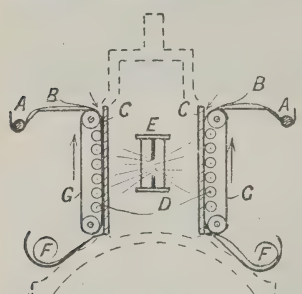


FIG. 9.

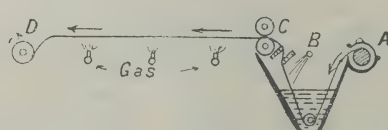


FIG. 10.

glass plate C and an endless blanket G pressed against it by the geared rollers D, passing several fixed arc lamps or one travelling one, E, after which it emerges and rolls up at F ready for trimming and washing.

Continuous printing machines, which are in two sizes, for paper either 30 or 40 ins. wide, are made single or double, the latter printing simultaneously two rolls of paper as in fig. 9. The speed can be varied from 4 feet per minute to 6 minutes per foot on either side so that fast and slow work can be done simultaneously. Fig. 10 shows a 'washing machine.' The roll of prints is placed in the trough A, passed through a running water-bath and spray B, between squeegees and rollers C, over a series of gas burners, and rolled up dry at D.

Trimming in quantities is usually effected with a guillotine for large work and a trimmer of the Merrett type for smaller sizes.

THE 'HALL-MARK' PHOTO-COPYER.

The 'Hall-Mark,' fig. 8 (p. 95), has a Hall-Thorkelin horizontal arc lamp specially designed to throw a richly actinic light downwards without causing a shadow. It is claimed for this lamp that it yields a higher percentage of actinic light per unit of current consumed than any arc lamp hitherto made, and also possesses the unique advantage that the carbons can be adjusted immediately after switching off and without waiting for the lamp to cool.

Beneath the lamp A, which travels automatically from side to side of the machine and across the width of the paper, is a curved glass plate, B, against the underside of which the tracing C, and sensitised paper D, are pressed and caused to travel by an endless canvas belt, E.

The close proximity of the arc light to the glass bend reduces the loss of light to a minimum and enables the machine to be operated at a very high rate of speed. Suitable mechanism provides for a range of from 4 ins. up to 8 ft. per minute.

The frame of the machine is of aluminium and wood for lightness, and everything, including the motor, is self-contained.

A machine taking continuous rolls of 42-in. (wide) paper occupies, including all projections, a floor space of about 5 ft. 6 ins. by 3 ft. 9 ins.

Tracing.

Place the sheet of paper over the drawing to be copied, and lightly rub it over with a ball of wool saturated with benzine; this renders the paper transparent temporarily, when the tracing can readily be made. The benzine by evaporation leaves the paper opaque as before. The benzine must be absolutely pure to secure good results.

SECTION IV

HEAT — SOUND — OPTICS (pp. 99-120)

SECTION IV

HEAT—SOUND—OPTICS.

HEAT.

Temperature Conversion Tables.

TABLE FOR THE CONVERSION OF DEGREES ON THE CENTIGRADE THERMOMETER INTO THOSE OF FAHRENHEIT'S SCALE.

°Cent.	° Fahr.	°Cent.	° Fahr.	°Cent.	° Fahr.	°Cent.	° Fahr.	°Cent.	° Fahr.	°Cent.	° Fahr.
−100	−148·0	− 55	− 67·0	−10	14·0	35	95·0	80	176·0	125	257·0
− 99	−146·2	− 54	− 65·2	− 9	15·8	36	96·8	81	177·8	126	258·8
− 98	−144·4	− 53	− 63·4	− 8	17·6	37	98·6	82	179·6	127	260·6
− 97	−142·6	− 52	− 61·6	− 7	19·4	38	100·4	83	181·4	128	262·4
− 96	−140·8	− 51	− 59·8	− 6	21·2	39	102·2	84	183·2	129	264·2
− 95	−139·0	− 50	− 58·0	− 5	23·0	40	104·0	85	185·0	130	266·0
− 94	−137·2	− 49	− 56·2	− 4	24·8	41	105·8	86	186·8	131	267·8
− 93	−135·4	− 48	− 54·4	− 3	26·6	42	107·6	87	188·6	132	269·6
− 92	−133·6	− 47	− 52·6	− 2	28·4	43	109·4	88	190·4	133	271·4
− 91	−131·8	− 46	− 50·8	− 1	30·2	44	111·2	89	192·2	134	273·2
− 90	−130·0	− 45	− 49·0	0	32·0	45	113·0	90	194·0	135	275·0
− 89	−128·2	− 44	− 47·2	+ 1	33·8	46	114·8	91	195·8	136	276·8
− 88	−126·4	− 43	− 45·4	2	35·6	47	116·6	92	197·6	137	278·6
− 87	−124·6	− 42	− 43·6	3	37·4	48	118·4	93	199·4	138	280·4
− 86	−122·8	− 41	− 41·8	4	39·2	49	120·2	94	201·2	139	282·2
− 85	−121·0	− 40	− 40·0	5	41·0	50	122·0	95	203·0	140	284·0
− 84	−119·2	− 39	− 38·2	6	42·8	51	123·8	96	204·8	141	285·8
− 83	−117·4	− 38	− 36·4	7	44·6	52	125·6	97	206·6	142	287·6
− 82	−115·6	− 37	− 34·6	8	46·4	53	127·4	98	208·4	143	289·4
− 81	−113·8	− 36	− 32·8	9	48·2	54	129·2	99	210·2	144	291·2
− 80	−112·0	− 35	− 31·0	10	50·0	55	131·0	100	212·0	145	293·0
− 79	−110·2	− 34	− 29·2	11	51·8	56	132·8	101	213·8	146	294·8
− 78	−108·4	− 33	− 27·4	12	53·6	57	134·6	102	215·6	147	296·6
− 77	−106·6	− 32	− 25·6	13	55·4	58	136·4	103	217·4	148	298·4
− 76	−104·8	− 31	− 23·8	14	57·2	59	138·2	104	219·2	149	300·2
− 75	−103·0	− 30	− 22·0	15	59·0	60	140·0	105	221·0	150	302·0
− 74	−101·2	− 29	− 20·2	16	60·8	61	141·8	106	222·8	151	303·8
− 73	− 99·4	− 28	− 18·4	17	62·6	62	143·6	107	224·6	152	305·6
− 72	− 97·6	− 27	− 16·6	18	64·4	63	145·4	108	226·4	153	307·4
− 71	− 95·8	− 26	− 14·8	19	66·2	64	147·2	109	228·2	154	309·2
− 70	− 94·0	− 25	− 13·0	20	68·0	65	149·0	110	230·0	155	311·0
− 69	− 92·2	− 24	− 11·2	21	69·8	66	150·8	111	231·8	156	312·8
− 68	− 90·4	− 23	− 9·4	22	71·6	67	152·6	112	233·6	157	314·6
− 67	− 88·6	− 22	− 7·6	23	73·4	68	154·4	113	235·4	158	316·4
− 66	− 86·8	− 21	− 5·8	24	75·2	69	156·2	114	237·2	159	318·2
− 65	− 85·0	− 20	− 4·0	25	77·0	70	158·0	115	239·0	160	320·0
− 64	− 83·2	− 19	− 2·2	26	78·8	71	159·8	116	240·8	161	321·8
− 63	− 81·4	− 18	− 0·4	27	80·6	72	161·6	117	242·6	162	323·6
− 62	− 79·6	− 17	+ 1·4	28	82·4	73	163·4	118	244·4	163	325·4
− 61	− 77·8	− 16	3·2	29	84·2	74	165·2	119	246·2	164	327·2
− 60	− 76·0	− 15	5·0	30	86·0	75	167·0	120	248·0	165	329·0
− 59	− 74·2	− 14	6·8	31	87·8	76	168·8	121	249·8	166	330·8
− 58	− 72·4	− 13	8·6	32	89·6	77	170·6	122	251·6	167	332·6
− 57	− 70·6	− 12	10·4	33	91·4	78	172·4	123	253·4	168	334·4
− 56	− 68·8	− 11	12·2	34	93·2	79	174·2	124	255·2	169	336·2

TABLE FOR THE CONVERSION OF DEGREES (*continued*).

°Cent.	° Fahr.	°Cent.	° Fahr.	°Cent.	° Fahr.	°Cent.	° Fahr.	°Cent.	° Fahr.	°Cent.	° Fahr.
170	338·0	210	410·0	250	482·0	290	554·0	330	626·0	440	824
171	339·8	211	411·8	251	483·8	291	555·8	331	627·8	450	842
172	341·6	212	413·6	252	485·6	292	557·6	332	629·6	460	860
173	343·4	213	415·4	253	487·4	293	559·4	333	631·4	470	878
174	345·2	214	417·2	254	489·2	294	561·2	334	633·2	480	896
175	347·0	215	419·0	255	491·0	295	563·0	335	635·0	482	900
176	348·8	216	420·8	256	492·8	296	564·8	336	636·8	490	914
177	350·6	217	422·6	257	494·6	297	566·6	337	638·6	500	932
178	352·4	218	424·4	258	496·4	298	568·4	338	640·4	538	1000
179	354·2	219	426·2	259	498·2	299	570·2	339	642·2	593	1100
180	356·0	220	428·0	260	500·0	300	572·0	340	644·0	600	1112
181	357·8	221	429·8	261	501·8	301	573·8	341	645·8	649	1200
182	359·6	222	431·6	262	503·6	302	575·6	342	647·6	700	1292
183	361·4	223	433·4	263	505·4	303	577·4	343	649·4	704	1300
184	363·2	224	435·2	264	507·2	304	579·2	344	651·2	760	1400
185	365·0	225	437·0	265	509·0	305	581·0	345	653·0	800	1472
186	366·8	226	438·8	266	510·8	306	582·8	346	654·8	815	1500
187	368·6	227	440·6	267	512·6	307	584·6	347	656·6	871	1600
188	370·4	228	442·4	268	514·4	308	586·4	348	658·4	900	1652
189	372·2	229	444·2	269	516·2	309	588·2	349	660·2	926	1700
190	374·0	230	446·0	270	518·0	310	590·0	350	662·0	982	1800
191	375·8	231	447·8	271	519·8	311	591·8	351	663·8	1000	1832
192	377·6	232	449·6	272	521·6	312	593·6	352	665·6	1037	1900
193	379·4	233	451·4	273	523·4	313	595·4	353	667·4	1093	2000
194	381·2	234	453·2	274	525·2	314	597·2	354	669·2	1100	2012
195	383·0	235	455·0	275	527·0	315	599·0	355	671·0	1148	2100
196	384·8	236	456·8	276	528·8	316	600·8	356	672·8	1200	2192
197	386·6	237	458·6	277	530·6	317	602·6	357	674·6	1300	2372
198	388·4	238	460·4	278	532·4	318	604·4	358	676·4	1400	2552
199	390·2	239	462·2	279	534·2	319	606·2	359	678·2	1500	2732
200	392·0	240	464·0	280	536·0	320	608·0	360	680·0	1600	2912
201	393·8	241	465·8	281	537·8	321	609·8	370	698	1700	3092
202	395·6	242	467·6	282	539·6	322	611·6	371	700	1800	3272
203	397·4	243	469·4	283	541·4	323	613·4	380	716	1900	3452
204	399·2	244	471·2	284	543·2	324	615·2	390	734	2000	3632
205	401·0	245	473·0	285	545·0	325	617·0	400	752	2100	3812
206	402·8	246	474·8	286	546·8	326	618·8	410	770	2200	3992
207	404·6	247	476·6	287	548·6	327	620·6	420	788	2300	4172
208	406·4	248	478·4	288	550·4	328	622·4	426	800	2400	4352
209	408·2	249	480·2	289	552·2	329	624·2	430	806	2500	4532

TABLE FOR THE CONVERSION OF DEGREES ON THE FAHRENHEIT THERMOMETER INTO THOSE OF THE CENTIGRADE SCALE.

°Fahr.	° Cent.	°Fahr.	° Cent.	°Fahr.	° Cent.	°Fahr.	° Cent.	°Fahr.	° Cent.	°Fahr.	° Cent.
212	100·0	193	89·4	174	78·8	155	68·3	136	57·7	117	47·2
211	99·4	192	88·8	173	78·3	154	67·7	135	57·2	116	46·6
210	98·8	191	88·3	172	77·7	153	67·2	134	56 6	115	46·1
209	98·3	190	87·7	171	77·2	152	66·6	133	56·1	114	45·5
208	97·7	189	87·2	170	76·6	151	66·1	132	55·5	113	45·0
207	97·2	188	86·6	169	76·1	150	65·5	131	55·0	112	44·4
206	96·6	187	86·1	168	75·5	149	65·0	130	54·4	111	43·8
205	96·1	186	85·5	167	75·0	148	64·4	129	53·8	110	43·3
204	95·5	185	85·0	166	74·4	147	63·8	128	53·3	109	42·7
203	95·0	184	84·4	165	73·8	146	63·3	127	52·7	108	42·2
202	94·4	183	83·8	164	73·3	145	62·7	126	52·2	107	41·6
201	93·8	182	83·3	163	72·7	144	62·2	125	51·6	106	41·1
200	93·3	181	82·7	162	72·2	143	61·6	124	51·1	105	40·5
199	92·7	180	82·2	161	71·6	142	61·1	123	50·5	104	40·0
198	92·2	179	81·6	160	71·1	141	60·5	122	50·0	103	39·4
197	91·6	178	81·1	159	70·5	140	60·0	121	49·4	102	38·8
196	91·1	177	80·5	158	70·0	139	59·4	120	48·8	101	38·3
195	90·5	176	80·0	157	69·4	138	58·8	119	48·3	100	37·7
194	90·0	175	79·4	156	68·8	137	58·3	118	47·7	99	37·2

TABLE FOR THE CONVERSION OF DEGREES (*continued*).

°Fahr.	°Cent.	°Fahr.	°Cent.	°Fahr.	°Cent.	°Fahr.	°Cent.	°Fahr.	°Cent.	°Fahr.	°Cent.
98	36·6	76	24·4	53	11·6	30	− 1·1	8	−13·3	−14	−25·5
97	36·1	75	23·8	52	11·1	29	− 1·6	7	−13·8	−15	−26·1
96	35·5	74	23·2	51	10·5	28	− 2·2	6	−14·4	−16	−26·6
95	35·0	73	22·7	50	10·0	27	− 2·7	5	−15·0	−17	−27·2
94	34·4	72	22·2	49	9·0	26	− 3·3	4	−15·5	−18	−27·7
93	33·8	71	21·6	48	8·8	25	− 3·8	3	−16·1	−19	−28·3
92	33·3	70	21·1	47	8·3	24	− 4·4	2	−16·6	−20	−28·8
91	32·7	69	20·5	46	7·7	23	− 5·0	1	−17·2	−21	−29·4
90	32·2	68	20·0	45	7·2	22	− 5·5	0	−17·7	−22	−30·0
89	31·6	67	19·4	44	6·6	21	− 6·1	− 1	−18·3	−23	−30·5
88	31·1	66	18·9	43	6·1	20	− 6·6	− 1	−18·8	−24	−31·1
87	30·5	65	18·3	42	5·5	19	− 7·2	− 3	−19·4	−25	−31·6
87	30·5	64	17·7	41	5·0	18	− 7·7	− 4	−20·0	−26	−32·2
86	30·0	63	17·2	40	4·4	17	− 8·3	− 5	−20·5	−27	−32·7
85	39·4	62	16·6	38	3·3	16	− 8.8	− 6	−21·1	−28	−33·3
84	28·8	61	16·1	37	2·7	15	− 9·5	− 7	−21·6	−29	−33·8
83	28·3	60	15·5	36	2·2	14	−10·0	− 8	−22·2	−30	−34·4
82	27·7	59	15·0	35	1·6	13	−10·5	− 9	−22·7	−31	−35·0
81	27·2	58	14·4	34	1·1	12	−11·1	−10	−23·3	−32	−35·5
80	26·6	57	13·8	33	0·5	11	−11·6	−11	−23·8	−33	−36·1
79	26·1	56	13·3	32	0·0	10	−12·2	−12	−24·4	−34	−36·6
78	25·5	55	12·7	31	−0·5	9	−12·7	−13	−25·0	−35	−37·2
77	25·0	54	12·2								

See also next page.

Miscellaneous Temperatures.

Description.	°F.	°C.	Description.	°F.	°C.
In the Bessemer furnace	4000	2204	A comfortable room	70	21·1
Puddling furnace	3500	1927	Mean temperature of ocean	62	16·6
Cupola	3000	1649	1 alcohol, 1 water freezes	− 7	−21·6
Heat of common fire	1100	593·3	Mean temperature of poles	− 13	−25·0
Red heat in daylight	1070	571·1	Interstellar space	Abs. 0	
Iron red in dark	752	400·0	Greatest natural cold	− 56	−48·5
Mean temp. of earth	50	10·0	Vinous fermentation	65	18·3
„ „ torrid zone	75	23·8	Acetous fermentation begins	78	25·5
„ „ temp. zone	50	10·0	Acetification ends	88	31·1
„ „ polar region	20	6·6	Phosphorus takes fire	112	44·4
Highest temp. of wind	117	47·2	Greatest artificial cold produced (*Kamerlingh Onnes*)	−457·9	−272·18
Temp. of human blood	98	36·6			

Ignition Temperatures.

Name of Substance.	°F.	°C.	Name of Substance.	°F.	°C.
Phosphorus, transparent	112	44·4	Rifle-powder	550	287·8
Bisulphide of carbon	300	143·3	Forced gunpowder	560	293·3
Fulminating powder	370	187·8	Picrate powder for torpedoes	570	299·0
Fulminate of mercury	390	198·9	Charcoal from willow wood	660	349·0
Gun-cotton	430	221·1	Picrate powder for cannon	720	382·2
Nitro-glycerine	490	254·4	Very dry pine wood	800	426·7
Phosphorus, amorphous	500	260·0	Dry oak wood	900	480·2

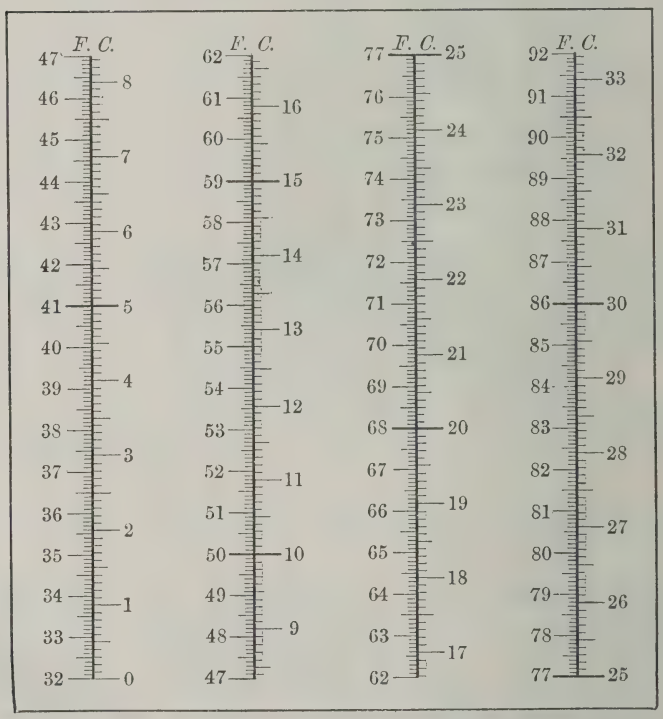
See also page 117, Flash Temperatures.

Values of Degrees.

°C. contain °F.			°F. contain °C.		
1 = 1·8	4 = 7·2	7 = 12·6	1 = 0·56	4 = 2·22	7 = 3·89
2 = 3·6	5 = 9·0	8 = 14·4	2 = 1·11	5 = 2·78	8 = 4·44
3 = 5·4	6 = 10·8	9 = 16·2	3 = 1·67	6 = 3·33	9 = 5·00

$$C^\circ = \frac{5}{9}(F^\circ - 32^\circ) = \frac{F^\circ - 32^\circ}{1{\cdot}8}. \qquad F^\circ = \frac{9}{5}C^\circ + 32^\circ = 1{\cdot}8\,C^\circ + 32^\circ.$$

Thermometer Scale Conversion Chart.



(R. E. Gould, American Bureau of Standards.)

Fig. 1.

Melting Points of Chemical Elements.

The following table gives the melting points of chemical elements and is reproduced from a circular issued by the U.S. Bureau of Standards. The value of the melting points used by the Bureau as standard temperatures for the calibration of thermometers and pyrometers are indicated in italics. The other values have been assigned after a careful survey of all the available data.

Element.	°F.	C.	Element.	°F.	°C.
Helium	<−456	<−271	Barium	1560	850
Hydrogen	−434	−259	Praseodymium	1725	940?
Neon	−423	−253?	Germanium	1756	958
Fluorine	−369	−223	*Silver*	1761	960·5
Oxygen	−360	−218	Glucinum	—	>Ag
Nitrogen	−346	−210	Radium	—	?
Argon	−306	−188	*Gold*	1945·5	1063·0
Krypton	−272	−169	*Copper*	1981·5	1083·0
Xenon	−220	−140	Manganese	2237	1225
Chlorine	−150·5	−101·5	Yttrium	—	?
Mercury	− 37·7	− 38·7	Samarium	2370–2550	1300–1400
Bromine	+ 18·9	− 7·3	Scandium	—	?
Cæsium	79	26	Silicon	2588	1420
Gallium	86	30	*Nickel*	2646	1452
Rubidium	100	38	Cobalt	2714	1490
Phosphorus	111·4	44	Chromium	2750	1510
Potassium	144	62·3	*Iron*	2768	1520
Sodium	207·5	97·5	*Palladium*	2820	1549
Iodine	236·5	113·5	Zirconium	3100	1700?
Sulphur	S_{I} 235·0	112·8	Thorium	>3090 <Pt.	>1700 <Pt.
	S_{II} 246·6	119·2			
	S_{III} 224·2	106·8	Vanadium	3150	1730?
Indium	311	155	*Platinum*	3191	1755
Lithium	367	186	Beryllium	>3270	>1800?
Selenium	422–428	217–220	Ytterbium	—	?
Tin	449·4	231·9	Titanium	3450	1900?
Bismuth	520	271	Rhodium	3525	1940
Thallium	576	302	Ruthenium	>3550	>1950
Cadmium	609·6	320·9	Columbium (Niobium)	4000	2200?
Lead	621·1	327·4	Boron	4000–4500	2200–2500
Zinc	786·9	419·4	Iridium	4170	2300?
Tellurium	846	452	Uranium	—	?
Antimony	1166	630·0	Molybdenum	4500	2500?
Cerium	1184	640	Osmium	4900	2700?
Magnesium	1204	651	Tantalum	5160	2850
Aluminium	1217·7	658·7	*Tungsten*	5430	3000
Calcium	1490	810	Carbon	>6500 for p=1 At.	>3600 for p=1 At
Lanthanum	1490	810?			
Strontium	—	>Ca<Ba?			
Neodymium	1544	840?			
Arsenic	1560	850?			

? Volatilise, but do not melt.

Melting Points of Various Substances.

Solids.	°F.	°C.	Solids.	°F.	°C.
Litharge	1739°	954°	Ice of milk	30°	− 1 1°
Slag of copper	2462	1350	„ sea-water	28	− 2·2
„ tin	2402	1318	„ vinegar	28	− 2·2
Puddle slag	2606	1430	„ strong wine	20	− 6·6
Sodic nitrate	591	310·5	„ „ brandy	7	− 13·9
Potassium nitrate	642	339	Ice of oil of turpentine	14	− 1
Beeswax (white)	155	68	1 snow, 1 salt	0·00	− 17.8
„ (yellow)	142	61	1 alcohol, 1 water	− 7	− 21·6
Spermaceti	142	61	Cyanogen	− 30	− 34·4
Yellow wax	143·6	62	Oil of vitriol	− 30	− 35
Olive oil	92	33	Sulphuric ether	− 47	− 43·9
Tallow	36	2·2	Sulphurous acid	− 105	− 76
Ice of water	32	0·000	Nitrous oxide	− 150	−101

MELTING POINTS OF ALLOYS.

Bismuth.	Lead.	Tin.	Melting Point.		Bismuth.	Lead.	Tin.	Melting Point.	
			° F.	° C.				° F.	° C.
50	31·2	18·8	201	94	16·6	33·2	50·2	316	158
47	35·5	17·7	208	98	16	36	48	311	155
42·1	42·1	15·8	226	108	15·3	38·8	45·9	309	154
40	40	20	235	113	14·8	40·2	45	307	153
36·5	36·5	27	243	117	14	43	43	309	154
33·3	33·3	33·3	253	123	13·7	44·8	41·5	320	160
30·8	38·4	30·8	266	130	13·3	46·6	40·1	329	165
28·5	43	28·5	270	132	12·8	49	38·2	342	172
25	50	25	300	149	12·5	50	37·5	352	178
23·5	47	29·5	304	151	11·7	46·8	41·5	333	167
22·2	44·4	33·4	289	143	11·4	45·6	43	329	165
21	42	57	289	143	11·2	44·4	44·4	320	160
20	40	40	293	145	10·8	43·2	46	318	159
19	38	43	298	148	10·5	42	47·5	320	160
18·1	36·2	45·7	304	151	10·2	41	49·8	322	161
17·3	34·6	48·1	311	155	10	40	50	324	162

The addition of cadmium gives alloys of still lower melting point. The following table includes the best known of these :—

Alloy.	Cadmium.	Lead.	Tin.	Bismuth.	Melting Point.	
					° F.	° C.
Fusible alloy . .	12·5	25	12·5	50	149	65
Lipowitz's alloy .	10	26·6	13·3	50·1	158	70
Woods' alloy . .	15·4	30·8	15·4	38·4	160	71
Fusible alloy . .	34·5	27·5	10	27·5	167	75
" " . .	6·2	34·5	9·3	50	171	77
" " . .	25	25	50	—	187	86
" " . .	16·6	—	33·3	50·1	203	95
" " . .	11·1	—	33·3	55·6	203	95
" " . .	25	—	25	50	203	95

(*Parkes and Martin.*)

The following are the melting points in C.° on the thermo-dynamic scale used by the Washington Bureau of Standards as standard temperatures in the standardisation of thermometers and pyrometers :—Mercury 38·9°, tin 231·9°, cadmium 320·9°, lead 327·4°, zinc 419·4°, antimony 630·0°, aluminium 658·7°, silver 960·5°, gold 1061·0°, copper 1083·0°, nickel 1452°, iron 1530°, palladium 1549°, platinum 1755°, tungsten 3000°. At temperatures of 1000° the uncertainty is of the order 0·1°, for platinum 5°, and for tungsten 100°. A further table of other standard temperatures is given in which the following are included :—Normal boiling point of oxygen 183·0°, sublimation of carbon dioxide in an inert liquid 78·5°, normal boiling points of water 100°, naphthalene 217·96°, benzo-phenone 305·9°, sulphur 444·6°, freezing point of sodium chloride 801°.

In the *Physical Review* the melting-point in C.° of rhodium is estimated to be at 1907° or 1968°, and that of iridium at 2292° or 2388°. That of silicon is placed at 1434° or 1452°, and that of palladium at 1548° or 1576°. The reason for two figures being given in each case is due to the fact that the calculations are based on different estimates of the melting-point of platinum, which has been placed at 1745° by some experimenters, and at 1789° by others.

MELTING POINTS OF REFRACTORY MATERIALS.*

The melting-points in C.° of various samples of fire-brick, as determined in an electric vacuum furnace, the temperature being measured with an optical pyrometer, were as follows :—Kaolin, 1740° ; pure alumina, 2010° ; pure silica, 1600° ; bauxite, 1820° ; bauxite clay, 1795° ; chromite, 2180° ; magnesite brick, 2165°. (*W. M. Barr, in 'Industrial Engineering.'*)

* See also 'Refractory Materials,' page 430.

Colour of Different Temperatures.

Colour.	F.°	C.°	Colour.	F.°	C.°	Colour.	F.°	C.°
Faint Red .	960	516	Bright Cherry Red	1830	999	Bright White Heat . .	2550	1399
Dull Red .	1290	699	Orange . .	2010	1099	Brilliant White Heat . .	2750	1510
Brilliant Red	1470	743	Bright Orange .	2190	1199			
Cherry Red .	1650	899	White Heat .	2370	1299			

(*Becquerel.*)

Freezing Mixtures.

Mixture.	Reduction in Temperature.
1 part nitre 1 part sal ammoniac 2 parts water	40° F.; 4·4° C.
Equal parts crystallised nitrate of ammonia and water	46° F.; 7·7° C.
Equal parts crystallised nitrate of ammonia (powdered), crystallised carbonate of soda (powdered), and water	57° F.; 13·8° C.
5 parts hydrochloric acid 8 parts crystallised sulphate of soda . .	50° F.; 10° C.
2 parts powdered ice or fresh snow . . . . 1 part common salt .	a temperature of −4° F.; −20° C.
3 parts crystallised chloride of calcium (cooled to 32°) . . . . 2 parts fresh snow . .	a temperature of −50° F.; −45·6° C.

Non-freezing Mixtures.

Common Salt (Sodium Chloride).				Commercial Calcium Chloride.			
Lbs. per Gal.	Freezing Point, F.°	Lbs. per Gal.	Freezing Point, F.°	Lbs. per Gal.	Freezing Point, F.°	Lbs. per Gal.	Freezing Point, F.°
½	24 above zero	2	6 above zero	½	29 above zero	3½	6 below zero
1	18 ,,	2½	3 ,,	1	27 ,,	4	17 ,,
1½	15 ,,	2½	1 ,,	1½	23 ,,	4½	27 ,,
1½	12 ,,	3	3 below zero	2	18 ,,	5	39 ,,
1¼	9 ,,	3½	8 ,,	2½	4 ,,	5½	54 ,,

The salt should be thoroughly dissolved, or the results will not be satisfactory. Calcium chloride is said to be superior to sodium chloride in that it does not corrode steel tanks and barrel hoops. Where calcium chloride is used, the wooden barrels should first be well coated inside with asphaltum, or a mixture of crude paraffin and resin, to prevent the shrinking of the staves and subsequent leakage.

Glycerine is recommended for use in chemical fire extinguishers. It has no effect upon metals, but has a tendency to disintegrate rubber. It has been stated, however, that the continued use of glycerine in water renders it liable to a decomposition that would develop compounds having a corrosive action on metals. The following proportions may be used :—

Glycerine, pounds per gallon . .	3½	5½	(One gall. of glycerine
Temp. solution will withstand . .	+ 10° F.	− 10° F.	weighs 10½ lbs.)

Denatured alcohol has advantages over both glycerine and calcium chloride, in water solutions, as it has no injurious effect on either metal or rubber. A solution of about 50 per cent. is inflammable, but it would rarely be necessary to have a solution of over 30 per cent.

A 20 per cent. solution freezes at + 10° F. | A 40 per cent. solution freezes at − 20° F.
30 ,, ,, ,, − 5° F. | 50 ,, ,, ,, − 35° F.

Freezing Temperatures.

Sea water, −2·5° C., 27·5° F.; Mercury, −38·8° C., −37·8° F.; Acetone, −95° C. −139° F.
Pratt's petrol spirit, −128° C., −198° F.

Temperature of the Atmosphere

diminishes according to the height above the earth's surface. In the Temperate Zone a diminution of 1° C. corresponds in the mean to an ascent of 180 yards. As the mean of a series of observations made during balloon ascents, a diminution of 1° C. corresponded to an increase in height of 232 yards.

According to Kropotkin, the temperature above the earth's surface decreases 3° to 5°F. for each 1,000 ft. The average temperature at an altitude of 20,000 ft. is 13° below 0° F.; at 25,000 ft. it is 35° below 0° F.; these low temperatures prevail all the year round. The ratio of decrease in temperature, which is 3° F. for each 1,000 ft. at the lower strata, is twice as great at the highest levels, but varies from summer to winter to the extent of about 20° F. (See also page 1181.)

The Mean Daily Temperature is that obtained by adding together twenty-four hourly observations and dividing by 24, the temperature being taken of the air, and not of the ground. The temperature of a month is the mean of those of thirty days, and the temperature of the year is the mean of those of twelve months. The mean temperature of London is 8·28° C., or 46·9° F.; the mean in summer is 62° F., and in winter 40° F.

Temperature of Seas and Lakes.

The temperature of the sea under the Torrid Zone is always about 26° to 27° C. at the surface; it diminishes as the depth increases, and in temperate as well as in tropical regions the temperature of the sea at great depths is between 2·5° and 3·5° C. The average winter temperature of the sea round the coast of England is higher than that of the land. The mean annual temperature of the surface of the sea round England is 49° F.; the mean surface temperature of the Indian Ocean is 89° F., and of the Red Sea 94° F. The temperature of lakes at their surface may be 20° to 25° C. in summer; the temperature of the bottom, 4° C., which is that of the maximum density of water. The Great Geyser, in Iceland, at a depth of 66 feet, has a temperature of 124° C.

Temperature of the Earth.

The average increase in temperature is 1° C. for every 27·4 metres of descent, which is at the rate of 100° F. per mile. Cordier says 1° C. for every 25 metres, or 1° F. in 45 feet.

Another statement is that the increase in temperature of the earth's strata shows a mean average for the entire earth surface of 1° F. for every 56 ft. of depth.

Specific Heat.

M = weight of body. T = temperature of body. C = specific heat of body.
m = weight of cold water. t = temperature of cold water.
θ = temperature of water after body has been plunged into it.

$$C = \frac{m(\theta - t)}{M(T - \theta)}$$

Specific Heat of Substances at Ordinary Temperatures.

Element.	Specific Heat of Equal Weights.	Element.	Specific Heat of Equal Weights.	Element.	Specific Heat of Equal Weights.
Diamond	·1468	Cast-iron (white)	·1298	Palladium	·0523
Graphite	·2018	„ (grey)	·1216	Rhodium	·0580
Wood charcoal	·2415	Cast-steel (hard)	·1185	Osmium	·0306
Coke	·2000	„ (soft)	·1166	Iridium	·0326
Average coal	·2412	Mild steel	·1158	Iodine	·0541
Clinkers	·1674	Wrought-iron	·1146	Bromine (solid)	·0843
Petroleum	·4340	Nickel	·1086	„ (liquid)	·1060
Silicon (fused)	·1750	Cobalt	·1070	Potassium	·1696
„ (crystallised)	·1767	Manganese	·1217	Sodium	·2934
Boron (crystallised)	·2500	Tin	·0562	Lithium	·9408
Firebrick	·2000	Tungsten	·0334	Phosphorus	·1887
Sulphur (native)	·1776	Molybdenum	·0722	Arsenic	·0814
„ (purified)	·2026	Copper	·0965	Antimony	·0508
Selenium	·0837	Gunmetal	·0952	Bismuth	·0308
Tellurium	·0474	Brass	·0940	Thallium	·0326
Magnesium	·2499	Lead	·0314	Silver	·0570
Zinc	·0956	Mercury (solid)	·0319	Gold	·0324
Cadmium	·0567	„ (liquid)	·0333	Water	1·0000
Aluminium	·2143	Platinum	·0324	Ice	·5040

Specific Heat of Gases at High Temperatures.

$CO_2 = 0{\cdot}1477 + 0{\cdot}000176t.$ | $N = 0{\cdot}170 + 0{\cdot}0000872t.$
$H_2O = 0{\cdot}3211 + 0{\cdot}000219t.$ | $O = 0{\cdot}1488 + 0{\cdot}0000763t.$

Where t = Centigrade temperature.

(*Mallard and Le Chatelier.*)

SPECIFIC HEAT OF GASES PER LB. AT HIGH TEMPERATURES.

(*Based on Mallard and Le Chatelier's results.*)

Temperature.		CO_2.	CO & N.	O.	H.
F.°	C.°				
200	932	·198	·241	·211	3·37
400	204	·210	·245	·214	3·41
600	316	·222	·247	·216	3·45
800	427	·235	·250	·218	3·49
1,000	538	·247	·252	·220	3·53
1,200	649	·260	·255	·223	3·57
1,400	760	·272	·257	·225	3·60
1,600	871	·285	·260	·227	3·64
1,800	981	·297	·261	·230	3·67
2,000	1,093	·310	·265	·231	3·70

(*G. H. Gill.*)

Latent Heat.

Equal weights of water, one quantity at 0° C., and the other at 100° C., when mixed have a temperature of 50° C.; but equal weights of ice and water, the former at 0° C., and the latter at 100° C., when mixed have a temperature of 10·6° C.

The conversion of a cube of ice 3-feet side into water at 0° would absorb the whole heat emitted by the combustion of a bushel of coal.

Absolute Zero of Temperature.

458° below Fahr. zero, or 272° below Cent. zero.

Heat Generated by the Combustion of Carbon.

HEAT GENERATED BY THE COMBUSTION OF ONE POUND OF CARBON.

State of Carbon.	Product of Combustion.	British Thermal Units per Pound.
Diamond	Carbon monoxide	3,915
,,	,, dioxide	14,146
Graphite	,, ,,	14,222
Amorphous	,, monoxide	4,415
,,	,, dioxide	14,647
Gaseous	,, monoxide	10,232
,,	,, dioxide	20,463
2⅓ lb. Carbon monoxide	,, ,,	10,231

HEAT ABSORBED IN THE METAMORPHIC CONVERSIONS OF CARBON.

State of Carbon.	Product of Metamorphism.	British Thermal Units per Pound.
Diamond	Gas	6,316
Graphite	,,	6,241
Amorphous	,,	5,817
Diamond	Amorphous	499
,,	Graphite	74·7
Graphite	Amorphous	424

HEAT UNIT TABLE.

(Compiled by Dr. F. L. Slocum, Pittsburgh.)

Name.	Molecular Symbol.	Atomic Weight.	Molecular Weight.	Products of Combustion.	Specific Gravity. Hydrogen = 2.	Specific Gravity. Air = 1.	Weight of 1 Litre in grams.	Weight of 1 cu. ft. in grains.	Weight of 1 cu. ft. in lbs.	Calories C–K.	Heat-Value. Carbon = 1.	Volume-Calorie. Hydrogen = 1.
Hydrogen	H_2	1	2	H_2O	2·000	0·06925	0·08955	39·1263	0·00559	34217·5	4·23	1·00
Marsh Gas	CH_4	—	15·97	H_2O–CO_2	15·974	0·55300	0·71506	312·4450	0·44640	13244·0	1·63	3·09
Carbon Monoxide	CO	—	27·93	CO_2	27·937	0·96715	1·25058	546·4397	0·07806	2441·7	0·302	1·00
Acetylene	C_2H_2	—	25·94	H_2O–CO_2	25·947	0·89829	1·16148	507·5338	0·07255	11923·0	1·48	4·53
Aethylene	C_2H_4	—	27·94	H_2O–CO_2	27·947	0·96749	1·25103	546·6318	0·07809	11884·0	1·47	4·86
Aethane	C_2H_6	—	29·94	H_2O–CO_2	29·947	1·03675	1·34058	585·7637	0·08368	12347·0	1·53	5·41
Propylene	C_3H_6	—	41·91	H_2O–CO_2	41·921	1·45124	1·87654	819·9506	0·11713	11731·0	1·45	7·21
Butylene	C_4H_8	—	55·89	H_2O–CO_2	55·894	1·93488	2·50190	1093·2072	0·15617	11619·0	1·44	9·51
Allylene	C_3H_4	—	39·92	H_2O–CO_2	39·921	1·38194	1·78692	780·7961	0·11154	11690·0	1·45	6·83
Benzole	C_6H_6	—	77·82	H_2O–CO_2	77·822	2·69463	3·48429	1522·4659	0·21749	10102·0	1·25	11·51
Napthalene	$C_{10}H_8$	—	127·70	H_2O–CO_2	127·722	4·39880	5·68783	2485·322	0·35505	9618·7	1·19	17·98
Sulphuretted Hydrogen	H_2S	—	33·98	SO_2–H_2O	33·981	1·17664	1·52147	664·802	0·09497	3488·0	0·43	1·73
Carbon Bi-Sulphide	CS_2	—	75·93	CO_2–SO_2	75·931	2·62580	3·39980	1483·577	0·21194	3404·0	0·421	3·79
Water Gas		* See analysis below.			15·562	0·53883	0·69678	304·438	0·04349	4839·7	0·5989	0·936
Coal Gas		* See analysis below.			11·332	0·39236	0·50739	186·620	0·02666	13817·0	1·710	1·924
Ammonia	NH_3	—	17·01	H_2O–N_2	17·010	0·58901	0·76163	332·790	0·47543	5332·0	0·659	1·33
Air	—	—	—	—	14·444	1·00000	1·29306	565·000	0·08071	—	—	—
Nitrogen	N_2	14·01	28·02	—	28·021	0·97026	1·25461	548·197	0·07831	—	—	—
Oxygen	O_2	15·96	31·92	—	31·920	1·10531	1·42923	624·500	0·08921	—	—	—
Carbon Dioxide	CO_2	—	43·89	—	43·892	1·51980	1·96519	858·687	0·12267	—	—	—
Carbon from Wood	C_2	11·97	23·94	CO_2	—	—	—	—	—	8080·0	1·000	—
Anthracite—Penna	—	—	—	CO_2–H_2O	—	—	—	—	—	7844·4	0·971	—
Bituminous Coal	—	—	—	CO_2–H_2O	—	—	—	—	—	8391·7	1·038	—
Cannel Coal	—	—	—	CO_2–H_2O	—	—	—	—	—	6365·5	0·788	—
Furnace Coke	—	—	—	CO_2–H_2O	—	—	—	—	—	7019·4	0·868	—
Gas House Coke	—	—	—	CO_2–H_2O	—	—	—	—	—	7000·0	0·866	—
Coal Tar	—	—	—	CO_2–H_2O	—	—	—	—	—	8667·0	1·073	—
Crude Petroleum	—	—	—	CO_2–H_2O	—	—	—	—	—	11094·1	1·373	—

* NOTE.

	Water Gas. (Uncarburetted.) Per cent.	Coal Gas. Per cent.
Hydrogen	49·2	50·28
Marsh gas	0·3	36·95
Illuminants	Trace.	6·34

	Water Gas (Uncarburetted.) Per cent.	Coal Gas. Per cent.
Carbon monoxide	43·8	4·37
Carbon dioxide	2·7	None.
Nitrogen	4·0	Trace.

Properties of Gases, &c.

(W. N. Booth.)

In the tables pages 112 and 113 are given many figures relative to the gases chiefly, which are of interest in questions of heat combustion, and the gas engine. The figures are principally based on Berthelot's thermal determinations of the calorific capacity of the gramme molecule, the figure for which, if divided by the molecular weight, gives the heat generated per gramme of the substance. An attempt is made to give the correct values of the specific heat at high temperatures, with a view to the bearing of this property on the theory of the gas engine, and the possibly erroneous ideas as to the supposed effects of dissociation, which have been much overrated. The thermo-physical properties of carbon are more fully tabulated than has hitherto been attempted, with a view to explain the variations due to allotropism.

Heat and Combustion.

There are in practice only two elementary fuels, carbon and hydrogen. They generate heat by union with oxygen, and their compounds are also available as fuels.

Heat is measured in units. The British thermal unit B.Th.U. is the amount of heat necessary to raise 1 pound of water from 60° F. to 61° F., and its mechanical equivalent is J = 778 pounds raised 1 foot in the latitude of Manchester; this last an unnecessary refinement in engineering.

The metrical or French calorie is the amount of heat necessary to raise 1 gramme of water 1° C. This is the 'petit calorie.' The great calorie is the heat necessary to raise 1 kilogram of water through 1° C. The mechanical equivalent is 427 kilograms raised 1 metre at the latitude of Paris; this latter refinement being unnecessary in engineering. The following table gives the ratios of English and French units:

1 Calorie = ·001 great calorie.	1 B.Th.U. = 1055 joules = 778 foot-pounds (until recently the accepted value was 772).
1 Great calorie = 3·968 B.Th.U.	1 Great calorie = 3087·3 foot-pounds.
1 B.Th.U. = 0·252 great calorie.	1 B.Th.U. = 107·6 kilog. metres.
1 Great calorie = 427 kilog. metres.	

Chemists use somewhat different terms and state calorific measurements in terms of the molecule. The gramme molecule of hydrogen is the unit employed. One gramme molecule of hydrogen, H_2, taken at 0° C. and 760 mm. of pressure of a mercury column, weighs 2 grammes and has a volume of 22·32 litres. The molecular weight of any other simple or compound gas is the weight which occupies the volume occupied by 2 grammes of hydrogen, that is 22·32 litres. The molecular specific heat of a gas is the heat of this volume in 'petit calories.' Then the molecular heat divided by the molecular weight will give the specific heat per unit weight or gramme; and the molecular heat divided by 22·32 will give the specific heat per unit of volume.

The specific heat of a gas is usually stated at constant pressure and at constant volume. In perfect gas the ratio of these two is about 1·4 at ordinary temperatures. This is an erroneous way of putting it. The difference is really due to work done on the atmosphere in expanding to constant pressure. This work is invariable at 1·94 calories. The true ratio of specific heat at constant pressure or Cp to that at constant volume Cv, is thus properly stated, $Cp = Cv + 1·94$, or usually $= Cv + 2$. Thus, if 22·32 litres of gas are heated 1° C. from 0° C. they will perform work against a column of mercury 760 mm. high to the extent of 1·94 calories at all temperatures. A gas expands when heated 1° C. to an additional bulk $= \frac{1}{273}$ of the bulk occupied at 0° C. Hence the invariable quantity 1·94.

The specific heat of a gas increases with temperature. The number of calories generated by the combustion of a unit weight of a substance if divided by the specific heat of the products of combustion and by the total weight of such products, will give the nominal temperature The 'theoretical' temperature of combustion is the nominal temperature found by using the specific heat at ordinary temperatures. These nominal temperatures are never attained. The ordinary specific heat of carbon dioxide is quadrupled at 4000° C., and the temperature that might theoretically reach 18000° C., in oxygen in a certain case would not go beyond 4500° C. or thereabouts.

The calorific capacity of fuel may be approximated by calculating the capacity of its constituents of carbon and hydrogen. This is absolutely accurate, subject to allowance for the heat of formation. Thus in the formation of some hydrocarbons, heat is evolved. When these hydrocarbons are burned, the heat which was produced in their original formation cannot be again produced, and the combustion is productive of so much less heat. Others, such, for example, as acetylene, absorb heat when formed, and this substance when burned gives out more heat than corresponds with its carbon and hydrogen. This is one reason for its brilliance, and accounts for its explosive tendency.

HEAT OF COMBUSTION OF SUBSTANCES.

	Calories.	B.Th.U.	Authority.
Cellulose	4,200	7,560	Berthelot
Soft resinous wood	5,050	9,090	Gottlieb
Hard wood	4,750	8,550	„
Peat	5,940	10,692	Bainbridge
Cane sugar	3,961	7,130	Berthelot
Asphalt	9,532	17,159	Slossom & Colburn
Pitch	8,400	15,120	Anon.
Naphthalin	9,690	16,842	Berthelot
Paraffin	11,000	19,800	Mahler
Tallow	9,500	17,100	Stohmann
Sulphur	2,500	4,500	Berthelot
Petroleum	9,600 to 11,000	17,280 to 19,800	Various
Schist oil	9,000 to 10,000	16,200 to 18,000	„
Heavy coal gas oil	8,900	16,020	Ste-Claire Deville
Cotton oil	9,500	17,100	Anon.
Rape oil	9,489	17,080	Stohmann
Olive oil	9,473	17,051	„
Sperm oil	10,000	18,000	Gibson

Combustion of Carbon.

Carbon exists only as a solid and is found in three forms or allotropic modifications : (*a*) the diamond ; (*b*) graphite ; (*c*) amorphous, as charcoal. The heat generated by the perfect combustion of each of these forms is peculiar to that form. Carbon exists in liquid form only in combination with other substances and does not vaporise except at 3600° C. The total heat of combustion of carbon is 11·646 calories per gramme from the state (*a*). Of this amount ·2783 calories disappear in converting the diamond to the amorphous state (*c*) ; 3·231 calories are absorbed in vaporising the carbon ; and 2·4533 calories are set free by its conversion into monoxide CO. 5·684 calories are then produced by the second oxidation of this CO into dioxide=CO_2. In practice the only available heat is that which is set free after vaporising the carbon, *i.e.* 8·137 calories. The following table gives these characteristics in brief, of converting 1 gramme of carbon.

	Calories.
1. Converting diamond to graphite	= ·041
2. „ graphite to amorphous	= ·237
3. „ diamond to amorphous (1)+(2)	= ·278
4. „ amorphous to monoxide CO	= 2·453
5. „ monoxide to dioxide	= 5·684
6. Vaporising amorphous carbon (5) − (4)	= 3·231
7. Total energy generated converting carbon vapour to dioxide (1)+(2)+(4)+(5)+(6)	= 11·646
8. Total available heat in practice (7) − (3) − (6)	= 8·137

The difference between (7) and (8) represents expenditure in unlocking the crystal form of the diamond (3) and in converting the carbon to gas (5). The latent heat of vaporisation of carbon is thus 3·231 or about half that of steam.

Hydrogen is known in three states, liquid, solid, and gaseous. The liquid and solid states are those of low temperature. All practical determinations start from the gaseous state. The following figures are found in the combustion of 1 gramme of hydrogen :

	Calories generated.
1. Converting hydrogen to water gas, H_2O,	29·15
2. „ „ „ liquid	34·50
3. „ „ „ solid	35·2

The ordinary available heat is that of (1) only ; (2) being only partially secured when the products of combustion are cooled considerably below 100° C.

Stated per pound, carbon yields 14,500 British heat units, and hydrogen 52,000 to 62,100 according to the final temperature. The first oxidation of 1 lb. of carbon to monoxide yields barely 4,500 heat units : the further oxidation of the resulting carbonic oxide yielding fully 10,000 units.

The difference of the two oxidations is due to the rendering latent of about half the heat of the first oxidation in gasifying the solid carbon. The following equivalent units are useful:

1 Calorie per square metre = 0·369 B.Th.U. per square foot.
1 B.Th.U. per square foot = 2·713 calorie per square metre.
1 Calorie per kilogram = 1·800 B.Th.U. per pound
1 B.Th.U. per pound = 0·556 calorie per kilogramme.

These calories are the great calories (p. 109.)

For thermo-chemical tables of gases see pp. 112–113.

The temperature produced by combustion is largely dependent upon the weight of the resultant product. Thus 1 lb. of hydrogen when burned to water vapour will produce 52,380 heat units, which have to be divided among the 9 lbs. of water. The nominal temperature is 14,375° F, which, however, is based on the supposititious constancy of the specific heat; 1 lb. of carbon, however, which only generates about 14,500 heat units, produces only $3\frac{2}{3}$ lbs. of gas, and the nominal resulting temperature is 18,457. These are for combustion in oxygen. With air the additional weight of nitrogen still further reduces the nominal temperatures to about 5,000° in each case.

The percentage of each constituent of a mixed product must be multiplied by the specific heat in order to find the true valency of that product in the calculation of temperature. To find the mean specific heat of a mixture of gases, the weight of each gas is to be multiplied by its specific heat. The products are to be added together and divided by the sum of the weights of all the gases. The result is the mean specific heat. Then the temperature of combustion is found by dividing the total heat of combustion of the combustible gases by the weight of products and by the mean specific heat.

Similarly the calorific value of a mixed gas is found by multiplying the weight of each gas by its calorific value per pound, adding the products together and dividing by the total weight. The result is the thermal value per pound of the mixed gas.

Boiling-points of Saturated Solutions of Various Salts.

Salt.	C.°	F.°	Salt.	C.°	F.°
Sodic nitrate	120·7	249·5	Hydrosodic phosphate	106·6	224
Potassic nitrate	115·7	240	Sal-ammoniac	114·2	237·6
,, chlorate	104	219·2	Calcic chloride	179·4	355
Sodic chloride	108·2	227	Potassic acetate	169	336
Potassic chloride	108	226·4	,, carbonate	134·7	275
Sodic carbonate	104·6	220	Calcic nitrate	151	304
,, acetate	124·5	256	Strontic chloride	118	244
Baric chloride	104·2	220	Potassic tartrate	114·4	238

Boiling-points of Water at Various Pressures.

Barometer in Inches.							
Boiling-point. F.°	Barometer. Inches.	Boiling-point. F.°	Barometer. Inches.	Boiling-point. F.°	Barometer. Inches.	Boiling-point. F.°	Barometer. Inches.
185	17·047	193	20·254	201	23·937	209	28·183
186	17·421	194	20·687	202	24·441	210	28·744
187	17·803	195	21·124	203	25·014	211	29·331
188	18·196	196	21·576	204	25·468	212	29·922
189	18·593	197	22·030	205	25·992	213	30·516
190	18·992	198	22·498	206	26·529	214	31·120
191	19·407	199	22·965	207	27·068	215	31·730
192	19·822	200	23·454	208	27·614	216	32·350

Barometer in Millimètres.							
Boiling-point. C.°	Barometer. Millimètres.	Boiling-point. C.°	Barometer. Millimètres.	Boiling-point. C.°	Barometer. Millimètres.	Boiling-point. C.°	Barometer. Millimètres.
85·5	441·62	89·5	515·53	93·5	599·49	97·5	694·56
86	450·34	90	525·45	94	610·74	98	707·26
86·5	459·21	90·5	535·53	94·5	622·17	98·5	720·15
87	468·22	91	545·78	95	633·78	99	733·21
87·5	477·38	91·5	556·19	95·5	645·57	99·5	746·50
88	486·69	92	566·76	96	657·54	100	760·00
88·5	496·15	92·5	577·50	96·5	669·69	100·5	773·71
89	505·76	93	588·41	97	682·03	101	787·63

Name.	Symbol.	Density. H = 1	Molecular Weight.	Lbs. per cubic ft.	Cubic ft. per lb.	Grams per litre.	Litres per gram.	Required to burn one unit.			
								Weight.		Volume.	
								Air.	Ox.	Air.	Ox.
Air	$\left\{\begin{matrix}O_{23}\\N_{76}\\X_1\end{matrix}\right\}$	14·44	—	·08073	12·385	1·29318	·773	—	—	—	—
Ammonia	NH_3	8·5	17	·05324	18·783	·761	1·313	6·13	1·41	3·58	·750
Carbon C. Diamond		—	—	12	—	—	—	to CO / „ CO_2	5·797 / 11·594	1·334 / 2·667	—	—
Carbon C. Graphite		—	—	„	—	—	—	to CO_2	do.	do.	—	—
Carbon C. Amorphous		—	—	„	—	—	—	to CO / „ CO_2	„	„	—	—
Carbon C. Vapour		—	12	„	·06696	14·930	1·0727	·932	„	„	9·54	2·00
Carbon dioxide	CO_2	22	44	·12344	8·147	1·967	·508	—	—	—	—
Carbonic oxide	CO	14	28	·07817	12·80	1·2515	·800	2·484	·571	2·381	·500
Carbon bisulphide	CS_2	38	76	·21242	4·706	3·4058	·2947	5·478	1·26	14·30	3·00
Hydrogen	H_2	1	2	·00559	178·83	·08961	11·16	34·785	8·000	2·39	·500
Oxygen	O_2	16	32	·08926	11·203	1·4298	·699	—	—	—	—
Nitrogen	N_2	14	28	·07845	12·763	1·25616	·796	—	—	—	—
Steam	H_2O	9	18	·05022	19·912	·8047	1·242	—	—	—	—
Steam at 2,000°	H_2O	—	„	—	—	—	—	—	—	—	—
Steam at 4,000°	H_2O	—	„	—	—	—	—	—	—	—	—
Acetylene	C_2H_2	13	26	·07267	13·456	1·190	·840	13·378	3·077	11·93	2·500
Benzene	C_6H_6	39	78	·208	4·808	3·333	·303	13·378	3·077	35·80	7·500
Ethylene	C_2H_4	14	28	·07814	12·797	1·2519	·799	14·903	3·428	14·30	3·000
Ethane	C_2H_6	15	30	·08565	11·950	1·3415	·746	16·484	3·733	16·70	3·500
Methane	CH_4	8	16	·04466	22·391	·7155	1·397	17·392	4·000	9·54	2·000
Ethylene chloride	$C_2H_4Cl_2$	49·5	99	·2767	3·631	4·4357	·22626	3·855	·808	11·93	2·500
Ethyl	C_2H_6O	23	46	·12857	7·775	2·061	·287	9·074	2·037	14·30	3·000
Methyl	CH_4O	16	32	·08926	11·203	1·4298	·699	6·521	1·500	7·15	1·500
Cyanogen	C_2N_2	26	52	·1453	6·88	2·338	·427	5·348	1·23	9·54	2·000
Glycerine	$C_3H_8O_3$	—	92	—	—	—	—	18·148	4·174	16·70	3·500
Blast Furnace Gas. $[CO]_{27}\ N_{65}\ (CO_2)_6\ H_2$		14±	—	·079	12·65	1·2515	·800	1·00 / ·721	·22 / ·166	·82	·164
Producer Gas, $[Sundry]_{1·6}$ $[CO]_{25}\ N_{69}\ [CO_2]2\ [CH_4]_2$		14±	—	·079	12·65	1·2515	·800	·99 / ·721	·21 / ·166	—	—
Water Gas, $[CO]_{76}\ [CH_4]_2\ [Sundry]_{7·5}\ (CO_2)_{10}\ N2\frac{1}{2}$		8±	—	·045	22·5	·726	1·40	3·878	·788	—	—
Coal Gas, &c., 16 $H_2\ [CH_4]_{57}\ [CO]_{15}\ N_4$		4·7	—	·032	31·6	·516	1·975	13·89	2·81	6·16	1·23
Natural Gas $[CH_4]_{90}\ N_6$ [Sundry]		8	—	·045	22·5	·726	1·40	15·00	3·06	—	—

Nominal temp. of combustion.				Heat generated by combustion of one					Heat of formation at 15° per Molecule
Air.		Oxygen.		Lb.	Cubic foot.	Gram.	Litre.	Molecule.	
F.°	C.°	F.°	C.°	B.Th.U.	B.Th.U.	Cal.	Cal.	Cal.	Cal.
—	—	—	—	—	—	—	—	—	—
3762	2090	8278	4539	9666	514·6	5·371	4·087	91·3	12·2 gas 16.6 liq.
2402	1317	6880	3804	3915	—	2·175	—	26·1	·50 [1]
4786	2718	17780	9879	14146	—	7·859	—	94·31	3·34 [2]
4812	2659	17875	9931	14222	—	7·901	—	94·81	2·84 [2] — ·50 [3]
2673	1485	7725	4292	4415	—	2·453	—	29·44	— 2·84 [1]
4988	2753	18440	10226	14647	—	8·137	—	97·65	— 3·34 [2]
6955	3846	25752	14290	20461	1410	11·367	12·193	136·41	−38·76 [2] −42·1 [3]
—	—	—	—	—	—	—	—	—	68·20 [4] 94·31 [3] 97·65 [2]
3494	1923	12892	7144	4383	342	2·436	3·047	68·2	26·1 [3] 29·4 [2]
6590	3661	24271	13484	9344	1985	5·197	17·66	394·5	−25·4 gas −19 liq.
—	(Water Vapour)			52290	293	29·15	2·612	58·3 gas	
4813	2674	12108	6727	62100	347	34·50	3·091	69 liq. 70·4 sol.	—
--	—	—	—	Water Liquid				—	—
—	—	—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—	Solid=70·4 Liq. =69·0 Gas =58·3	per H_2
—	—	—	—	—	—	—	—	—	50·6
—	—	—	—	—	—	—	—	—	37·1
6120	3400	20340	11300	21856	1624	12·142	14·46	315·7	−58·1
5022	2790	16830	9350	18094 17930	3764	10·052 9·960	33·496	784·1 gas 776·9 liq.	− 1·8 sol. − 4·1 liq. −11·3 gas
5400	3000	16886	9381	21927	1744	12·182	15·250	341·1	−14·6
4354	2419	14848	8249	22338	1912	12·410	16·641	372·3	23·3
4036	2245	14348	7971	24017	1073	13·343	9·547	213·5	18·9
5144	2858	13179	7321	5490	1511·8	3·050	13·52	302·	34·4 gas 41 liq.
4630	2573	12583	6690	12744	1639·1	7·080	14·54	325·7	59·8 gas 69·9 liq.
4186	2325	10216	5675	9526	856·5	5·331	7·627	170·6	53·3 gas 61·7 liq.
6099	3388	18222	10125	9686	1320·6	5·048	12·02	262·5	−73·9 gas −68·5 liq.
4000	2222	8078	4488	7770	—	4·317	—	397·2	161·7 liq. 165·6 sol.
2160	1200	4500	2500	1223 to 1237	96·7 97·8	·700	·900	—	—
3440 2160	1910 1200	4500	2500	1265 to 2530	100 to 200	·773 to 1·370	·9674 to 1·713	—	—
4850	2700	—	—	4230 to 5458	330 to 700	2·35 to 3·03	3·00 to 6·35	—	—
4500	2500	—	—	21400	685	11·9	6·099	—	—
4200	2333	—	—	24444	1100	13·58	10·0	—	—

[1] From Graphite. [2] From Amorphous Carbon. [3] From Diamond. [4] From Carbonic Oxide.

Name.	Critical T.° and Pressure (Atmospheres).				Temp.		Density of liquid.
	Lower limit.		Upper limit.		Boil.	Fusion.	
	T.°C. Absol.	Atmos.	T.°C.	Atmos.	C.°	C.°	Water =1.
Air	133	39	−192·2	—	−192·2	—	·933
Ammonia	403	115	—	—	− 38·5	− 75	·636
Carbon C. Diamond	—	—	—	—	3600	—	—
Carbon C. Graphite	—	—	—	—	,,	—	—
Carbon C. Amorphous	—	—	—	—	,,	—	—
Carbon C. Vapour	—	—	—	—	,,	—	—
Carbon dioxide	306	77	—	—	− 78·2	−115	·83
Carbonic oxide	137·5	35·5	—	—	− 190	−207	—
Carbon bisulphide	546	77·9	277·55	95·86	46·2	−116	1·298
Hydrogen	39	20	—	—	− 243	—	·0714
Oxygen	154·2	50·8	—	—	−181·4	−211+	1·124
Nitrogen	127	35	—	—	−194·4	−203	·855
Steam	—	—	370	195·5	100	0	1·00
Steam at 2,000°	—	—	—	—	—	—	—
Steam at 4,000°	—	—	—	—	—	—	—
Acetylene	310	68	—	—	− 85	− 81	·451
Benzene	569·7	60·5	—	—	80·8	4·5	·899
Ethylene	283	51	—	—	−103	−169	—
Ethane	307	50·2	—	—	− 93	−151+	—
Methane	200	56·8	—	—	−164	−185·5	·415
Ethylene chloride	—	—	283	—	84·9	—	1·28
Ethyl	235·39	66·78	—	—	78·3	− 90+	·809
Methyl	202·76	72·85	233	69·7	66·8	—	·814
Cyanogen	7·2	3·6	124	61·7	− 20·7	− 40	·87
Glycerine	—	—	—	—	290·4	17	1·267
Blast Furnace Gas	—	—	—	—	—	—	—
Producer Gas &c.	—	—	—	—	—	—	—

[The + sign indicates that the actual temperature is lower than that given.]

Water Gas, &c.	—	—	—	—	—	—	—
Coal Gas, &c.	—	—	—	—	—	—	—
Natural Gas	—	—	—	—	—	—	—

Specific heat. Water =1 Liquid.	Constant Press.	Constant Vol.	Molecular. Constant Press.	Constant Vol.	Elementary sp. heat at high temp.	Temp. of ignition.	Heat of solution.	a ☆
—	·2375	·1686	—	—	—	— —	—	·0010
Sol.	·508	·391	9·64 gas 15·1 liq.	7·6	8·51+·0053T	— —	4·4	·0071
·1468	—	—	6·85 at	200°	11·3 at 1000° 14·2 „ 2000°	— —	—	—
·285								
·2018	—	—	7·12 at	200°	18·8 „ 3600°	— —	—	—
·2415	—	—	4·6 at	100°	—	— —	—	—
—	·285	—	6·85	—	8·42+·00144 T at 1000° to 3600°	— —	—	—
—	·216	·171	8·4 + ·0106 T	6·4 + ·0106T	19·1+·0030 (T − 2000°)=25·1 at 4000°	— —	5·6	·0084
—	·245	·173	6·8	4·8	4·8+·0032(T−1600°)	— —	—	·0010
·20884	·157	·130	21·7 gas 18·2 liq.	—	10+·0746 T 80° to 230°	— —	—	—
—	3·410	2·4146	6·82	4·8	4·8+·0032 (T − 1600°)=14·1 at 4500°	— —	—	·0010
—	·217	·1548	6·95	4·95	4·95+·00324(T−1600°)	— —	—	·0010
—	·244	·173	6·83	4·8	4·8+·0032(T−1600°)	— —	—	·0010
1·0 liq. ·504 solid	·479	·370	8·65	6·65	16·2+·038(T − 2000°)	— —	—	—
—	—	·9	—	16·2	—	— —	—	—
—	—	1·32	—	23·8	—	— —	—	—
—	·373	—	9·7	—	—	580 —	5·3	—
·43602	·3754	·350	34 liq. 29·2 gas (116° to 218°)	—	17·45+·0798T	— —	—	·0510
—	·404	·332	9·42 at 0° 14·0 at 200°	—	9·42+·0231T	— —	—	·0137
—	—	—	—	—	—	616 —	—	—
—	·593	·468	9·5 100° to 200°	—	—	667 —	—	—
—	·23	liq.=28·9+·0044T gas=22·7 at 111° to 211°		—	—	— —	—	—
·60 liq. ·50 gas	·451	·320	51·2 liq. 20·8° (110° to 220°		—	— —	2·5	—
·66 liq. ·46 gas	—	—	21·4 liq. 14·7 gas (101° to 225°		—	— —	2·0	—
—	—	—	—	—	—	— —	6·8 gas 1·4 liq.	—
—	—	—	—	—	—	— —	5·4	—

[NOTE.—Gases expand by heat to the extent of $\frac{1}{273}$ of their bulk at 0° C. for each degree Centigrade, or $\frac{1}{491{\cdot}4}$ of their bulk at 32° F. for each degree Fahr.

☆ The specific heat of gases varies with the temperature, being more for higher temperatures. At the absolute zero the values of the molecular heat of all gases seem to converge at 6·5 for constant pressure values. Lechatelier therefore gives a formula for specific heat $Cp = 6{\cdot}5 + aT$, where T is the absolute temperature Centigrade and a is a coefficient greater according to the complexity of the molecule. For values of a see table. This has an important bearing on the theory of the gas engine.]

[T = Temperature Centigrade.]

—	—	—	—	—	—	— —	—	—	—

Dilatation or Expansion of Substances per Degree of Fahrenheit Scale.

Temperatures.	Solids.	Linear. (a.)	Surface. (b.)	Volume. (c.)
32° to 212°	Glass	0·00000478	0·00000956	0·00001434
212 „ 392		0·00000546	0·00001093	0·00001639
392 „ 572		0·00000660	0·00001320	0·00001980
32 „ 212	Wrought iron	0·00000656	0·00001312	0·00001968
212 „ 572		0·00000895	0·00001790	0·00002686
32 „ 212	Soft, good iron	0·00000680	0·00001360	0·00002040
32 „ 212	Cast iron	0·00000618	0·00001236	0·00001854
32 „ 212	„ steel	0·00000600	0·00001200	0·00001800
32 „ 212	Hardened steel	0·00000689	0·00001378	0·00002067
32 „ 212	Copper	0·00000955	0·00001910	0·00002865
212 „ 572		0·00001092	0·00002184	0·00003276
32 „ 212	Lead	0·00001580	0·00003160	0·00004740
32 „ 212	Gold, pure	0·00000815	0·00001630	0·00002445
32 „ 212	„ hammered	0·00000830	0·00001660	0·00002490
32 „ 212	Silver, pure	0·00001060	0·00002120	0·00003180
32 „ 212	„ hammered	0·00001116	0·00002232	0·00003348
32 „ 212	Brass, common cast	0·00001043	0·00002086	0·00003129
32 „ 212	„ wire or sheet	0·00001075	0·00002150	0·00003225
32 „ 212	Platinum, pure	0·00000491	0·00000982	0·00001473
212 „ 572		0·00000520	0·00001040	0·00001560
32 „ 212	Palladium	0·00000555	0·00001110	0·00001665
32 „ 212	Roman cement	0·00000797	0·00001594	0·00002391
32 „ 212	Platinum, hammered	0·00000530	0·00001060	0·00001590
32 „ 212	Zinc, pure or cast	0·00001633	0·00003266	0·00004899
32 „ 212	„ hammered	0·00001722	0·00003444	0·00005166
32 „ 212	Tin, cast	0·00001207	0·00002414	0·00003621
32 „ 212	„ hammered	0·00001500	0·00003000	0·00004500
32 „ 212	Fire-brick	0·00000275	0·00000470	0·00000705
32 „ 212	Good red brick	0·00000305	0·00000610	0·00000915
32 „ 212	Marble	0·00000613	0·00001226	0·00001839
32 „ 212	Granite	0·00000438	0·00000876	0·00001314
32 „ 212	Bismuth	0·00000773	0·00001546	0·00002319
32 „ 212	Antimony	0·00000602	0·00001204	0·00001806
32 „ 212	Palladium	0·00000555	0·00001110	0·00001665
32 „ 212	Mercury	0·00003333	0·00006666	0·00010000
212 „ 392		0·00003416	0·00006833	0·00010250
392 „ 572		0·00003500	0·00007000	0·00010500
32 „ 212	Water	0·00008806	0·00017612	0·00026420
212 „ 392		0·00017066	0·00034133	0·00051020
392 „ 572		0·00018904	0·00037808	0·00056713
32 „ 212	Salt, dissolved	0·00009250	0·00018560	0·00027780
32 „ 212	Sulphuric acid	0·00011111	0·00022222	0·00033333
32 „ 212	Turpentine and ether	0·00012966	0·00025933	0·00038900
32 „ 212	Oil, common	0·00014814	0·00029629	0·00044444
32 „ 212	Alcohol and nitric acid	0·00015151	0·00030302	0·00055555
32 „ 212	All permanent gases	0·00069416	0·00138832	0·00208250

The superficial expansion is twice the linear, and the cubical expansion three times the linear. The coefficient of expansion of an alloy is equal to the mean coefficients of expansion of the volumes of the metals composing it.

If a bar is *increased* in length from l to L by an *increase* of temperature of $t°$, then

$$L = l(1 + at°).$$

If a bar is *decreased* in length from L to l by a *decrease* of temperature $t°$, then

$$l = \frac{L}{1 + at°}$$

Change in Density of a Body by Change in Heat.

If *decrease* in density be from d to d' by an *increase* of temperature of $t°$, then

$$d' = \frac{d}{1 + ct°}$$

If *increase* in density be from d' to d by a *decrease* of temperature of $t°$, then

$$d = d'(1 + \alpha t°).$$

1. Solids have a much smaller coefficient of expansion than liquids.

2. The coefficient of expansion of liquids increases with the temperature.

3. The coefficient of expansion of a liquid which only preserves its state under very great pressure is probably very great.

Expansion of Air by Heat.

Dry air expands or contracts uniformly ·002039 its volume per F.° difference under *constant pressure.*

If the *volume is constant* and the pressure varies, then the coefficient per F.° is ·002036.

Reduction of Volume of Liquids under Pressure.

An ordinary liquid under a pressure of 200,000 lbs. per sq. in. loses in volume about 33 per cent.

The following shows the percentage of reduction of volume under a pressure of 1 atmosphere (14·7 lbs. per square inch) :—

	Per Cent.		Per Cent.		Per Cent.		Per Cent.
Alcohol . . .	·00216	Ether . . .	·006158	Mercury . .	·000265	Water . . .	·004663

Flash Temperatures.

Dr. H. Holm finds that the temperature at which substances in contact with atmospheric air at ordinary pressures will catch fire are, among others :—Lighting gas, 600° C ; benzine, 415° C. ; petroleum, 380° C. ; gas oil, 350° C. ; machine oil, 380° C. ; coal tar, 500° C. ; tar oil, 580° C. ; and benzol, 520° C. These temperatures are greatly influenced by the presence of other substances, which act as catalysers. This is especially the case with hydrogen, methane, and ethane, not ethylene or acetylene. Among solids it is the substances with the biggest molecules which most readily burst into flame. As a broad rule, the nearer a substance is to a gaseous condition, the higher the temperature to which it must be exposed before it will kindle : but this depends also on the chemical constitution.

See also page 101.

SOUND.

The intensity of sound decreases as the square of the distance.

The intensity of sound increases as the square of the amplitude of the vibrations of the sonorous body.

The intensity of sound depends on the density of the air in the place at which it is produced.

The intensity of sound is greater in the direction of the wind than in the contrary direction.

Velocity of Sound through Air.

Sound travels at the rate of 1126 feet (·213 miles) (343 metres) in a second through air at a temperature of 16° C.

If

v = velocity in feet per second ;
$t°$ = temperature of the air, Fahr. scale ;
D = distance in feet the sound travels in the time, T ;

then,

$$v = 1089{\cdot}42\ \sqrt{1 + 0{\cdot}00208\ (t° - 32)};\quad D = 1089{\cdot}42T\ \sqrt{1 + {\cdot}00208\ (t° - 32)},$$

$$T = D \div v.$$

EXAMPLE.—A ship at sea was seen to fire a cannon, and 6·5 seconds (T) afterwards the report was heard ; the temperature ($t°$) of the air was 60°. Required, the distance (D) to the ship.

$$D = 1089{\cdot}42 \times 6{\cdot}5\ \sqrt{1 + {\cdot}00208\ (60° - 32)} = 7{,}300 \text{ feet, or } 1{\cdot}38 \text{ mile.}$$

The velocity of sound in water is about four times that in air.

Velocity of sound in sea water $= 4756 + 13{\cdot}8\,t - 0{\cdot}12\,t^2 + 3{\cdot}73\ (S-35)$ (ft./sec.), where $t°$ = temperature, and S = salinity in parts per thousand. This formula holds over the temperature range 6°–17° C. and for salinities of about 35 per cent.

(*F. E. Smith, F.R.S., Royal Institution Lecture, Feb.* 25, 1924.)

Distances at which Sounds can be Heard.

Descriptions of Sounds.	Audible at a Distance of	
	Feet.	Miles.
A powerful human voice in the open air, no wind	460	0·087
Report of a musket	16,000	3·02
Drum	10,500	2
Music, strong brass band	15,840	3
Cannonading, very strong	475,000	90
In a barely observable breeze a strong human voice with the wind can be heard	15,840	3

Velocity of Sound through Solid Substances in Feet per Second.

Metals.				Woods.			
Lead	4,000	Copper	11,700	Pine	10,900	Elm	13,500
Gold	5,700	Steel wire	15,500	Oak	12,600	Fir	15,200
Silver	8,550	Iron	16,800	Ash	13,300	Aspen	16,700

Speaking-Tubes.

The greatest distance at which a sound signal, strongly and clearly given, can be heard through a straight line of tube, without branches, is about 1,500 ft., and cannot exceed 1,650 ft.

The best sizes for speaking-tubes are 1·2 in. for distances below, and 2·1 ins. for those above 650 ft., in single lines; and for branched systems up to 160 ft., ·80 in.; 160 to 500 ft., 1 in.; 500 to 1,000 ft., 1·3 in.; above 1,000 ft., 2·1 ins. Below 1·2 in. the frictional resistance to the sound waves is too great, and above 2·1 ins. it is beyond the power of the average human voice to set the enclosed air in vibration with the necessary intensity.

The greater the diameter of the tube the louder must the voice and the clearness of the articulation of the speaker be.

Vowel sounds are more readily heard than consonants, and longer rather than short accented ones.

Sheet zinc, on account of its low elasticity, is the best material for tubes, although in most cases galvanised iron may be preferred on account of the readiness with which it transmits knocker signals.

Screwed collars are preferable to flanges for joining the tubes.

The tubes must be as smooth as possible inside, and sudden changes of section are to be avoided.

Although the best method of isolating the tubes is to inclose them in earth or masonry, it will be more convenient in most cases to use earthenware pipes as casings suspended from the roof of the levels, which need only be solidly clamped at the stations.

Changes in the direction of the line and branch connections are best made by curves of not less than 16 ft. radius. When there are many branches, these should be only signalling points, the others being all receiving stations. If, however, a line is required for giving signals from both ends, it may be utilised for intermediate points by short receiving branches, about 16 ft. long, connected with the main line by proper curves. The number of these branches should not, however, be more than two for a 1·6-in. tube of 1,000 ft., or three for one of 2·1 ins. and 1,300 ft. In cases of greater complexity it is better to establish repeating stations at intervals of about 650 ft.

The best call signals for great distances are made by striking the side of the tube; but a bugle may be used, if it is pressed close to the mouthpiece. Whistles are inaudible except at short distances.

The oval or elliptical mouthpieces commonly used on short lines are quite useless. The most convenient arrangement is to make the end of the tube slightly conical, widened to about 1·5 in. in diameter, and cover it with a flat wooden ring, like a telephone, which can be used either for speaking or hearing with the minimum loss of sound.

OPTICS.

LIGHT.

Decomposition of Light in the Spectrum.

Colours.	Maximum Ray.	Combination of Colours.		
		Primary.	*Secondary.*	*Tertiary.*
Violet	Chemical			
Indigo		Blue + Yellow	Green	
Blue	Electrical			Dark Green
Green		Blue + Red	Purple	
Yellow	Light			Brown
Orange		Yellow + Red	Orange	
Red	Heat			

All the colours of the spectrum mixed together make white, which is proved by the decomposition of white light, which makes the seven colours.

Velocity of Light.

Velocity of light = $3 \cdot 004 \times 10^{10}$ cms. per sec. Mean wave length = $5 \cdot 3 \times 10^{-5}$ cms.

The velocity of light through the atmosphere is 186,330 miles per second (*Newcomb*). The velocity of light through transparent bodies is not known, but probably varies inversely as the square root of the specific gravity of the transparent substance.

Light passes from the sun to the earth ($1 \cdot 487 \times 10^{13}$ cms. = 92,390,000 miles [*Everett*]) in eight minutes, at which velocity light can pass around the earth in one-eighth of a second.

To find the Focal Length of a Convex Mirror.

Use as object an opaque screen, with a hole and pin-point, and painted white, or covered with white paper.

Set up on the bench, in line, say, with the left edge of the hole, the convex mirror and an auxiliary biconvex lens of short focal length (6 ins. or so), and adjust the lens so that the image of the hole and pin-point is formed side by side with the object. The centre of the mirror is now at the point at which the image would be formed by the lens alone; this position may either be calculated or found (after noting the position of the mirror and then removing it) by means of a screen. Thus the radius is easily measured.

If the focal length of the mirror be greater than f, that of the lens, the simplest way of adjusting is to put the lens as close as possible to the mirror, put the object at principal focus of lens, and move the object back until the image is formed as above.

If, however, the focal length be less, we can be sure of finding the position by putting the mirror at a distance of 4 f from the object, and the lens at 2 f, and moving the lens back until the desired position is reached.

To find the Focal Length of a Concave Lens.

Use an object like the one mentioned above, an auxiliary convex lens (say, 6 ins. focal length) to produce a convergent beam, and an auxiliary plane mirror, placed beyond the concave lens.

Adjust until the image is formed side by side with the object, as before; then the rays must be emerging parallel to one another from the concave lens, and hence the convergent beam from the convex lens will (when the concave lens and mirror are removed) form an image at the principal focus of the concave lens. A direct measure can thus be made of the focal length.

(*E. Budden.*)

Photographic Lens Apertures.

Stops or apertures are openings in the lens diaphragms which determine the amount of light admitted to the sensitive plate in a camera. They are usually expressed as fractions of the lens focus, *i.e.* with an 8-in. focus lens a stop or aperture 1 in. in diameter would be known as $f8$—*viz.* it would divide into the focal length eight times; $f16$ would be $\frac{1}{2}$ in. diameter, and so on.

The exposure required varies inversely as the area of the opening. The apertures mostly in use are $f5 \cdot 6$, $f8$, $f11$, $f16$, $f22$, $f32$, $f45$, and $f64$, and each will be half the area and require twice the exposure of the next larger aperture. *E.g.*, if $f11$ requires 1 second $f16$ would require 2 seconds and $f8$ only half a second.

If the diameter of any one aperture be known the remainder can be ascertained by the method shown in fig. 2.

Say the diameter of *f*11 is ¾ in., set off *f*11 as one side of a right-angled triangle ABC: then BD, at right angles to the hypotenuse, would give the diameter of *f*16, DE would give *f*22, FA the diameter of *f*8, and so on as indicated.

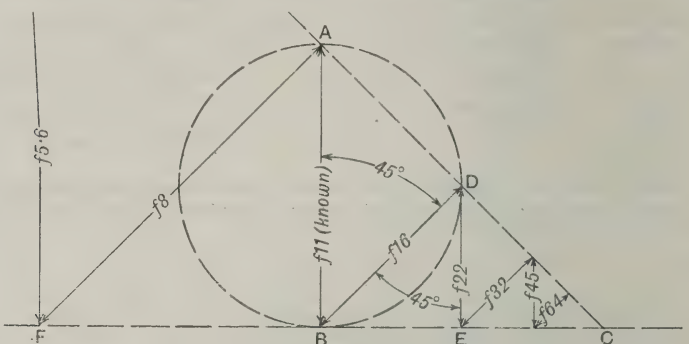


FIG. 2.

Theoretically *f*8 in one lens should be of the same exposure value as *f*8 in any other lens of whatever make or focus, but loss due to refraction, etc., must vary according to the number of component parts, and a compound lens containing many surfaces would obviously be slightly slower than one of the simplest form.

Luminous Energy.

In a series of lectures on 'Flame,' at the Royal Institution, Sir James Dewar illustrated in a striking form a fact which has long occupied the attention of scientists in relation to the energy expended in the production of light. The following figures show how utterly imperfect are the various lighting devices now employed on a basis of scientific efficiency:—

Lighting Device.	Percentage of Light.	Non-Luminous Energy.
Candle	2	98
Oil	2	98
Coal Gas	2	98
Incandescent lamp	3	97
Arc lamp	10	90
Magnesium lamp	15	85
Cuban firefly	99	1

Helium.

Properties of helium are: complete chemical inertness, a specific gravity of less than one-seventh that of air, thermal conductivity over six times as great as air, high specific heat, the lowest boiling point known, namely, — 267·8° C., and the ability to conduct sound nearly two and a half times as rapidly as air.

Next to hydrogen, helium is the lightest gas known. It is both non-inflammable and non-explosive, and possesses 92 per cent. of the lifting power of hydrogen, so that it is a most suitable filling for airship envelopes. By the use of helium, the engines of airships can be placed within the gas envelope if desired. A further advantage of helium over hydrogen is that the buoyancy can be increased at will by heating or cooling the gas by electric or other means. Moreover, the loss of gas from diffusion through the envelope is less with helium than with hydrogen by about 30 per cent.

The weight of 1,000 cu. ft. of dry air (reckoned at a barometric pressure of 29·5 ins. and a temperature of 55·5° Fahr.) is 76·03 lbs.; of the same bulk of pure hydrogen, 5·284 lbs., and of helium 10·46 lbs. The 'lift' of hydrogen per 1,000 cu. ft. is, therefore, 76·03 less 5·284 lbs., or 70·746 lbs. and of helium 76·03 less 10·46—*i.e.* 65·57 lbs.

SECTION V

PART I

WEIGHTS AND SPECIFIC GRAVITIES—WEIGHT OF SHEET, BAR, AND BALL METALS—WEIGHT OF PIPES AND TUBES —WEIGHT OF WIRE—WEIGHT OF NAILS AND SCREWS— THICKNESSES OF PIPES — SPECIFICATION FOR CAST IRON PIPES (pp. 123-167)

PART II

STRENGTH OF MATERIALS—STRENGTH OF WIRE, CHAINS, AND ROPES—BRITISH STANDARD SPECIFICATIONS FOR STRUCTURAL STEEL—ADMIRALTY TESTS FOR STEEL AND IRON—STRENGTH OF TUBES, RINGS, CHAINS, SLINGS, AND ROPES—FATIGUE OF METALS (pp. 169-204)

SECTION V

PART I

WEIGHTS AND SPECIFIC GRAVITIES WEIGHT OF SHEET, BAR, AND BALL METALS—WEIGHT OF PIPES AND TUBES—WEIGHT OF WIRE—WEIGHT OF NAILS AND SCREWS — THICKNESSES OF PIPES — SPECIFICATION FOR CAST IRON PIPES.

WEIGHTS AND SPECIFIC GRAVITIES.

ATOMIC WEIGHTS.

Name.	Symbol.	Atomic Weight. O = 16.	Atomic Weight. H = 1.	Name.	Symbol.	Atomic Weight. O = 16.	Atomic Weight. H = 1.
Aluminium	Al	27·1	26·78	Glucinum (*Beryllium*)	Gl	9·1	9·03
Antimony	Sb	120·2	119·24	Gold	Au	197·2	195·63
Argon	A	39·88	39·56	Helium	He	3·99	3·93
Arsenic	As	74·96	74·21	Holmium	Ho	163·5	162·2
Barium	Ba	137·37	136·27	Hydrogen	H	1·008	1·00
Bismuth	Bi	208·0	206·34	Indium	In	114·8	113·8
Boron	B	11·0	10·9	Iodine	I	126·92	125·91
Bromine	Br	79·92	79·28	Iridium	Ir	193·1	191·56
Cadmium	Cd	112·4	111·5	Iron	Fe	55·84	55·40
Cæsium	Cs	132·81	131·75	Krypton	Kr	82·92	82·26
Calcium	Ca	40·07	39·77	Lanthanum	La	139·0	137·9
Carbon	C	12·00	11·9	Lead	Pb	207·10	205·45
Cerium	Ce	140·25	139·0	Lithium	Li	6·94	6·88
Chlorine	Cl	35·46	35·17	Lutetium	Lu	174·0	172·5
Chromium	Cr	52·0	51·58	Magnesium	Mg	24·32	24·12
Cobalt	Co	58·97	58·4	Manganese	Mn	54·93	54·4
Columbium (*Niobium*)	Cb	93·5	92·75	Mercury	Hg	200·6	198·41
Copper	Cu	63·57	63·06	Molybdenum	Mo	96·0	95·23
Dysprosium	Dy	162·5	161·2	Neodymium	Nd	144·3	142·5
Erbium	Er	167·7	166·07	Neon	Ne	20·2	20·04
Europium	Eu	152·0	150·8	Nickel	Ni	58·68	58·21
Fluorine	F	19·0	18·8	Niton (radium emanation)	Nt	222·4	220·6
Gadolinium	Gd	157·3	156·05	Nitrogen	N	14 01	13·9
Gallium	Ga	69·9	69·34	Osmium	Os	190·9	189·38
Germanium	Ge	72·5	71·9				

ATOMIC WEIGHTS (*continued*).

Name.	Sym-bol.	Atomic Weight. O = 16.	Atomic Weight. H = 1.	Name.	Sym-bol.	Atomic Weight. O = 16.	Atomic Weight. H = 1.
Oxygen	O	16·00	15·87	Sulphur	S	32·07	31·82
Palladium	Pd	106·7	105·85	Tantalum	Ta	181·5	179·6
Phosphorus	P	31·04	30·79	Tellurium	Te	127·5	126·49
Platinum	Pt	195·2	193·65	Terbium	Tb	159·2	158·0
Potassium	K	39·10	38·79	Thallium	Tl	204·0	202·38
Praseodymium	Pr	140·6	139·48	Thorium	Th	232·4	230·57
Radium	Ra	226·4	224·1	Thulium	Tu	168·5	166·8
Rhodium	Rh	102·9	102·08	Tin	Sn	119·0	118·05
Rubidium	Rb	85·45	84·77	Titanium	Ti	48·1	47·7
Ruthenium	Ru	101·7	100·9	Tungsten	W	184·0	182·52
Samarium	Sa	150·4	149·2	Uranium	U	238·5	236·6
Scandium	Sc	44·1	43·75	Vanadium	V	51·0	50·65
Selenium	Se	79·2	78·6	Xenon	X	130·2	129·16
Silicon	Si	28·3	28·07	Ytterbium	Yb	172·0	170·73
Silver	Ag	107.88	107·02	Yttrium	Y	89·0	88·3
Sodium	Na	23.0	22·81	Zinc	Zn	65·37	64·85
Strontium	Sr	87·63	86·93	Zirconium	Zr	90·6	89·88

Weight of Water.

Pure water weighs 62·288 lbs. avoir. per cubic foot at 62° F. (16·6° C.) (*International Bureau of Standards, Paris*, 1921).

The following shows the principal weights, etc., of water:

Detail.	Fresh.	Salt.
Greatest density at	39·2° F.	Freezing-point
1 cubic foot at 62° F. weighs	62·288 lbs.	64 lbs.
1 cubic inch „ „	0·036047 lb.	0·037037 lb.
1 cubic foot „ equals	6·2288 gals.	
1 gallon 62° at F. weighs	10 lbs.	10·3 lbs.
1 ton equals	35·962 cub. ft.	35 cub. ft.
1 ton contains	224 gals.	217 gals.
Freezes at	32° F.	27·4° F.

From 4° C. water expands regularly as the temperature rises; so that 1,000 parts at 0° C become at 0° = 1,000·00, at 4° = 999·88, at 10° = 1,000·14, at 20° = 1,001·67, at 30° = 1,004·21 at 40° = 1,007·61, at 50° = 1,011·93, at 60° = 1,016·86, at 70° = 1,022·43, at 80° = 1,028·73 at 90° = 1,035·58, at 100° = 1,043·03.

See also Section XXV., HYDRAULICS.

To obtain Weight of a Gas.

If

P = Absolute pressure in lbs. per sq. in.;
V = Volume in cu. ft.;
W = Weight in lbs.;
T = Absolute temperature = F.° + 460;
a = Atomic weight.

then,

$$W = \frac{PVa}{10{\cdot}7\,T} = 0{\cdot}00935\,\frac{PVa}{T}.$$

Thus to get the weight of 10·3 cu. ft. of CO_2 ($a = 12{\cdot}00 + (16 \times 2) = 44$) at 180 F.° and 14 lbs. absolute pressure:

$$W = 0{\cdot}00935\,\frac{14 \times 10{\cdot}3 \times 44}{180 + 460} = 0{\cdot}926 \text{ lbs.}$$

Specific Gravity.

The specific gravity of any substance in the C.G.S. system is its weight in grammes per cubic centimetre.

The relative specific gravity of a body is its weight as compared with that of an equal bulk of some other body which is adopted as a standard of comparison. For other substances than air and gases generally, pure water is the usual standard; and since the weight of a given bulk of water varies somewhat with its temperature, and also with the state of the air, the former is assumed to be 62° F., and the latter at 30 ins. at sea-level.

To find Specific Gravity.

Solids.

W = weight of a body in the air;
S = specific gravity of the body,
W = weight of the body (heavier than water) immersed in water;

then,

$$W - W : W = 1 : S. \quad S = \frac{W}{W - W};$$

When the body is lighter than water, annex to it a heavier body that is able to sink the lighter one.

S = specific gravity of the heavier annexed body;
s = specific gravity of the lighter body;
W = weight of the two bodies in air;
W = weight of the two bodies in water;
w = weight of the heavier body in air;
w = weight of the lighter body in air;

then,

$$s = \frac{w}{W - W - \frac{w}{S}};$$

A simple way to obtain the specific gravity of a wood is to form it into a parallel rod, and place it vertically in water; then, when in equilibrium, the immersed end is to the whole rod as the specific gravity is to 1.

Liquids.

Measure the weight of a stoppered-glass bottle; then fill the same with distilled water and weigh it again; the difference between the two measurements gives the weight of the distilled water. Then completely empty out the water and refill the bottle with the liquid whose specific gravity is required and again weigh; this weight less the weight of the bottle gives the weight of the liquid; this weight divided by the weight of the water gives the specific gravity of the liquid.

In practice it is convenient to employ a bottle (called a specific gravity bottle) that holds exactly 10 grammes of distilled water at 4° C., because, when such a bottle is filled with the liquid under trial the weight in centigrammes of the liquid taken at 0° C. represents the specific gravity at once, without calculation. For convenience a counterpoise of brass is adjusted to the weight of the bottle.

Gases.

Measure in the same way as liquids, taking Air as the standard. A large and very light, *i.e.* thin, glass vessel must be used for the purpose, otherwise the weight of the glass will swamp, as it were, the weight of the enclosed gas, and accurate results cannot be obtained. Certain corrections have to be made in practice, for great accuracy.

To find the Specific Gravity of a mixture of given volumes of any number of fluids whose specific gravities are given.

If V_1, V_2, V_3, . . . V_n be the volumes of the fluids whose respective specific gravities are s_1, s_2, s_3, . . . s_n, and S the specific gravity of the mixture, then

$$S = \frac{V_1 s_1 + V_2 s_2 + V_3 s_3 + \ldots V_n s_n}{V_1 + V_2 + V_3 + \ldots V_n}.$$

To find the Specific Gravity of a mixture of given weights of any number of fluids whose specific gravities are given.

If W_1, W_2, W_3, . . . W_n be the weights of the fluids whose respective specific gravities are s_1, s_2, s_3, . . . s_n, and S the specific gravity of the mixture, then

$$S = \frac{W_1 + W_2 + W_3 + \ldots W_n}{\frac{W_1}{s_1} + \frac{W_2}{s_2} + \frac{W_3}{s_3} + \ldots \frac{W_n}{s_n}}.$$

To Discover the Adulteration in Metals, or to find the Proportions of Two Ingredients in a Compound.

$$w = \frac{W - s(W - W)}{1 - \frac{s}{S}}.$$

EXAMPLE.—A metal compounded of silver and gold weighs, W=6 pounds in the air, and in water, W=5·636 pounds. Required the proportions of silver and gold.

S=19·36 specific gravity of gold. s=10·51 specific gravity of silver.

$$\text{Weight } w = \frac{6 - 10{\cdot}51(6 - 5{\cdot}636)}{1 - \frac{10{\cdot}51}{19{\cdot}36}} = 4{\cdot}755 \text{ pounds of gold, and } 1{\cdot}245 \text{ pounds of silver.}$$

SPECIFIC GRAVITIES AND WEIGHTS OF VARIOUS SOLIDS AND LIQUIDS.

(Specific gravity × 62·288 = weight of cubic foot in lbs.
" " × ·03604 = " " inch ")

Substance	Sp. Gr.	Weight of a Cubic Foot.	Substance.	Sp. Gr.	Weight of a Cubic Foot.	Substance.	Sp. Gr.	Weight of a Cubic Foot.
Metals.								
		Lbs.			Lbs.			Lbs.
Aluminium, wrought	2·67	167	Iron, white	7·50	467	Palladium	11·40	71
" cast	2·56	160	" grey	7·20	449	Phosphor bronze, cast	8·60	53
Antimony	6·71	418	" Eglinton hot-blast	6·97	435	Platinum, hammered	20·30	1,26
Arsenic	5·76	359		6·98	435	" native	16·00	99
Barium	·47	29	" Rennie	to	to	" rolled	22·10	1,38
Bismuth	9·90	617		7·11	444	Potassium, 50°	·87	5
Boron	2·00	125	" Mean for ordinary calculations	7·22	450	Red-lead	8·94	55
Brass, cast	8·10	505	" wrought, least and greatest density	7·47	466	Rhodium	10·65	66
" sheet, 75 Cu., 25 Zn.	8·45	527		to	to	Ruthenium	8·60	53
" yellow, 66 Cu., 34 Zn.	8·30	518		7·81	487	Silicium	4·50	28
" Muntz metal, 60 Cu., 40 Zn.	8·20	511	" common bar	7·55	471	Silver, pure cast	10·47	65
Bromine	3·00	187		7·53	469·5	" " hammered	10·50	65
Bronze (gun-metal) 84 Cu., 16 Tin	8·56	534	" puddled slab	to 7·60	to 474	Sodium	·97	6
83 " 17 "	8·46	528	" various irons, tested by Mr. Kirkaldy	7·5 to 7·8	468 to 486	Steel, least and greatest density	7·73 to 7·90	48 to 49
81 " 19 "	8·46	528				" homogeneous metal	7·90	49
" (mill bearings) 79 Cu., 21 Tin	8·73	544	" " average	7·65	477	" blistered	7·82	48
90 " 10 "	8·50	530·6	" Yorkshire iron bar	7·76	484	" crucible, average	7·84	48
Cadmium	8·65	539	" Lowmoor plates, 1½–3 ins. thick	7·81	487	" cast, average	7·85	489·
Calcium	1·58	98				" Bessemer, average	7·85	489·
Chromium	5·00	311	" Mean for ordinary calculations	7·70	480	" Mean for ordinary calculations	7·85	489·
Cinnabar	8·10	504				Strontium	2·54	15
Cobalt	8·60	536	Lead, cast	11·40	710	Tellurium	6·11	38
Columbium	6·00	374	" rolled	11·40	710	Thallium	11·85	73
Copper, cast	8·79	547	Lithium	·59	37	Tin, Cornish, hammered	7·39	46
" sheet	8·81	549	Magnesium	1·75	109	" Cornish, pure	7·29	45
" hammered	8·92	556	Manganese	8·00	498	Titanium	5·30	33
" wire	8·91	555	Mercury — 40°	15·60	972	Tungsten	17·00	1,06
Gold, pure cast	19·30	1,200	" + 32°	13·60	847	Uranium	18·33	1,14
" hammered	19·40	1,210	" 60°	13·60	847	Wolfram	7·12	44
" 22 carats fine	17·50	1,090	" 212°	13·40	835	Zinc, cast	6·86	42
" 20 " "	15·71	978	Molybdenum	8·60	536	" rolled	7·19	44
Iridium	18·71	1,170	Nickel, hammered	8·67	541			
hammered	23·00	1,430	" cast	8·28	516			
Iron, cast, least and greatest density	6·90 to 7·50	430 to 467	Osmium	21·4	1,334			

SPECIFIC GRAVITIES AND WEIGHTS OF VARIOUS SOLIDS AND LIQUIDS (*continued*).

Substance.	Specific Gravity.	Weight of a Cubic Foot.	Substance.	Specific Gravity.	Weight of a Cubic Foot.	Substance.	Specific Gravity.	Weight of a Cubic Foot.
			Stones, Earths, &c.					
		Lbs.			Lbs.			Lbs.
Agate	2·59	161	Emery	4·00	249	Pearl, Oriental	2·65	165
Alabaster, white	2·73	170	Flint, black	2·58	161	Peat	·60	37
„ yellow	2·70	168	„ white	2·50	156		1·33	83
Alum	1·71	106	Fluorine	1·32	82	Phosphorus	1·77	110
Amber	1·08	67	Garnet	4·19	261	Plaster of Paris	1·18	73
Ambergris	·87	54	„ black	3·75	233	Plumbago	2·10	131
Asbestos, starry	3·07	191	Glass, bottle	2·73	170	Porcelain, China	2·30	143
Asphaltum	·91	57	„ Crown	2·49	155	Porphyry, red	2·77	173
	1·65	103	„ flint	2·93	182	Pumice-stone	·92	57
Barytes, sulphate	4·00	249		3·20	199	Quartz	2·66	166
	4·87	303	„ green	2·64	164	Red-lead	8·94	557
Basalts	2·74	171	„ optical	3·45	215	Resin	1·09	68
	2·86	178	„ white	2·89	180	Rock, crystal	2·74	171
Borax	1·71	106	„ window	2·64	164	Rotten-stone	1·98	123
Brick	1·90	118	Granite, Egyptian red	2·65	165	Ruby	4·28	267
	1·37	85	„ Patapsco	2·64	164	Salt, common	2·13	133
„ fire	2·20	137	„ Quincy	2·65	165	Saltpetre	2·09	130
„ (pressed) in cement	2·17	135	„ Scotch	2·63	164	Sand, coarse	1·80	112
„ work in mortar	1·80	112	„ Susquehannah	2·70	168	„ common	1·67	104
	2·06	125	Gravel, common	1·75	109	„ damp and loose	1·39	87
Carbon	3·50	219	Grindstone	2·14	133	„ dried and loose	1·56	97
Cement, Portland	1·30	81	Gypsum, opaque	2·17	135	„ dry	1·42	88
„ Roman	1·56	97	Hone, white, razor	2·88	179	„ siliceous	1·70	106
Chalk	1·52	95	Hornblende	3·54	220	Sapphire	3·99	248
	2·78	173	Iodine	4·94	308	Schorl	3·17	197
Chrysolite	2·78	173	Jet	1·30	81	Shale	2·60	162
Clay	1·93	120	Lime, hydraulic	2·75	171	Slate	2·90	181
„ with gravel	2·48	154	„ quick	·80	50		2·67	166
Coal, anthracite	1·44	90	Limestone, green	3·18	198	„ purple	2·78	173
	1·64	102	„ white	3·16	197	Smalt	2·44	152
„ Cannel	1·24	77	Magnesia, carbonate	2·40	149	Spar, calcareous	2·74	171
	1·32	82	Marble, Adelaide	2·72	169	„ Feld, blue	2·69	168
„ Derbyshire	1·29	80	„ African	2·71	169	„ „ green	2·70	168
„ Newcastle	1·27	79	„ Biscayan, black	2·70	168	„ Fluor	3·40	212
	1·26	79	„ Carrara	2·72	169	Stalactite	2·42	151
„ Scotch	1·30	81	„ common	2·69	168	Stone, Bath	1·96	122
„ Wales, mean	1·32	82	„ Egyptian	2·67	166	„ Basalt	2·63	164
Coke	1·00	62	„ French	2·65	165	„ Bristol	2·51	156
Concrete, mean	2·00	124	„ Italian, white	2·71	169	„ Caen, Normandy	2·07	129
Copal	1·05	65	„ Parian	2·84	177	„ common	2·52	157
Coral, red	2·70	168	Marl, mean	1·75	109	„ Craigleith	2·32	144
„ white	2·55	159	Mica	2·80	174	„ Kentish rag	2·65	165
Cornelian	2·61	163	Millstone	2·48	154	„ Portland	2·37	148
Diamond, Oriental	3·52	219	Mortar	1·38	86	„ sandstone	2·20	137
„ Brazilian	3·44	214		1·75	109	Sulphur, native	2·03	126
Earth, common soil	2·19	136	Mud	1·63	102	Talc, mean	2·50	156
„ loose	1·50	93	Nitre	1·90	118	„ black	2·90	181
„ moist sand	2·05	128	Opal	2·11	131	Tile	1·82	113
„ mould, fresh	2·05	128	Oyster-shell	2·09	130	Topaz, Oriental	4·01	250
„ rammed	1·60	100	Paving-stone	2·42	151	Trap	2·72	169
„ rough sand	1·92	120				Turquoise	2·75	171
„ with gravel	2·02	126						

SPECIFIC GRAVITIES AND WEIGHTS OF VARIOUS SOLIDS AND LIQUIDS (*continued*).

Miscellaneous.

Substance.	Specific Gravity.	Weight of a Cubic Foot.	Substance.	Specific Gravity.	Weight of a Cubic Foot.	Substance.	Specific Gravity.	Weight of a Cubic Foot.
		Lbs.			Lbs.			Lbs.
Asphaltum	·91	57	Gum Arabic	1·45	90	Mastic	1·07	67
	1·65	103	Gunpowder, loose	·90	56	Myrrh	1·36	85
Atmospheric air	·00121	·075	„ shaken	1·00	62	Opium	1·34	83
Beeswax	·97	60	„ solid	1·55	97	Soap, Castile	1·07	57
Butter	·94	59		1·80	112	Spermaceti	·94	59
Camphor	·99	62	Gutta-percha	·98	61	Starch	·95	59
Caoutchouc	·90	56	Horn	1·69	105	Sugar	1·61	100
Egg	1·09	68	Ice, at 32°	·92	57		1·33	83
Fat of beef	·92	57	Indigo	1·01	63	„ ·66	·97	60
„ hogs	·94	59	Isinglass	1·11	69	Tallow	·94	59
„ mutton	·92	57	Ivory	1·83	114	Wax	·96	60
Gamboge	1·22	76	Lard	·95	59		·97	60

Liquids.

Acid, acetic	1·06	66	Alcohol, proof spirit, 50 per cent., 80°	·88	55	Oil, olive	·92	57
„ benzoic, solid	·67	42				„ palm	·97	60
„ citric, solid	1·03	64				„ petroleum	·88	55
„ oxalic „	1·64	102	Ammonia, 27·9 per cent.	·89	55	„ rape	·91	57
„ fluoric	1·50	93				„ sunflower	·93	58
„ hydrochloric	1·20	75	Aquafortis, double	1·30	81	„ turpentine	·87	54
„ nitric	1·22	76	„ single	1·20	75	„ whale	·92	57
„ phosphoric	1·56	97	Beer	1·03	64	Spirit, rectified	·82	51
„ phosphoric, solid	2·80	174	Bitumen, liquid	·85	53	Tar	1·02	64
„ sulphuric	1·85	115	Blood (human)	1·05	65	Vinegar	1·08	67
Alcohol, pure, 60°	·79	49	Brandy, $\frac{5}{6}$ of spirit	·92	57	Water, Dead Sea	1·24	77
„ 95 per cent.	·82	51	Cider	1·02	64	„ 60°	1·00	62
„ 80 „	·86	54	Ether, acetic	·87	54	„ 212°	·96	59
„ 50 „	·93	58	„ muriatic	·85	53	„ distilled, 39°	1·00	62
„ 40 „	·95	59	„ sulphuric	·72	45	„ Mediterranean	1·03	64
„ 25 „	·97	60	Glycerine	1·28	79·5	„ rain	1·00	62
„ 10 „	·99	61	Honey	1·45	90	„ sea	1·03	64
„ 5 „	·99	62	Milk	1·03	64	Wine, Burgundy	·99	62
„ proof spirit, 50 per cent., 60°	·93	58	Oil, anise-seed	·99	62	„ Champagne	1·00	62
			„ codfish	·92	57	„ Madeira	1·04	65
			„ linseed	·94	59	„ port	1·00	62
			„ naphtha	·85	53			

Woods (dry). (*H. Stone.*)

Acacia or Locust (Robinia)	·58–·83	36–52	Ash, Mountain or Rowan	·57–·94	35½–58½	Blackbutt	·74	46
Alder, black or common	·42–·64	26–39¾	„ Mountain (of Queensland)	·73	45½	Blackthorn	·67–·71	41¾–44½
Almond	·85–·88	53–56	„ Nova Scotia	·71	44½	Boxwood, Cape (Gonioma Kamassi)	·93	58
Apple	·66–·84	41–52½	Assegai wood (Cape)	·96	60	„ Cape or East London (Buxus Macowani)	1·19	74
Arbor vitæ	·38–·54	23¾–33¾	Bamboo	·40	25	„ English	·99–1·02	61¾–63½
Ash, American (Fraxinus Americana, L.)	·65–·83	40¾–52	Bay	·82	51	„ Turkey	1·08	67
			Beech	·66–·83	41–52	„ West Indian	·79	49
„ American, Arkansas or white	·36–·57	22½–35¾	„ Cape (Benkenhout)	·63–·74	29½–46	Bulletwood (Mimasops Balata)	·84	67½
„ European or common	·52–·95	32½–59	„ Australian white	·53	32¾			
			Birch, American	·60–·77	37½–48			
			„ European	·47–·77	29½–48			

SPECIFIC GRAVITIES AND WEIGHTS OF VARIOUS SOLIDS AND LIQUIDS (*continued*).

Substance.	Specific Gravity.	Weight of a Cubic Foot.	Substance.	Specific Gravity.	Weight of a Cubic Foot.	Substance.	Specific Gravity.	Weight of a Cubic Foot.
Woods (dry) (continued).						(*H. Stone.*)		
		Lbs.			Lbs.			Lbs.
Bulletwood, species unknown, but more commonly met with	1·20	75	Filbert, *see* Hazel			Mahogany, East Indian	·69	43
			Fir, Danzig . .	·48–·75	29⅔–47	„ Cuba . .	·72–·82	44¾–51
			„ Riga . . .	·51–·66	31⅔–41	„ Honduras	·56–·69	35–42¾
			„ Silver . .	·36–·60	23–37½	„ Mexican .	·61–·79	38½–49½
			„ Spruce . .	·35–·60	21¾–37½	„ Panama .	·47	29½
Butternut, *see* Hickory			Fustic . . .	·63	39	„ red (Australia) .	1·11	69
Campeachy, *see* Logwood			Greenheart, Trop. (America)	1·03–1·20	64–75	„ Spanish .	·86	53½
			„ Demerara	1·02–1·20	63¾–74½	Maple, Bird's-eye	·58	36
Camphor wood (Borneo) . .	·90–1·06	55¾–65¾	Gum, blue . .	·69–1·11	43–69	„ hard . .	·53–·80	33–50
Cedar, Cuba . .	·37–·53	23¼–33¼	„ grey (Australia) .	1·00	62¼	„ soft . .	·53–·71	33¾–44
„ Pencil or Virginian	·40–·60	25–37½	„ red (America) *see* Satin Walnut			„ Sugar . .	·58–·81	36–50½
„ Indian . .	1·32	82	„ Sweet, *see* Satin Walnut			Mastic	·85	53
„ red (Australia) .	·43–·59	26½–36½	Hackmatack (American Larch) . .	·63	39	Mora	1·08–1·10	67½–68½
„ white(Australia) .	·33–·53	20½–33	Hackia	·82–·96	51–59½	Mulberry . . .	·56–·90	35–56
Charcoal, pine .	·44	27½	Hawthorn . . .	·81–·91	50½–56¾	Oak, African . .	·83–1·09	51½–68
„ fresh burned	·38	23¾	Hazel	·56–·71	35–44¼	„ American,red	·84–·88	52¼–54½
„ oak . .	1·57	98	Hemlock . . .	·69	23	„ „ white	·94–1·05	58¾–65½
„ soft wood	·28	17½	Hickory, Bitternut	·78	47	„ Baltimore .	·70–·82	43½–51¼
„ triturated .	1·38	86¼	„ Butternut or Shellbark	·69–·83	43–52	„ Canadian .	—	54½
Cherry, American	·59	36½	„ Pig-nut	·79–·82	49½–51¼	„ Dantzig . .	·76–·90	47½–56
„ English .	·54–·71	33½–44¼	Holly	·76–·94	47½–58¾	„ Desert or She-oak (Australia)	1·14	71¼
Chestnut, sweet or Spanish	·60–·72	37½–45	Honeysuckle (New Zealand) .	·74–·81	46¼–50½	„ Dutch or Rhenish . .	·94–1·10	58¾–68½
„ Horse . .	·51–·63	32–39¼	Hornbeam . . .	·62–·82	38¾–51¼	„ English . .	·69–1·04	43–64½
Citron . . .	·72	45	Ironbark, red (Australia) . .	·98–1·16	61–72½	„ Forest or She-oak (Australia)	·78–·85	48½–53
Cocus or Cocos .	1·48–1·61	92–100	Ironwood, Pynkado . .	1·15–1·27	71¾–79	„ Italian or Modena . .	1·06–1·15	66–71¾
Cogwood, Jamaica	1·08	67	Jackwood . . .	·56–·71	35–44½	„ Italian or Tuscan .	1·07–1·08	66½–67½
Cork	·24	15	Jarrah	·81–1·05	50½–65¾	„ Sardinian .	·95–1·03	59–64
Cottonwood (N. America). . .	·55	34½	Jasmine	·77	48¼	„ Silky (Australia) . .	·60	37¼
Crabwood (British Guiana)	·63	39½	Juniper	·53–·70	33–43¾	„ Spanish or white (America)	·87–1·00	54¼–62
Cypress, Spanish .	·65	40¼	Karri	·85–1 02	53–63¾	„ Stettin . .	·56	35
Damson	·76–·84	47½–52½	Kauri or Cowrie Pine	·51–·60	32–37½	Olive	·87	54
Degame wood . .	·79	49	Lancewood (Jamaica orCuba)	·83–1·01	52–63	Orange	·71–·93	44–58
Dogwood, common .	·77–·83	48–51½	Larch	·44–·80	27½–50	Orham	·51	32
„ red (American)	·77–·81	48–50½	„ Russian .	·59–·70	36¾–43½	Padouk	·69–1·00	43–62
Ebony, black or Billetwood . .	·98–1·14	61–71	Lemon	·71–·80	44–50	Pear	·66–·72	41¼–45
Ebony, Indian .	1·21	75½	Lignum vitæ Guayaquil	·79	49	„ white (Cape)	·79–·98	49–61
„ Macassar .	·98–1·24	61–77	„ Jamaica .	1·61–1·43	72½–89	Persimmon . . .	·71–·85	44½–53
„ Purple (Ceylon)	·86	53½	„ Maracaibo	1·11	69	Pine, American Pitch . .	·50–·92	31¼–57½
Elder	·54–·76	33¾–47¼	Lime or Linden .	·32–·59	20–36¾	„ Corsican . .	·38–·76	23¾–47½
Elm, American or Rock	·69–·73	43–45¼	Locust, *see* Acacia			„ red or yellow (Pinus sylvestris or Scots fir)	·31–·74	19¼–46¼
„ „ or white	·65	40½	Logwood . . .	·92	57	„ red(Canada)	·51–·59	32–36¾
„ common or European	·56–·82	35–51¼	Mahoe, blue . .	·71–·79	44½–49½	„ white (Pinus strobus) .	·31–·56	19¼–35

SPECIFIC GRAVITIES AND WEIGHTS OF VARIOUS SOLIDS AND LIQUIDS (*continued*).

Substance.	Specific Gravity.	Weight of a Cubic Foot.	Substance.	Specific Gravity.	Weight of a Cubic Foot.	Substance.	Specific Gravity.	Weight of a Cubic Foot.
			Woods (*dry*) (continued).			(*H. Stone.*)		
		Lbs.			Lbs.			Lbs.
Pine, white (Australia)	·43–·49	27–30½	Quince	·71	44	Teak	·62	38½
„ yellow (Canada)	·43–·60	26½–37½	Rosewood, Indian	·73–1·03	45½–64	Tewart	1·16–1·20	72–74½
Plane, Eastern or European	·49–·68	30½–42½	Sabien	·70	43½	Waibaima	·92	57
„ Western or American	·57	35¼	„ Cuba	·89–·94	55½–58⅜	Wallaba	1·05	65½
Plum	·64–·90	40–56	„ fromTunas de Zaza	1·00	62½	Walnut, American or black	·46–·88	28¾–55
Pomegranate	1·36	84½	Sappan	·96–·98	60–61	„ Cape or Stinkwood	—	51⅜
Poon	·58	36¼	Sassafras	·48	30	„ English or Italian	·63–·71	39–44½
Poplar, American	·39–·48	24½–30	Satin-walnut, *see* Walnut			„ Satin, Red Gum, or Californian Red Gum	·59–·96	36¾–59½
„ aspen	·43–·56	26¾–35	Satinwood(Ceylon)	·98–1·04	61–65			
„ black	·39–·52	24¼–32½	„ (EastIndia)	79–·98	49–61			
„ white	·40–·57	25–35½	Siniaruba, Bitterwood	·48	30			
Purpleheart (BritishGuiana, Kooroobooilli)	·85	53	Spruce, European (Canada)	·35–·60	21¾–37½	Washiba or Bowwood	·88	55
Purpleheart (BritishGuiana, Marawinaroo)	1·00	62	Stonewood	·93	58	Whitewood, Canadian or Canary	·45–·62	28–38⅜
Purpleheart (Demerara)	·81	50½	Sneezewood	1·08	67½	Willow, common or Goat	·43–·63	27–39¼
Quassia	·54	33⅝	Sycamore, American, *see* Plane			Willow, white	·43–·50	26¾–31¼
			„ European	·47–·79	29–49	Yew	·74–·94	46–58½
			Tamarack, *see* Hackmatack					

SPECIFIC GRAVITIES OF WOODS.

The specific gravity or weight per cubic inch or cubic foot of woods can never be expressed by a fixed value. Wood varies in the same tree from the root to the branches, becoming lighter upwards, and again varies from the lower branches to the top in the reverse way, *i.e.* it becomes heavier upwards. It also varies from within outwards, and also varies according to the time of year, starch being accumulated at one season and used up at another. No two pieces of wood can be found of the same specific gravity except by accident, or by taking two small pieces from the same annual ring, which would rarely occur in practice. Hence it is misleading to work out specific gravities of woods to several places of decimals. Moreover, wood is hygroscopic, and however dry a specimen may be, a different result will be obtained with every movement of the barometer. The term 'dry weight' is merely an expression of convenience, and most of the figures given in the foregoing table have been calculated from specimens which have been kept in a dry room at ordinary temperatures, and which have ceased to lose weight. Wood dried by artificial means at high temperatures gains weight by absorbing moisture, this absorption taking place from the moment the wood is withdrawn from the desiccator.

Drying and Seasoning of Woods.

There is no difference between 'drying' and 'seasoning,' and the terms may be regarded as synonymous. Too rapid drying may be injurious to a wood, and hence seasoning has come to be regarded as something different from drying, but it is only so in certain cases, and is disputed even in these. Some foresters maintain that a few rings of wood nearest the heart pass from the condition of sapwood into that of heartwood during seasoning, and that therefore the amount of useable timber is increased thereby. (*H. Stone.*)

See also pp. 359–366.

SPECIFIC GRAVITIES OF EBONITE.

Black qualities, approx., 1·2 : Red qualities, approx., 1·6; Mottled quality, 1·33.

HYDROMETER SCALES.

(Compiled by The Kestner Evaporator and Engineering Co., Ltd.)

Degrees Twaddell.	Sp. Gr.	Degrees Baumé.	Approximate Degrees Brix at 15° C.	Degrees Twaddell.	Sp. Gr.	Degrees Baumé.	Approximate Degrees Brix at 15° C.
1	1·005	0·7	1·3	52	1·260	29·7	55·2
2	1·010	1·4	2·5	54	1·270	30·6	56·6
3	1·015	2·1	3·7	56	1·280	31·5	58·5
4	1·020	2·7	4·9	58	1·290	32·4	60·2
5	1·025	3·4	5·9	60	1·300	33·3	62·0
6	1·030	4·1	7·2	62	1·310	34·2	63·8
7	1·035	4·7	8·3	64	1·320	35·0	65·4
8	1·040	5·4	9·5	66	1·330	35·8	66·8
9	1·045	6·0	10·6	68	1·340	36·6	68·6
10	1·050	6·7	12·0	70	1·350	37·4	70·0
11	1·055	7·4	13·4	72	1·360	38·2	71·8
12	1·060	8·0	14·5	74	1·370	39·0	73·5
13	1·065	8·7	15·8	76	1·380	39·8	75·0
14	1·070	9·4	17·0	78	1·390	40·5	76·4
15	1·075	10·0	18·2	80	1·400	41·2	77·8
16	1·080	10·6	19·3	82	1·410	42·0	79·5
17	1·085	11·2	20·4	84	1·420	42·7	81·0
18	1·090	11·9	21·6	86	1·430	43·4	82·5
19	1·095	12·4	22·5	88	1·440	44·1	83·8
20	1·100	13·0	23·6	90	1·450	44·8	85·4
21	1·105	13·6	24·6	92	1·460	45·4	86·6
22	1·110	14·2	25·8	94	1·470	46·1	88·0
23	1·115	14·9	26·8	96	1·480	46·8	89·5
24	1·120	15·4	27·6	98	1·490	47·4	90·8
25	1·125	16·0	28·6	100	1·500	48·1	92·2
26	1·130	16·5	29·5	102	1·510	48·7	93·8
27	1·135	17·1	30·5	104	1·520	49·4	95·2
28	1·140	17·7	31·5	106	1·530	50·0	96·8
29	1·145	18·3	32·5	108	1·540	50·6	—
30	1·150	18·8	33·4	110	1·550	51·2	—
31	1·155	19·3	34·4	112	1·560	51·8	—
32	1·160	19·8	35·4	114	1·570	52·4	—
33	1·165	20·3	36·2	116	1·580	53·0	—
34	1·170	20·9	37·4	118	1·590	53·6	—
35	1·175	21·4	38·4	120	1·600	54·1	—
36	1·180	22·0	39·4	122	1·610	54·7	—
37	1·185	22·5	40·4	124	1·620	55·2	—
38	1·190	23·0	41·4	126	1·630	55·1	—
39	1·195	23·5	42·2	128	1·640	56·3	—
40	1·200	24·0	43·2	130	1·650	56·9	—
41	1·205	24·5	44·2	132	1·660	57·4	—
42	1·210	25·0	45·2	134	1·670	57·9	—
43	1·215	25·5	46·2	136	1·680	58·4	—
44	1·220	26·0	47·0	138	1·690	58·9	—
45	1·225	26·4	47·8	140	1·700	59·5	—
46	1·230	26·9	48·8	142	1·710	60·0	—
47	1·235	27·4	49·8	144	1·720	60·4	—
48	1·240	27·9	50·6	146	1·730	60·9	—
49	1·245	28·4	52·0	148	1·740	61·4	—
50	1·250	28·8	53·4	150	1·750	61·8	—

BAUMÉ GRAVITY.

The formula for liquids lighter than water now recommended by the Bureau of Standards and accepted by the petroleum trade generally for converting Baumé degrees to specific gravity is

$$\text{Sp. Gr.} = \frac{141{\cdot}5}{131{\cdot}5 + \text{Baumé degs.}}$$

Specific Gravities of Elastic Fluids

1 cubic foot of dry atmospheric air at 62° F. and at normal pressure weighs 532·5 grains. Its assumed gravity of 1 is the unit for elastic fluids.

Fluid	Specific Gravity Compared with Air.	Fluid.	Specific Gravity Compared with Air.
Atmospheric air	1	Sulphuretted hydrogen	1·17
Ammonia	·589	Sulphurous acid	2·21
Carbonic anhydride	1·52	Steam, 212°	·4883
„ oxide	·972	Smoke of bituminous coal	·102
Carburetted hydrogen	·559	„ coke	·105
Chlorine	2·47	„ wood	·09
Chloro-carbonic	3·389	Vapour of alcohol	1·613
Cyanogen	1·815	„ bisulphide of carbon	2·64
Gas, coal	·4 to ·752	„ bromine	5·1
		„ chloric ether	3·44
Hydrogen	·069	„ ether	2·586
Hydrochloric acid	1·278	„ hydrochloric ether	2·255
Hydrocyanic „	·942	„ iodine	8·675
Nitrogen	·972	„ nitric acid	3·75
Nitric oxide	1·094	„ spirits of turpentine	4·763
Nitrous acid	2·638	„ sulphuric acid	2·7
„ oxide	1·527	„ „ ether	2·586
Oxygen	1·102	„ sulphur	2·214
Phosphuretted hydrogen	1·77	„ water	·623

Weight of Metals

Weight in Lbs. of 1 Cubic Inch of Metals.

Aluminium = 0·09	Copper = 0·32	Tin	= 0·263
Brass = 0·3	Lead = 0·41	Wrought iron	= 0·28
Cast iron = 0·26	Steel = 0·282	Zinc	= 0·253
	Water = 0·03617		

Example.—Cast iron block 4 × 2 × 3 ins. = 4 × 2 × 3 × 0·26 = 6·24 lbs.

Weight of Sheet, Bar, and Ball Metals.

Weight of a Square Foot of Sheet Metals.

Thickness.	Weight in Lbs.			Thickness.	Weight in Lbs.		
Inch.	Iron.	Copper.	Brass.	Inch.	Iron.	Copper.	Brass.
1/2	20·21	23·12	21·87	1/16	2·52	2·89	2·73
·450	18·19	20·80	19·68	·055	2·22	2·54	2·40
·437	17·66	20·20	19·11	·048	1·94	2·22	2·10
7/16	15·15	17·34	16·40	·042	1·69	1·94	1·83
·340	13·74	15·72	14·87	·035	1·41	1 62	1·53
5/16	12·63	14·45	13·67	·033	1·33	1·52	1·44
·284	11·47	13·13	12·42	·029	1·17	1·34	1·27
·261	10·55	12·06	11·41	·028	1·13	1·29	1·22
1/4	10·10	11·56	10·94	·025	1·01	1·15	1·09
·217	8·77	10·03	9·49	·021	·85	·97	·92
·208	8·40	9·62	9·10	·020	·81	·92	·87
3/16	7·57	8·67	8·20	·018	·73	·83	·79
·166	6·71	7·67	7·26	·015	·60	·69	·65
·158	6·38	7·30	6·91	·013	·52	·60	·57
·137	5·54	6·33	5·99	·012	·48	55	·52
1/8	5·05	5·78	5·47	·010	·40	·46	·43
·109	4·40	5·04	4·77	·009	·36	·41	·40
·094	3·80	4·34	4·11	·008	·32	·37	·35
·080	3·23	3·70	3·50	·007	·28	·32	·30
·072	2·91	3·32	3·15	·005	·20	·23	·22

Weight of Sheet, Bar, and Ball Cast Iron.

(1 cubic foot weighs 450 lbs.)

Sheet.			Bar.		Ball.	Sheet.			Bar.		Ball.
Thickness or Diameter.		Weight of a Square Foot.	Weight of a *Square* Bar, 1 Foot long.	Weight of a *Round* Bar, 1 Foot long.	Weight of a Ball.	Thickness or Diameter.		Weight of a Square Foot.	Weight of a *Square* Bar, 1 Foot long.	Weight of a *Round* Bar, 1 Foot long.	Weight of a Ball.
Ins.	Ft.	Lbs.	Lbs.	Lbs.	Lbs.	Ins.	Ft.	Lbs.	Lbs.	Lbs.	Lbs.
$\frac{1}{32}$	·0026	1·173	·003	·002	—	$3\frac{1}{8}$	·2604	117·3	30·52	23·97	4·162
$\frac{1}{16}$	·0052	2·344	·012	·010	—	$\frac{1}{4}$	·2708	121·8	33·01	25·93	4·681
$\frac{3}{32}$	·0078	3·516	·027	·021	·0001	$\frac{9}{32}$	·2813	126·5	35·60	27·95	5·243
$\frac{1}{8}$	·0104	4·687	·048	·038	·0003	$\frac{5}{16}$	·2917	131·2	38·28	30·07	5·846
$\frac{5}{32}$	·0130	5·861	·076	·060	·0005	$\frac{11}{32}$	·3021	135·9	41·07	32·25	6·498
$\frac{3}{16}$	·0156	7·032	·110	·086	·0009	$\frac{3}{4}$	·3125	140·6	43·95	34·51	7·193
$\frac{7}{32}$	·0182	8·203	·150	·118	·0014	$\frac{7}{8}$	·3229	145·3	46·93	36·85	7·934
$\frac{1}{4}$	·0208	9·375	·195	·154	·0021	4	·3333	150·0	50·01	39·27	8·726
$\frac{9}{32}$	·0234	10·54	·247	·194	·0030	$\frac{1}{4}$	·3438	154·7	53·18	41·77	9·572
$\frac{5}{16}$	·0260	11·73	·305	·240	·0042	$\frac{1}{2}$	·3542	159·3	56·46	44·33	10·47
$\frac{11}{32}$	·0287	12·89	·370	·290	·0056	$\frac{33}{64}$	·3646	164·0	59·82	46·99	11·42
$\frac{3}{8}$	·0313	14·06	·440	·346	·0072	$\frac{1}{2}$	·3750	168·7	63·33	49·71	12·43
$\frac{13}{32}$	·0339	15·24	·516	·400	·0092	$\frac{35}{64}$	·3854	173·4	66·86	52·52	13·49
$\frac{7}{16}$	·0365	16·41	·598	·470	·0114	$\frac{3}{4}$	·3958	178·1	70·52	55·39	14·62
$\frac{15}{32}$	·0391	17·56	·687	·540	·0140	$\frac{7}{8}$	·4063	182·8	74·28	58·34	15·81
$\frac{1}{2}$	·0417	18·75	·781	·610	·0170	5	·4167	187·5	78·12	61·37	17·05
$\frac{9}{16}$	·0469	21·10	·989	·777	·0243	$\frac{1}{8}$	·4271	192·2	82·10	64·47	18·35
$\frac{17}{32}$	·0521	23·44	1·221	·959	·0334	$\frac{1}{4}$	·4375	196·9	86·14	67·65	19·73
$\frac{9}{16}$	·0573	25·79	1·478	1·161	·0444	$\frac{37}{64}$	·4479	201·6	90·29	70·52	21·18
$\frac{5}{8}$	·0625	28·12	1·758	1·381	·0575	$\frac{1}{2}$	·4583	206·2	94·54	74·26	22·68
$\frac{11}{16}$	·0677	30·47	2·064	1·621	·0732	$\frac{39}{64}$	·4688	210·9	98·89	77·66	24·27
$\frac{7}{8}$	·0729	32·81	2·393	1·880	·0913	$\frac{3}{4}$	·4792	215·6	103·3	81·16	25·93
$\frac{13}{16}$	·0781	35·16	2·747	2·158	·1124	$\frac{7}{8}$	·4896	220·3	107·9	84·72	27·41
1	·0833	37·50	3·125	2·455	·1363	6	·5000	225·0	112·5	88·36	29·44
$\frac{1}{16}$	·0885	39·84	3·528	2·771	·1636	$\frac{1}{4}$	·5208	234·4	122·1	95·89	33·28
$\frac{1}{8}$	·0938	42·19	3·955	3·107	·1942	$\frac{1}{2}$	·5417	243·8	132·0	103·7	37·44
$\frac{3}{16}$	·0990	44·53	4·407	3·461	·2284	$\frac{3}{4}$	·5625	253·1	142·4	111·9	41·94
$\frac{1}{4}$	·1042	46·87	4·883	3·835	·2664	7	·5833	262·5	153·2	120·2	46·77
$\frac{5}{16}$	·1094	49·22	5·384	4·229	·3084	$\frac{1}{4}$	·6042	271·9	164·2	129·0	51·97
$\frac{3}{8}$	·1146	51·57	5·909	4·640	·3546	$\frac{1}{2}$	·6250	281·3	175·8	138·1	57·54
$\frac{7}{16}$	·1198	53·91	6·461	5·073	·4058	$\frac{3}{4}$	·6458	290·7	187·7	147·4	63·47
$\frac{1}{2}$	·1250	56·26	7·033	5·523	·4603	8	·6667	300·0	200·1	157·0	69·82
$\frac{9}{16}$	·1302	58·60	7·632	5·993	·5204	$\frac{1}{4}$	·6875	309·4	212·7	167·0	76·58
$\frac{5}{8}$	·1354	60·94	8·253	6·484	·5852	$\frac{1}{2}$	·7083	318·8	225·8	177·3	83·74
$\frac{11}{16}$	·1406	63·28	8·900	6·991	·6555	$\frac{3}{4}$	·7292	328·2	239·3	187·9	91·35
$\frac{3}{4}$	·1458	65·63	9·572	7·518	·7310	9	·7500	337·4	253·1	198·8	99·42
$\frac{13}{16}$	·1510	67·97	10·27	8·064	·8122	$\frac{1}{4}$	·7708	346·8	267·4	210·0	107·9
$\frac{7}{8}$	·1563	70·32	10·99	8·630	·8991	$\frac{1}{2}$	·7917	356·2	282·1	221·5	116·8
$\frac{15}{16}$	·1615	72·66	11·73	9·215	·9920	$\frac{3}{4}$	·8125	365·6	297·0	233·3	126·3
2	·1667	75·01	12·50	9·821	1·073	10	·8333	375·0	312·5	245·5	136·3
$\frac{1}{8}$	·1771	79·70	14·11	11·09	1·308	$\frac{1}{4}$	·8542	384·4	328·4	257·8	146·8
$\frac{1}{4}$	·1875	84·40	15·83	12·43	1·554	$\frac{1}{2}$	·8750	393·7	344·5	270·6	157·9
$\frac{3}{8}$	·1979	89·07	17·63	13·85	1·827	$\frac{3}{4}$	·8958	403·1	361·2	283·7	169·3
$\frac{1}{2}$	·2083	93·75	19·54	15·34	2·131	11	·9167	412·5	378·2	297·0	181·5
$\frac{5}{8}$	·2188	98·44	21·54	16·56	2·467	$\frac{1}{4}$	·9375	421·9	395·5	310·6	194·2
$\frac{3}{4}$	·2292	103·2	23·64	18·56	2·835	$\frac{1}{2}$	·9583	431·2	413·3	324·6	207·3
$\frac{7}{8}$	·2396	107·8	25·84	20·29	3·241	$\frac{3}{4}$	·9792	440·6	431·4	338·8	219·2
3	·2500	112·6	28·13	22·10	3·682	12	1 Foot	450·0	450·0	353·4	235·6

Weight of Wrought Iron and Steel.

(The Table gives the weights for wrought iron, 1 cubic foot of wrought iron being taken to weigh 485 lbs. For steel an addition of about 1 lb. in 100 lbs. must be made to the tabular amounts.)

Thickness or Diameter in Inches.	Thickness or Diameter in Decimals of a Foot.	Weight of a *Square Foot* in Pounds.	Weight of a *Square* Bar, 1 Foot long, in Pounds.	Weight of a *Round* Bar, 1 Foot long, in Pounds.	Weight of *Balls* in Pounds.
1/32	·0026	1·263	·0033	·0026	—
1/16	·0052	2·526	·0132	·0104	—
3/32	·0078	3·789	·0296	·0233	·0001
1/8	·0104	5·052	·0526	·0414	·0003
5/32	·0130	6·315	·0823	·0646	·0005
3/16	·0156	7·578	·1184	·0930	·0009
7/32	·0182	8·841	·1612	·1266	·0015
1/4	·0208	10·10	·2105	·1653	·0023
9/32	·0234	11·37	·2665	·2093	·0033
5/16	·0260	12·63	·3290	·2583	·0045
11/32	·0287	13·89	·3980	·3126	·0060
3/8	·0313	15·16	·4736	·3720	·0078
13/32	·0339	16·42	·5558	·4365	·0098
7/16	·0365	17·68	·6446	·5063	·0123
15/32	·0391	18·95	·7400	·5813	·0151
1/2	·0417	20·21	·8420	·6613	·0184
9/16	·0469	22·73	1·066	·8370	·0262
5/8	·0521	25·26	1·316	1·033	·0359
11/16	·0573	27·79	1·592	1·250	·0478
3/4	·0625	30·31	1·895	1·488	·0620
13/16	·0677	32·84	2·223	1·746	·0788
7/8	·0729	35·37	2·579	2·025	·0985
15/16	·0781	37·89	2·960	2·325	·1211
1	·0833	40·42	3·368	2·645	·1470
1 1/16	·0885	42·94	3·803	2·986	·1763
1 1/8	·0938	45·47	4·263	3·348	·2093
1 3/16	·0990	48·00	4·750	3·730	·2461
1 1/4	·1042	50·52	5·263	4·133	·2870
1 5/16	·1094	53·05	5·802	4·557	·3323
1 3/8	·1146	55·57	6·368	5·001	·3820
1 7/16	·1198	58·10	6·960	5·466	·4365
1 1/2	·1250	60·63	7·578	5·952	·4960
1 9/16	·1302	63·15	8·223	6·458	·5606
1 5/8	·1354	65·68	8·893	6·985	·6306
1 11/16	·1406	68·20	9·591	7·533	·7062
1 3/4	·1458	70·73	10·31	8·101	·7876
1 13/16	·1510	73·26	11·07	8·690	·8750
1 7/8	·1563	75·78	11·84	9·300	·9688
1 15/16	·1615	78·31	12·64	9·930	1·069
2	·1667	80·83	13·47	10·58	1·176
2 1/8	·1771	85·89	15·21	11·95	1·410
2 1/4	·1875	90·94	17·05	13·39	1·674
2 3/8	·1979	95·99	19·00	14·92	1·969
2 1/2	·2083	101·0	21·05	16·53	2·296
2 5/8	·2188	106·1	23·21	18·23	2·658
2 3/4	·2292	111·2	25·47	20·01	3·056
2 7/8	·2396	116·2	27·84	21·87	3·492
3	·2500	121·3	30·31	23·81	3·968
3 1/8	·2604	126·3	32·89	25·83	4·484
3 1/4	·2708	131·4	35·57	27·94	5·045
3 3/8	·2813	136·4	38·37	30·13	5·649
3 1/2	·2917	141·5	41·26	32·41	6·301
3 5/8	·3021	146·5	44·26	34·76	7·000
3 3/4	·3125	151·6	47·37	37·20	7·750
3 7/8	·3229	156·6	50·57	39·72	8·550
4	·3333	161·7	53·89	42·33	9·405
4 1/8	·3438	166·7	57·31	45·01	10·32
4 1/4	·3542	171·8	60·84	47·78	11·28
4 3/8	·3646	176·8	64·47	50·63	12·31
4 1/2	·3750	181·9	68·20	53·57	13·39
4 5/8	·3854	186·9	72·05	56·59	14·54
4 3/4	·3958	192·0	75·99	59·69	15·75
4 7/8	·4063	197·0	80·05	62·87	17·03
5	·4167	202·1	84·20	66·13	18·37
5 1/8	·4271	207·1	88·47	69·48	19·78
5 1/4	·4375	212·2	92·83	72·91	21·26
5 3/8	·4479	217·2	97·31	76·43	22·82
5 1/2	·4583	222·3	101·9	80·02	24·45
5 5/8	·4688	227·3	106·6	83·70	26·16
5 3/4	·4792	232·4	111·4	87·46	27·94
5 7/8	·4896	237·5	116·3	91·31	29·80
6	·5000	242·5	121·3	95·23	31·74
6 1/4	·5208	252·6	131·6	103·3	35·88
6 1/2	·5417	262·7	142·3	111·8	40·36
6 3/4	·5625	272·8	153·5	120·5	45·19
7	·5833	282·9	165·0	129·6	50·40
7 1/4	·6042	293·0	177·0	139·0	56·00
7 1/2	·6250	303·1	189·5	148·8	62·00
7 3/4	·6458	313·2	202·3	158·9	68·40
8	·6667	323·3	215·6	169·3	75·24
8 1/4	·6875	333·4	229·3	180·1	82·52
8 1/2	·7083	343·5	243·4	191·1	90·25
8 3/4	·7292	353·6	247·9	202·5	98·45
9	·7500	363·8	272·8	214·3	107·1
9 1/4	·7708	373·9	288·2	226·3	116·3
9 1/2	·7917	384·0	304·0	238·7	126·0
9 3/4	·8125	394·1	320·2	251·5	136·2
10	·8333	404·2	336·8	264·5	146·9
10 1/4	·8542	414·3	353·9	277·9	158·2
10 1/2	·8750	424·4	371·3	291·6	170·1
10 3/4	·8958	434·5	389·2	305·7	182·6
11	·9167	444·6	407·5	320·1	195·6
11 1/4	·9375	454·7	426·3	334·8	209·2
11 1/2	·9583	464·8	445·4	349·8	223·5
11 3/4	·9792	474·9	465·0	365·2	238·4
12	1 Foot	485·0	485·0	380·9	253·9

WEIGHT OF BALLS OF VARIOUS METALS.

Diameter	Cast Lead.	Cast Copper.	Cast Brass.	Cast Iron.	Diameter	Cast Lead.	Cast Copper.	Cast Brass.	Cast Iron.
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Ins.	Lbs.	Lbs.	Lbs.	Lbs.
$\frac{1}{8}$	·026	·021	·019	·017	5	26·0	20·8	18·6	17·0
$\frac{3}{8}$	·088	·070	·063	·058	$\frac{1}{4}$	30·1	24·1	21·5	19·8
1	·209	·167	·148	·136	$\frac{1}{2}$	34·7	27·7	24·7	22·7
$\frac{1}{4}$	·408	·325	·290	·266	$\frac{3}{4}$	39·6	31·7	28·3	25·9
$\frac{1}{2}$	·705	·562	·501	·460	6	45·0	36·0	32·0	29·4
$\frac{3}{4}$	1·12	·893	·795	·731	$\frac{1}{2}$	57·2	45·8	40·8	37·4
2	1·67	1·33	1·19	1·07	7	71·5	57·2	50·9	46·8
$\frac{1}{4}$	2·38	1·90	1·69	1·55	$\frac{1}{2}$	88·0	70·3	62·6	57·5
$\frac{1}{2}$	3·25	2·60	2·32	2·13	8	106	85·3	76·0	69·8
$\frac{3}{4}$	4·34	3·47	3·09	2·83	$\frac{1}{2}$	127	102	91·2	83·7
3	5·63	4·50	4·01	3·68	9	151	121	108	99·4
$\frac{1}{4}$	7·15	5·72	5·10	4·68	$\frac{1}{2}$	178	143	127	117
$\frac{1}{2}$	8·94	7·14	6·36	5·85	10	208	167	148	136
$\frac{3}{4}$	11·0	8·79	7·83	7·19	$\frac{1}{2}$	241	193	172	158
4	13·4	10·7	9·50	8·73	11	277	222	198	182
$\frac{1}{4}$	16·0	12·8	11·4	10·5	$\frac{1}{2}$	317	253	226	207
$\frac{1}{2}$	18·9	15·2	13·5	12·4	12	360	288	257	236
$\frac{3}{4}$	22·7	17·9	15·9	14·6					

WEIGHT OF HEXAGON ROD METAL.

Weight of hexagon rod metal, measured across sides = weight of square metal × ·86602.

WEIGHT OF OCTAGON ROD METAL.

Weight of octagon rod metal, measured across sides = weight of square metal × ·82843.

WEIGHT OF FLAT ROLLED IRON PER FOOT RUN.

Width.	Thickness of Metal in Parts of an Inch.											
	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1
Ins.	Lb.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
1	·2083	·4166	·6250	·8333	1·042	1·250	1·458	1·666	2·083	2·500	2·916	3·333
$\frac{1}{8}$	·2344	·4688	·7033	·9375	1·172	1·406	1·640	1·875	2·344	2·812	3·280	3·75
$\frac{1}{4}$	·2605	·5210	·7810	1·042	1·303	1·563	1·823	2·083	2·605	3·125	3·646	4·166
$\frac{3}{8}$	·2865	·5730	·8595	1·146	1·432	1·719	2·006	2·292	2·864	3·438	4·012	4·583
$\frac{1}{2}$	·3125	·6250	·9375	1·250	1·562	1·875	2·188	2·500	3·125	3·750	4·375	5·000
$\frac{5}{8}$	·3385	·6771	1·015	1·354	1·692	2·031	2·370	2·708	3·384	4·062	4·740	5·416
$\frac{3}{4}$	·3646	·7292	1·094	1·458	1·823	2·188	2·550	2·916	3·646	4·375	5·105	5·833
$\frac{7}{8}$	·3906	·7812	1·172	1·562	1·953	2·344	2·735	3·125	3·906	4·688	5·470	6·25
2	·4166	·8333	1·25	1·666	2·083	2·500	2·916	3·333	4·166	5·000	5·833	6·666
$\frac{1}{8}$	·4427	·8855	1·328	1·771	2·214	2·656	3·098	3·542	4·428	5·312	6·196	7·083
$\frac{1}{4}$	·4688	·9375	1·406	1·875	2·344	2·812	3·281	3·750	4·688	5·624	6·562	7·500
$\frac{3}{8}$	·4948	·9895	1·484	1·979	2·474	2·968	3·463	3·958	4·948	5·936	6·926	7·916
$\frac{1}{2}$	·5210	1·042	1·562	2·083	2·605	3·125	3·646	4·166	5·210	6·250	7·291	8·333
$\frac{5}{8}$	·5470	1·094	1·641	2·187	2·735	3·282	3·829	4·375	5·470	6·564	7·658	8·750
$\frac{3}{4}$	·5730	1·146	1·719	2·292	2·865	3·438	4·011	4·583	5·730	6·876	8·022	9·166
$\frac{7}{8}$	·5990	1·198	1·797	2·396	2·995	3·594	4·193	4·792	5·990	7·188	8·386	9·583
3	·625	1·250	1·875	2·500	3·125	3·750	4·375	5·000	6·250	7·500	8·750	10·00
$\frac{1}{8}$	·6515	1·303	1·954	2·605	3·257	3·908	4·560	5·210	6·514	7·816	9·120	10·42
$\frac{1}{4}$	·6770	1·354	2·031	2·708	3·385	4·062	4·739	5·416	6·770	8·124	9·478	10·83
$\frac{3}{8}$	·7031	1·406	2·109	2·812	3·516	4·218	4·921	5·625	7·032	8·436	9·842	11·25
$\frac{1}{2}$	·7291	1·458	2·188	2·916	3·646	4·375	5·105	5·833	7·291	8·750	10·21	11·66
$\frac{5}{8}$	·7555	1·511	2·266	3·021	3·777	4·533	5·288	6·042	7·554	9·066	10·58	12·08
$\frac{3}{4}$	·7812	1·562	2·343	3·125	3·906	4·686	5·468	6·25	7·812	9·372	10·94	12·50
$\frac{7}{8}$	8070	1·614	2·421	3·229	4·035	4·842	5·65	6·458	8·070	9·684	11·30	12·92

WEIGHT OF FLAT ROLLED IRON PER FOOT RUN (*continued*).

Width.	Thickness of Metal in Parts of an Inch.											
	1/16	1/8	3/16	1/4	5/16	3/8	7/16	1/2	5/8	3/4	7/8	1
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
4	·8333	1·666	2·500	3·333	4·166	5·000	5·833	6·666	8·333	10·00	11·66	13·33
1/32	·8595	1·719	2·578	3·438	4·297	5·166	6·016	6·875	8·594	10·31	12·03	13·75
1/16	·8855	1·771	2·656	3·542	4·427	5·312	6·198	7·083	8·854	10·62	12·40	14·16
3/32	·9115	1·823	2·734	3·646	4·557	5·468	6·380	7·291	9·114	10·94	12·76	14·58
1/8	·9375	1·875	2·812	3·750	4·687	5·624	6·562	7·500	9·374	11·25	13·12	15·00
5/32	·9636	1·927	2·891	3·854	4·818	5·782	6·745	7·708	9·636	11·56	13·49	15·42
3/16	·9895	1·979	2·968	3·958	4·947	5·936	6·926	7·917	9·894	11·87	13·85	15·83
5	1·042	2·083	3·125	4·166	5·210	6·25	7·291	8·333	10·42	12·50	14·58	16·66
1/32	1·068	2·136	3·204	4·271	5·340	6·408	7·476	8·542	10·68	12·81	14·95	17·08
1/16	1·094	2·188	3·282	4·375	5·470	6·564	7·658	8·750	10·94	13·13	15·31	17·50
3/32	1·120	2·240	3·360	4·479	5·600	6·720	7·840	8·958	11·20	13·44	15·68	17·92
1/8	1·146	2·292	3·438	4·584	5·730	6·876	8·022	9·167	11·46	13·75	16·04	18·33
5/32	1·172	2·344	3·516	4·687	5·860	7·032	8·204	9·375	11·72	14·06	16·40	18·75
3/16	1·198	2·396	3·594	4·791	5·990	7·188	8·386	9·583	11·98	14·37	16·77	19·16
6	1·224	2·448	3·672	4·896	6·120	7·344	8·568	9·792	12·24	14·69	17·13	19·58
1/32	1·250	2·500	3·750	5·000	6·250	7·500	8·750	10·00	12·50	15·00	17·50	20·00
1/16	1·276	2·552	3·828	5·104	6·380	7·656	8·932	10·21	12·76	15·31	17·86	20·42
3/32	1·302	2·604	3·906	5·208	6·510	7·812	9·114	10·42	13·02	15·62	18·23	20·83
1/8	1·328	2·657	3·984	5·313	6·640	7·968	9·297	10·63	13·28	15·93	18·59	21·25
5/32	1·354	2·708	4·063	5·417	6·770	8·126	9·480	10·83	13·54	16·25	18·96	21·66
3/16	1·381	2·761	4·143	5·521	6·906	8·286	9·668	11·04	13·81	16·57	19·33	22·08
7	1·406	2·813	4·218	5·625	7·030	8·436	9·843	11·25	14·06	16·87	19·69	22·50
1/32	1·432	2·864	4·296	5·729	7·160	8·592	10·02	11·46	14·32	17·18	20·04	22·92
1/16	1·458	2·916	4·375	5·833	7·291	8·750	10·20	11·66	14·58	17·50	20·42	23·33
3/32	1·484	2·969	4·452	5·938	7·420	8·904	10·39	11·87	14·84	17·81	20·78	23·75
1/8	1·511	3·021	4·533	6·042	7·555	9·066	10·58	12·08	15·11	18·13	21·16	24·16
5/32	1·536	3·073	4·608	6·146	7·680	9·216	10·75	12·29	15·36	18·43	21·50	24·58
3/16	1·562	3·125	4·686	6·250	7·810	9·372	10·93	12·50	15·62	18·74	21·86	25·00
7/32	1·588	3·177	4·764	6·354	7·940	9·528	11·12	12·71	15·88	19·05	22·24	25·42
1/4	1·615	3·229	4·845	6·458	8·075	9·690	11·31	12·92	16·15	19·38	22·62	25·83
9/32	1·641	3·281	4·923	6·562	8·205	9·846	11·48	13·13	16·41	19·69	22·96	26·25
8	1·666	3·333	5·000	6·666	8·333	10·00	11·66	13·33	16·66	20·00	23·33	26·66
1/32	1·693	3·386	5·079	6·771	8·455	10·15	11·85	13·54	16·91	20·30	23·70	27·08
1/16	1·719	3·438	5·157	6·875	8·595	10·31	12·03	13·75	17·19	20·61	24·06	27·50
3/32	1·745	3·489	5·235	6·979	8·725	10·47	12·21	13·96	17·45	20·94	24·42	27·92
1/8	1·771	3·542	5·313	7·083	8·855	10·63	12·40	14·17	17·71	21·26	24·80	28·33
5/32	1·797	3·594	5·391	7·188	8·985	10·78	12·58	14·37	17·97	21·56	25·16	28·75
3/16	1·823	3·646	5·469	7·292	9·115	10·94	12·76	14·58	18·23	21·88	25·52	29·17
7/32	1·849	3·698	5·547	7·396	9·245	11·09	12·94	14·79	18·49	22·18	25·88	29·58
9	1·875	3·750	5·625	7·500	9·375	11·25	13·12	15·00	18·75	22·50	26·24	30·00
1/32	1·901	3·802	5·703	7·604	9·505	11·41	13·31	15·21	19·00	22·81	26·62	30·42
1/16	1·927	3·854	5·781	7·708	9·635	11·56	13·49	15·42	19·27	23·12	26·98	30·83
3/32	1·953	3·906	5·859	7·812	9·765	11·72	13·67	15·62	19·53	23·44	27·34	31·25
1/8	1·979	3·958	5·937	7·916	9·895	11·87	13·85	15·84	19·79	23·74	27·70	31·67
5/32	2·005	4·010	6·015	8·021	10·02	12·03	14·04	16·04	20·04	24·06	28·08	32·08
3/16	2·031	4·062	6·093	8·125	10·16	12·18	14·21	16·25	20·32	24·36	28·42	32·50
7/32	2·057	4·114	6·171	8·229	10·29	12·34	14·40	16·46	20·58	24·68	28·80	32·92
10	2·083	4·166	6·250	8·333	10·41	12·50	14·58	16·66	20·82	25·00	29·16	33·33
1/32	2·109	4·219	6·327	8·438	10·55	12·65	14·76	16·87	21·10	25·30	29·52	33·75
1/16	2·135	4·270	6·405	8·541	10·67	12·81	14·94	17·08	21·34	25·62	29·88	34·17
3/32	2·162	4·323	6·486	8·646	10·81	12·97	15·13	17·29	21·62	25·94	30·26	34·58
1/8	2·188	4·375	6·564	8·750	10·94	13·13	15·31	17·50	21·88	26·26	30·62	35·00
5/32	2·214	4·427	6·642	8·854	11·07	13·28	15·50	17·71	22·14	26·56	31·00	35·42
3/16	2·239	4·479	6·717	8·958	11·20	13·43	15·67	17·92	22·40	26·86	31·34	35·83
7/32	2·266	4·531	6·798	9·062	11·33	13·59	15 86	18·12	22·66	27·18	31·72	36·25
11	2·291	4·583	6·873	9·166	11·46	13·75	16·04	18·33	22·90	27·50	32·08	36·66
1/32	2·318	4·636	6·954	9·271	11·59	13·91	16·22	18·54	23·18	27·82	32·44	37·08
1/16	2·344	4·688	7·032	9·375	11·72	14·06	16·40	18·75	23·44	28·12	32·80	37·50
3/32	2·370	4·740	7·110	9·479	11·85	14·22	16·59	18·96	23·70	28·44	33·18	37·92
1/8	2·395	4·791	7·185	9·582	11·97	14·37	16·76	19·16	23·94	28·74	33·52	38·33
5/32	2·422	4·844	7·266	9·688	12·11	14·53	16·95	19·37	24·22	29·06	33·90	38·75
3/16	2·448	4·896	7·344	9·792	12·24	14·68	17·13	19·58	24·48	29·36	34·26	39·16
7/32	2·474	4·948	7·422	9·896	12·37	14·84	17·32	19·79	24·74	29·68	34·64	39·58
12	2·500	5·000	7·500	10·00	12·50	15·00	17·50	20·00	25·00	30·00	35·00	40·00

WEIGHT OF FLAT STEEL PER FOOT RUN.

Width.	Thickness of Metal in Parts of an Inch.							
	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
$\frac{1}{8}$	·0532	·1063	·1595	·2127	·2658	·3190	·3722	·4253
$\frac{1}{2}$	·2127	·4253	·6380	·8507	1·063	1·276	1·489	1·701
$\frac{5}{8}$	·2658	·5317	·7975	1·063	1·329	1·595	1·861	2·127
$\frac{3}{4}$	·3190	·6380	·9570	1·276	1·595	1·914	2·233	2·552
$\frac{7}{8}$	·3722	·7443	1·117	1·489	1·861	2·233	2·605	2·977
1	·4253	·8507	1·276	1·701	2·127	2·552	2·976	3·403
$1\frac{1}{8}$	·4785	·9570	1·436	1·914	2·393	2·871	3·350	3·828
$1\frac{1}{4}$	·5317	1·063	1·595	2·127	2·658	3·190	3·722	4·253
$1\frac{3}{8}$	·5849	1·170	1·755	2·339	2·924	3·509	4·094	4·679
$1\frac{1}{2}$	·6380	1·276	1·914	2·552	3·190	3·828	4·466	5·104
$1\frac{3}{4}$	·7444	1·489	2·233	2·977	3·722	4·466	5·211	5·955
2	·8507	1·701	2·552	3·403	4·253	5·104	5·955	6·806
$2\frac{1}{4}$	·9570	1·914	2·871	3·828	4·785	5·742	6·699	7·656
$2\frac{1}{2}$	1·063	2·127	3·190	4·253	5 317	6·380	7·444	8·507
$2\frac{3}{4}$	1·170	2·339	3·509	4·679	5·849	7·018	8·188	9·358
3	1·276	2·552	3·828	5·104	6·380	7·656	8·932	10·208
$3\frac{1}{4}$	1·382	2·765	4·147	5·530	6·912	8·294	9·677	11·059
$3\frac{1}{2}$	1·489	2·977	4·466	5·955	7·444	8·932	10·421	11·910
$3\frac{3}{4}$	1·595	3·190	4·785	6·380	7·975	9·570	11·165	12·760
4	1·701	3·403	5·104	6·806	8·507	10·208	11·910	13·611
$4\frac{1}{4}$	1·808	3·615	5·423	7·231	9·039	10·846	12·654	14·462
$4\frac{1}{2}$	1·914	3·828	5·742	7·656	9·570	11·484	13·398	15·313
$4\frac{3}{4}$	2·020	4·041	6·061	8·082	10·102	12·122	14·143	16·163
5	2·127	4·254	6·380	8·507	10·634	12·760	14·887	17·014
$5\frac{1}{2}$	2·339	4·679	7·018	9·358	11·697	14·037	16·376	18·715
6	2·552	5·104	7·656	10·208	12·761	15·313	17·865	20·417

WEIGHT OF ROUND, OCTAGONAL, AND SQUARE STEEL PER FOOT RUN.

Diam.	Round.	Octag.	Square.	Diam.	Round.	Octag.	Square.	Diam.	Round.	Octag.	Square.
In.	Lb.	Lb.	Lb.	Ins.	Lbs.	Lbs.	Lbs.	Ins.	Lbs.	Lbs.	Lbs.
$\frac{1}{16}$	·010	·011	·013	$\frac{5}{8}$	1·044	1·101	1·329	$1\frac{3}{8}$	5·053	5·230	6.433
$\frac{1}{8}$	·041	·044	·053	$\frac{11}{16}$	1·263	1·332	1·608	$1\frac{7}{16}$	6·013	6·343	7·656
$\frac{3}{16}$	·094	·099	·119	$\frac{3}{4}$	1·503	1·586	1·914	$1\frac{1}{2}$	7·057	7·444	8·985
$\frac{1}{4}$	·167	·176	·213	$\frac{13}{16}$	1·764	1·861	2·246	$1\frac{9}{16}$	8·185	8·633	10·421
$\frac{5}{16}$	·261	·275	·332	$\frac{7}{8}$	2·046	2 158	2·605	$1\frac{5}{8}$	9·396	9·910	11·963
$\frac{3}{8}$	·376	·396	·478	$\frac{15}{16}$	2·349	2·478	2·991	2	10·690	11·276	13·611
$\frac{7}{16}$	·511	·539	·651	1	2·673	2·819	3·403	$2\frac{1}{4}$	13·530	14·271	17·227
$\frac{1}{2}$	·668	·706	·851	$1\frac{1}{8}$	3·382	3·568	4·307	$2\frac{1}{2}$	16·703	17·618	21·267
$\frac{9}{16}$	·845	·892	1·077	$1\frac{1}{4}$	4·176	4·405	5·317	$2\frac{3}{4}$	20·211	21·318	25·734

(Diameter of octagon steel is measured across sides.)

WEIGHT OF ROLLED LEAD, COPPER, AND BRASS: SHEETS AND BARS.

Thickness or Diameter, or Side.	Lead.			Copper.			Brass.			Thickness or Diameter, or Side.
	Sheets, per Square Foot.	Square Bars, 1 Foot long.	Round Bars, 1 Foot long.	Sheets, per Square Foot.	Square Bars, 1 Foot long.	Round Bars, 1 Foot long.	Sheets, per Square Foot.	Square Bars, 1 Foot long.	Round Bars, 1 Foot long.	
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Ins.
$\frac{1}{32}$	1·86	·005	·004	1·44	·004	·003	1·36	·004	·003	$\frac{1}{32}$
$\frac{1}{16}$	3·72	·019	·015	2·89	·015	·012	2·71	·014	·011	$\frac{1}{16}$
$\frac{3}{32}$	5·58	·044	·034	4·33	·034	·027	4·06	·032	·025	$\frac{3}{32}$
$\frac{1}{8}$	7·44	·078	·061	5·77	·060	·047	5·42	·056	·044	$\frac{1}{8}$
$\frac{5}{32}$	9·30	·121	·095	7·20	·094	·074	6·75	·088	·069	$\frac{5}{32}$
$\frac{3}{16}$	11·2	·174	·137	8·66	·135	·106	8·13	·127	·100	$\frac{3}{16}$
$\frac{7}{32}$	13·0	·237	·187	10·1	·184	·144	9·50	·173	·136	$\frac{7}{32}$
$\frac{1}{4}$	14·9	·310	·244	11·5	·240	·189	10·8	·226	·177	$\frac{1}{4}$
$\frac{5}{16}$	18·6	·485	·381	14·4	·376	·295	13·5	·353	·277	$\frac{5}{16}$
$\frac{3}{8}$	22·3	·698	·548	17·3	·541	·425	16·3	·508	·399	$\frac{3}{8}$

WEIGHT OF ROLLED LEAD, COPPER, AND BRASS: SHEETS AND BARS—*continued.*

Thickness or Diameter, or Side.	LEAD.			COPPER.			BRASS.			Thickness or Diameter, or Side.
	Sheets, per Square Foot.	Square Bars, 1 Foot long.	Round Bars, 1 Foot long.	Sheets, per Square Foot.	Square Bars, 1 Foot long.	Round Bars, 1 Foot long.	Sheets, per Square Foot.	Square Bars, 1 Foot long.	Round Bars, 1 Foot long.	
Ins.	Lbs.	Lbs	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Ins.
7/16	26·0	·950	·746	20·2	·736	·578	19·0	·691	·543	7/16
1/2	29·8	1·24	·974	23·1	·962	·755	21·7	·903	·709	1/2
9/16	33·5	1·57	1·23	26·0	1·22	·955	24·3	1·14	·900	9/16
5/8	37·2	1·94	1·52	28·9	1·50	1·18	27·1	1·41	1·11	5/8
11/16	40·9	2·34	1·84	31·7	1·82	1·43	29·8	1·70	1·34	11/16
3/4	44·6	2·79	2·19	34·6	2·16	1·70	32·5	2·03	1·60	3/4
13/16	48·3	3·27	2.57	37 5	2·55	1·99	35·2	2·38	1·87	13/16
7/8	52·1	3·80	2·98	40·4	2·94	2·31	37·9	2·76	2·17	7/8
15/16	56·0	4·37	3·42	43·3	3·38	2·65	40·6	3·18	2·49	15/16
1	59·5	4·96	3·90	46·2	3·85	3·02	43·3	3·61	2·84	1
1 1/8	66·9	6·27	4·92	52·0	4·87	3·82	48·7	4·57	3·60	1 1/8
1 1/4	74·4	7·75	6·09	57·7	6·01	4·72	54·2	5·64	4·43	1 1/4
1 3/8	81·8	9·37	7·37	63·5	7·28	5·72	59·6	6·82	5·37	1 3/8
1 1/2	89·3	11·2	8·77	69 3	8·65	6·80	65 0	8·12	6·38	1 1/2
1 3/4	104	15·2	11·9	80·8	11·8	9·25	75·9	11·1	8·68	1 3/4
2	119	19·8	15·6	92·3	15·4	12·1	86·7	14·4	11·3	2

WEIGHT OF SHEET LEAD.

Lbs. per Sq. Ft.	Thickness. Inch.	Lbs. per Sq. Ft.	Thickness. Inch.	Lbs. per Sq. Ft.	Thickness. Inch.	Lbs. per Sq. Ft.	Thickness. Inch.	Lbs. per Sq. Ft.	Thickness. Inch.	Lbs. per Sq. Ft.	Thickness. Inch.
2·5	·042	4	·068	6	102	8	·136	10	·170	14	·237
3	·051	5	·085	7	119	9	·153	12	·203	16	·271

WEIGHT OF SHEET ZINC.

Gauge No.	Approximate Thickness.		Approximate weight per square foot.			Approximate weight of Sheets. 36 ins. × 72 ins.			36 ins. × 84 ins.			36 ins. × 96 ins.		
	In.	Mm.	lbs.	oz.	drms.	lbs.	oz.	drms.	lbs.	oz.	drms.	lbs.	oz.	drms.
3	0·007	0·171	...	3	15	4	6	14	5	2	11	5	14	8
4	0·008	0·209	...	4	13	5	6	10	6	5	1	7	3	8
5	0·010	0·247	...	5	11	6	6	6	7	7	7	8	8	8
6	0·011	0·291	...	6	11	7	8	6	8	12	7	10	0	8
7	0·013	0·337	...	7	12	8	11	8	10	2	12	12	10	0
8	0·015	0·386	...	8	14	9	15	12	11	10	6	13	5	0
9	0·018	0·450	...	10	5	11	9	10	13	8	9	15	7	8
10	0·020	0·500	...	11	7	12	13	14	15	0	3	17	2	8
12	0·026	0·660	...	15	2	17	0	4	19	13	10	22	11	0
14	0·032	0·820	1	2	12	21	1	8	24	9	12	28	2	0
16	0·043	1·080	1	8	12	27	13	8	32	7	12	37	2	0
18	0·053	1·340	1	14	11	34	8	6	40	4	7	46	0	8
20	0·063	1·600	2	4	10	41	3	4	48	1	2	54	15	0
22	0·077	1·960	2	12	14	50	7	12	58	14	6	67	5	0
24	0·091	2·320	3	5	3	59	13	6	69	12	15	79	12	8
26	0·105	2·680	3	13	7	69	1	14	80	10	3	92	2	8

SPECIFICATION OF SHEET METALS.

When specifying for sheet metals, care should be taken that the nature of the gauge is stated, *i.e.* whether it is S.W.G., B.W.G., or Trade Gauge. In the case of 'Sheet Zinc,' for example, the Trade Gauge No. 20 is 0·063 in. (see above, and Index, 'Sheet Zinc Trade Gauge'), but this if expressed in S.W.G. is 0·36 in. (see Index, 'Standard Wire Gauge').

WEIGHT AND ELECTRICAL VALUE OF ALUMINIUM BAR AND STRIP.

Dimensions.		Sectional Area.	Weight.	Resistance at 20° C.	Current-carrying Capacity.	
Width.	Thickness.	Sq. ins.	Lbs. per ft.	Microhms per ft.	Amps. for 30° C. rise.	Amps. at 650 per sq. in.
6	½	3·000	3·510	4·47	1,940	1,950
6	⅜	2·250	2·633	5·96	1,692	1,461
6	¼	1·500	1·755	8·95	1,397	975
5	½	2·500	2·935	5·36	1,649	1,625
5	⅜	1·875	2·195	7·15	1,430	1,219
5	¼	1·250	1·463	10·72	1,179	812
5	3/16	0·937	1·096	14·3	1,029	609
4½	½	2·250	2·633	5·96	1,500	1,461
4½	⅜	1·687	1·972	7·95	1,300	1,097
4½	¼	1·125	1·318	11·9	1,070	731
4½	3/16	0·844	0·988	15·9	933	549
4	½	3·000	3·510	4·47	1,660	1,950
4	⅜	2·000	2·340	6·70	1,348	1,300
4	¼	1·500	1·755	8·95	1,169	975
4	5/16	1·000	1·170	13·4	960	650
4	3/16	0·750	0·878	17·9	836	487
4	⅛	0·500	0·585	26·8	692	325
3½	½	1·750	2·050	7·67	1,197	1,138
3½	¼	0·875	1·022	15·3	849	569
3½	5/16	1·625	1·901	8·25	1,122	1,057
3½	3/16	0·812	0·950	16·5	793	527
3	⅜	1·125	1·318	11·9	902	731
3	¼	0·750	0·878	17·9	737	487
3	3/16	0·562	0·658	23·9	641	365
2⅝	¼	0·687	0·804	19·5	680	446
2 9/16	⅜	0·961	1·124	14·0	783	625
2½	½	1·250	1·462	10·72	890	812
2½	⅜	0·937	1·096	14·30	766	609
2½	5/16	0·781	0·914	17·2	698	508
2½	¼	0·625	0·731	21·5	624	406
2¼	½	1·687	1·972	7·95	1,020	1,097
2¼	⅜	1·125	1·318	11·9	814	731
2¼	⅜	0·844	0·988	15·9	698	549
2¼	5/16	0·703	0·823	19·1	636	457
2¼	¼	0·562	0·658	23·9	568	365
2¼	3/16	0·422	0·494	31·8	493	274
2	½	1·500	1·755	8·95	926	975
2	⅜	1·250	1·463	10·72	834	812
2	⅜	1·000	1·170	13·4	736	650
2	⅜	0·750	0·878	17·9	630	487
2	5/16	0·625	0·731	21·5	672	406
2	¼	0·500	0·585	26·8	511	325
2	3/16	0·375	0·439	35·8	442	244
2	⅛	0·250	0·293	53·7	364	162
1 3/16	5/16	0·566	0·662	23·7	525	368
1⅞	⅜	1·093	1·280	12·25	747	710
1¾	½	0·875	1·022	15·32	657	569
1¾	¼	0·437	0·512	30·7	453	284
1¾	5/16	0·328	0·384	40·9	392	213
1¾	⅛	0·219	0·256	61·3	322	142
1⅝	¼	0·406	0·475	33·0	425	264
1⅝	3/16	0·305	0·357	44·0	368	198
1⅝	⅛	0·203	0·238	66·1	301	132
1½	½	0·750	0·878	17·9	577	487
1½	⅜	0·562	0·658	23·9	492	365
1½	¼	0·375	0·439	35·8	396	244
1½	9/128	0·281	0·329	47·7	342	183
1¼	⅛	0·188	0·220	71·4	280	122

(The British Aluminium Co. Ltd.)

WEIGHT OF SHEET ALUMINIUM.

Size S.W.G.	Birmingham Sheet Gauge. Old Style	Birmingham Sheet Gauge. Legalized. 1914.	Thickness. Inches.	Mm.	Weight in lbs. per Sq. Ft.
			0·375	9·53	5·26
		1	0·353	8·96	4·96
1/0			0·324	8·23	4·55
			0·312	7·93	4·38
	39		0·289	7·34	4·06
	38		0·278	7·06	3·90
	37		0·270	6·86	3·79
	36	4	0·250	6·35	3·51
4			0·232	5·89	3·26
	34		0·216	5·49	3·03
	33		0·200	5·08	2·81
6			0·192	4·88	2·70
	32		0·182	4·62	2·56
	31		0·166	4·22	2·33
		8	0·157	3·99	2·20
9			0·144	3·66	2·02
	29		0·136	3·45	1·91
		10	0·125	3·18	1·76
11			0·116	2·95	1·63
		11	0·111	2·82	1·56
	26		0·100	2·54	1·41
13			0·092	2·34	1·29
		13	0·088	2·24	1·24
14			0·080	2·03	1·12
	23		0·077	1·96	1·08
		15	0·070	1·78	0·982
	20		0·065	1·65	0·914
	19		0·063	1·60	0·885
	18		0·060	1·53	0·843
	17		0·055	1·40	0·773
		18	0·050	1·27	0·702
	15		0·047	1·19	0·660
	14		0·042	1·07	0·590
		20	0·039	0·991	0·548

Size S.W.G.	Birmingham Sheet Gauge. Old Style.	Birmingham Sheet Gauge. Legalized 1914.	Thickness. Inches.	Mm.	Weight in lbs. per Sq. Ft.
20			0·036	0·914	0·506
21			0·032	0·813	0·449
22		23	0·028	0·711	0·393
		24	0·025	0·635	0·351
	9		0·023	0·584	0·323
	8		0·021	0·533	0·295
	7		0·019	0·483	0·267
			0·017	0·432	0·239
	6	27	0·0160	0·406	0·225
28			0·0148	0·376	0·208
		29	0·0139	0·353	0·195
30			0·0124	0·315	0·174
	4		0·0120	0·305	0·169
		31	0·0110	0·279	0·155
33	3		0·0105	0·267	0·147
34			0·0092	0·233	0·129
		33	0·0087	0·221	0·122
	1		0·0080	0·203	0·112
36			0·0076	0·193	0·107
37			0·0068	0·172	0·0916
38			0·0060	0·152	0·0843
39			0·0052	0·132	0·0731
41			0·0044	0·112	0·0618
42			0·0040	0·102	0·0562
43			0·0036	0·0915	0·0506
44			0·0032	0·0813	0·0450
45			0·0028	0·0711	0·0394
46		44	0·0024	0·0610	0·0337
47			0·0020	0·0508	0·0281
		47	0·0017	0·0432	0·0239
		48	0·0015	0·0381	0·0211
		50	0·0012	0·0305	0·0169
49			0·0010	0·0254	0·0140
50					

NOTE.—The above weights are subject to the usual manufacturing tolerances.

	Aluminium.	Brass.	Copper.	Steel.
Specific Gravity	2·71	8·45	8·89	7·8
Ratio of Weights	1·00	3·12	3·28	2·88

(*The British Aluminium Co. Ltd.*)

WEIGHT OF ANGLE AND TEE IRON PER FOOT RUN.

Breadth (Flanges Added).	Thickness in Fractions of an Inch.													
	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	1
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
2	1·13	1·46	1·76	2·03										
2¼	1·29	1·67	2·02	2·34										
2½	1·45	1·88	2·28	2·66	3·01									
2¾	1·60	2·08	2·54	2·97	3·37									
3	1·76	2·29	2·80	3·28	3·74	4·17								
3¼	1·91	2·50	3·06	3·59	4·10	4·58								
3½	2·07	2·71	3·32	3·91	4·47	5·00	5·51							
3¾	2·23	2·92	3·58	4·22	4·83	5·42	5·98							
4	2·38	3·13	3·84	4·53	5·20	5·83	6·45	7·03						
4¼	2·54	3·33	4·10	4·84	5·56	6·25	6·91	7·55						
4½	2·70	3·54	4·36	5·16	5·92	6·67	7·38	8·07	8·74					
4¾	2·85	3·75	4·62	5·47	6·29	7·08	7·85	8·59	9·31					
5	3·01	3·96	4·88	5·78	6·65	7·50	8·32	9·11	9·88	10·63				
5¼	3·16	4·17	5·14	6·09	7·02	7·92	8·79	9·64	10·46	11·25				
5½	3·32	4·38	5·40	6·41	7·38	8·33	9·26	10·16	11·03	11·88	12·70			
5¾	3·48	4·58	5·66	6·72	7·75	8·75	9·73	10·68	11·60	12·50	13·37			
6	3·63	4·79	5·92	7·03	8·11	9·17	10·20	11·20	12·17	13·13	14·05	14·95		
6¼	3·79	5·00	6·18	7·34	8·48	9·58	10·66	11·72	12·75	13·75	14·73	15·68		
6½	3·95	5·21	6·45	7·66	8·84	10·00	11·13	12·24	13·32	14·38	15·40	16·41	17·38	
6¾	4·10	5·42	6·71	7·97	9·21	10·42	11·60	12·76	13·89	15·00	16·08	17·13	18·16	
7	4·26	5·63	6·97	8·28	9·57	10·83	12·07	13·28	14·47	15·63	16·76	17·86	18·95	20·00
7¼	4·41	5·83	7·23	8·59	9·93	11·25	12·54	13·80	15·04	16·25	17·43	18·59	19·73	20·83
7½	4·57	6·04	7·49	8·91	10·30	11·67	13·01	14·32	15·61	16·88	18·11	19·32	20·51	21·67
7¾	4·73	6·25	7·75	9·22	10·66	12·08	13·48	14·84	16 18	17·50	18·79	20·05	21·29	22·50
8	4·88	6·46	8·01	9·53	11·03	12·50	13·95	15·36	16·76	18·13	19·47	20·78	22·07	23·33
8¼	5·04	6·67	8·27	9·84	11·39	12·92	14·41	15·89	17·33	18·75	20·14	21·51	22·85	24·17
8½	5·20	6·88	8·53	10·16	11·76	13·33	14·88	16·41	17·90	19·38	20·82	22·24	23·63	25·00
8¾	5·35	7·08	8·79	10·47	12·12	13·75	15·35	16·93	18·48	20·00	21·50	22·97	24·41	25·83
9	5·51	7·29	9·05	10·78	12·49	14·17	15·82	17·45	19·05	20·63	22·17	23·70	25·20	26·67
9¼	5·66	7 50	9·31	11·09	12·85	14·58	16·29	17·97	19·62	21·25	22·85	24·43	25·98	27·50
9½	5·82	7·71	9·57	11·41	13·22	15·00	16·76	18·49	20·19	21·88	23·53	25·16	26·76	28·33
9¾	5·98	7·92	9·83	11·72	13·58	15·42	17·23	19·01	20·77	22·50	24·20	25·88	27·54	29·17

WEIGHT OF ANGLE AND TEE IRON PER FOOT RUN (*continued*).

Breadth (Flanges Added).	Thickness in Fractions of an Inch.												
	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	1
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
10	6·13	8·13	10·09	12·03	13·95	15·83	17·70	19·53	21·34	23·13	24·88	26·61	28·32	30·00
10½	6·29	8·33	10·35	12·34	14·31	16·25	18·16	20·05	21·91	23·75	25·56	27·34	29·10	30·83
10½	6·45	8·54	10·61	12·66	14·67	16·67	18·63	20·57	22·49	24·38	26·24	28·07	29·88	31·67
10¾	6·60	8·75	10·87	12·97	15·04	17·08	19·10	21·09	23·06	25·00	26·91	28·80	30·66	32·50
11	6·76	8·96	11·13	13·28	15·40	17·50	19·57	21·61	23·63	25·63	27·59	29·53	31·45	33·33
11¼	..	9·17	11·39	13·59	15·77	17·92	20·04	22·13	24·21	26·25	28·27	30·26	32·23	34·17
11½	..	9·38	11·65	13·91	16·13	18·33	20·51	22·66	24·78	26·88	28·95	30·99	33·01	35·00
11¾	..	..	11·91	14·22	16·50	18·75	20·98	23·18	25·35	27·50	29·62	31·72	33·79	35·83
12	..	..	12·17	14·53	16·86	19·17	21·45	23·70	25·92	28·13	30·30	32·45	34·57	36·67
12¼	..	..	..	14·84	17·23	19·58	21·91	24·22	26·50	28·75	30·98	33·18	35·35	37·50
12½	..	..	..	15·16	17·59	20·00	22·38	24·74	27·07	29·38	31·65	33·91	36·13	38·33
12¾	..	..	..	..	17·96	20·42	22·85	25·26	27·64	30·00	32·33	34·64	36·91	39·17
13	..	..	..	..	18·32	20·83	23·32	25·78	28·22	30·63	33·01	35·36	37·70	40·00
13½	..	..	..	..	..	21·67	24·26	26·82	29·36	31·88	34·36	36·82	39·26	41·67
14	..	..	..	..	..	22·50	25·20	27·86	30·51	33·13	35·72	38·28	40·82	43·33
14½	..	..	..	..	..	23·33	26·13	28·91	31·65	34·38	37·07	39·74	42·38	45·00
15	..	..	..	..	..	24·17	27·07	29·95	32·80	35·63	38·42	41·20	43·95	46·67

(Henderson & Glass.)

WEIGHT OF ANGLE AND TEE STEEL PER FOOT RUN.

Breadth (Flanges added.)	Thickness in Fractions of an Inch.											
	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
2	1·16	1·49	1·79	2·07								
2½	1·32	1·70	2·06	2·39								
2½	1·47	1·91	2·32	2·71	3·07							
2¾	1·63	2·13	2·59	3·03	3·44							
3	1·79	2·34	2·86	3·35	3·81	4·25						
3¼	1·95	2·55	3 12	3·67	4·18	4·68						
3½	2·11	2·76	3·39	3·98	4 56	5·10	5·62					
3¾	2·27	2·98	3·65	4·30	4·93	5·53	6·10					
4	2·43	3·19	3·92	4·62	5·30	5·95	6·57	7·17				
4½	2·59	3·40	4·18	4·94	5·67	6·38	7·05	7·70				

WEIGHT OF ANGLE AND TEE STEEL PER FOOT RUN—*continued.*

Breadth (Flanges added)	Thickness in Fractions of an Inch.											
	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
4½	2·75	3·61	4·45	5·26	6·04	6·80	7·53	8·23	8·91			
4¾	2·91	3·83	4·72	5·58	6·42	7·23	8·01	8·77	9·50			
5	3·07	4·04	4·98	5·90	6·79	7·65	8·49	9·30	10·08	10·84		
5¼	3·23	4·25	5·25	6·22	7·16	8·08	8·97	9·83	10·67	11·48		
5½	3·39	4·46	5·51	6·53	7·53	8·50	9·44	10·36	11·25	12·11	12·95	
5¾	3·55	4·68	5·78	6·85	7·90	8·93	9·92	10·89	11·83	12·75	13·64	
6	3·71	4·89	6·04	7·17	8·27	9·35	10·40	11·42	12·42	13·39	14·33	15·25
6¼	3·87	5·10	6·31	7·49	8·65	9·78	10·88	11·95	13·00	14·03	15·02	15·99
6½	4·02	5·31	6·57	7·81	9·02	10·20	11·36	12·48	13·59	14·66	15·71	16·73
6¾	4·18	5·53	6·84	8·13	9·39	10·63	11·83	13·02	14·17	15·30	16·40	17·48
7	4·34	5·74	7·11	8·45	9·76	11·05	12·31	13·55	14·76	15·94	17·09	18·22
7¼	4·50	5·95	7·37	8·77	10·13	11·48	12·79	14·08	15·34	16·58	17·78	18·97
7½	4·66	6·16	7·64	9·08	10·51	11·90	13·27	14·61	15·92	17·21	18·47	19·71
7¾	4·82	6·38	7·90	9·40	10·88	12·33	13·75	15·14	16·51	17·85	19·17	20·45
8	4·98	6·59	8·17	9·72	11·25	12·75	14·22	15·67	17·09	18·49	19·86	21·20
8¼	5·14	6·80	8·43	10·04	11·62	13·18	14·70	16·20	17·68	19·13	20·55	21·94
8½	5·30	7·01	8·70	10·36	11·99	13·60	15·18	16·73	18·26	19·76	21·24	22·68
8¾	5·46	7·23	8·97	10·68	12·37	14·03	15·66	17·27	18·85	20·40	21·93	23·43
9	5·62	7·44	9·23	11·00	12·74	14·45	16·14	17·80	19·43	21·04	22·62	24·17
9¼	5·78	7·65	9·50	11·32	13·11	14·88	16·62	18·33	20·02	21·68	23·31	24·92
9½	5·94	7·86	9·76	11·63	13·48	15·30	17·09	18·86	20·60	22·31	24·00	25·66
9¾	6·10	8·08	10·03	11·95	13·85	15·73	17·57	19·39	21·18	22·95	24·69	26·40
10	6·26	8·29	10·29	12·27	14·22	16·15	18·05	19·92	21·77	23·59	25·38	27·15
10¼	6·42	8·50	10·56	12·59	14·60	16·58	18·53	20·45	22·35	24·23	26·07	27·89
10½	6·57	8·71	10·82	12·91	14·97	17·00	19·01	20·98	22·94	24·86	26·76	28·63
10¾	6·73	8·93	11·09	13·23	15·34	17·43	19·48	21·52	23·52	25·50	27·45	29·38
11	6·89	9·14	11·36	13·55	15·71	17·85	19·96	22·05	24·11	26·14	28·14	30·12
11¼	—	9·35	11·62	13·87	16·08	18·38	20·44	22·58	24·69	26·78	28·83	30·87
11½	—	9·56	11·89	14·18	16·46	18·70	20·92	23·11	25·27	27·41	29·52	31·61
11¾	—	—	12·15	14·50	16·83	19·13	21·40	23·64	25·86	28·05	30·22	32·36
12	—	—	12·42	14·82	17·20	19·55	21·87	24·17	26·44	28·69	30·91	33·10

(*Henderson & Glass.*)

EBONITE SHEET.

Thickness.			Approx. Weight per Sheet.					
			36 × 18 ins. (914× 457 m/m.)		24 × 24 ins. (610 × 610 m/m.)		36 × 24 ins. (914 × 610 m/m.)	
Ins.	Ins.	m/m.	lbs.	kgs.	lbs.	kgs.	lbs.	kgs.
...	0·015	0·38	0·42	0·19	0·37	0·17	0·56	0·25
...	0·025	0·64	0·70	0·32	0·63	0·29	0·94	0·43
...	0·040	1·02	1·12	0·51	1·00	0·45	1·50	0·68
...	0·060	1·52	1·68	0·76	1·50	0·68	2·25	1·02
...	0·090	2·29	2·52	1·14	2·25	1·02	3·37	1·53
1/8	0·125	3·18	3·51	1·59	3·12	1·42	4·69	2·13
3/16	0·187	4·75	5·26	2·39	4·67	2·12	7·01	3·18
1/4	0·250	6·35	7·02	3·18	6·25	2·84	9·38	4·26
5/16	0·312	7·93	8·77	3·98	7·80	3·54	11·70	5·31
3/8	0·375	9·53	10·55	4·79	9·37	4·25	14·06	6·38
7/16	0·437	11·10	12·29	5·58	10·92	4·95	16·40	7·44
1/2	0·500	12·70	14·04	6·37	12·50	5·67	18·75	8·51
5/8	0·625	15·88	17·55	7·96	15·62	7·09	23·43	10·63
3/4	0·750	19·05	21·06	9·55	18·74	8·50	28·12	12·76
7/8	0·875	22·23	24·60	11·16	21·86	9·91	32·80	14·88
1	1·000	25·40	28·12	12·76	25·00	11·34	37·50	17·01

EBONITE ROD.

Diameter.			Approx. Weight per 100 metres.	Diameter.			Approx. Weight per 100 metres.
Ins.	Ins.	m/m.	Lbs.	Ins.	Ins.	m/m.	Lbs.
3/32	0·0937		1·18	5/8	0·625		52·35
	0·0984	2½	1·29		0·630	16	53·17
1/8	0·125		2·09	21/32	0·656		57·67
	0·137	3½	2·54		0·669	17	60·00
5/32	0·157	4	3·23	11/16	0·687		63·25
3/16	0·187		4·69		0·709	18	67·30
7/32	0·217	5½	6·28	23/32	0·719		69·27
1/4	0·250		8·37		0·728	18½	71·08
	0·256	6½	8·77	3/4	0·750		75·37
9/32	0·282		10·65		0·768	19½	78·98
	0·295	7½	11·68	25/32	0·781		81·72
5/16	0·312		13·05		0·787	20	83·08
	0·315	8	13·29	13/16	0·812		88·35
11/32	0·344		15·86		0·827	21	91·60
3/8	0·375	9½	18·85	27/32	0·844		95·45
13/32	0·406		22·09		0·866	22	100·5
	0·413	10½	22·90	7/8	0·875		102·6
7/16	0·437		25·60	29/32	0·905	23	110·0
	0·453	11½	27·47	15/16	0·937		117·6
15/32	0·468		29·35		0·945	24	120·0
	0·472	12	29·90	1	1·000	25·4	134·0
1/2	0·500		33·50	1 1/8	1·125	28 5	170
17/32	0·532	13½	37·85	1 1/4	1·25	31·8	210
9/16	0·562		42·33	1 3/8	1·375	35·0	253
	0·571	14½	43·66	1 1/2	1·5	38·0	302
19/32	0·594		47·28	1 3/4	1·75	44·5	410
	0·610	15½	49·90	2	2	50·8	536

(*Siemens Bros. & Co.*)

Weight of Pipes and Tubes.

D = outer diameter ; b = bore ; t = thickness ; W = weight of 1 ft. length ; k = ·02182 × weight of 1 cubic foot of metal.

$$W = t\,(D - t)\,k = t\,(b + t)\,k.$$

Weight of Wrought-Iron Pipes per Foot Run (W).

$k = 10{\cdot}55$.

Bore. (b)	Thickness of Metal in Parts of an Inch (t).							
	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
$\frac{1}{4}$	·21	·5	·9	1·3	1·9	2·5	3·1	3·9
$\frac{3}{8}$	·3	·7	1·1	1·6	2·3	2·9	3·8	4·6
$\frac{1}{2}$	·4	·83	1·4	2·0	2·7	3·5	4·3	5·3
$\frac{5}{8}$	·46	1·1	1·6	2·3	3·1	3·9	4·9	5·9
$\frac{3}{4}$	·54	1·2	1·9	2·6	3·5	4·5	5·5	6·6
$\frac{7}{8}$	·6	1·3	2·1	2·9	3·9	4·9	6·1	7·3
1	·7	1·5	2·4	3·3	4·3	5·5	6·7	7·9
$1\frac{1}{4}$	·87	1·8	2·9	3·9	5·2	6·4	7·8	9·3
$1\frac{1}{2}$	1·0	2·1	3·3	4·7	5·9	7·4	8·9	10·6
$1\frac{3}{4}$	1·2	2·5	3·8	5·3	6·8	8·4	10·1	11·9
2	1·4	2·8	4·3	5·9	7·6	9·5	11·3	13·2
$2\frac{1}{4}$	1·53	3·1	4·8	6·6	8·5	10·4	12·4	14·5
$2\frac{1}{2}$	1·7	3·5	5·3	7·3	9·3	11·4	13·6	15·9
$2\frac{3}{4}$	1·9	3·8	5·8	7·9	10·1	12·4	14·7	17·2
3	2·03	4·1	6·3	8·6	10·9	13·3	15·9	18·5

Weights of Gas and Steam Pipe.

Gas Sizes.	Gas Pipe.	Steam Pipe.	Gas Sizes	Gas Pipe.	Steam Pipe.
Ins.	Lbs. per Ft.	Lbs. per Ft.	Ins.	Lbs. per Ft.	Lbs. per Ft.
$\frac{1}{8}$	·27	·32	$1\frac{3}{4}$	3·642	4·356
$\frac{1}{4}$	·40	·486	2	4·21	5·07
$\frac{3}{8}$	·56	·68	$2\frac{1}{4}$	4·614	5·49
$\frac{1}{2}$	·84	1·01	$2\frac{1}{2}$	5·349	6·333
$\frac{3}{4}$	1·20	1·43	$2\frac{3}{4}$	5·874	6·953
1	1·67	2·01	3	6·347	7·52
$1\frac{1}{4}$	2·34	2·80	$3\frac{1}{2}$	7·213	8·551
$1\frac{1}{2}$	3·10	3·76	4	8·097	9·601

Standard Weights and Dimensions of Cast-iron Water-main Pipes.

Internal Diam. of Pipe.	Length of each Pipe.	Depth of Socket.	Thickness of Metal.	Mean Weight of each Pipe.	Internal Diam. of Pipe.	Length of each Pipe.	Depth of Socket.	Thickness of Metal.	Mean Weight of each Pipe.
Ins.	Ft. Ins.	Ins.	Ins.	cwt. qrs. lbs.	Ins.	Ft. Ins.	Ins.	Ins.	cwt. qrs. lbs.
2	6 3	3	$\frac{5}{16}$	0 2 0	18	12 4½	4½	$\frac{9}{16}$	14 2 16
3	9 4	3	$\frac{3}{8}$	1 0 14	19	12 4½	4½	—	—
4	9 4	3	$\frac{3}{8}$	1 2 0	20	12 4½	4½	$\frac{3}{4}$	20 0 0
5	9 4	3½	$\frac{7}{16}$	2 0 0	21	12 4½	4½	—	—
6	9 4	3½	$\frac{7}{16}$	2 2 0	22	12 4½	4½	$\frac{3}{4}$	22 3 14
7	9 4	3½	$\frac{1}{2}$	3 1 14	23	12 4	5	—	—
8	9 4	4	$\frac{1}{2}$	3 2 0	24	12 4	5	$\frac{7}{8}$	24 0 0
9	9 4	4	$\frac{1}{2}$	4 2 0	26	12 5	5	—	—
10	9 4	4	$\frac{9}{16}$	5 1 0	28	12 5	5	—	—
11	12 4	4½	$\frac{5}{8}$	—	30	12 5	5	1	35 3 0
12	12 4½	4½	$\frac{5}{8}$	8 2 0	32	12 5	5	—	—
13	12 4½	4½	—	—	34	12 5	5	—	—
14	12 4½	4½	$\frac{5}{8}$	10 2 0	36	12 5	5	1	41 0 0
15	12 4½	4½	$\frac{5}{8}$	11 2 0	40	12 5	5	1⅛	48 2 0
16	12 4½	4½	$\frac{5}{8}$	13 2 0	42	12 5	5	1⅛	50 2 0
17	12 4½	4½	—	—	48	12 5½	5½	1⅛	64 0 0

Weight of Ordinary Copper Pipes per Foot Run (W).

$k = 12 \cdot 105$.

Bore. (b)	Thickness of Metal in Parts of an Inch (t).						
	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
$\frac{1}{2}$	·22	·53	·94	1·43	2·01	2·68	3·44
$\frac{1}{2}$	·40	·89	1·47	2·15	2·91	3·75	4·70
$\frac{3}{4}$	·58	1·25	2·01	2·86	3·80	4·83	5·95
1	·76	1·61	2·55	3·58	4·70	5·92	7·25
$1\frac{1}{4}$	·94	1·96	3·09	4·31	5·64	6·98	8·46
$1\frac{1}{2}$	1·12	2·34	3·67	5·01	6·49	8·05	9·71
$1\frac{3}{4}$	1·33	2·66	4·14	5·70	7·36	9·11	10·94
2	1·48	3·04	4·69	6·44	8·27	10·20	12·21
$2\frac{1}{4}$	1·65	3·40	5·23	7·16	9·17	11·27	13·46
$2\frac{1}{2}$	1·83	3·75	5·77	7·87	10·06	12·35	14·72
$2\frac{3}{4}$	2·01	4·11	6·31	8·59	10·96	13·42	15·97
3	2·19	4·47	6·84	9·31	11·85	14·69	17·42

Weight of Ordinary Brass Pipes per Foot Run (W).

$k = 11 \cdot 452$.

Bore. (b)	Thickness of Metal in Parts of an Inch (t).						
	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
$\frac{1}{4}$	·23	·56	·99	1·51	2·13	2·83	3·64
$\frac{1}{2}$	·42	·94	1·56	2·27	3·07	3·97	4·96
$\frac{3}{4}$	·61	1·32	2·13	3·02	4·02	5·11	6·29
1	·80	1·70	2·69	3·78	4·96	6·24	7·61
$1\frac{1}{4}$	·99	2·08	3·26	4·54	5·91	7·38	8·94
$1\frac{1}{2}$	1·18	2·46	3·83	5·29	6·85	8·51	10·26
$1\frac{3}{4}$	1·37	2·84	4·40	6·05	7·80	9·64	11·58
2	1·56	3·22	4·96	6·81	8·75	10·78	12·91
$2\frac{1}{4}$	1·75	3·59	5·53	7·56	9·69	11·92	14·23
$2\frac{1}{2}$	1·94	3·97	6·10	8·32	10·64	13·06	15·56
$2\frac{3}{4}$	2·13	4·35	6·67	9·08	11·59	14·19	16·88
3	2·31	4·73	7·24	9·83	12·53	15·32	18·21

Weight of Lead Pipes, Sizes and Weights usually Manufactured.

Bore.	Length.	Weight and Thickness of each Length.					
		Common.		Middling.		Strong.	
		Thickness.	Weight.	Thickness.	Weight.	Thickness.	Weight.
Ins.	Ft.	In.	Lbs.	In.	Lbs.	In.	Lbs.
$\frac{1}{2}$	15	·11	16	·15	22	·17	26
$\frac{3}{4}$	15	·12	24	·13	27	·15	30
1	15	·12	30	·15	39	·16	42
$1\frac{1}{4}$	12	·14	36	·16	44	·20	53
$1\frac{1}{2}$	12	·16	48	·18	56	·21	67
2	10	·15	50	·18	60	·21	70
$2\frac{1}{2}$	10	·17	70	·21	86	·24	100

Formulæ for Weight of Lead Piping.

$15{\cdot}517\,(D-t)\,t$ lbs. per foot; $81930\,(D-t)\,t$ lbs. per mile; $36{\cdot}576\,(D-t)\,t$ tons per mile; where D = external diameter in inches, t = thickness in inches.

Weight of Cast-Iron Pipes per Foot Run.

Wt = Thickness (Bore + Thickness) 9·85.

Inner Diam. or Bore in Inches.	Thickness of Metal in Inches.												
	¼	⅜	½	⅝	¾	⅞	1	1⅛	1¼	1⅜	1½	1¾	2
	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.
1	3·07	5·07	7·38	9·99	12·9	16·2	19·7	23·5	27·7	32·1	36·9	47·4	59·1
¼	3·69	6·00	8·61	11·5	14·8	18·3	22·2	26·3	30·8	35·5	40·6	51·7	64·0
½	4·30	6·92	9·84	13·1	16·6	20·5	24·6	29·1	33·8	38·9	44·3	56·0	68·9
¾	4·92	7·84	11·1	14·6	18·5	22·6	27·1	31·8	36·9	42·3	48·0	60·3	73·8
2	5·53	8·76	12·3	16·2	20·3	24·8	29·5	34·6	40·0	45·7	51·7	64·6	78·7
¼	6·15	9·69	13·5	17·7	22·2	26·9	32·0	37·4	43·1	49·0	55·4	68·9	86·7
½	6·76	10·6	14·8	19·2	24·0	29·1	34·5	40·1	46·1	52·4	59·1	73·2	88·6
¾	7·37	11·5	16·0	20·8	25·9	31·2	36·9	42·9	49·2	55·8	62·7	77·5	93·5
3	7·98	12·5	17·2	22·3	27·7	33·4	39·4	45·7	52·3	59·2	66·4	81·8	98·4
¼	8·60	13·4	18·5	23·8	29·5	35·5	41·8	48·3	55·4	62·3	70·1	86·1	103
½	9·21	14·3	19·7	25·4	31·4	37·7	44·3	51·2	58·4	65·9	73·8	90·4	108
¾	9·83	15·2	20·9	26·9	33·2	39·8	46·8	54·0	61·5	69·3	77·5	94·7	113
4	10·3	16·1	22·2	28·5	35·1	42·0	49·2	56·7	64·6	72·7	81·2	99·0	118
¼	11·1	17·1	23·4	30·0	36·9	44·1	51·7	59·5	67·7	76·1	84·9	103	123
½	11·7	18·0	24·6	31·5	38·8	46·3	54·1	62·3	70·7	79·5	88·6	108	128
¾	12·3	18·9	25·8	33·1	40·6	48·5	56·6	65·0	73·8	83·9	92·3	112	133
5	12·9	19·8	27·1	34·6	42·5	50·6	59·1	67·8	76·9	87·2	96·0	116	138
¼	13·5	20·8	28·3	36·1	44·3	52·8	61·5	70·6	80·0	90·6	99·6	121	143
½	14·2	21·7	29·5	37·7	46·1	54·9	64·0	73·3	83·0	94·0	103	125	148
¾	14·8	22·6	30·8	39·2	48·0	57·1	66·4	76·1	86·1	97·4	107	129	153
6	15·4	23·5	32·0	40·8	49·8	59·2	68·9	78·9	89·2	99·8	111	134	158
½	16·6	25·4	34·5	43·8	53·5	63·5	73·8	84·4	95·3	107	118	142	167
7	17·8	27·2	36·9	46·9	57·2	67·8	78·7	89·4	102	113	126	151	177
½	19·1	29·1	39·4	50·0	60·9	72·1	83·7	95·5	108	120	133	159	187
8	20·3	30·9	41·8	53·1	64·6	76·4	88·6	101	114	127	140	168	197
½	21·5	32·8	44·3	56·1	68·3	80·7	93·5	107	120	134	148	177	207
9	22·8	34·6	46·8	59·2	72·0	85·1	98·4	112	126	140	155	185	217
½	24·0	36·4	49·2	62·3	75·7	89·3	103	118	132	147	163	194	226
10	25·1	38·3	51·7	65·3	79·4	93·6	108	123	138	154	170	202	235
½	26·4	40·1	54·1	68·4	83·0	97·9	113·2	129	145	161	177	211	245
11	27·6	42·0	56·6	71·5	86·7	102	118	134	151	168	185	220	255
½	28·8	43·8	59·1	74·6	90·4	107	123	140	157	174	192	228	265
12	30·0	45·7	61·5	77·7	94·1	111	128	145	163	181	199	237	275
13	32·5	49·4	66·4	83·8	102	120	138	156	175	195	214	254	294
14	35·0	53·1	71·4	89·4	109	128	148	168	188	208	229	271	314
15	37·4	56·7	76·3	96·1	116	137	158	179	200	222	244	289	334
16	39·1	60·4	81·2	102	124	145	167	190	212	235	258	306	353
17	42·3	64·1	86·1	108	131	154	177	201	225	249	273	323	373
18	44·8	67·8	91·0	115	139	163	187	212	237	262	288	340	393
19	47·3	71·5	96·0	121	146	171	197	223	249	276	303	357	412
20	49·7	75·2	101	127	153	180	207	234	261	289	317	375	432
21	52·2	78·9	106	133	161	188	217	245	274	303	332	392	452
22	54·6	82·6	111	139	168	196	227	256	286	316	347	409	471
23	57·1	86·3	116	145	175	206	236	267	298	330	362	426	491
24	59·6	89·9	121	152	183	214	246	278	311	343	375	444	511
25	62·0	93·6	126	158	190	223	256	289	323	357	391	461	531
26	64·5	97·3	131	164	198	231	266	300	335	370	406	478	550
27	66·9	101	135	170	205	240	276	311	348	384	421	495	570
28	69·4	105	140	176	212	249	286	323	360	397	436	512	590
29	71·8	109	145	182	220	257	295	334	372	411	450	530	609
30	74·2	112	150	188	227	266	305	345	384	424	465	547	629
31	76·7	116	155	195	234	275	315	356	397	438	480	564	649
32	79·1	120	160	201	242	283	325	367	409	451	495	581	668
33	81·6	123	165	207	249	292	335	378	421	465	509	598	688
34	84·1	127	170	213	257	300	345	389	434	479	524	616	709
35	86·5	131	175	219	264	309	354	400	446	492	539	633	726
36	89·0	134	180	225	271	318	364	411	458	506	554	650	746
42	104	156	210	262	315	370	423	478	532	588	644	753	864
48	119	178	239	298	359	422	482	544	605	669	733	856	982

APPROXIMATE WEIGHTS
(Based on the Imperial

Thickness, I.W.G. Ins. (*t*) m/m.	15 ·072 1·829	14 ·080 2·032	13 ·092 2·337	12 ·104 2·642	11 ·116 2·946	10 ·128 3·251	9 ·144 3·658	8 ·160 4·064	7 ·176 4·470	6 ·192 4·877	5 ·212 5·385
External Diam. Ins. (*D*).											WEIGHT PER
1	0·700	0·771	0·875	0·976	1·074	1·169	1·291	1·407	1·519	1·624	1·749
1⅛	0·794	0·875	0·995	1·112	1·226	1·336	1·479	1·617	1·749	1·876	2·027
1¼	0·888	0·980	1·116	1·248	1·377	1·504	1·668	1·826	1·979	2·127	2·304
1⅜	0·982	1·085	1·236	1·384	1·529	1·671	1·856	2·036	2·210	2·378	2·582
1½	1·077	1·190	1·356	1·520	1·681	1·839	2·045	2·245	2·440	2·630	2·859
1⅝	1·171	1·294	1·477	1·656	1·833	2·007	2·233	2·455	2·671	2·881	3·137
1¾	1·265	1·399	1·597	1·793	1·985	2·174	2·422	2·664	2·901	3·132	3·414
1⅞	1·359	1·504	1·718	1·929	2·137	2·342	2·610	2·873	3·131	3·384	3·692
2	—	1·608	1·838	2·065	2·288	2·509	2·799	3·083	3·362	3·635	3·969
2⅛	—	1·713	1·959	2·201	2·440	2·677	2·987	3·292	3·592	3·886	4·247
2¼	—	1·818	2·079	2·337	2·592	2·844	3·176	3·502	3·822	4·138	4·524
2⅜	—	—	2·199	2·473	2·744	3·012	3·364	3·711	4·053	4·389	4·802
2½	—	—	2·320	2·609	2·896	3·179	3·553	3·921	4·283	4·640	5·079
2⅝	—	—	2·440	2·745	3·048	3·347	3·741	4·130	4·514	4·892	5·357
2¾	—	—	2·561	2·882	3·200	3·514	3·930	4·339	4·744	5·143	5·634
2⅞	—	—	2·681	3·018	3·351	3·682	4·118	4·549	4·974	5·394	5·912
3	—	—	2·802	3·154	3·503	3·850	4·307	4·758	5·205	5·646	6·189
3⅛	—	—	—	3·290	3·655	4·017	4·495	4·968	5·435	5·897	6·467
3¼	—	—	—	3·426	3·807	4·185	4·684	5·177	5·665	6·148	6·744
3⅜	—	—	—	3·562	3·959	4·352	4·872	5·387	5·896	6·400	7·022
3½	—	—	—	3·698	4·111	4·520	5·061	5·596	6·126	6·651	7·299
3⅝	—	—	—	3·835	4·262	4·687	5·249	5·806	6·357	6·902	7·577
3¾	—	—	—	3·971	4·414	4·855	5·438	6·015	6·587	7·154	7·854
3⅞	—	—	—	4·107	4·566	5·022	5·626	6·224	6·817	7·405	8·132
4	—	—	—	4·247	4·718	5·190	5·815	6·434	7·048	7·656	8·410
4⅛	—	—	—	4·506	5·022	5·525	6·192	6·853	7·509	8·159	8·965
4¼	—	—	—	4·776	5·325	5·860	6·569	7·272	7·969	8·662	9·520
4⅜	—	—	—	5·056	5·631	6·195	6·946	7·690	8·430	9·164	10·075
5	—	—	—	5·336	5·937	6·530	7·323	8·109	8·891	9·667	10·630
5⅛	—	—	—	5·600	6·243	6·866	7·700	8·528	9·352	10·170	11·185
5¼	—	—	—	5·876	6·549	7·201	8·077	8·947	9·812	10·672	11·740
5⅜	—	—	—	6·152	6·857	7·536	8·454	9·366	10·273	11·175	12·295
6	—	—	—	6·432	7·164	7·871	8·831	9·785	10·734	11·678	12·850
6½	—	—	—	6·715	7·471	8·213	9·208	10·204	11·195	12·180	13·405
6½	—	—	—	7·000	7·778	8·555	9·585	10·623	11·655	12·683	13·960
6¾	—	—	—	7·273	8·087	8·897	9·962	11·042	12·116	13·186	14·515
7	—	—	—	7·556	8·393	9·239	10·339	11·460	12·577	13·688	15·070
7¼	—	—	—	7·834	8·700	9·581	10·716	11·880	13·038	14·191	15·625
7½	—	—	—	8·115	9·007	9·923	11·083	12·300	13·499	14·694	16·180
7¾	—	—	—	8·383	9·314	10·265	11·460	12·720	13·959	15·196	16·735
8	—	—	—	8·664	9·621	10·607	11·847	13·135	14·420	15·699	17·290
8½	—	—	—	8·943	9·928	10·949	12·224	13·554	14·880	16·204	17·845
8½	—	—	—	9·227	10·235	11·291	12·601	13·973	15·340	16·710	18·400
8¾	—	—	—	9·516	10·542	11·633	12·978	14·400	15·800	17·216	18·955
9	—	—	—	9·807	10·850	11·985	13·355	14·818	16·260	17·722	19·510
9¼	—	—	—	10·073	11·158	12·327	13·732	15·230	16·720	18·230	20·065
9½	—	—	—	10·342	11·466	12·669	14·109	16·656	17·180	18·738	20·620
9¾	—	—	—	10·604	11·774	13·011	14·486	16·070	17·640	19·230	21·175
10	—	—	—	10·878	12·082	13·353	14·863	16·507	18·100	19·735	21·730
10¼	—	—	—	11·156	12·390	13·695	15·240	16·922	18·563	20·242	22·285
10½	—	—	—	11·428	12·698	14·042	15·617	17·344	19·024	20·753	22·840
10¾	—	—	—	11·687	13·006	14·384	15·994	17·768	19·482	21·255	23·395
11	—	—	—	11·952	13·314	14·726	16·371	18·185	19·940	21·756	23·950
11½	—	—	—	12·516	13·930	15·410	17·125	19·016	20·860	22·758	25·053
12	—	—	—	13·070	14·550	16·100	17·878	19·855	21·780	23·760	26·200

OF WROUGHT-IRON TUBES.

Standard Wire Gauge.) $W = t(D - t)\,10{\cdot}472.$

4 ·232 5·893	3 ·252 6·401	2 ·276 7·010	1 ·300 7·620	⅛ ·125 3·175	3/16 ·187 4·762	¼ ·250 6·350	5/16 ·313 7·937	⅜ ·375 9·525	7/16 ·437 11·112	½ ·500 12·700	Thickness, I.W.G. Ins. (*t*) m/m.
FOOT IN POUNDS (W).											External Dia. Ins. (*D*).
1·866	1·974	2·092	2·199	1·145	1·595	1·963	2·250	2 454	2·577	2·618	1
2·169	2·304	2·454	2·592	1·309	1·841	2·291	2·659	2·945	3·150	3·272	1⅛
2·473	2·634	2·815	2·984	1·473	2·086	2·618	3·068	3·436	3·723	3·927	1¼
2·777	2·963	3·176	3·377	1·636	2·332	2·945	3·477	3·927	4·295	4·581	1⅜
3·081	3·293	3·538	3·770	1·800	2·577	3·272	3·886	4·418	4·868	5·236	1½
3·384	3·623	3·899	4·163	1·963	2·822	3·600	4·295	4·909	5·440	5·890	1⅝
3·688	3·953	4·260	4·555	2·127	3·068	3·927	4·704	5·400	6·013	6·545	1¾
3·992	4·283	4·621	4·948	2·291	3·313	4·254	5·113	5·890	6·586	7·199	1⅞
4·295	4·613	4·983	5·341	2·454	3·559	4·581	5·522	6·381	7·159	7·854	2
4·599	4·943	5·344	5·733	2·618	3·804	4·909	5·931	6·872	7·731	8·508	2⅛
4·903	5·273	5·705	6·126	2·782	4·050	5·236	6·340	7·363	8·304	9·163	2¼
5·206	5·602	6·067	6·519	2·945	4·295	5·563	6·749	7·854	8·877	9·817	2⅜
5·510	5·932	6·428	6·911	3·109	4·541	5·890	7·159	8·345	9·449	10·472	2½
5·814	6·262	6·789	7·304	3·272	4·786	6·218	7·568	8·836	10·022	11·126	2⅝
6·117	6·592	7·150	7·697	3·436	5·031	6·545	7·977	9·327	10·595	11·781	2¾
6·421	6·922	7·512	8·090	3·600	5·277	6·872	8·386	9·818	11·167	12·435	2⅞
6·725	7·252	7·873	8·482	3·763	5·522	7·200	8·795	10·308	11·740	13·090	3
7·028	7·582	8 234	8·875	3 927	5·768	7·527	9·204	10·799	12·313	13·744	3⅛
7·332	7·911	8·596	9·268	4·091	6·013	7·854	9·613	11·290	12·885	14·399	3¼
7·636	8·241	8·957	9·660	4·254	6·259	8·181	10·022	11·781	13·458	15·053	3⅜
7·940	8 571	9·318	10·053	4·418	6·504	8·508	10·431	12·272	14·031	15·708	3½
8·243	8·901	9·679	10·446	4·581	6·750	8·836	10·840	12·763	14·604	16·362	3⅝
8·547	9·231	10·041	10·838	4·745	6·995	9·163	11·249	13·254	15·176	17·017	3¾
8·851	9 561	10·402	11·231	4·908	7·240	9·490	11·658	13·745	15·749	17·671	3⅞
9·154	9·891	10·763	11·624	5·072	7·486	9·817	12·067	14·235	16·322	18·326	4
9·762	10·550	11·486	12·409	5·400	7·977	10·472	12·885	15·217	17·467	19·635	4¼
10·369	11·210	12·208	13·195	5·727	8·468	11·126	13·704	16·199	18·612	20·944	4½
10·976	11·870	12·931	13·980	6·054	8·959	11·781	14·522	17·181	19·758	22·253	4¾
11·584	12·530	13·654	14·765	6·381	9·450	12·435	15·340	18·162	20·903	23·562	5
12·191	13·189	14·376	15·551	6·709	9·940	13·090	16·158	19·144	22·048	24·871	5¼
12·798	13·849	15·099	16·336	7·036	10·431	13·744	16·976	20·126	23·194	26·180	5½
13·406	14·509	15·821	17·122	7·363	10·922	14·399	17·794	21·108	24·339	27·489	5¾
14·013	15·169	16·544	17·907	7·690	11·413	15·053	18·612	22·090	25·485	28·798	6
14·621	15·828	17·266	18·692	8·017	11·904	15·708	19·430	23·071	26·630	30·107	6¼
15·228	16·488	17·989	19·478	8·345	12·395	16·362	20·249	24·053	27·775	31·416	6½
15·835	17·184	18·712	20·263	8·672	12·886	17·017	21·067	25·035	28·921	32·725	6¾
16·443	17·807	19·434	21·048	8·999	13·376	17·671	21·885	26·017	30·066	34·034	7
17·050	18·467	20·157	21·834	9·327	13·867	18·326	22·703	26·998	31·212	35·343	7¼
17·658	19 127	20·879	22·619	9·654	14·358	18·980	23·521	27·980	32·357	36·652	7½
18·265	19·787	21·602	23·405	9·981	14·849	19·635	24 339	28·962	33·502	37·961	7¾
18·872	20·446	22·324	24·190	10·308	15·340	20·289	25·157	29·944	34·648	39·270	8
19·480	21·106	23·047	24·976	10·636	15·831	20·944	25·975	30·925	35·793	40·579	8¼
20·087	21·766	23·769	25·761	10·963	16·322	21·598	26·794	31·907	36·938	41·888	8½
20·694	22·426	24·492	26·546	11·290	16·813	22·253	27·612	32·889	38·084	43·197	8¾
21·302	23·085	25·215	27·332	11·617	17·303	22·907	28·430	33·871	39 229	44·506	9
21·909	23·745	25·937	28·117	11·945	17·794	23·562	29·248	34·852	40·375	45·815	9¼
22·517	24·405	26·660	28·903	12·272	18·285	24·216	30·066	35·834	41·520	47·124	9½
23·124	25·065	27·382	29·688	12·599	18·776	24·871	30·884	36·816	42·665	48·433	9¾
23·731	25·724	28·105	30·473	12·926	19·267	25·525	31·702	37·708	43·811	49·742	10
24·338	26·383	28·828	31·258	13·253	19·758	26·179	32·520	38·780	45·957	51·052	10¼
24·945	27 042	29·551	32·043	13·580	20·249	26·833	33·338	39·762	46·103	52·361	10½
25·535	27·702	30·274	32·828	13·910	20·740	27·487	34·156	40·744	47·250	53·670	10¾
26·162	28·362	31·002	33·613	14·237	21·231	28·141	34·974	41·726	48·396	54·980	11
27·376	29·682	32·445	35·183	14·897	22 213	29·450	36·610	43·690	50·690	57·600	11½
28·590	31·002	33·891	36·753	15·550	23·195	30·758	38·246	45·654	52·984	60·215	12

(*John Spencer, Ltd.*)

WEIGHTS AND DIMENSIONS OF CAST-IRON PIPES, FOR GAS, WATER, AND STEAM.

Dimensions.			Spigot and Faucet Joints.			Turned and Bored Joints.			Flanged Joints.					
Inside Diameter.	(A) Thickness of Metal.	Length exclusive of Faucet.	Average Weight per Pipe.			Average Weight per Pipe.			Inside Diameter.	(B) Thickness of Metal.	Length over Flanges.	Average Weight per Pipe.		
Ins.	Ins.	Ft.	Cwts.	Qrs.	Lbs.	Cwts.	Qrs.	Lbs.	Ins.	Ins.	Ft.	Cwts.	Qrs.	Lbs.
$\frac{3}{4}$	$\frac{7}{16}$	$4\frac{1}{2}$			$14\frac{1}{2}$				2	$\frac{5}{16}$	6		1	21
1	$\frac{1}{4}$	6			22		1	0	$2\frac{1}{2}$	$\frac{5}{16}$	9		3	2
$1\frac{1}{4}$	$\frac{1}{4}$	6			26		1	5	$2\frac{1}{2}$	$\frac{1}{2}$	9	1	1	7
$1\frac{1}{2}$	$\frac{1}{4}$	6		1	2		1	7	3	$\frac{5}{16}$	9	1	0	0
$1\frac{1}{2}$	$\frac{9}{32}$	6		1	7		1	13	3	$\frac{3}{8}$	9	1	0	13
2	$\frac{1}{4}$	6		1	14		1	20	3	$\frac{1}{2}$	9	1	2	0
2	$\frac{5}{16}$	6		1	22		2	0	3	$\frac{5}{8}$	9	2	0	0
2	$\frac{11}{32}$	6		2	0		2	8	$3\frac{1}{2}$	$\frac{3}{8}$	9	1	1	3
$2\frac{1}{2}$	$\frac{5}{16}$	6		2	0		2	7	$3\frac{1}{2}$	$\frac{1}{2}$	9	1	3	0
$2\frac{1}{2}$	$\frac{5}{16}$	9		3	0		3	10	4	$\frac{3}{8}$	9	1	1	21
$2\frac{1}{2}$	$\frac{11}{32}$	9		3	14		3	21	4	$\frac{7}{16}$	9	1	2	24
3	$\frac{5}{16}$	9		3	14		3	21	4	$\frac{1}{2}$	9	2	0	0
3	$\frac{11}{32}$	9	1	0	0	1	0	7	4	$\frac{5}{8}$	9	2	2	4
3	$\frac{3}{8}$	9	1	0	10	1	0	17	$4\frac{1}{2}$	$\frac{1}{2}$	9	2	0	21
$3\frac{1}{2}$	$\frac{5}{16}$	9	1	0	14	1	0	21	$4\frac{1}{2}$	$\frac{9}{16}$	9	2	2	0
4	$\frac{11}{32}$	9	1	1	14	1	1	21	5	$\frac{1}{2}$	9	2	1	21
4	$\frac{3}{8}$	9	1	2	0	1	2	7	5	$\frac{5}{8}$	9	3	0	14
$4\frac{1}{2}$	$\frac{11}{32}$	9	1	2	14	1	2	21	$5\frac{1}{2}$	$\frac{1}{2}$	9	2	2	14
5	$\frac{3}{8}$	9	1	3	14	1	3	21	6	$\frac{7}{16}$	9	2	2	0
5	$\frac{13}{64}$	9	2	0	0	2	0	7	6	$\frac{1}{2}$	9	2	3	14
$5\frac{1}{2}$	$\frac{3}{8}$	9	2	0	11	2	0	18	6	$\frac{5}{8}$	9	3	2	14
6	$\frac{13}{32}$	9	2	1	14	2	1	14	7	$\frac{1}{2}$	9	3	1	7
6	$\frac{7}{16}$	9	2	2	0	2	2	9						
7	$\frac{7}{16}$	9	2	3	14	2	3	24						
7	$\frac{1}{2}$	9	3	1	4	3	1	14						

(Macfarlane, Strang & Co., Ltd.)

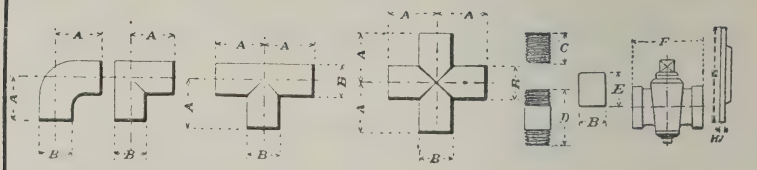
THICKNESSES AND WEIGHTS OF SCREWED AND SOCKETED TUBES.

Nominal Bore.	External Diameter.	Length of Screw on Tubes.	Pitch of Screw, No. of Threads, per in.	Thickness of Tubes Wire Gauge No.			Approximate Weight in lbs. per Foot.		
Ins.	Ins.	Ins.		Gas.	Water.	Steam.	Gas.	Water.	Steam.
$\frac{1}{8}$	$\frac{13}{32}$	$\frac{3}{8}$	28	14	13	12	0·278	0·306	0·337
$\frac{1}{4}$	$\frac{17}{32}$	$\frac{7}{16}$	19	14	13	12	0·385	0·428	0·472
$\frac{3}{8}$	$\frac{11}{16}$	$\frac{1}{2}$	19	13	12	11	0·582	0·644	0·703
$\frac{1}{2}$	$\frac{27}{32}$	$\frac{5}{8}$	14	12	11	10	0·818	0·896	0·973
$\frac{3}{4}$	$1\frac{3}{16}$	$\frac{3}{4}$	14	11	10	9	1·165	1·268	1·403
1	$1\frac{11}{32}$	$\frac{7}{8}$	11	10	9	8	1·653	1·833	2·008
$1\frac{1}{4}$	$1\frac{11}{16}$	1	11	9	8	7	2·367	2·598	2·827
$1\frac{1}{2}$	$1\frac{29}{32}$	1	11	8	7	6	2·973	3·237	3·500
$1\frac{3}{4}$	$2\frac{5}{32}$	$1\frac{1}{8}$	11	8	7	6	3·406	3·710	4·019
2	$2\frac{3}{8}$	$1\frac{1}{8}$	11	8	7	6	3·786	4·128	4·473
$2\frac{1}{4}$	$2\frac{3}{8}$	$1\frac{1}{4}$	11	7	6	5	4·602	4·980	5·447
$2\frac{1}{2}$	3	$1\frac{1}{4}$	11	7	6	5	5·338	5·779	6·323
3	$3\frac{1}{2}$	$1\frac{3}{8}$	11	7	6	5	6·309	6·834	7·483
$3\frac{1}{2}$	4	$1\frac{1}{2}$	11	7	6	5	7·265	7·873	8·627
4	$4\frac{1}{2}$	$1\frac{5}{8}$	11	7	6	5	8·253	8·945	9·803
$4\frac{1}{2}$	5	$1\frac{5}{8}$	11	7	6	5	9·230	10·006	10·969
5	$5\frac{1}{2}$	$1\frac{3}{4}$	11	7	6	5	10·232	11·091	12·159
6	$6\frac{1}{2}$	2	11	7	6	5	12·305	13·332	14·609

The above dimensions are subject to the usual working allowances.

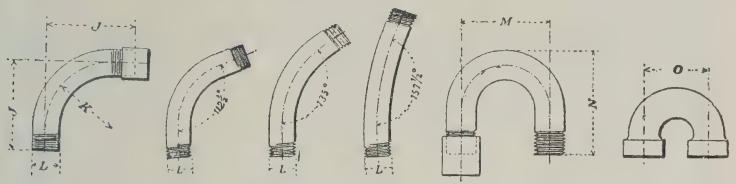
(*Stewarts & Lloyds, Ltd.*)

APPROXIMATE DIMENSIONS OF



			Size, ins.	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{3}{4}$
A	Centre to Face of Elbows, Tees, and Crosses .	GAS . .	ins.	$\frac{3}{4}$	$\frac{7}{8}$	$1\frac{5}{16}$	$1\frac{1}{8}$	$1\frac{3}{8}$
		STEAM . .	,,	$\frac{7}{8}$	$1\frac{5}{16}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$
B	Outside Diam. of Elbows, Tees, Crosses, and Sockets . .	GAS . .	,,	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	$1\frac{1}{16}$	$1\frac{5}{16}$
		STEAM . .	,,	$1\frac{1}{16}$	$1\frac{3}{16}$	$1\frac{5}{16}$	$1\frac{1}{8}$	$1\frac{3}{8}$
C	Length of Nipples . .	GAS & STEAM .	,,	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1
D	Length of Barrel Nipples .	GAS & STEAM .	,,	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2
E	Length of Sockets . .	GAS & STEAM .	,,	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$
F	Length of Main Cocks .	GAS . .	,,	—	$1\frac{5}{16}$	$2\frac{5}{8}$	3	$3\frac{1}{2}$
G	Diameter of Flanges .	GAS & STEAM .	,,	$2\frac{1}{4}$	$2\frac{1}{2}$	3	3	4
H	Thickness of Flanges	GAS . .	,,	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{3}{16}$
		STEAM . .	,,	—	$\frac{7}{32}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$
Drilling	Pitch Circle . . .	GAS & STEAM .	,,	$1\frac{5}{8}$	$1\frac{3}{4}$	$2\frac{1}{8}$	$2\frac{3}{8}$	$2\frac{7}{8}$
	Number and Size of Bolts.	GAS .	,,	$4 \cdot \frac{5}{16}$	$4 \cdot \frac{5}{16}$	$4 \cdot \frac{3}{8}$	$4 \cdot \frac{3}{8}$	$4 \cdot \frac{1}{2}$
	Number and Size of Bolts.	STEAM . .	,,	—	$4 \cdot \frac{5}{16}$	$4 \cdot \frac{3}{8}$	$4 \cdot \frac{1}{2}$	$4 \cdot \frac{1}{2}$
J	Centre to Face of Bends .	GAS & STEAM .	,,	$2\frac{1}{8}$	$2\frac{1}{8}$	$2\frac{7}{8}$	$3\frac{1}{4}$	4
K	Radii of Bends . .	GAS & STEAM .	,,	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{7}{8}$	$2\frac{1}{4}$	$2\frac{7}{8}$
L	Outside Diam. of Bends and Springs . .	GAS & STEAM .	,,	$1\frac{13}{32}$	$1\frac{17}{32}$	$1\frac{11}{16}$	$2\frac{27}{32}$	$1\frac{1}{16}$
Nos. 6, 7, 8, 9	Length of tube in Bends and Springs . . .	GAS & STEAM .	,,	$3\frac{3}{4}$	$4\frac{1}{4}$	$4\frac{7}{8}$	$5\frac{3}{8}$	$6\frac{5}{8}$
M	Centre to Centre of W.I. Double Bends . .	STEAM .	,,	$1\frac{3}{4}$	2	$2\frac{1}{2}$	$3\frac{1}{2}$	$5\frac{1}{2}$
N	Back to Face of W.I. Double Bends . .	STEAM . .	,,	2	$2\frac{1}{2}$	$2\frac{3}{4}$	$3\frac{1}{2}$	4
O	Centre to Centre of Malleable Double Bends .	STEAM . .	,,	—	—	—	$1\frac{1}{2}$	$1\frac{3}{4}$

STEAM AND GAS FITTINGS.



1	1¼	1½	1¾	2	2¼	2½	2¾	3	3½	4	4½	5	5½	6	
1⅝ 1¾	1 13/16 1 15/16	2 1/16 2 3/16	2 3/16 2 5/16	2½ 2⅝	2⅞ 3	3⅛ 3¼	3⅜ 3½	3⅝ 3⅝	4 4⅛	4⅛ 4⅜	5¼ 5¼	5¾ 5¾	6 6	6½ 6½	A
1 9/16 1⅝	1 13/16 2 1/16	2¼ 2 5/16	2⅜ 2 7/16	2⅝ 2 13/16	3 1/16 3⅛	3⅜ 3 7/16	3 11/16 3¾	3 13/16 4	4 7/16 4½	5 5 1/16	5⅝ 5⅝	6⅛ 6⅛	6⅝ 6⅝	7⅛ 7⅛	B
1½	1½	1⅝	1⅝	1⅞	2⅛	2⅜	2⅝	2⅝	2⅞	2⅞	2⅞	3	3⅛	3¼	C
2¼	2⅜	2½	2⅝	2⅝	3	3¼	3½	3½	3¾	4¼	4¼	4½	4½	5½	D
2	2⅛	2⅜	2¼	2⅜	2⅝	2⅞	2⅞	3¼	3¼	3½	3½	4	4	4½	E
4⅛	4⅞	5½	6¼	6½	7	7⅝	8½	8⅝	10⅜	11½	—	—	—	—	F
4½	4⅝	5¼	5½	6	—	6½	—	7⅛	8	8½	9	10	—	11	G
3/16 9/32	1/4 5/16	1/4 11/32	1/4 11/32	5/16 3/8	— —	5/16 13/32	— —	3/8 7/16	3/8 15/32	3/8 1/2	7/16 17/32	1/2 9/16	— —	1/2 11/16	H
3½ 4·1½ 4·1½	3 7/16 4·1⅝ 4·1⅝	3⅞ 4·1⅞ 4·1⅞	4⅛ 4·1⅞ 4·1⅞	4½ 4·2⅛ 4·2⅛	— — —	5 4·2⅜ 4·2⅜	— — —	5⅝ 4·2⅝ 8·2⅝	6½ 4·2⅝ 8·2⅝	7 4·2⅝ 8·2⅝	7½ 8·2⅝ 8·2⅝	8¼ 8·2⅝ 8·2⅝	— — —	9½ 8·2⅝ 8·2¾	Drilling
4⅝	6	6⅝	7⅛	8	8⅞	9⅝	10⅜	11½	13½	15	16½	18	20	22	J
3⅜	4½	5	5½	6½	7	7⅝	8½	9⅛	10⅝	12⅛	14	15	16½	18	K
1 11/32	1 1/16	1 29/32	2 5/32	2⅜	2⅝	3	3¼	3½	4	4½	5	5½	6	6½	L
7⅞	10⅛	11⅛	11¾	13	14½	15⅞	17¼	18½	22	24¾	28	30	33	36	Nos. 6, 7, 8, 9
4⅛	4½	5½	5¾	6	8½	9½	10	12	13	15	—	—	—	—	M
4¾	5⅛	6	6¼	7	8½	9	9½	11	11	12¼	—	—	—	—	N
—⅜	3¾	4⅛	4½	4½	4⅜	4⅜	5	5⅜	7	9½	—	—	—	—	O

(Stewarts & Lloyds, Ltd.)

Standard Weights of Cast-Iron Water-Pipe Fittings.

Internal Diameter of Pipes and Fittings in Inches.

Description.	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.
Pipes	0 2 0	1 0 14	1 2 0	2 0 0	2 2 0	3 1 14	3 2 0	4 1 0	6 3 0		8 2 0		10 2 0	11 2 0	13 2 0	
Bends $\frac{1}{16}$-in.	0 1 6	0 2 0	0 3 4	1 0 0	1 1 7	1 3 7	2 2 0	3 0 4	3 2 0		4 1 9		5 3 0	6 0 0	7 0 0	
Bends $1\frac{1}{8}$ in.	0 1 7	0 2 2	0 3 0	0 3 27	1 1 7	1 3 25	2 2 7	3 0 4	3 1 14		4 1 7		5 2 0	6 1 0	6 3 0	
Bends $\frac{1}{4}$ in.	0 1 7	0 2 7	0 3 7	1 0 16	1 2 0	1 3 14	2 2 7	3 0 21	3 1 21		4 1 0		5 3 0	6 2 0	7 1 0	
Cross-pipe	0 2 21	0 3 18	1 0 14	1 1 7	1 3 21	2 1 21	2 3 0	4 0 0	4 3 0		6 2 6		6 2 0	10 0 0	10 2 0	
S bend	0 1 21	0 2 0	0 3 21	1 0 4	1 3 0	2 0 0	2 3 0	3 1 0	3 3 0		5 2 0					
Bevel arm-piece or Y-piece spigots	0 2 0	0 3 0	1 0 7	1 1 14	1 2 7	2 0 9	2 3 0	3 1 0	4 1 0		6 2 0	Made to Drawings.				
Bevel arm-piece or Y-piece sockets	0 2 7	0 3 14	1 0 21	1 2 0	1 3 0	2 1 7	2 3 0	3 1 0	4 3 0		6 2 0					
Arm-pipe or branch	0 3 0	0 3 21	1 1 0	1 1 21	1 3 7	2 1 14	3 0 0	3 2 0	4 3 14		5 2 0		8 0 0	9 1 0	9 2 14	
Tee-pipe	0 1 7	0 3 0	1 0 0	1 1 13	1 2 4	2 1 0	3 1 0	3 2 14	4 0 0		5 1 7		7 2 14	9 0 0	9 1 0	
Wash-out pipe	0 2 14	0 2 21	0 3 21	1 0 16	1 1 14	1 2 14	1 3 20	2 2 0	3 0 0		1 2 0		1 3 0	2 0 0	2 3 0	
Collar or thimble	0 0 20	0 0 24	0 1 4	0 1 14	0 2 0	0 2 21	0 3 14	1 0 0	1 1 10		1 2 2		1 3 14	1 3 25	2 0 18	
Cap	0 0 12	0 0 18	0 0 24	0 1 2	0 1 12	0 2 0	0 2 14	0 3 0	0 3 9		1 0 4		1 1 21	1 2 7	1 3 0	
Reducers	3″–2″	4″–3″	5″–4″	6″–5″	7″–6″	8″–7″	9″–8″	10″–9″	12″–10″		14″–12″	Made to Drawings.				
	0 1 0	0 2 0	0 2 7	0 3 14	1 0 0	1 1 10	1 3 21	2 0 0	2 3 21		3 3 18					

Description.	18	19	20	21	22	23	24	26	28	30	32	34	36	40	42	48
	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.
Pipes	14 2 16		20 0 0		22 3 14		24 0 0			35 3 0			41 0 0	48 2 0	50 2 0	64 0 0
Bends $\frac{1}{16}$ in.	7 3 7		9 1 21													
Bends $\frac{1}{8}$ in.	7 2 7		11 1 14				13 0 7			18 3 0						
Bends $\frac{1}{4}$ in.	7 3 16		10 3 21				13 1 6									
Cross-pipe	11 2 0		14 0 0				17 0 0									
S bend																
Bevel arm-piece or Y-piece spigots								Made Specially to Drawings.								
Bevel arm-piece or Y-piece sockets																
Arm-pipe or branch	10 2 0		12 0 0				15 0 0									
Tee-pipe	10 1 0		11 2 0				14 1 10									
Wash-out pipe	3 1 0		4 0 0													
Collar or thimble	3 0 25		3 2 10				5 2 10									
Cap	2 2 0		3 0 0				4 2 10									
Reducers																

Pipes 2-in. bore in 6 ft. 3 ins. lengths; 3-in. bore to 9 ins., 9 ft. 4 ins. lengths; 9-in. bore to 48 ins., 12 ft. 4 ins. lengths.

The pipes are vertically cast in dry sand, socket down, and are tested to a pressure of 500 feet head (216 lbs. per sq. in.) of water; the fittings are tested to the same pressure.

(*The Staveley Coal and Iron Co.*)

APPROXIMATE WEIGHT OF SOCKET PIPES AND FITTINGS.

Socket Pipes.

Inside Diam.	Length ex. Socket.	Depth of Socket.	Diam. of Socket.	Weight per Length.	Inside Diam.	Length ex. Socket.	Depth of Socket.	Diam. of Socket.	Weight per Length.
ins.	ft. ins.	ins.	ins.	c. q. lbs.	ins.	ft. ins.	ins.	ins.	c. q. lbs.
$1\frac{1}{2}$	6 0	$2\frac{3}{4}$	$2\frac{3}{4}$	1 0 to 1 10	6	9 0	$4\frac{1}{2}$	$7\frac{1}{2}$	2 1 14 to 2 1 21
2	6 0	3	$3\frac{1}{4}$	1 14 to 1 21	7	9 0	5	$8\frac{3}{4}$	2 2 0 to 2 3 14
$2\frac{1}{2}$	6 0	$3\frac{1}{2}$	$3\frac{3}{4}$	2 0 to 2 14	8	9 0	5	$9\frac{3}{4}$	3 0 0 to 3 1 14
3	9 0	$3\frac{1}{2}$	$4\frac{1}{4}$	3 14 to 3 21					3 3 0 to 3 3 14
$3\frac{1}{2}$	9 0	4	$4\frac{3}{4}$	1 0 0 to 1 1 0	9	9 0	5	$10\frac{3}{4}$	4 2 0
4	9 0	4	$5\frac{1}{4}$	1 1 14 to 1 2 0	10	9 0	5	$11\frac{3}{4}$	4 2 0 to 5 0 0
$4\frac{1}{2}$	9 0	4	6	1 3 2 to 1 3 14	12	9 0	5	$13\frac{3}{4}$	5 3 0 to 6 2 0
5	9 0	$4\frac{1}{2}$	$6\frac{1}{2}$	1 3 21 to 2 0 0	12	12 0	5	$13\frac{3}{4}$	8 3 0

Connections.

Description.	$1\frac{1}{2}$ in.	2 in.	$2\frac{1}{2}$ in.	3 in.	$3\frac{1}{2}$ in.	4 in.	5 in.	6 in.	7 in.	8 in.	9 in.	10 in.	12 in.
	q. lbs.	q. lbs.	q. lbs.	q. lbs.	q. lbs.	q. lbs.	q. lbs.	q. lbs.	q. lbs.	q. lbs.	q. lbs.	q. lbs.	q. lbs.
Elbow	10	19	26	1 6	1 16	2 0	3 0	3 18	1 1 24	1 2 14	2 0 10	2 3 0	—
½ bend	13	—	26	1 6	1 22	2 8	1 0	0 1	1 14	2 0	20 2 3	21 3 0	14 4 1
⅛ ,,	13	—	24	25	1 7	—	2 0	3 26	1 1	8 1	3 22	2 3	20 3 1
⅛ ,,	—	—	20	25	1 8	—	1 27	2 25	1 1	0	—	—	3 0
Tee	1 7	1 18	2 5	2 24	3 6	3 19	1 0	21 1	2 10	2 2	7 3	0 18	4 0
Branch	1 9	1 24	2 12	3 0	—	1 0	10 1	1 14	1 2	14 3	0 0	3 1	6 3
Collar	11	16	20	26	1 2	1 16	1 20	2 10	3 0	3 17	3 10	1 0	14 1
Syphon bend													
,,													
Plug pipe	—	1 10	1 20	1 25	—	2 12	2 27						
End plug pipe	—	—	—	2 20	—	2 26	3 12						
,, ,, New Style													
Ducksfoot bend	—	1 16	—	2 7	—	3 16							
Syphon and cover	3 26	1 2 0	—	2 0 0	—	2 1 0	2 2 14	3 1 21	5 0 21	5 1 14	6 0 14	7 0 0	8 1 0
,, ,, new style	—	1 1 17	—	1 3 16	—	2 0 25	3 0 15	3 0 21	—	3 2 7	—	—	9 3 0
Cross	—	1 27	—	3 16	—	1 0 16	1 2 12	2 0 4					
Double bend 4½ cen.	—	—	—	1 16	—	2 7							
,, 6 ,,	—	—	—	1 17	—	2 10							
,, 7 ,,													
,, 9 ,,	—	—	—	1 18	—	2 14							
,, 12 ,,	—	—	—	1 25	—	3 0							
,, 13½ ,,	—	—	—	2 0	—	3 0							
,, 15 ,,													
,, 18 ,,													
,, 21 ,,													
Cap	8	8	9	15	18	23	1 1	1 8	1 24	2 10	2 22	3 12	1 1 20
Angle branch	—	1 24	—	3 0	—	3 24	1 1 14	1 3 14					

Reducing Pipes.

Size.	c. q. lb.	Size.	c. q. lb.	Size.	q. lb.	Size.	q. lb.	Size.	lbs.
12 in. to 10 in.	3 0 7	8 in. to 6 in.	1 3 14	5 in. to 4 in.	3 10	2 to 1½ in.	1 6	3 to 2 in., 2 in.	
10 ,, 9	3 0 2	7 ,, 6	1 2 0	5 ,, 3	3 0	6 ,, 5 in., 5 in. socket	2 14	socket	25
10 ,, 8	2 2 25	7 ,, 5	1 1 21	4 ,, 3	2 18	6 ,, 4 ,,	1 25		
9 ,, 8	2 1 20	6 ,, 5	1 0 8	4 ,, 2	2 8	5 ,, 4 ,,	1 18		
9 ,, 7	2 0 8	6 ,, 4	1 0 4	3 ,, 2½	2 6	4 ,, 3 ,,	1 9		
8 ,, 7	2 0 0	6 ,, 3	1 0 0	3 ,, 1	1 24	4 ,, 2 ,,	1 3		

(Bailey, Pegg & Co.)

WEIGHT OF SEAMLESS COPPER TUBES.

S.W.G.		Thickness of Copper.																							
		00 0	000	00	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Inches.		·400	·372	·348	·324	·300	·276	·252	·232	·212	·192	·176	·160	·144	·128	·116	·104	·092	·080	·072	·064	·056	·048	·040	·036
F=full. B=bare.		13/32B	3/8B	11/32F	21/64B	19/64F	9/32B	1/4F	15/64B	7/32B	3/16F	11/64F	5/32F	9/64F	1/8F	7/64F	7/64B	3/32	5/64F	5/64B	1/16F	1/16B	3/64F	3/64B	1/32F
Millimètres.		10·160	9·449	8·839	8·229	7·620	7·010	6·401	5·893	5·385	4·877	4·470	4·064	3·658	3·251	2·946	2·262	2·337	2·032	1·829	1·626	1·422	1·219	1·016	0·914
Internal Diam.		Weight of a Lineal Foot in Pounds.																							
Inches.	Mm.																								
$\frac{1}{8}$	3·2	2·54	2·24	1·99	1·76	1·54	1·34	1·15	1·00	·86	·74	·64	·55	·47	·39	·34	·29	·24	·20	·17	·15	·12	·10	·08	·07
$\frac{1}{4}$	6·3	3·14	2·80	2·52	2·25	2·00	1·76	1·53	1·35	1·18	1·03	·91	·79	·69	·58	·51	·44	·38	·32	·28	·24	·21	·17	·14	·12
$\frac{3}{8}$	9·5	3·75	3·36	3·04	2·74	2·45	2·17	1·91	1·70	1·50	1·32	1·17	1·04	·90	·78	·69	·60	·52	·44	·39	·34	·29	·25	·20	·18
$\frac{1}{2}$	12·7	4·35	3·92	3·57	3·23	2·90	2·59	2·29	2·05	1·83	1·61	1·44	1·28	1·12	·97	·86	·76	·66	·56	·50	·44	·38	·32	·26	·23
$\frac{5}{8}$	15·9	4·96	4·49	4·10	3·72	3·36	3·01	2·67	2·40	2·15	1·90	1·70	1·52	1·34	1·17	1·04	·92	·80	·68	·61	·53	·46	·39	·32	·29
$\frac{3}{4}$	19·0	5·56	5·05	4·62	4·21	3·81	3·43	3·05	2·76	2·47	2·19	1·97	1·76	1·56	1·36	1·21	1·07	·94	·80	·72	·63	·55	·46	·38	·34
$\frac{7}{8}$	22·2	6·17	5·61	5·15	4·70	4·26	3·84	3·44	3·11	2·79	2·48	2·24	2·00	1·77	1·55	1·39	1·23	1·08	·92	·82	·73	·63	·54	·44	·40
1	25·4	6·77	6·17	5·67	5·19	4·72	4·26	3·82	3·46	3·11	2·77	2·50	2·24	1·99	1·75	1·57	1·39	1·21	1·04	·93	·82	·71	·61	·50	·45
$1\frac{1}{8}$	28·6	7·38	6·74	6·20	5·68	5·17	4·68	4·20	3·81	3·43	3·06	2·77	2·49	2·21	1·94	1·74	1·55	1·35	1·17	1·04	·92	·80	·68	·56	·51
$1\frac{1}{4}$	31·7	7·98	7·30	6·73	6·17	5·62	5·09	4·58	4·16	3·75	3·35	3·04	2·73	2·43	2·13	1·92	1·70	1·49	1·29	1·15	1·02	·88	·75	·62	·56
$1\frac{3}{8}$	34·9	8·59	7·86	7·25	6·66	6·08	5·51	4·96	4·51	4·07	3·64	3·30	2·97	2·65	2·33	2·09	1·86	1·63	1·41	1·26	1·11	·97	·83	·68	·61
$1\frac{1}{2}$	38·1	9·19	8·42	7·78	7·15	6·53	5·93	5·34	4·86	4·39	3·93	3·57	3·21	2·86	2·52	2·27	2·02	1·77	1·53	1·37	1·21	1·05	·90	·74	·67
$1\frac{5}{8}$	41·3	9·80	8·99	8·31	7·64	6·99	6·35	5·72	5·21	4·71	4·22	3·83	3·45	3·08	2·71	2·44	2·17	1·91	1·65	1·48	1·31	1·14	·97	·81	·72
$1\frac{3}{4}$	44·4	10·40	9·55	8·83	8·13	7·44	6·76	6·10	5·56	5·03	4·51	4·10	3·70	3·30	2·91	2·62	2·33	2·05	1·77	1·59	1·40	1·22	1·04	·87	·78
$1\frac{7}{8}$	47·6	11·01	10·11	9·36	8·62	7·89	7·18	6·48	5·91	5·35	4·80	4·37	3·94	3·52	3·10	2·79	2·49	2·19	1·89	1·70	1·50	1·31	1·12	·93	·83
2	50·8	11·61	10·67	9·88	9·11	8·35	7·60	6·86	6·26	5·67	5·09	4·63	4·18	3·73	3·29	2·97	2·65	2·33	2·01	1·80	1·60	1·39	1·19	·99	·89
$2\frac{1}{8}$	54·0	12·23	11·24	10·41	9·60	8·80	8·02	7·25	6·61	5·99	5·38	4·90	4·42	3·95	3·49	3·14	2·80	2·47	2·13	1·91	1·69	1·48	1·26	1·05	·94
$2\frac{1}{4}$	57·1	12·82	11·80	10·94	10·09	9·25	8·48	7·63	6·97	6·31	5·67	5·16	4·66	4·17	3·68	3·32	2·96	2·61	2·25	2·02	1·79	1·56	1·33	1·11	1·00
$2\frac{3}{8}$	60·3	13·43	12·36	11·46	10·58	9·71	8·85	8·01	7·32	6·63	5·96	5·43	4·91	4·39	3·88	3·50	3·12	2·75	2·38	2·13	1·89	1·65	1·41	1·17	1·05
$2\frac{1}{2}$	63·5	14·03	12·92	11·99	11·07	10·16	9·27	8·39	7·67	6·95	6·25	5·70	5·15	4·61	4·07	3·67	3·28	2·88	2·50	2·24	1·98	1·73	1·48	1·23	1·10
$2\frac{5}{8}$	66·7	14·64	13·49	12·52	11·56	10·62	9·69	8·77	8·02	7·28	6·54	5·96	5·39	4·82	4·26	3·85	3·43	3·02	2·62	2·35	2·08	1·82	1·55	1·29	1·16
$2\frac{3}{4}$	69·8	15·24	14·05	13·04	12·05	11·07	10·10	9·15	8·37	7·60	6·83	6·23	5·63	5·04	4·46	4·02	3·59	3·16	2·74	2·46	2·18	1·90	1·62	1·35	1·21
$2\frac{7}{8}$	73·0	15·85	14·61	13·57	12·54	11·52	10·52	9·53	8·72	7·92	7·12	6·50	5·87	5·26	4·65	4·20	3·75	3·30	2·86	2·57	2·27	1·99	1·70	1·41	1·27
3	76·2	16·45	15·17	14·09	13·03	11·98	10·94	9·91	9·07	8·24	7·41	6·76	6·12	5·48	4·84	4·37	3·90	3·44	2·98	2·68	2·37	2·07	1·77	1·47	1·32
$3\frac{1}{4}$	82·5	17·66	16·30	15·15	14·01	12·88	11·77	10·68	9·77	8·88	7·99	7·29	6·60	5·91	5·23	4·72	4·22	3·72	3·22	2·89	2·57	2·24	1·91	1·59	1·43
$3\frac{1}{2}$	88·9	18·87	17·42	16·20	14·99	13·79	12·61	11·44	10·47	9·52	8·58	7·83	7·08	6·35	5·62	5·07	4·53	4·00	3·46	3·11	2·76	2·41	2·06	1·71	1·54
$3\frac{3}{4}$	95·2	20·08	18·55	17·25	15·97	14·70	13·44	12·20	11·18	10·16	9·16	8·36	7·57	6·78	6·00	5·42	4·85	4·28	3·71	3·33	2·95	2·58	2·20	1·83	1·65
4	101·6	21·29	19·67	18·30	16·95	15·61	14·28	12·96	11·88	10·80	9·74	8·89	8·05	7·22	6·39	5·78	5·16	4·55	3·95	3·55	3·15	2·75	2·35	1·95	1·76
$4\frac{1}{4}$	107·9	22·50	20·80	19·36	17·93	16·51	15·11	13·72	12·58	11·44	10·32	9·42	8·54	7·65	6·78	6·13	5·48	4·83	4·19	3·76	3·34	2·92	2·50	2·08	1·87
$4\frac{1}{2}$	114·3	23·71	21·93	20·41	18·91	17·42	15·95	14·49	13·28	12·08	10·90	9·96	9·02	8·09	7·17	6·48	5·79	5·11	4·43	3·98	3·53	3·09	2·64	2·20	1·97
$4\frac{7}{8}$	120·6	24·92	23·05	21·46	19·89	18·33	16·78	15·25	13·98	12·73	11·48	10·49	9·50	8·53	7·55	6·83	6·11	5·39	4·67	4·20	3·73	3·26	2·79	2·32	2·08
5	127·0	26·13	24·18	22·51	20·87	19·23	17·62	16·01	14·68	13·37	12·06	11·02	9·99	8·96	7·94	7·18	6·42	5·67	4·92	4·42	3·92	3·42	2·93	2·44	2·19
$5\frac{1}{4}$	133·3	27·34	25·30	23·57	21·85	20·14	18·45	16·77	15·39	14·01	12·64	11·55	10·47	9·40	8·33	7·53	6·74	5·95	5·16	4·64	4·11	3·59	3·08	2·56	
$5\frac{1}{2}$	139·7	28·55	26·43	24·62	22·83	21·05	19·29	17·54	16·09	14·65	13·22	12·08	10·96	9·83	8·71	7·88	7·05	6·22	5·40	4·85	4·31	3·76	3·22	2·68	
$5\frac{7}{8}$	146·0	29·76	27·55	25·67	23·81	21·96	20·12	18·30	16·79	15·29	13·80	12·62	11·44	10·27	9·10	8·23	7·36	6·50	5·64	5·07	4·50	3·93	3·37	2·80	
6	152·4	30·97	28·68	26·72	24·79	22·86	20·95	19·06	17·49	15·93	14·38	13·15	11·93	10·70	9·49	8·58	7·68	6·78	5·88	5·29	4·69	4·10	3·51	2·92	

S.W.G.	0000	000	00	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
	Thickness of Copper.																							
Inches.	·400	·372	·348	·324	·300	·276	·252	·232	·212	·192	·176	·160	·144	·128	·116	·104	·092	·080	·072	·064	·056	·048	·040	·036
F=full. B=bare.	13/32B	3/8B	11/32F	21/64B	19/64F	9/32B	1/4F	15/64B	7/32B	3/16F	11/64F	5/32F	9/64F	1/8F	7/64F	7/64B	3/32	5/64F	5/64B	1/16F	1/16B	3/64F	3/64B	1/32F
Millimètres.	10·160	9·449	8·889	8·229	7·620	7·010	6·401	5·893	5·385	4·877	4·470	4·064	3·658	3·251	2·946	2·642	2·337	2·032	1·829	1·626	1·422	1·219	1·016	·914
Internal Diam.	Weight of a Lineal Foot in Pounds.																							
Inches.	Mm.																							
6¼	158·7	32·18	29·80	27·78	25·77	23·77	21·79	19·82	18·19	16·57	14·96	13·68	12·41	11·14	9·88	8·93	7·99	7·06	6·13	5·51	4·89	4·27	3·66	
6½	165·1	33·39	30·93	28·83	26·75	24·68	22·62	20·58	18·89	17·21	15·54	14·21	12·89	11·57	10·26	9·28	8·31	7·34	6·37	5·72	5·08	4·44	3·80	
6¾	171·4	34·60	32·05	29·88	27·78	25·59	23·46	21·35	19·60	17·85	16·12	14·75	13·37	12·01	10·65	9·63	8·62	7·61	6·61	5·94	5·28	4·61	3·95	
7	177·8	35·81	33·18	30·93	28·71	26·49	24·29	22·11	20·30	18·50	16·70	15·28	13·86	12·44	11·04	9·99	8·94	7·89	6·85	6·16	5·47	4·78	4·09	
7¼	184·1	37·02	34·30	31·99	29·69	27·40	25·13	22·87	21·00	19·14	17·29	15·81	14·34	12·88	11·42	10·34	9·25	8·17	7·09	6·38	5·66	4·95		
7½	190·5	38·23	35·43	33·04	30·67	28·31	25·96	23·63	21·70	19·78	17·87	16·34	14·83	13·32	11·81	10·69	9·57	8·45	7·34	6·60	5·86	5·12		
7¾	196·8	39·44	36·55	34·09	31·65	29·22	26·80	24·39	22·40	20·42	18·45	16·88	15·31	13·75	12·20	11·04	9·88	8·73	7·58	6·81	6·05	5·29		
8	203·2	40·65	37·68	35·14	32·63	30·12	27·63	25·16	23·10	21·06	19·03	17·41	15·79	14·19	12·59	11·39	10·20	9·01	7·82	7·03	6·24	5·46		
8¼	209·5	41·86	38·80	36·20	33·61	31·03	28·47	25·92	23·81	21·70	19·61	17·94	16·28	14·62	12·97	11·74	10·51	9·28	8·06	7·25	6·44			
8½	215·9	43·07	39·93	37·25	34·59	31·94	29·30	26·68	24·51	22·34	20·19	18·47	16·76	15·06	13·36	12·09	10·82	9·56	8·30	7·47	6·63			
8¾	222·2	44·28	41·05	38·30	35·57	32·84	30·14	27·44	25·21	22·98	20·77	19·00	17·25	15·49	13·75	12·44	11·14	9·84	8·55	7·68	6·82			
9	228·6	45·49	42·18	39·35	36·55	33·75	30·97	28·21	25·91	23·63	21·35	19·54	17·73	15·93	14·13	12·79	11·45	10·12	8·79	7·90	7·02			
9¼	234·9	46·70	43·30	40·41	37·53	34·66	31·81	28·97	26·61	24·27	21·93	20·07	18·21	16·36	14·52	13·14	11·77	10·40	9·08	8·12				
9½	241·3	47·91	44·43	41·46	38·51	35·57	32·64	29·73	27·31	24·91	22·51	20·60	18·70	16·80	14·91	13·49	12·08	10·68	9·27	8·34				
9¾	247·6	49·12	45·55	42·51	39·49	36·47	33·48	30·49	28·02	25·55	23·09	21·13	19·18	17·24	15·30	13·84	12·40	10·95	9·51	8·55				
10	254·0	50·33	46·68	43·56	40·47	37·38	34·31	31·25	28·72	26·19	23·67	21·67	19·67	17·67	15·68	14·20	12·71	11·23	9·76	8·77				
10¼	260·3	51·54	47·80	44·62	41·45	38·29	35·15	32·02	29·42	26·83	24·25	22·20	20·15	18·11	16·07	14·55	13·03	11·51	10·00					
10½	266·7	52·75	48·93	45·67	42·43	39·20	35·98	32·78	30·12	27·47	24·83	22·73	20·63	18·54	16·46	14·90	13·34	11·79	10·24					
10¾	273·0	53·96	50·05	46·72	43·41	40·10	36·81	33·54	30·82	28·11	25·42	23·26	21·12	18·98	16·84	15·25	13·66	12·07	10·48					
11	279·4	55·16	51·18	47·77	44·39	41·01	37·65	34·30	31·52	28·75	26·00	23·80	21·60	19·41	17·23	15·60	13·97	12·34	10·72					
11¼	285·7	56·37	52·30	48·83	45·37	41·92	38·48	35·06	32·28	29·40	26·58	24·33	22·08	19·85	17·62	15·95	14·28	12·62						
11½	292·1	57·58	53·43	49·88	46·35	42·83	39·32	35·88	32·95	30·04	27·16	24·86	22·57	20·28	18·01	16·30	14·60	12·90						
11¾	298·4	58·79	54·55	50·93	47·33	43·73	40·15	36·59	33·63	30·68	27·74	25·39	23·05	20·72	18·39	16·65	14·91	13·18						
12	304·8	60·00	55·68	51·98	48·31	44·64	40·99	37·35	34·33	31·32	28·32	25·92	23·54	21·16	18·78	17·00	15·23	13·46						
12¼	311·1	61·21	56·80	53·04	49·29	45·55	41·82	38·11	35·03	31·96	28·90	26·46	24·02	21·59	19·17	17·35	15·54							
12½	317·5	62·42	57·93	54·09	50·26	46·45	42·66	38·88	35·73	32·60	29·48	26·99	24·50	22·03	19·55	17·70	15·86							
12¾	323·8	63·63	59·05	55·14	51·24	47·36	43·49	39·64	36·44	33·24	30·06	27·52	24·99	22·46	19·94	18·05	16·17							
13	330·2	64·84	60·18	56·19	52·22	48·27	44·33	40·40	37·14	33·88	30·64	28·05	25·47	22·90	20·33	18·41	16·49							
13¼	336·5	66·05	61·30	57·25	53·20	49·18	45·16	41·16	37·84	34·53	31·22	28·59	25·96	23·33	20·72	18·76								
13½	342·9	67·26	62·43	58·30	54·18	50·08	46·00	41·92	38·54	35·17	31·80	29·12	26·44	23·77	21·10	19·11								
13¾	349·2	68·47	63·55	59·35	55·16	50·99	46·83	42·69	39·24	35·81	32·38	29·65	26·92	24·20	21·49	19·46								
14	355·6	69·68	64·68	60·40	56·14	51·90	47·67	43·45	39·94	36·45	32·96	30·18	27·41	24·64	21·88	19·81								
		3·87	3·35	2·93	2·54	2·18	1·84	1·54	1·30	1·09	·89	·75	·62	·50	·40	·33	·26	·20	·15	·12	·10	·08	·06	·04	·03

If the external diameter is given, subtract figure in black type at bottom of column; for example, the weight per lineal foot of a copper tube 2 inches external diameter, 12 S.W.G., is 2·65 − 0·26 = 2·39 lbs. The weights are calculated from 12 samples of tubes made from tough copper (specific gravity at 60° F. 8·8917, or ·3299 lb. per cubic inch). Brazed copper tubes weigh more than seamless; but an exact multiple cannot be given. Mandril-drawn brazed tubes weigh the same as seamless tubes. In practice a slight deviation from the above theoretical weights must be expected. To ascertain the weight of a seamless tube of other metal, multiply the weight of a similar copper tube by ·9626 for brass (10 and 30 alloy), by ·86 for wrought iron, by ·81 for cast iron, or by 1·28 for lead. (*The Broughton Copper Co., Ltd.*)

Weight of Seamless Brass Tubes.

(Containing 70 % of copper.)

S.W.G.	Thickness of Brass															
	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Inches . .	·212	·192	·176	·160	·144	·128	·116	·104	·092	·080	·072	·064	·056	·048	·040	·036
Millimètres	5·385	4·877	4·470	4·064	3·658	3·251	2·946	2·642	2·337	2·032	1·829	1·626	1·422	1·219	1·016	·914
External Diameter.	Weight of a Lineal Foot in Pounds.															
Ins. Mm.																
9·5	..	..	..	..	..	..	..	..	..	..	..	..	·21	·18	·16	·14
11·1	..	..	..	..	..	..	..	..	..	..	..	·28	·25	·22	·18	·17
12·7	..	..	..	..	..	..	..	..	..	..	·36	·32	·29	·25	·21	·19
15·9	..	..	..	..	..	..	..	..	..	·51	·46	·42	·37	·32	·27	·25
19·0	..	..	..	..	..	..	..	..	·70	·62	·57	·51	·45	·39	·33	·30
22·2	..	..	..	..	..	..	..	·93	·84	·74	·67	·60	·53	·46	·39	·35
1 25·4	..	..	..	..	..	..	1·19	1·08	·97	·86	·78	·70	·62	·53	·45	·40
1⅛ 28·6	..	..	..	..	..	1·49	1·36	1·24	1·11	·97	·88	·79	·70	·60	·50	
1¼ 31·7	..	..	..	..	..	1·67	1·53	1·39	1·24	1·09	·99	·88	·78	·67	·56	
1⅜ 34·9	..	..	..	..	2·06	1·86	1·70	1·54	1·37	1·21	1·09	·98	·86	·74	·62	
1½ 38·1	..	..	..	..	2·27	2·04	1·87	1·69	1·51	1·32	1·20	1·07	·94	·81	·68	
1⅝ 41·3	..	..	..	2·73	2·48	2·23	2·04	1·84	1·64	1·44	1·30	1·16	1·02	·88		
1¾ 44·4	..	..	..	2·96	2·69	2·42	2·21	1·99	1·78	1·56	1·41	1·26	1·10	·95		
1⅞ 47·6	..	..	3·48	3·19	2·90	2·60	2·38	2·14	1·91	1·67	1·51	1·35	1·19	1·02		
2 50·8	..	..	3·74	3·43	3·11	2·79	2·54	2·30	2·04	1·79	1·62	1·44	1·27	1·09		
2⅛ 54·0	..	4·32	3·99	3·66	3·32	2·98	2·71	2·45	2·18	1·90	1·72	1·54	1·35			
2¼ 57·1	..	4·60	4·25	3·89	3·53	3·16	2·88	2·60	2·31	2·02	1·83	1·63	1·43			
2⅜ 60·3	5·34	4·88	4·51	4·13	3·74	3·35	3·05	2·75	2·45	2·14	1·93	1·72	1·51			
2½ 63·5	5·65	5·16	4·76	4·36	3·95	3·54	3·22	2·90	2·58	2·25	2·04	1·81	1·59			
2⅝ 66·7	5·96	5·44	5·02	4·59	4·16	3·72	3·39	3·05	2·71	2·37	2·14	1·91	1·67			
2¾ 69·8	6·27	5·72	5·28	4·83	4·37	3·91	3·56	3·20	2·85	2·49	2·24	2·00	1·76			
2⅞ 73·0	6·57	6·00	5·53	5·06	4·58	4·09	3·73	3·36	2·98	2·60	2·35	2·09	1·84			
3 76·2	6·88	6·28	5·79	5·29	4·79	4·28	3·90	3·51	3·11	2·72	2·45	2·19	1·92			
3⅛ 79·3	7·19	6·56	6·04	5·52	5·00	4·47	4·06	3·66	3·25	2·84	2·56	2·28				
3¼ 82·5	7·50	6·84	6·30	5·76	5·21	4·65	4·23	3·81	3·38	2·95	2·66	2·37				
3⅜ 85·7	7·81	7·12	6·56	5·99	5·42	4·84	4·40	3·96	3·52	3·07	2·77	2·47				
3½ 88·9	8·12	7·40	6·81	6·22	5·63	5·03	4·57	4·11	3·65	3·19	2·87	2·56				
3⅝ 92·0	8·43	7·68	7·07	6·46	5·84	5·21	4·74	4·26	3·78	3·30	2·98	2·65				
3¾ 95·2	8·73	7·95	7·32	6·69	6·05	5·40	4·91	4·42	3·92	3·42	3·08	2·75				
3⅞ 98·4	9·04	8·23	7·58	6·92	6·26	5·58	5·08	4·57	4·05	3·53	3·19	2·84				
4 101·6	9·35	8·51	7·84	7·15	6·47	5·77	5·25	4·72	4·19	3·65	3·29	2·93				
4¼ 107·9	9·97	9·07	8·35	7·62	6·88	6·14	5·58	5·02	4·45	3·88	3·50					
4½ 114·3	10·59	9·63	8·86	8·09	7·30	6·52	5·92	5·32	4·72	4·12	3·71					
4¾ 120·6	11·20	10·19	9·37	8·55	7·72	6·89	6·26	5·63	4·99	4·35						
5 127·0	11·82	10·75	9·89	9·02	8·14	7·26	6·60	5·93	5·26	4·58						
5¼ 133·3	12·44	11·31	10·40	9·48	8·56	7·63	6·93	6·23	5·53							
5½ 139·7	13·05	11·87	10·91	9·95	8·98	8·01	7·27	6·53	5·79							
5¾ 146·0	13·67	12·43	11·42	10 41	9·40	8·38	7·61	6·84								
6 152·4	14·29	12·99	11·94	10·88	9·82	8·75	7·95									
Per cent .	*103·4*	*105·2*	*102·1*	*102·9*	*102·6*	*104·4*	*103·3*	*104·6*	*103·1*	*103·6*	*100·0*	*101·5*	*103·5*	*102·0*	*104·8*	*97·3*

The figures in italics show the difference in percentage of weight of brass tubes made to the B.W.G. and the S.W.G., the latter being taken at 100.

The above weights, multiplied by ·994, give the weight of a brass tube 2 and 1 alloy.

These weights are calculated from the specific gravity of 16 samples of tubes, from various makers, each of the samples being 1 foot long × 1¼ inch external diameter × 14 B.W.G., which was found to be 8·558, or ·3089 lb. per cubic inch, for the alloy of 70 and 30, and 8·508 specific gravity, or ·3071 lb. per cubic inch, for the alloy of 2 and 1 at a temperature of 60° F.

To ascertain the weight of a seamless tube of copper, multiply the weight of a similar brass tube by 1·0388.

S.W.G N	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
Thickness ins.	0·300	0·276	0·252	0·232	0·212	0·192	0·176	0·160	0·144	0·128	0·116	0·104	0·092	0·080	0·072	0·064	0·056	0·048	0·040	0·036	0·032	0·028	0·024
Thickness mm.	7·62	7·00	6·40	5·89	5·38	4·87	4·46	4·06	3·66	3·25	2·94	2·64	2·34	2·03	1·83	1·62	1·42	1·22	1·02	0·914	0·812	0·711	0·610
Outside Diameter ins. mm.	Weight in lbs. per Linear foot.																						
4·76	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	0·029	0·028	0·025	0·022	0·020	0·018	0·016	0·014
¼ 6·35	—	—	—	—	—	—	—	—	—	—	—	—	—	0·050	0·047	0·044	0·040	0·036	0·031	0·028	0·026	0·023	0·020
9·52	—	—	—	—	—	—	—	—	—	—	0·110	0·104	0·096	0·087	0·080	0·073	0·066	0·058	0·049	0·045	0·040	0·036	0·031
12·7	—	—	—	—	—	—	—	—	0·189	0·175	0·164	0·152	0·138	0·124	0·113	0·103	0·091	0·080	0·068	0·061	0·055	0·049	0·042
⅝ 15·9	—	—	—	—	—	0·308	0·291	0·274	0·255	0·234	0·217	0·199	0·180	0·160	0·147	0·132	0·117	0·102	0·086	0·078	0·070	0·061	0·053
¾ 19·0	—	—	—	—	0·419	0·394	0·372	0·347	0·321	0·293	0·270	0·247	0·223	0·197	0·180	0·162	0·143	0·124	0·104	0·095	0·085	0·074	0·064
⅞ 22·2	—	—	0·578	0·549	0·517	0·483	0·453	0·421	0·387	0·352	0·324	0·295	0·265	0·234	0·213	0·191	0·169	0·146	0·123	0·111	0·099	0·087	0·075
1 25·4	0·773	0·735	0·694	0·656	0·615	0·571	0·533	0·494	0·453	0·411	0·377	0·343	0·308	0·271	0·246	0·220	0·195	0·168	0·141	0·128	0·114	0·100	0·086
1⅛ 28·6	0·910	0·862	0·809	0·762	0·713	0·659	0·615	0·568	0·520	0·469	0·431	0·391	0·350	0·308	0·279	0·251	0·220	0·190	0·160	0·144	0·129	0·113	0·097
1¼ 31·8	1·05	0·989	0·926	0·870	0·810	0·748	0·696	0·642	0·586	0·529	0·484	0·438	0·392	0·345	0·312	0·279	0·246	0·213	0·178	0·161	0·143	0·126	0·108
1⅜ 34·9	1·19	1·11	1·04	0·975	0·907	0·836	0·777	0·716	0·653	0·587	0·538	0·487	0·435	0·381	0·345	0·309	0·272	0·235	0·196	0·178	0·158	0·139	0·120
1½ 38·1	1·32	1·24	1·16	1·08	1·01	0·924	0·858	0·789	0·719	0·646	0·591	0·534	0·476	0·418	0·378	0·338	0·298	0·257	0·215	0·194	0·173	0·152	0·130
1⅝ 41·3	1·46	1·37	1·27	1·19	1·10	1·01	0·939	0·863	0·785	0·705	0·645	0·582	0·519	0·455	0·411	0·368	0·323	0·279	0·234	0·211	0·188	—	—
1¾ 44·5	1·60	1·50	1·39	1·30	1·20	1·10	1·02	0·936	0·851	0·764	0·699	0·630	0·560	0·491	0·444	0·397	0·349	0·301	0·252	0·227	0·202	—	—
1⅞ 47·6	1·74	1·62	1·50	1·40	1·30	1·19	1·10	1·01	0·917	0·822	0·751	0·678	0·603	0·528	0·478	0·426	0·375	0·323	—	—	—	—	—
2 50·8	1·88	1·75	1·62	1·51	1·39	1·28	1·18	1·08	0·982	0·882	0·805	0·726	0·645	0·565	0·510	0·455	0·400	0·345	—	—	—	—	—
2⅛ 53·9	2·01	1·88	1·74	1·62	1·49	1·36	1·26	1·16	1·05	0·941	0·858	0·773	0·687	0·602	0·543	0·485	0·426	0·370	—	—	—	—	—
2¼ 57·1	2·15	2·00	1·85	1·72	1·59	1·45	1·34	1·23	1·12	1·00	0·910	0·821	0·730	0·639	0·577	0·515	0·452	0·389	—	—	—	—	—
2⅜ 60·3	2·29	2·13	1·97	1·83	1·69	1·54	1·42	1·30	1·18	1·06	0·965	0·866	0·775	0·676	0·610	0·544	0·478	0·411	—	—	—	—	—
2½ 63·5	2·43	2·26	2·08	1·94	1·78	1·63	1·51	1·38	1·25	1·12	1·02	0·916	0·815	0·712	0·643	0·573	0·504	0·433	—	—	—	—	—
2⅝ 66·6	2·57	2·39	2·20	2·04	1·88	1·72	1·59	1·45	1·32	1·18	1·07	0·965	0·857	0·750	0·675	0·603	0·530	0·455	—	—	—	—	—
2¾ 69·8	2·70	2·51	2·32	2·15	1·98	1·81	1·67	1·53	1·38	1·24	1·12	1·01	0·899	0·785	0·708	0·632	0·555	0·477	—	—	—	—	—
2⅞ 73·0	2·84	2·64	2·43	2·26	2·08	1·90	1·75	1·60	1·44	1·29	1·18	1·06	0·942	0·822	0·743	0·662	0·581	0·500	—	—	—	—	—
3 76·2	2·98	2·77	2·55	2·37	2·17	1·98	1·83	1·67	1·51	1·35	1·23	1·11	0·985	0·860	0·776	0·690	—	—	—	—	—	—	—
3⅛ 82·6	3·26	3·02	2·78	2·57	2·37	2·16	1·99	1·82	1·65	1·47	1·34	1·20	1·07	0·932	0·842	0·750	—	—	—	—	—	—	—
3¼ 88·9	3·53	3·27	3·02	2·78	2·57	2·34	2·15	1·97	1·78	1·59	1·45	1·30	1·15	1·01	0·909	0·809	—	—	—	—	—	—	—
3⅜ 95·2	3·81	3·53	3·24	3·00	2·76	2·51	2·32	2·12	1·91	1·71	1·55	1·40	1·24	1·08	—	—	—	—	—	—	—	—	—
4 102	4·08	3·78	3·48	3·22	2·96	2·69	2·48	2·26	2·04	1·82	—	—	—	—	—	—	—	—	—	—	—	—	—
4⅛ 108	4·36	4·03	3·70	3·43	3·15	2·87	2·64	2·41	2·18	1·94	—	—	—	—	—	—	—	—	—	—	—	—	—
4¼ 114	4·63	4·28	3·93	3·64	3·34	3·04	2·80	2·56	2·31	2·06	—	—	—	—	—	—	—	—	—	—	—	—	—
4⅜ 121	4·91	4·54	4·17	3·85	3·54	3·22	2·96	2·71	2·44	2·18	—	—	—	—	—	—	—	—	—	—	—	—	—
5 127	5·19	4·79	4·40	4·06	3·73	3·40	3·12	2·85	2·57	2·30	—	—	—	—	—	—	—	—	—	—	—	—	—
5⅛ 133	5·46	5·05	4·62	4·27	3·92	3·57	3·29	3·00	2·70	2·41	—	—	—	—	—	—	—	—	—	—	—	—	—
5¼ 140	5·75	5·30	4·86	4·49	4·12	3·74	3·45	3·15	2·84	2·53	—	—	—	—	—	—	—	—	—	—	—	—	—
5⅜ 146	6·01	5·55	5·09	4·71	4·32	3·92	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
6 152	6·30	5·81	5·33	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—

	Aluminium.	Brass.	Copper.	Steel.
Specific Gravity ...	2·71	8·45	8·89	7·80
Ratio of Weight ...	1·00	3·12	3·28	2·88

NOTE.—The above weights are subject to the usual manufacturing tolerances. (*The British Aluminium Co., Ltd.*)

TINPLATES.

Principal Denominations and Sizes.

Mark.	Size.	Sheets per Box.	Weight per Box.	Thicknesses of Sheets.		Mark.	Size.	Sheets per Box.	Weight per Box.	Thicknesses of Sheets.	
	Ins.		Lbs.	Mm.	In.		Ins.		Lbs.	Mm.	In.
IC	14×10	225	108	0·313	0·0123	IC	28×20	56	108	0·315	0·0123
IX	,,	,,	136	0·395	0·0155	IX	,,	,,	136	0·396	0·0155
IXX	,,	,,	156	0·453	0·0179	IC	20×10	225	154	0·313	0·0123
IXXX	,,	,,	176	0·511	0·0201	IX	,,	,,	194	0·394	0·0155
IC	20×14	112	108	0·315	0·0123	IC	14×18¾	124	110	0·309	0·0122
ICL	,,	,,	100	0·292	0·0114	IC	14×19¼	120	110	0·311	0·0122
ICL	,,	,,	95	0·277	0·0109	IC	30×21	112	243	0·315	0·0124
ICL	,,	,,	90	0·262	0·0103	CL	,,	,,	224	0·290	0·0114
ICL	,,	,,	85	0·248	0·0097	CLL	,,	,,	190	0·246	0·0097
ICL	,,	,,	80	0·233	0·0091	CLLL	,,	,,	176	0·228	0·0090
IX	,,	,,	136	0·396	0·0155	CLLLL	,,	,,	160	0·207	0·0081
IXX	,,	,,	156	0·455	0·0179	DC	17×12½	100	94	0·404	0·0160
IXXX	,,	,,	176	0·513	0·0201	DX	,,	,,	122	0·525	0·0206
IXXXX	,,	,,	196	0·571	0·0223	DXX	,,	,,	143	0·615	0·0242
IC	28×20	,,	216	0·315	0·0124	DXXX	,,	,,	164	0·706	0·0278
IX	,,	,,	272	0·396	0·0156	DXXXX	,,	,,	185	0·796	0·0313

Weights in Lbs. per Foot Super and Equivalent S.W.G.

Mark.	Weight.	S.W.G.	Mark.	Weight.	S.W.G.
IC	0·514	30	DXXX	1·138	22
IX	0·643	27	DXXXX	1·281	21
IXX	0·739	26	SDC	0·733	26
IXXX	0·836	25	SDX	0·825	25
IXXXX	0·932	23	SDXX	0·916	24
DC	0·664	27	SDXXX	1·008	23
DX	0·854	24	SDXXXX	1·100	22
DXX	0·996	23			

Approximate Weight of Single Plates

Mark.	Weight per Plate.	Size of Plate, Ins.	Weight per Plate.	Size of Plate, Ins.	Weight per Plate.	Size of Plate, Ins.
	lb. oz.		lb. oz.		lb. oz.	
IC	0 8	14 × 10	1 0	14 × 20	2 0	28 × 20
IX	0 10		1 4		2 8	
IXX	0 11½		1 7		2 14	
IXXX	0 13		1 10		3 4	
IXXXX	0 14½		1 13		3 10	
DC	0 15½	17 × 12½	1 15½	17 × 25	3 14½	34 × 25
DX	1 4½		2 8½		5 0½	
DXX	1 7½		2 15		5 14	
DXXX	1 10½		3 5½		6 11½	
DXXXX	1 14½		3 12½		7 9	
SDC	0 13½	15 × 11	1 10½	15 × 22	—	—
SDX	0 15½		1 14½			
SDXX	1 0½		2 1½			
SDXXX	1 2½		2 5			
SDXXXX	1 4½		2 8½			

WEIGHT AND DIMENSIONS OF HOOP-IRON PER FOOT RUN.

B. W. Gauge.	Width.	Weight.	B. W. Gauge.	Width.	Weight.	B. W. Gauge.	Width.	Weight.	B. W. Gauge.	Width.	Weight.
	Ins.	Lbs.		Ins.	Lb.		Ins.	Lb.		Ins.	Lb.
11	2¾	1·1577	13	2	·6332	15	1½	·3637	18	1	·1616
12	2½	·9178	13	1⅞	·5540	15	1⅜	·3334	19	⅞	·1237
12	2	·7342	14	1¾	·4715	16	1¼	·2652	20	¾	·0884
13	2¼	·7123	14	1½	·4042	17	1⅛	·2084	21	⅝	·0694

Weight of Wire per 100 Feet.

B. W. Gauge.	Diam. in Decimals of an Inch.	Iron.	Steel.	Brass.	Copper.	B. W. Gauge.	Diam. in Decimals of an Inch.	Iron.	Steel.	Brass.	Copper.
		Lbs.	Lbs.	Lbs.	Lbs.			Lbs.	Lbs.	Lbs.	Lbs.
0	·340	30·582	30·919	33·427	35·170	19	·042	·466	·471	·510	·536
1	·312	25·753	26·036	28·148	29·616	20	035	·324	·327	·344	·372
2	·284	21·338	21·573	23·322	24·539	21	·033	·288	·291	·314	·331
3	·261	18·022	18·220	19·698	20·725	22	·029	·222	·224	·243	·255
4	·239	15·112	15·278	16·517	17·378	23	·028	·207	·209	·226	·238
5	·217	12·458	12·595	13·616	14·326	24	·025	·164	·166	·179	·189
6	·208	11·446	11·572	12·510	13·173	25	·021	·116	·118	·127	·134
7	·187	9·251	9·353	10·112	10·639	26	·020	·105	·107	·115	·121
8	·166	7·290	7·370	7·968	8·384	27	·018	·085	·086	·093	·098
9	·158	6·604	6·677	7·219	7·595	28	·015	·059	·060	·065	·068
10	·137	4·965	5·020	5·427	5·710	29	·013	·044	·045	·048	·051
11	·125	4·134	4·179	4·518	4·754	30	·012	·038	·038	·041	·043
12	·109	3·143	3·178	3·435	3·615	31	·010	·026	·026	·028	·030
13	·094	2·338	2·363	2·555	2·688	32	·009	·021	·021	·023	·024
14	·080	1·693	1·712	1·851	1·947	33	·008	·016	·017	·018	·019
15	·072	1·371	1·387	1·499	1·577	34	·007	·013	·013	·014	·014
16	·063	1·050	1·062	1·148	1·208	35	·005	·006	·006	·007	·007
17	·055	·800	·809	·874	·920	36	·004	·004	·004	·004	·005
18	·048	·609	·616	·666	·701						

Table of Weight of Castings in Proportion to the Make of Patterns of Different Materials.

A Pattern weighing 1 lb.	Will Weigh when Cast in				
	Cast Iron.	Zinc.	Copper.	Yellow Brass.	Gun-metal.
	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
Mahogany	8	8	10	9·8	10
White pine	14	14·5	18	17·5	17·8
Yellow pine	13	12·6	16	15·5	16
Cedar	11·5	11·4	14·5	14	14·5
Maple	10	9·8	12·5	12	12·4

Number of Cold Punched Nuts per 100 Pounds.

Size.	Square.	Hexagon.	Size.	Square.	Hexagon.	Size.	Square.	Hexagon.
Size ⅝ inch	2,694	2,944	¾ inch	412	496	1¼ inch	126	156
Size ¾ ,,	1,200	1,440	⅞ ,,	258	314	1½ ,,	104	121
Size ⅞ ,,	758	900	1 ,,	171	201	1½ ,,	60	68

Weight of Nut and Bolt-heads.

Head and Nut.	Diameter of Bolt in Inches.												
	¼	⅜	½	⅝	¾	⅞	1	1¼	1½	1¾	2	2½	3
	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
Hexagon .	·017	·057	·128	·267	·43	·73	1·1	2·14	3·77	5·62	8·75	17·2	28·8
Square .	·021	·070	·164	·321	·553	·882	1·31	2·56	4·42	7·00	10·5	21·0	36·4

WEIGHT OF WASHERS PER 100.

Size.	Weight.	Size.	Weight.	Size.	Weight.	Size.	Weight.
In.	Lbs.	In.	Lbs.	Ins.	Lbs.	Ins.	Lbs.
$\frac{3}{8}$	$1\frac{3}{4}$	$\frac{5}{8}$	$6\frac{3}{4}$	1	$10\frac{1}{2}$	$1\frac{1}{4}$	24
$\frac{1}{2}$	$2\frac{1}{2}$	$\frac{7}{8}$	$8\frac{1}{2}$	$1\frac{1}{8}$	$18\frac{3}{4}$	$1\frac{1}{2}$	30
$\frac{7}{8}$	$4\frac{1}{2}$						

WEIGHT OF WROUGHT-IRON NAILS PER 1,000.

Name.	Lgth	Weight per 1000.	Name.	Lgth	Weight per 1000.	Name.	Lgth	Weight per 1000.
	Ins.	Lbs. Ozs.		Ins.	Lbs. Ozs.		Ins.	Lbs. Ozs.
Spike . .	5	100 0	Clasp, strong .	$2\frac{1}{2}$	12 0	Tacks, clout .	$\frac{5}{8}$	1 12
" . .	6	200 0	" " .	$2\frac{3}{4}$	14 0	Tacks, cut .	$\frac{11}{16}$	14
" . .	7	300 0	" " .	$3\frac{1}{8}$	20 0	" " .	$\frac{3}{4}$	12
" . .	8	450 0	" " .	$3\frac{3}{8}$	25 0	" " .	$\frac{13}{16}$	10
" . .	9	600 0	" " .	$3\frac{7}{8}$	32 0	Trunk . .	$\frac{7}{8}$	1 4
" . .	10	750 0	" " .	4	40 0	" . .	1	1 12
Rose . .	$\frac{5}{8}$	2 0	Clasp, cut .	$1\frac{1}{4}$	3 0	" . .	$1\frac{1}{4}$	2 12
" . .	$1\frac{1}{4}$	4 0	" " .	$1\frac{1}{2}$	3 8	" . .	$1\frac{1}{2}$	5 0
" . .	$1\frac{3}{4}$	7 0	" " .	2	8 0	Brads, wrought	$\frac{3}{4}$	8
" . .	2	10 0	" " .	$2\frac{1}{2}$	12 0	" " .	$\frac{7}{8}$	12
" . .	$2\frac{1}{2}$	16 0	Clout . .	$\frac{3}{4}$	2 8	" " .	1	14
" . .	$2\frac{3}{4}$	20 0	Clout, fine .	1	2 0	" " .	$1\frac{1}{4}$	1 12
" . .	$3\frac{1}{4}$	28 0	" " .	$1\frac{1}{2}$	3 0	" " .	$1\frac{1}{2}$	2 12
" . .	$3\frac{3}{4}$	36 0	" " .	$1\frac{1}{2}$	4 0	" " .	2	4 0
" . .	4	50 0	" " .	$1\frac{5}{8}$	5 0	Brads, flooring	$2\frac{1}{4}$	10 0
" . .	$4\frac{1}{2}$	70 0	Clout, strong .	$1\frac{5}{8}$	5 0	" " .	$2\frac{1}{2}$	15 0
" . .	5	90 0	" " .	$1\frac{5}{8}$	9 0	" " .	$3\frac{1}{4}$	20 0
Rose, fine drawn	4	40 0	" " .	$1\frac{3}{4}$	16 0	Brads, cut .	$\frac{1}{2}$	5
" " " .	$5\frac{1}{4}$	80 0	" " .	2	23 0	" " .	$\frac{3}{4}$	12
Rose, stamped .	$1\frac{1}{4}$	4 0	" " .	$2\frac{1}{4}$	32 0	" " .	1	1 0
" " .	$1\frac{1}{2}$	5 0	" " .	$2\frac{1}{2}$	40 0	" " .	$1\frac{1}{4}$	1 4
" " .	2	7 0	" " .	3	50 0	Glaziers' sprigs	$\frac{1}{2}$	4
" " .	$2\frac{1}{2}$	11 0	" " .	$3\frac{1}{4}$	60 0	" "	$\frac{3}{4}$	14
" " .	3	25 0	" " .	$3\frac{3}{4}$	80 0	Horseshoe .	2	5 8
Clasp, fine .	1	1 12	Clout, tinned .	$1\frac{1}{2}$	4 0	" . .	$2\frac{1}{4}$	7 0
" " .	$1\frac{1}{4}$	2 8	" " .	$1\frac{3}{4}$	7 0	" . .	$2\frac{1}{2}$	8 8
" " .	$1\frac{1}{2}$	4 0	Tacks, Flemish	$\frac{1}{4}$	4	" . .	$2\frac{3}{4}$	10 0
" " .	2	6 0	" " .	$\frac{3}{8}$	6	" . .	$2\frac{1}{4}$	11 0
" " .	$2\frac{1}{2}$	10 0	" " .	$\frac{1}{2}$	8	" . .	$2\frac{3}{4}$	13 8
" " .	$3\frac{1}{4}$	18 0	" " .	$\frac{9}{16}$	14	Cart-tyre .	$3\frac{1}{2}$	187 8
Clasp, strong .	$1\frac{3}{4}$	7 0	Tacks, clout .	$1\frac{1}{8}$	3 8	" " .	4	218 12
" " .	$2\frac{1}{4}$	10 0	" " .	$\frac{15}{16}$	2 12	Mop . .	$5\frac{1}{2}$	58 0

WEIGHT OF WROUGHT COPPER NAILS PER 1,000.

Name.	Length.	Weight per 1000.	Name.	Length.	Weight per 1000.	Name.	Length.	Weight per 1000.
	Ins.	Lbs. Ozs.		Ins.	Lbs. Ozs.		Ins.	Lbs. Ozs.
Diehead .	5	80 6	Rose, strong	1	3 4	Rose, fine .	$2\frac{3}{4}$	18 11
" .	5	116 4	Rose, fine .	$\frac{7}{8}$	1 12	" " .	3	30 1
" .	4	80 1	" " .	1	2 5	" " .	4	50 6
" .	3	40 0	" " .	$1\frac{1}{4}$	4 7	Jagged .	$1\frac{3}{4}$	8 4
" .	$2\frac{1}{2}$	29 4	" " .	$1\frac{1}{2}$	5 5	Brads .	2	6 5
" .	$2\frac{1}{4}$	11 12	" " .	$1\frac{3}{4}$	6 3	" .	$1\frac{1}{2}$	3 0
Rose, strong	2	11 4	" " .	2	10 2	Tacks .	$1\frac{3}{4}$	1 15
" "	$1\frac{1}{2}$	5 9	" " .	$2\frac{1}{4}$	10 3	" . .	$\frac{1}{2}$	13
" "	$1\frac{1}{4}$	5 3	" " .	$2\frac{1}{2}$	14 6	" . .	$\frac{3}{8}$	10

Weight of Screws per 1,000.

Name.	Length.	Weight per 1000.	Name.	Length.	Weight per 1000.	Name.	Length.	Weight per 1000.
	Ins.	Lbs. Ozs.		Ins.	Lbs. Ozs.		Ins.	Lbs. Ozs.
Strong .	$3\frac{3}{4}$	28 0	Middling .	$2\frac{3}{4}$	16 0	Fine . .	$2\frac{3}{4}$	14 0
,, .	3	24 0	,, .	$2\frac{1}{2}$	16 0	,, . .	$2\frac{1}{2}$	14 0
,, .	$2\frac{3}{4}$	24 0	,, .	2	14 0	,, . .	$2\frac{1}{4}$	13 0
,, .	$2\frac{1}{2}$	22 0	,, .	$1\frac{3}{4}$	14 0	,, . .	2	13 0
,, .	2	22 0	,, .	$1\frac{1}{2}$	13 0	,, . .	$1\frac{3}{4}$	12 0
,, .	$1\frac{3}{4}$	20 0	,, .	$1\frac{1}{4}$	13 0	,, . .	$1\frac{1}{2}$	11 0
,, .	$1\frac{1}{2}$	18 0	,, .	1	12 0	,, . .	$1\frac{1}{4}$	10 0
,, .	1	16 0	,, .	$\frac{7}{8}$	11 0	,, . .	1	8 0
,, .	$\frac{7}{8}$	14 0	,, .	$\frac{3}{4}$	10 0	,, . .	$\frac{7}{8}$	8 0
,, .	$\frac{3}{4}$	13 0	,, .	$\frac{5}{8}$	9 0	,, . .	$\frac{3}{4}$	7 0
,, .	$\frac{5}{8}$	13 0	,, .	$\frac{1}{2}$	9 0	,, . .	$\frac{5}{8}$	6 0
Middling .	5	24 0	,, .	$\frac{3}{8}$	7 0	,, . .	$\frac{1}{2}$	6 0
,, .	4	20 0	Fine . .	$3\frac{1}{2}$	18 0	,, . .	$\frac{3}{8}$	3 0
,, .	$3\frac{1}{2}$	20 0	,, . .	$3\frac{1}{4}$	16 0	,, . .	$\frac{1}{4}$	3 0
,, .	3	16 0	,, . .	3	16 0			

Composition Nails.

Gauge	Thick.	Length.	In 1 lb.	Gauge	Thick.	Length.	In 1 lb.	Gauge	Thick.	Length.	In 1 lb.
No.	In.	Ins.	Num.	No.	In.	Ins.	Num.	No.	In.	Ins.	Num.
1	·04	$\frac{5}{8}$	290	7	·1	$1\frac{1}{8}$	184	12	·12	2	64
2	·05	$\frac{5}{8}$	260	8	·1	$1\frac{1}{4}$	168	13	·13	$2\frac{1}{4}$	59
3	·06	1	212	9	·11	$1\frac{1}{2}$	110	14	·14	$2\frac{1}{2}$	51
4	·07	$1\frac{1}{8}$	201	10	·11	$1\frac{5}{8}$	101	15	·15	$2\frac{3}{4}$	43
5	·08	$1\frac{1}{4}$	199	11	·12	$1\frac{3}{4}$	74	16	·16	3	35
6	·09	1	190								

Brazers' Copper Rivets.

Gauge	Diameter.	Length.	In 10 lbs.	Gauge	Diameter.	Length.	In 10 lbs.	Gauge	Diameter.	Length.	In 10 lbs.
No.	In.	In.	Num.	No.	In.	Ins.	Num.	No.	In.	Ins.	Num.
1	$\frac{3}{16}$	$\frac{1}{2}$	2,384	7	$\frac{3}{8}$	$\frac{15}{16}$	334	12	$\frac{11}{16}$	$1\frac{9}{16}$	53·8
2	$\frac{1}{4}$	$\frac{1}{2}$	1,018	8	$\frac{7}{16}$	1	210	13	$\frac{3}{4}$	$1\frac{1}{2}$	41·6
3	$\frac{1}{4}$	$\frac{9}{16}$	983	9	$\frac{7}{16}$	$1\frac{3}{16}$	141	14	$\frac{13}{16}$	$1\frac{13}{16}$	32·8
4	$\frac{5}{16}$	$\frac{9}{16}$	573	10	$\frac{9}{16}$	$1\frac{5}{16}$	99·5	15	$\frac{7}{8}$	$2\frac{1}{16}$	26·3
5	$\frac{5}{16}$	$\frac{5}{8}$	516	11	$\frac{5}{8}$	$1\frac{7}{16}$	71·9	16	1	$2\frac{3}{8}$	16·7
6	$\frac{3}{8}$	$\frac{7}{8}$	357								

Iron Rivets.

Gauge	Diameter.	Length.	In 10 lbs.	Gauge	Diameter.	Length.	In 10 lbs.	Gauge	Diameter.	Length.	In 10 lbs.
No.	In.	In.	Num.	No.	In.	Ins.	Num.	No.	In.	Ins.	Num.
0	$\frac{3}{16}$	$\frac{1}{2}$	3280	6	$\frac{3}{8}$	$\frac{15}{16}$	380	11	$\frac{11}{16}$	$1\frac{9}{16}$	61·3
1	$\frac{1}{4}$	$\frac{1}{2}$	1276	7	$\frac{7}{16}$	1	239	12	$\frac{3}{4}$	$1\frac{1}{2}$	47·3
2	$\frac{1}{4}$	$\frac{9}{16}$	1130	8	$\frac{7}{16}$	$1\frac{3}{16}$	160	13	$\frac{13}{16}$	$1\frac{13}{16}$	37·3
3	$\frac{5}{16}$	$\frac{9}{16}$	654	9	$\frac{9}{16}$	$1\frac{5}{16}$	112	14	$\frac{7}{8}$	$2\frac{1}{16}$	30
4	$\frac{5}{16}$	$\frac{5}{8}$	589	10	$\frac{9}{16}$	$1\frac{7}{16}$	81·7	15	1	$2\frac{3}{8}$	19
5	$\frac{3}{8}$	$\frac{7}{8}$	407								

APPROXIMATE WEIGHT OF PIPE JOINTING MATERIALS PER JOINT.

Spigot and Faucet Pipes.			Flanged Pipes.						
			India Rubber.			Bolts and Nuts.			
Inside Diam.*	Lead	Rope Yarn.	Inside Diam.*	Faced all over.	Faced on Strips.	No.	Diam.	Length.	Weight.
Ins.	Lbs.	Ozs.	Ins.	Ozs.	Ozs.		Ins.	Ins.	Lbs.
$\frac{3}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	2	$1\frac{3}{8}$	$\frac{7}{8}$	4	$\frac{1}{2}$	$2\frac{3}{8}$	$1\frac{1}{4}$
1	$\frac{3}{4}$	1	$2\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{1}{8}$	4	$\frac{9}{16}$	$2\frac{1}{2}$	$1\frac{1}{2}$
$1\frac{1}{4}$	1	1	$2\frac{1}{2}$	2	$1\frac{1}{3}$	4	$\frac{9}{16}$	$2\frac{1}{2}$	$1\frac{1}{2}$
$1\frac{1}{2}$	$1\frac{1}{4}$	1	3	2	2	4	$\frac{9}{16}$	$2\frac{5}{8}$	$1\frac{1}{2}$
$1\frac{1}{2}$	$1\frac{1}{4}$	1	3	2	2	4	$\frac{9}{16}$	$2\frac{5}{8}$	$1\frac{1}{2}$
2	$1\frac{1}{2}$	$1\frac{1}{4}$	3	2	2	4	$\frac{9}{16}$	$2\frac{5}{8}$	$1\frac{1}{2}$
2	$1\frac{1}{2}$	$1\frac{1}{4}$	3	2	2	4	$\frac{9}{16}$	$2\frac{5}{8}$	$1\frac{1}{2}$
2	$1\frac{1}{2}$	$1\frac{1}{4}$	$3\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{2}{3}$	4	$\frac{9}{16}$	$2\frac{5}{8}$	$1\frac{1}{2}$
$2\frac{1}{2}$	2	$1\frac{1}{2}$	$3\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{2}{3}$	4	$\frac{9}{16}$	$2\frac{5}{8}$	$1\frac{1}{2}$
$2\frac{1}{2}$	2	$1\frac{1}{2}$	4	$2\frac{7}{8}$	$3\frac{1}{4}$	4	$\frac{9}{16}$	$2\frac{3}{4}$	$1\frac{3}{4}$
$2\frac{1}{2}$	2	$1\frac{1}{2}$	4	$2\frac{7}{8}$	$3\frac{1}{4}$	4	$\frac{9}{16}$	$2\frac{3}{4}$	$1\frac{3}{4}$
3	$2\frac{1}{2}$	2	4	$2\frac{7}{8}$	$3\frac{1}{4}$	4	$\frac{9}{16}$	$2\frac{3}{4}$	$1\frac{3}{4}$
3	$2\frac{1}{2}$	2	4	$2\frac{7}{8}$	$3\frac{1}{4}$	4	$\frac{9}{16}$	$2\frac{3}{4}$	$1\frac{3}{4}$
3	$2\frac{1}{2}$	2	$4\frac{1}{2}$	$3\frac{1}{4}$	4	4	$\frac{5}{8}$	3	$2\frac{1}{4}$
$3\frac{1}{2}$	$2\frac{3}{4}$	$2\frac{1}{2}$	$4\frac{1}{2}$	$3\frac{1}{4}$	4	4	$\frac{5}{8}$	3	$2\frac{1}{4}$
4	$3\frac{3}{4}$	$2\frac{1}{2}$	5	$3\frac{1}{2}$	5	6	$\frac{5}{8}$	3	$3\frac{1}{4}$
4	$3\frac{3}{4}$	$2\frac{1}{2}$	5	$3\frac{1}{2}$	5	6	$\frac{5}{8}$	3	$3\frac{1}{4}$
$4\frac{1}{2}$	$3\frac{3}{4}$	$3\frac{1}{2}$	$5\frac{1}{2}$	$3\frac{3}{4}$	$5\frac{1}{3}$	6	$\frac{5}{8}$	3	$3\frac{1}{4}$
5	6	4	6	$4\frac{1}{4}$	6	6	$\frac{5}{8}$	3	$3\frac{1}{4}$
5	6	4	6	$4\frac{1}{4}$	6	6	$\frac{5}{8}$	3	$3\frac{1}{4}$
$5\frac{1}{2}$	7	$4\frac{1}{2}$	6	$4\frac{1}{4}$	6	6	$\frac{5}{8}$	3	$3\frac{1}{4}$
6	8	$5\frac{1}{2}$	7	$5\frac{1}{4}$	8	6	$\frac{5}{8}$	$3\frac{1}{8}$	$3\frac{1}{2}$
6	8	$5\frac{1}{2}$							
7	$8\frac{1}{2}$	$6\frac{1}{4}$							
7	$8\frac{1}{2}$	$6\frac{1}{4}$							

(*Macfarlane, Strang & Co., Ltd.*)

* For thickness of metal, see columns (A) and (B), page 150.

THICKNESSES OF PIPES.

1. Iron Water Piping.			2. Hydraulic Piping. Wrought Iron. To Stand 4,000 lbs. per Square Inch Pressure.			3. Hydraulic Piping. Cast Iron. To stand 200 lbs. per Square Inch Pressure.	
Internal Diameter of Pipe, in Inches.	External Diameter of Pipe, in Inches.	Number of Threads per Inch Whitworth Standard.	Internal Diameter of Pipe, in Inches.	External Diameter of Pipe, in Inches.	Number of Threads per Inch.	Internal Diameter of Pipe, in Inches.	Thickness of Pipe, in Inches.
$\frac{3}{8}$	·665	19	$\frac{3}{8}$	$\frac{3}{4}$	14	$1\frac{1}{2}$	$\frac{7}{16}$
$\frac{1}{2}$	·822	14	$\frac{1}{2}$	1	14	$1\frac{3}{4}$	$\frac{7}{16}$
$\frac{3}{4}$	1·034	14	$\frac{5}{8}$	$1\frac{1}{4}$	14	2	$\frac{1}{2}$
1	1·302	11	$\frac{3}{4}$	$1\frac{1}{4}$	11	$2\frac{1}{4}$	$\frac{9}{16}$
$1\frac{1}{8}$	1·492	11	$\frac{7}{8}$	$1\frac{3}{8}$	11	$2\frac{1}{2}$	$\frac{9}{16}$
$1\frac{1}{4}$	1·650	11	1	$1\frac{1}{2}$	11	3	$\frac{5}{8}$
$1\frac{3}{8}$	1·745	11	$1\frac{1}{8}$	$1\frac{5}{8}$	11	$3\frac{1}{4}$	$\frac{5}{8}$
$1\frac{1}{2}$	1·882	11	$1\frac{1}{4}$	$1\frac{3}{4}$	11	4	$\frac{3}{4}$
$1\frac{5}{8}$	2·021	11	$1\frac{3}{8}$	$1\frac{7}{8}$	11	5	$\frac{7}{8}$
$1\frac{3}{4}$	2·047	11	$1\frac{1}{2}$	2	11	6	1
$1\frac{7}{8}$	2·245	11	$1\frac{5}{8}$	$2\frac{1}{8}$	11		
2	2·347	11					
$2\frac{1}{8}$	2·467	11					
$2\frac{1}{4}$	2·587	11					
$2\frac{3}{8}$	2·794	11					
$2\frac{1}{2}$	3·001	11					

British Standard Specification for Cast-Iron Pipes and Special Castings for Water, Gas, and Sewage.*

(No. 78—1917.)

(*Abstract.*)

2. The metal used for casting the pipes shall be pig iron alone, or pig iron and scrap, and shall be remelted in the cupola or air furnace. The pig iron shall be best tough grey foundry pig iron, and the scrap shall be clean, unburnt, and of good quality. There shall be no admixture of cinder iron or any material calculated to render the metal inferior in quality, and the resulting casting shall not be white or vitreous on fracture.

7. The tensile breaking strength of the metal shall be not less than 9½ tons per square inch.

11. The straight pipes shall be cast vertically, in dry sand moulds formed from turned iron patterns and in accurately faced and truly jointed boxes, and without the use of core nails, chaplets, or thickness pieces, or any substitute therefor. They shall be cast with a sufficient head of metal to ensure soundness, which head shall be afterwards cut off in a lathe and the finished pipe left the length and shape required. All straight spigot and socket pipes shall be cast socket downwards.

22. The straight pipes shall, before being coated, respectively withstand the following hydraulic test pressures without showing any leakage, sweating, or other defect of any kind :—

	Gas.	Water.		
	Class A.	Class B.	Class C.	Class D.
Test pressure in feet head . .	200	400	600	800

While under the test pressure each pipe shall be smartly struck with a suitable hand hammer weighing not less than one and a half pounds.

British Standard Cast-Iron Pipes for Water, Gas, and Sewage.

Standard Thicknesses and External Diameters for Spigot and Socket and Flange Straight Pipes.

Nominal Internal Diameter of Pipe (ins.)	Gas. Class A. Test Pressure 200 ft. hd.		Water and Sewage. Class B. Test Pressure 400 ft. hd.		Class C. Test Pressure 600 ft. hd.		Class D. Test Pressure 800 ft. hd.		Nominal Internal Diameter of Pipe (ins.)
	Thick-ness. (ins.)	External Diameter. (ins.)	Thick-ness. (ins.)	External Diameter. (ins.)	Thick-ness. (ins.)	External Diameter. (ins.)	Thick ness. (ins.)	External Diameter. (ins.)	
3	·38	3·76	·38	3·76	·38	3·76	·40	3·76	3
4	·39	4·80	·39	4·80	·40	4·80	·46	4·80	4
5	·41	5·90	·41	5·90	·45	5·90	·52	5·90	5
6	·43	6·98	·43	6·98	·49	6·98	·57	6·98	6
7	·45	8·06	·45	8·06	·53	8·06	·61	8·06	7
8	·47	9·14	·47	9·14	·57	9·14	·65	9·14	8
9	·49	10·20	·49	10·20	·60	10·20	·69	10·20	9
10	·52	11·26	·52	11·26	·63	11·26	·73	11·26	10
12	·55	13·14	·57	13·14	·69	13·60	·80	13·60	12
14	·57	15·22	·61	15·22	·75	15·72	·86	15·72	14
15	·59	16·26	·63	16·26	·77	16·78	·89	16·78	15
16	·60	17·30	·65	17·30	·80	17·84	·92	17·84	16
18	·63	19·38	·69	19·38	·85	19·96	·98	19·96	18
20	·65	21·46	·73	21·46	·89	22·06	1·03	22·06	20
21	·67	22·50	·75	22·50	·92	23·12	1·06	23·12	21

* By permission of the British Engineering Standards Association.

BRITISH STANDARD CAST-IRON PIPES FOR WATER, GAS AND SEWAGE—(*contd.*).

Nominal Internal Diameter of Pipe (ins.)	Gas. Class A. Test Pressure 200 ft. hd.		Water and Sewage. Class B. Test Pressure 400 ft. hd.		Class C. Test Pressure 600 ft. hd.		Class D. Test Pressure 800 ft. hd.		Nominal Internal Diameter of Pipe (ins.)
	Thickness. (ins.)	External Diameter. (ins.)	Thickness. (ins.)	External Diameter. (ins.)	Thickness. (ins.)	External Diameter. (ins.)	Thickness. (ins.)	External Diameter. (ins.)	
22	·68	23·54	·77	23·54	·94	24·16	1·08	24·16	22
24	·71	25·60	·80	25·60	·98	26·26	1·13	26·26	24
26	·74	27·66	·83	27·66	1·02	28·36	1·18	28·36	26
27	·75	28·70	·85	28·70	1·04	29·40	1·20	29·40	27
28	·76	29·72	·86	29·72	1·06	30·44	1·22	30·44	28
30	·79	31·78	·89	31·78	1·09	32·52	1·26	32·52	30
32	·82	33·84	·92	33·84	1·13	34·62	1·31	34·62	32
33	·83	34·88	·94	34·88	1·15	35·66	1·33	35·66	33
36	·87	37·96	·98	37·96	1·20	38·76	1·38	38·76	36
38	·90	40·02	1·01	40·02	1·23	40·84	1·42	40·84	38
40	·92	42·06	1·03	42·06	1·26	42·92	1·46	42·92	40
42	·95	44·12	1·06	44·12	1·30	45·00	1·50	45·00	42
44	·98	46·16	1·08	46·16	1·33	47·06	1·53	47·06	44
46	1·00	48·22	1·11	48·22	1·36	49·14	1·57	49·14	46
48	1·03	50·26	1·13	50·26	1·38	51·20	1·60	51·20	48

NOTE.—It is suggested that the working pressures should not exceed one-half of the respective test pressures.

Weight of Steel Wire Ropes.

The weight per fathom (6ft.) in lbs. of the usual engineering wire rope is approximately the square of the circumference, *i.e.*, for example, a 3-in. rope would weigh 9 lbs. per fathom, approximately.

Sorting out Mixed Steel Bars.

In the majority of large engineering works who carry a fairly large stock of different kinds of steel bar, it frequently happens that the material inadvertently becomes mixed up so that no one can vouch for any particular bar being of a certain composition. This mixing up invariably leads to trouble, and so it becomes imperative that the material be correctly sorted out into its correct categories again.

Classes of steel may be broadly divided into two distinct classes, which are ' straight carbon steels ' and ' alloy steels.' These two classes may be again subdivided into low, medium, and high carbon steels, and nickel, nickel-chrome, and high-speed alloy steels.

To ascertain if the steel is of the ' straight carbon ' variety or of the ' alloy steel ' variety, dissolve a few drillings or chippings from the bar in dilute nitric acid ; then all straight carbon steels will produce a brownish-coloured solution—the brown colour varying in intensity with the amount of carbon in the steel—whilst nickel steels will produce a light green-coloured solution, and nickel-chrome steels a dark green solution of a characteristic hue.

To determine what class of carbon steel any particular bar consists of, supposing it has been found to be a straight carbon steel, grind a piece of the steel on any ordinary grinding wheel and carefully note the type of spark produced. The higher the carbon content of the steel the more brilliant and lively are the sparks produced and, therefore, file steel produces brilliant, lively, and forked sparks, whilst case-hardening steel produces dull and dead-looking sparks. With repeated practice it is easily possible to discriminate between low, medium, and high carbon steels by this method. This test is one of the best for high-speed steel, this steel producing very characteristic sparks of a dull reddish colour.

To ascertain the carbon content of a steel very accurately, dissolve 0·20 gram of the steel in 2 c.c. of dilute nitric acid, and at the same time dissolve (in a separate vessel) the same quantity

of steel of known carbon content. Place the two solutions in graduated tubes and dilute the unknown sample with water until it is exactly the same colour as the standard steel which has previously been diluted to any convenient degree. From this reading of the graduated tube the carbon content of the unknown steel may be immediately found. For example, supposing the standard steel contains 0·49 per cent. carbon: this may be diluted to 9·8 c.c., when, if the unknown sample of steel requires being diluted to 6·8 c.c. before it is the same colour as the standard, then it will contain 0·34 per cent. carbon. The method is accurate to about 0·02 per cent., and twenty or more tests may be carried out in about half an hour.

Should the steel prove to be an alloy steel the colour test will ascertain whether it is a nickel or nickel-chrome alloy steel, and to discover if it is an oil or air-hardening nickel-chrome steel, take a small piece of the steel and heat it to 1,550° F., and allow it to cool freely in air, then test the hardness of the steel by means of a Brinell or scleroscope hardness tester.

(*Met.*, '*Machinery*,' *July* 5, 1923.)

Identification of Nickel-Steel Bars.

Place one drop of nitric acid having a specific gravity of 1·20 on a spot on the bar from which the scale has been removed, and allow it to work for ten seconds. Press a piece of white blotting paper lightly on the spot to soak up the nitrates of iron and nickel (if present). Put two drops of dimethyl-glyoxime solution on the nitrates on the blotting paper.

If the colour appears red, the steel contains nickel, but if it is brown then the steel contains no nickel. This method is sensitive to 0·17 per cent. nickel. The time required to make the test is about three minutes. (*Machinery*.)

Identification of Steel Pipes.

Steel pipe may be distinguished from wrought-iron pipe by testing the material in the pipe for manganese, which is present in the steel pipe, but not in the wrought iron.

The test is as follows:—Place a clean, bright chip or filing of the metal to be tested, about the size of a pin-head, in a porcelain crucible; add six drops of pure nitric acid, and heat; add two drops of silver nitrate solution, then one crystal of ammonium persulphate not greater than $\frac{1}{8}$ in. in diameter; warm the solution, but do not let it boil. If the metal is steel, a pink colour will begin to develop, and at this point the crucible should be removed from the source of heat, when a very decided red coloration will result. If no colour develops, but a small amount of dark residue remains in the dish, the metal is wrought iron.

SECTION V

PART II

TENSILE STRENGTH OF MATERIALS — STRENGTH OF WIRE, CHAINS, AND ROPES—LATERAL STRENGTH OF MATERIALS—TRANSVERSE STRENGTH OF MATERIALS —BRITISH STANDARD SPECIFICATIONS FOR STRUCTURAL STEEL—ADMIRALTY TESTS FOR STEEL AND IRON—STRENGTH OF TUBES, RINGS, CHAINS, SLINGS, AND ROPES—FATIGUE OF METALS.

STRENGTH OF MATERIALS.

This Table shows the weight a column can bear with *safety* when the weight presses through the length of the column. The tabular number is the weight in pounds or tons per square inch on the transverse section of a column of a length less than twelve times its smallest thickness.

Resistance to Compression.

FIG. 1.

Kind of Material.	Lbs.	Tons.
Oak of good quality	432	·1885
„ common	280	·125
Spruce, red (Sapin rouge)	540	·241
„ white (Sapin blanc)	140	·6256
Iron, wrought	14,400	6·43
„ cast	28,750	12·85
Basalt	2,875	1·285
Granite, hard	1,000	·446
„ common	575	·256
Marble, hard	1,435	·640
„ common	431	·192
Sandstone, hard	1,295	·577
„ loose	5·6	·0025
Brick, good quality	175	·078
„ common	58	·0259
Limestone of hardest kind	720	·321
„ common	432	·193
Plaster of Paris	86	·0384
Mortar, good quality, and 18 months old	58	·0259
„ common	36	·016

When the length or height of the column is more than twelve times its smallest thickness, divide the tabular weight by the corresponding number in the following Table :

Length ÷ thickness	12	18	24	30	36	42	48	54	60
Divide by	1·2	1·6	2	2·8	4	5	6	8	12

TENSILE STRENGTH OF MATERIALS.

Weight or Power required to tear Asunder One Square Inch.

TENSILE STRENGTH OF WOODS.

Woods.	Lbs.	Woods.	Lbs.	Woods.	Lbs.
Ash	14000	Lignum-vitæ	11800	Pear	9800
Bay	14000	Locust	20500	Pine, pitch	12000
Beech	11500	Mahogany	21000	„ larch	9500
Box	20000	„ Spanish	12000	„ American, white	11800
Cedar	11400	„ „	8000	Poplar	7000
Chestnut, sweet	10500	Maple	10500	Spruce, white	10290
Cypress	6000	Oak, American, white	11500	Sycamore	13000
Deal, Christiania	12400	„ English	10000	Teak	14000
Elm	13400	„ seasoned	13600	Walnut	7800
Lance	23000	„ African	14500	Willow	13000

TENSILE STRENGTH OF RUBBER.

Plantation rubber of good average quality should have a tensile strength of from 2,300 lbs. to 2,400 lbs. per square inch.

TENSILE STRENGTH OF METALS.

Metals.	Lbs.	Metals.	Lbs.	Metals.	Lbs.
Copper, wrought	34000	Iron, English bar	56000	Lead, milled	3320
„ rolled	36000	„ rivets, American	53300	„ wire	2580
„ cast, American	24250	„ „ English	65000	Platinum wire	53000
„ wire	61200	„ bolts	52250	Silver, cast	40000
„ bolt	36800	„ hammered	53913	Steel, cast, maximum	142000
Iron, cast, Low Moor		„ „ mean of		„ „ mean	88657
No. 2	14076	English	53900	„ blistered, soft	133000
„ „ Clyde No. 1	16125	„ crank-shaft	44750		104000
„ „ „ No. 3	23468	„ turnings	55800	„ shear	124000
„ „ Calder No. 1	13735	„ plates, boiler, American	48000	„ chrome, mean	170980
„ „ Stirling, mean	25764		62000	„ puddled, extreme	173817
„ „ mean of American	31829	„ plates, mean of English	51000	„ plates, lengthwise	96300
„ „ mean of English	19484	„ plates, lengthwise	53800	„ crosswise	93700
„ wrought wire	103000	„ „ crosswise	48800	„ razor	150000
„ best Swedish bar	72000	„ inferior, bar	30000	Tin, cast, block	5000
„ Russian bar	59500	„ wire, American	73600	„ Banca	2122
		„ scrap	53400	Zinc	3500
		Lead, cast	1800	„ sheet	16000

Lake Superior and Iron Mountain charcoal bloom iron has resisted 90,000 lbs. per square inch.

TENSILE STRENGTH OF ALLOYS.*

Alloys.	Lbs.	Alloys.	Lbs.
Gold 5, copper 1	50000	Copper 10, tin 1	32000
Brass	42000	„ 8, „ 1 (gun-metal)	30000
„ yellow	18000	„ 8, „ 1 (gun-metal, small bars)	50000
Bronze, least	17698	Tin 10, antimony 1	11000
„ greatest	56788	Delta bronze, No. I alloy	111600

* See also METALLURGY, Section XL.

THE METALLURGIST.

Supplement issued free with the last issue in each month of The Engineer.

The only paper wholly and solely devoted to scientific Metallurgy, it is indispensable to the works laboratory, and to all who desire to keep pace with the rapid advances in the knowledge of ferrous and non-ferrous metals and alloys. Its critical reviews of British and foreign researches and investigations, written by eminent metallurgists, form an invaluable feature.

A sample copy will be sent to any genuine metallurgist at home or abroad.

Copies of the issues of The Engineer containing The Metallurgist may be ordered from Newsagents, or from

The Manager,

THE ENGINEER

28 Essex Street, Strand, London, W.C.2.

Price, including Postage, **15/- per annum.**

"DELTA" BRAND

(Registered Trade Mark)

Strongest Malleable Corrosion-Resisting Bronzes

High Conductivity and Firebox Copper

Naval Brass, Yellow Metal, Screwing Brass

Zinc, Aluminium, Anti-Friction Metals

Castings, Forgings, Stampings, Pressings, Sheets
Tubes, Wire, Rods of any Section

THE FASTEST TURNING AND
SCREWING RODS IN
THE WORLD

Original and Largest Manufacturers of
EXTRUDED RODS and SECTIONS

Registered Trade Marks
DELTA · DELTOID · DIXTRUDO · DIXTAMPO

The Delta Metal Company, Limited

Dartmouth Street
BIRMINGHAM

East Greenwich
LONDON, S.E.10

TENSILE TEST OF DELTA BRONZE No. I, EXTRUDED.

Made by Mr. R. H. Harry Stanger, Broadway Testing Works, Westminster.

Description.	Diameter. Inches.	Area. Square Inches.	Ultimate Strength. Tons per Sq In.	Elongation per Cent. on 2 inches.	Reduction of Area. Per Cent.
Extruded bar. Delta bronze No. 1	0·502	0·1979	49·82	26·0	24·0

TENSILE TESTS OF DELTA BRONZE No. I, CAST.

Made by Mr. W. H. Stanger, M.I.C.E., F.C.S., Broadway Testing Works, Westminster.

Diameter. Inches.	Area. Square Inches.	Ultimate Strength. Tons per Square Inch.	Elongation per Cent. on 2 Inches.	Reduction of Area per Cent.
0·500	0·1963	40·75	18·6	23·9
0·500	0·1963	40·75	19·4	20·1
0·500	0·1963	40·80	23·0	22·2
0·500	0·1963	41·30	20·0	20·4
0·501	0·1971	41·40	—	—
0·501	0·1971	41 50	20·5	19·3
0·500	0·1963	40·50	20·5	20·8
0·491	0·1893	40·46	18·0	17·9

Tensile strength, mean of 8 tests 40·93 tons per square inch.

Tensile Strength of Metals at Different Temperatures.

TENSILE STRENGTH OF IRON, STEEL, AND CAST IRON AT HIGH AND LOW TEMPERATURES.

The tensile strength of wrought iron or steel appears to be a minimum at about 70° F., and increases as the temperature is changed, becoming 20 per cent. greater at about 500° F. There are indications of a brittle region at the temperature of steam at 5 to 10 lbs. pressure. Iron at low temperatures, while increasing in strength, retains its ductility.

The tensile strength of cast iron begins to decrease at as low a temperature as 200° F., and at a dull red heat, say about 1,000° F., its strength is reduced by one-third; at 1,240° F. it is reduced to zero.

Effect of Superheated Steam Temperatures on Steel.

Based upon the experiments made in the Brown-Firth Research Laboratories, Sheffield.

(*Contributed by W. H. Hatfield, D.Met.*)

Steam at 100 lbs. per sq. in. pressure and 350° of superheat probably represents the highest temperature at which steam is likely to be used. This corresponds to about 845° F. or 452° C. As the testing temperature increases from atmospheric to about 500° C. most types of steel show first an increase of strength, then a decrease. A mild carbon steel which at ordinary temperatures gave 27·2 tons per sq. in. maximum stress with 38 per cent. elongation gave at 200° C. 30 tons per sq. in. and 27 per cent. elongation, at 300° C. 32·3 tons per sq. in. with 23 per cent. elongation, but at 400° C. fell to 25·1 tons per sq. in. maximum stress with 37·6 per cent. elongation. At 550° C. the maximum stress was only 16·8 tons per sq. in. Carbon steel of higher carbon content shows a similar order of changes, but there is in some cases a slight decrease in strength at about 100° C. before the increase takes place. The temperatures at which

the steel has its maximum strength varies with the steel, a higher carbon generally bringing the maximum at a lower temperature. A 52-ton carbon steel gave 50 tons per sq. in. at 100° C., 53 tons per sq. in. at 200° C., 58 tons at 310° C., 48·5 tons per sq. in. at 420° C., and so on. A nickel chrome steel, which gave 37 tons per sq. in. and 30 per cent. elongation at ordinary temperatures, gave only 32·4 tons per sq. in. at 200° C.; 29 per cent. elongation followed by an increase of strength to 34·2 tons per sq. in., with 33 per cent. elongation at 410° C. Beyond this temperature the strength fell off rapidly.

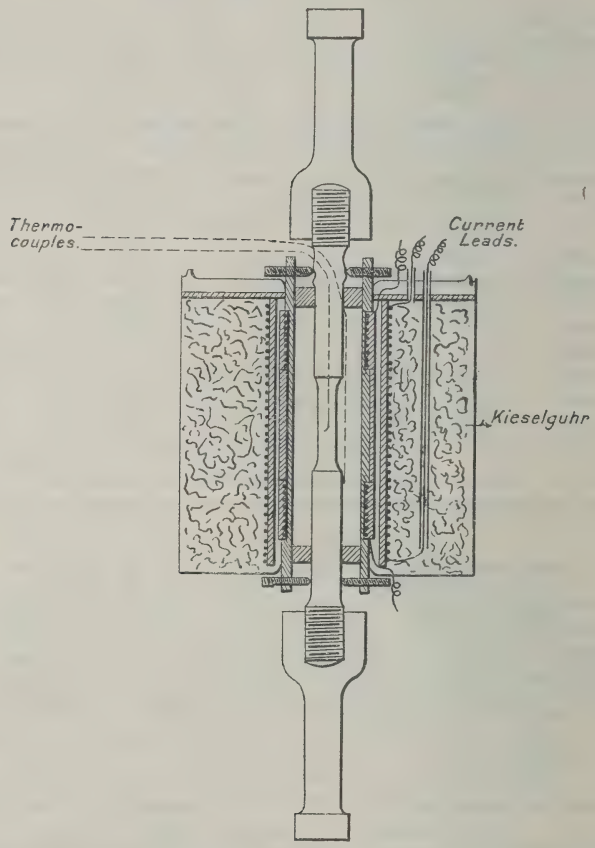


FIG. 2.

Tests in Firth's stainless steel, which had a strength of 55·0 tons per sq. in. and 21 per cent. elongation at ordinary temperature, retained a strength of 45·0 tons per sq. in. with 22 per cent elongation at 410° C., after falling slightly below this strength at 300° C. A specimen of high-speed tool steel in an annealed condition showed a gradual reduction in strength as the temperature increased. The values were 50 tons per sq. in., and 21·6 per cent. elongation at ordinary temperature, and 36·4 tons per sq. in., 19·2 per cent. elongation at 460° C. This stainless steel and high-speed steel retain their strength to a considerable degree up to 600° C.

A further test on nickel steel showed a decrease of strength from 48·6 tons per sq. in. to 39·2 tons per sq. in. at 250° C., followed by an increase to 43 tons per sq. in. at 350° C., and a further decrease to 32·7 tons per sq. in. at 500° C. The elongations per cent. at these temperatures were, ordinary 23·2 per cent. ; 250° C., 21 per cent. : 350° C., 19 per cent. ; 500° C., 24 per cent.

For use in steam turbines, the above properties require careful consideration, since, of course, the design and strength of the parts should be based on the properties of the material at the actual working temperature.

For making the above determinations, a special small electric furnace (fig. 2) was constructed and used for heating the specimens. An iron tube about 7¼ ins. long and 1½ ins. internal diameter and $\frac{3}{16}$ in. thick was turned down, except about 1½ in. length at each end, to about $\frac{3}{8}$ in. thickness. The main heating coil was a spiral of nichrome wire wrapped on a silica tube of 2¼ ins. diameter and fixed concentrically with the iron tube. Auxiliary heating coils were added at each end of the turned portion of the central tube, being separated from the latter by a layer of mica, and covered with a coating of refractory cement. A cylindrical metal box surrounded the whole, the space being filled with refractory material (kieselguhr). The object of the metal tube was to equalise the temperatures over the central portion of the furnace, and the end heating coils were added to compensate for the heat lost by radiation at the ends of the furnace. The tests were carried out in the Olsen testing machine. The test piece was 10 ins. long, having the test portion 1¼ in. parallel, by 357 mils diameter (i.e. $\frac{1}{10}$ sq. in. cross section). This was located axially in the furnace by three set screws carried at each end of the iron tube. The screwed ends of the test piece were connected to the shackles of the machine by special gluts of 25 per cent. nickel steel, which has fairly low heat-conducting properties. This was found to be necessary in order to prevent the shackles and attachments of the machine from heating up unduly. The weight of the furnace was carried by the test piece itself from its upper end, a groove being turned on the test piece so that the upper set screws could be screwed tightly on it. The lower set screws were turned so as to only bear lightly on the test piece, thus permitting the lower portion of the latter to pass through easily, when elongation and fracture took place during the test. The heating coils, in series with a suitable resistance, were connecting across the 200-volt. A.C. supply mains. The temperature was measured by means of two thermo-electric couples, which had been previously calibrated. One was placed with the junction at the centre of the parallel portion of the test piece, and the other just above the end of the 'radius' so as to detect any variation in the temperature. The couples were tied in position on the test piece by fine wire previous to inserting in the furnace. To prevent any air current passing through the furnace the ends of the tube were packed with asbestos packing.

Strength of Steel at very High Temperatures.

(W. H. Hatfield.)

It frequently happens that engine parts have to work at still higher temperatures than those corresponding to superheated steam. Valves, pistons, and other parts of internal combustion engines, boiler flues, etc., are examples of such parts.

It is generally found that as the temperature rises above 500° C. the mechanical strength falls off fairly rapidly, whatever the composition of the steel. A mild steel of ·14 per cent. carbon content showed a strength of 9 tons per sq. in. at 640° C. and a strength of only 2·5 tons per sq. in. at 850° C. Under such conditions, however, the strength exhibited depends largely on the speed of deformation. A relatively small load will frequently cause a gradual breakdown of the material if the load is maintained. The order of strengths exhibited at 850° C. is somewhat as follows :— Mild carbon steel, higher carbon steel, nickel chrome steel, Firth's stainless steel, tungsten high-speed steel. A sample of the latter gave a strength of 11 tons per sq. in. when tested at 850° C. when suitably hardened and tempered.

It might be of interest to note that a quality of high speed steel having the condition of hardness in which it is employed as a tool (and in which it retains its cutting properties while working at a red-heat) had a strength of 78 tons per sq. in. at a temperature of 625° C.

TENSILE STRENGTH OF COPPER AT DIFFERENT TEMPERATURES.

Temperature, °F . . .	122°	212°	302°	482°	545°	602°	801°	912°	1010°
Lbs. per square inch . .	33,100	32,200	30,900	27,000	25,400	22,300	18,900	14,800	11,100

Copper may suffer considerable loss in strength when exposed for a long period to variable temperatures, even if as low as 400° F. Le Chatelier found that at this temperature the ultimate tenacity of good copper thoroughly annealed is not more than 10 tons per square inch, while this figure cannot be depended on in the case of brazed pipes, as the metal is often seriously injured in the neighbourhood of the joint. Hard copper suffers more than soft annealed copper

from the effects of heat. Le Chatelier's test showed that with a hard copper having a tenacity of 31·5 tons per sq. in., at 60° F. the strength was reduced to 11·4 tons after being exposed to a temperature of 482° F. for 30 minutes. Under the same conditions the tenacity of soft copper was reduced from 15·9 to 10·4 tons per sq. in. Prof. Roberts-Austen showed that minute quantities of impurities in copper seriously affect its tenacity. The presence of one-tenth of 1 per cent. of bismuth reduced the breaking load of a sample from 8 tons per sq. in. at 64° F. to 1·7 tons at 390° F. (*Philip Dawson, M.Inst.C.E.*)

TENSILE STRENGTH OF DELTA BRONZE NO. IV. AT DIFFERENT TEMPERATURES.

Delta Bronze, No. IV. Cast, tested at increased temperatures by Professor W. C. Unwin had lost :—

At 410° F.	. . . .	5·9 per cent. of its original strength
„ 506° F.	. . . .	17·6 „ „ „
„ 590° F.	. . . .	33·0 „ „ „

that is to say, at a temperature as high as 590° F. it still retained two-thirds of its original strength.

TENSILE STRENGTH OF GUN-METAL AT DIFFERENT TEMPERATURES.

Gun-metal (9 of copper and 1 of tin) becomes perceptibly softer at 400° F., and at 550° F. its tenacity is reduced one-half ; at 1380° F. (750° C.) it almost reaches zero. This is no doubt due to a large extent to the tin element, the melting-point of which is 446° F. But it is not altogether the melting-point of the individual constituents which appear to influence the matter. Copper, for instance (see p. 173), has a high melting-point (1929° F.), but its strength is seriously affected by prolonged exposure to a temperature as low as 400° F. Superheated steam will affect gun-metal (and other alloys) by chemical action, apart from the question of temperature.

REINFORCED SHEET LEAD.

A reinforcement of lead with perforated iron sheet has been found to increase the elastic limit four- to five-fold, and the tensile strength from two- to three-fold, the variations being due to differences in gauge of wire and mesh used. Lateral extensions of 20 per cent. and elongation of 20 per cent. have been obtained. Sheets 4 ft. 6 ins. by 5 ft. 6 ins. have been made. These reinforced sheets may be bent or shaped almost at will, as the lead does not fracture when bent. The sheets may be cut and joints burned together as in ordinary sheet lead. A pipe, 8 ins. in diameter, made up of reinforced lead 0·2 in. thick, when tested for strength, resisted pressures up to 85 lbs. per sq. in.

Stress due to Temperature.

A change of temperature = 16° C. (29° F.) for cast iron, and of 6½° C. (11·7° F.) for wrought iron, will produce 1 ton stress per square inch of section. For 45° C. (81° F.) wrought iron varies $\frac{1}{1800}$ of its length ; this will produce 7 tons stress per square inch of section.

Corrosion of Metals.*

RELATIVE CORROSION OF SOFT STEEL, WROUGHT IRON, AND NICKEL STEEL, TAKING WROUGHT IRON AS A STANDARD.

Metal.	Sea Water.	Fresh Water.	Weather.	Average.
Wrought iron . . .	100	100	100	100
Soft steel	114	94	103	103
3 per cent. nickel steel .	83	80	67	77
26 per cent. nickel steel .	32	32	30	31

(*H. M. Howe.*)

COMPARATIVE TESTS WITH WROUGHT IRON, STEEL, AND DELTA BRONZE NO. IV., IMMERSED DURING A PERIOD OF 6½ MONTHS IN ACID MINE WATER.

Details.	Wrought Iron.	Steel.	DELTA BRONZE No. IV.
Weight of bar when put in . . .	1·1805	1·2125	1·2787
Weight of bar after 6½ months . . .	0·6393	0·6614	1·2633
Loss in 6½ months	46·3 per cent.	45·45 per cent.	1·2 per cent.

* See also METALLURGY, Section XL.

Strength of Brickwork.

(Royal Institute of British Architects' Experiments.)

Nature of Brickwork.	Crushing Strength. Tons per Sq. Ft.
London stocks in lime mortar (average workmanship)	12
" " (specially good workmanship)	18½
" cement mortar (average workmanship)	17
" " (specially good workmanship)	39
Gault bricks in lime mortar (average workmanship)	21
" " (specially good workmanship)	31
" cement mortar (average workmanship)	24
" " (specially good workmanship)	50½
Leicester reds in lime mortar (average workmanship)	32
" " (specially good workmanship)	45½
" cement mortar (average workmanship)	54½
" " (specially good workmanship)	84
Staffordshire blues in lime mortar (average workmanship)	74
" " (specially good workmanship)	114
" cement mortar (average workmanship)	77½
" " (specially good workmanship)	119
Fletton bricks in lime mortar (specially good workmanship)	30½
" cement mortar (specially good workmanship)	56

The following table gives the strength of these bricks when tested separately :—

Strength of Bricks.

Description.	London Stocks.	Burham Gaults.	Leicester Reds.	Staffordshire Blues.	Fletton.
	Tons per Sq. Ft.	Tons per Sq. Ft.	Tons per Sq. Ft.	Tons per Sq. Ft.	Tons per Sq. Ft.
Cracked at	76·2	102·3	248·1	471·6	—
Crushed at	84·3	182·2	382·1	701	220·8

Strength of Masonry.

Masonry will carry safely 2 to 10 tons per square foot, and carefully-built masses of cut and dressed granite may carry 4 times the higher figure.

Adhesive Strength of Mortars.

Nature of Substance.	Tensile Strength, in Pounds per Square Inch.	After Mixture.
		Months.
Common mortar to compact limestone	15	6
" " brick	33	6
Good hydraulic lime	170	12
Ordinary " "	140 to 100	12
Fat lime	40	12
Good hydraulic mortar	140	12
Ordinary " "	85	12
Good common mortar	50	12
Bad " "	20	12
Cement from chalk lime and blue clay a few days after mixture	125	
Portland cement from compact limestone and clay 30 to 50 days after mixture	1.200 to 1.500	

Holding Power of Glue.

Wood.	Pounds per Square Inch across the Grain End to End.	Pounds per Square Inch with the Grain.
Beech	2,133	1,095
Elm	1,436	1,124
Oak	1,735	568
White wood	3,149	341
Maple	1,422	896

[1] Some elaborate tests carried out in Germany proved that joints made with ordinary joiners' glue are strongest when the glue is dissolved in 150 per cent. of water and applied to wood heated to 104° Fah. The joint was dried under a pressure of 284 lbs. per square inch for twenty-four hours, and had a breaking strength of 984 lbs. per square inch. Increases in the proportion of water or the temperature reduced the strength of the joint.

Strength of Soldered Joints.

It is generally recognised that the strength of soft-soldered joints is increased if the film of solder between the surfaces of the join is kept as thin as possible. Some experiments carried out by T. B. Crow and described in *The Journal of the Society of Chemical Industry*, No. 13, vol. xliii, showed that by keeping the film as thin as possible a tensile strength 2½ times as strong as the original solder is obtainable in a joint between copper pieces, although in no case was the strength up to the yield point of the copper. It was demonstrated that the tin of the solder diffuses into the copper. With solder films 0·1 to 0·15 mm. in thickness the strongest joints, in tension, were those made at 265° C., while those made at a temperature above 280° C. were noticeably weak and brittle.

Breaking Stresses of Wire and Wire Ropes.

SIZES, WEIGHTS, LENGTHS, AND BREAKING STRESSES OF ANNEALED AND BRIGHT IRON WIRE (*Rylands'*).

Standard Wire Gauge.	Diameter.		Sectional Area.	Weight of		Length of 1 Cwt.	Breaking Stresses.		Standard Wire Gauge.
				100 Yards.	Mile.		Annealed.	Bright.	
	In.	Mm.	Sq. In.	Lbs.	Lbs.	Yards.	Lbs.	Lbs.	
7/0	·500	12·7	·1963	193·4	3,404	58	10,470	15,700	7/0
6/0	·464	11·8	·1691	166·5	2,930	67	9,017	13,525	6/0
5/0	·432	11·0	·1466	144·4	2,541	78	7,814	11,725	5/0
4/0	·400	10·2	·1257	123·8	2,179	91	6,702	10,052	4/0
3/0	·372	9·4	·1087	107·1	1,885	105	5,796	8,694	3/0
2/0	·348	8·8	·0951	93·7	1,649	120	5,072	7,608	2/0
1/0	·324	8·2	·0824	81·2	1,429	138	4,397	6,595	1/0
1	·300	7·6	·0707	69·6	1,225	161	3,770	5,655	1
2	·276	7·0	·0598	58·9	1,037	190	3,190	4,785	2
3	·252	6·4	·0499	49·1	864	228	2,660	3,990	3
4	·232	5·9	·0423	41·6	732	269	2,254	3,381	4
5	·212	5·4	·0353	34·8	612	322	1,883	2,824	5
6	·192	4·9	·0290	28·5	502	393	1,544	2,316	6
7	·176	4·5	·0243	24·0	422	467	1,298	1,946	7
8	·160	4·1	·0201	19·8	348	566	1,072	1,608	8
9	·144	3·7	·0163	16·0	282	700	869	1,303	9
10	·128	3·3	·0129	12·7	223	882	687	1,030	10
11	·116	3·0	·0106	10·4	183	1,077	564	845	11
12	·104	2·6	·0085	8·4	148	1,333	454	680	12
13	·092	2·3	·0066	6·5	114	1,723	355	532	13
14	·080	2·0	·0050	5·0	88	2,240	268	402	14
15	·072	1·8	·0041	4·0	70	2,800	218	326	15
16	·064	1·6	·0032	3·2	56	3,500	172	257	16
17	·056	1·4	·0025	2·4	42	4,667	131	197	17
18	·048	1·2	·0018	1·8	32	6,222	97	145	18
19	·040	1·0	·0013	1·2	21	9,333	67	100	19
20	·036	0·9	·0010	1·0	18	11,200	55	82	20

BREAKING STRESSES OF STEEL WIRE (*Rylands'*).

S.W.G.	Annealed.	Bright.	S.W.G.	Annealed.	Bright.	S.W.G.	Annealed.	Bright.
	Lbs.	Lbs.		Lbs.	Lbs.		Lbs.	Lbs.
7/0	13,611	20,310	3	3,458	5,187	12	590	884
6/0	11,722	17,583	4	2,930	4,395	13	461	691
5/0	10,159	15,243	5	2,447	3,672	14	349	523
4/0	8,712	13,067	6	2,007	3,011	15	284	424
3/0	7,534	11,302	7	1,668	2,530	16	223	334
2/0	6,593	9,891	8	1,393	2,091	17	170	256
1/0	5,726	8,573	9	1,130	1,694	18	128	188
1	4,901	7,351	10	893	1,339	19	87	130
2	4,127	6,221	11	734	1,099	20	72	106

BREAKING STRESSES OF GALVANISED HIGH-PURITY IRON LINE WIRE.
(*British Engineering Standards Association Specification No.* 182—1927.)*

Weight per Statute Mile.			Diameter.			Minimum Breaking Load of Wire of Standard Weight.	Minimum Number of Twists.		Weight of each piece (or coil)*.	
Standard.	Minimum.	Maximum.	Standard.	Minimum.	Maximum.				Minimum.	Maximum.
lbs.	lbs.	lbs.	inch.	inch.	inch.	lbs.			lbs.	lbs.
800	780	820	0·240	0·236	0·242	2,400	12		90	120
600	585	615	0·207	0·205	0·210	1,800	15		90	120
400	380	420	0·170	0·165	0·174	1,200	22	In 6 ins.	90	120
300	285	315	0·147	0·143	0·150	900	24		70	120
200	190	210	0·120	0·117	0·123	600	26		50	120
100	95	105	0·085	0·083	0·087	300	18	In 3 ins.	50	120
60	57	63	0·066	0·064	0·067	220	15		15	30

BREAKING STRESSES OF HARD-DRAWN COPPER WIRE.
(*British Engineering Standards Association Specification No.* 174—1927.)*

Weight per Statute Mile.			Diameter.			Minimum Breaking Load of Wire of Standard Weight.	Minimum Number of Twists.		Weight of each piece (or coil).	
Standard.	Minimum.	Maximum.	Standard.	Minimum.	Maximum.				Minimum.	Maximum.
lbs.	lbs.	lbs.	inch.	inch.	inch.	lbs.			lbs.	lbs.
800	784	816	0·2237	0·2215	0·2259	2,400	15			
600	588	612	0·1937	0·1918	0·1956	1,800	20	In		
400	392	408	0·1582	0·1566	0·1598	1,250	25	6 ins.	100	140
300	294	306	0·1370	0·1356	0·1384	945	30			
200	196	204	0·1119	0·1108	0·1130	640	20		75	140
150	147	153	0·0969	0·0959	0·0979	490	25	In 3 ins.	75	140
100	98	102	0·0791	0·0783	0·0799	330	30			
70	68·5	71·5	0·0662	0·0655	0·0669	230	35		50	140

* By permission of the British Engineering Standards Association.

TABLE OF ROUND STEEL WIRE ROPES FOR MINING, HAULING, WINDING AND SIMILAR PURPOSES.

Size Circumference.	Approximate Diameter.	Approximate Weight per 100 Feet.	Actual Breaking Stress.			
			Best Patent Steel.	Special Improved Patent Steel.	Best Plough Steel.	Special Improved Plough Steel.
Ins.	Ins.	Lbs.	Tons.	Tons.	Tons.	Tons.
1½	13/32	27	4·5	4·9	5·4	6·0
1¾	½	40	6·5	7·1	7·9	8·6
1¾	9/16	54	8·7	9·8	10·8	11·9
2	5/8	72	11·5	12·8	14·2	15·6
2¼	11/16	87	14·2	16·9	17·6	19·3
2½	13/16	107	17·7	19·8	21·8	23·9
2¾	7/8	132	21·9	24·5	27·1	29·6
3	15/16	154	25·6	28·7	31·7	34·7
3¼	1 1/32	184	30·2	33·7	37·3	48·8
3½	1 1/16	217	35·1	39·2	43·3	47·5
3¾	1 3/16	247	40·4	45·1	49·9	54·6
4	1¼	275	45·4	50·7	56·0	61·4
4¼	1 5/16	308	51·4	57·5	63·5	69·5
4½	1 7/16	350	57·8	64·5	71·3	78·1
4¾	1½	392	64·4	72·0	79·6	87·2
5	1⅝	420	70·7	79·0	87·4	97·7

BRITISH STANDARD SPECIFICATION FOR GALVANISED STEEL WIRE STRAND FOR SIGNALLING PURPOSES.*

(No. 163—1924.) (*Abstract.*)

7. British Standard Galvanised Steel Wire Strand for Signalling Purposes shall consist of 7 wires of No. 17 Standard Wire Gauge (S.W.G.) stranded together.

8. The wire shall be of a homogeneous steel of one of the four grades specified in the table (col. 1), and shall be galvanised with virgin spelter of good ordinary brands, containing not less than 97·75 per cent. pure zinc.

9. The wire shall be drawn in continuous pieces and shall be, after galvanising, 56 mils (1·42 mm.) in diameter, with a tolerance above or below of 3 mils (0·076 mm.).

11. The lay shall be right-handed and 2¾ inches in length.

The wires shall be so stranded together that when an evenly distributed pull is applied at the ends of the completed strand, each wire will take an equal share of the pull.

12. Each piece shall be warranted to contain no weld, splice or joint whatever, either in the rod before it is drawn or in the finished wire.

18. (*a*) The quality of the galvanising will be tested by immersing samples of the wire before stranding in a freshly made neutral solution of sulphate of copper saturated at 60° F. (15·6° C.) and allowing them to remain in the solution for one minute, when they will be withdrawn and wiped clean. The galvanising shall admit of this process being performed twice with each sample without there being any sign of a reddish deposit of metallic copper on the wire.

(*b*) The galvanised wire before stranding shall also stand being coiled round a bar having a diameter ten times the diameter of the wire without the galvanised surface cracking.

* By permission of the British Engineering Standards Association.

TABLE OF WIRE STRAND FOR SIGNALLING PURPOSES.

1	2	3	4	5	6	7	8
Designation.	Single 17 S.W.G. Wire.						
	Tensile Strength.		Percentage Elongation in 10 in.		Minimum Breaking Load.	Minimum Number of Twists in 5·6 in. (142 mm.)	7/17 Wire Strand. Minimum Breaking Load.
	Tons per sq. in.	lb. per sq. in.	Before Stranding.	After Stranding.	lb.		lb.
W.S. 1	25–30	56,000– 67,200	10	8	138	30	966
W.S. 2	40–50	89,600–112,000	7	5	220	25	1540
W.S. 3	60–70	134,400–156,800	5	4	330	25	2310
W.S. 4	80–90	179,200–201,600	5	4	440	25	3080

Col. 7. 5·6 in. (142 mm.) is equal to one hundred times the diameter of No. 17 S.W.G. wire.
Col. 8. A tolerance of 5 per cent. below the figures shown in the column will be allowed.

Resistance to Torsional Rupture.

RELATIVE POWER OF RESISTANCE TO TORSIONAL RUPTURE BY SHOCK OF VARIOUS WOODS.

White pine . .	1·00	Ash . . .	2·25	Locust . . .	5·80
Spruce . . .	1·50	Chestnut . .	2·40	Oak . . .	6·60
Red cedar . .	1·61	Yellow pine . .	3·87	Hickory . .	6·90
Spanish mahogany .	1·65	Black walnut . .	3·95		

In a report issued by the Forest Service of the United States Department of Agriculture, it is stated that a wooden beam can take twice as much deflection under impact as under ordinary rates of loading without showing loss of elasticity. Increased rate of loading is accompanied by an increase in strength.

Stiffness of Woods.

(Ratio of twisting moment to total angle of torsion.)

White pine . .	1·00	Ash . . .	1·87	Oak . . .	2·53
Yellow pine, sap .	2·25	Black walnut . .	2·63	Hickory . .	4·1
„ „ heart .	2·25	Red cedar . .	4·00	Locust . . .	5·50
Spruce . . .	0·67	Spanish mahogany .	3·00	Chestnut . .	1·60

Resistance of Wood to Reversals of Stress.

The limiting resistance of spruce (used for aeroplane wing spars) to alternate tension and compression was proved by experiment to be above 1,600 lbs. per sq. inch, and below 1,970 lbs. per sq. inch. For spruce of this quality the safe range of stress under reversals of bending would be approximately ± 1,800 lbs. per sq. inch, or one-quarter of the ultimate stress, which was found to be 6,800 lbs. per sq. inch. In this respect spruce appears to be at a slight disadvantage compared with mild steel, in which the safe limit of stress under similar circumstances is generally about one-third of the ultimate stress. (*The National Physical Laboratory.*)

Lateral Strength of Materials.

l = length of beam, in feet.
h = height, b = breadth or thickness in inches, of the beam where the load is acting.
k = coefficient for the different materials and sections of beams.*
x = coefficient of elasticity of materials (table, page 182) = modulus of elasticity† ÷ 7,000.
f = elastic deflection, in inches.
W = weight, in lbs., which the beam can bear with safety, being about one-quarter of the ultimate load at which the beam would break.

* The value of k varies according to the shape of the beam section (see pages 181–182). For rectangular beams the values are those given in the table, page 182. † See page 184.

EXAMPLE 1 (fig. 3).—A rectangular beam of English Oak fastened in a wall projects out $l = 6$ feet 4 inches; $h = 8$ inches, and $b = 5$ inches. Required, what weight, W, it can bear with safety on the end. For English oak, coefficient $k = 25$.

$$W = \frac{25 \times 5 \times 8^2}{6{\cdot}333} = 1{,}263 \text{ lbs.}$$

EXAMPLE 2 (fig. 4).—A cast-iron beam of section fig. 14, with thickness $b = 1{\cdot}25$ inches height $h = 22{\cdot}5$ inches, supported at the two ends in a length $l = 25$ feet. Required, what weight, W, it can bear with safety in the middle. For cast iron, coefficient $k = 266$.

$$W = \frac{4 \times 266 \times 1{\cdot}25 \times 22{\cdot}5^2}{25} = 26{,}930 \text{ lbs.} = 12{\cdot}03 \text{ tons}$$

EXAMPLE 3.—Required the elastic deflection for the same beam and condition as in the last example. Modulus of elasticity $x = 2{,}285$ for cast iron.

$$f = \frac{26325 \times 25^3}{16 \times 2285 \times 1{\cdot}25 \times 22{\cdot}5^3} = {\cdot}79 \text{ inches.}$$

EXAMPLE 4 (fig. 7).—A wrought-iron girder of section fig. 20, consisting of four angle irons of $a = 3{\cdot}5 \times 0{\cdot}5 \times 2 \times 4 = 14$ square inches, the plate being $0{\cdot}5 : 1{\cdot}35 = 0{\cdot}37$ inches thick, and $h = 18$ inches deep by $l = 22$ feet. Required, how much weight, W, evenly distributed, the girder can bear with safety. For wrought iron, coefficient $k = 1040$.

$$W = \frac{8 \times 1040 \times 14 \times 18}{22} = 95{,}302 \text{ lbs.} = 42{\cdot}55 \text{ tons.}$$

If plates are riveted to the angle iron at top and bottom, add that area to a.

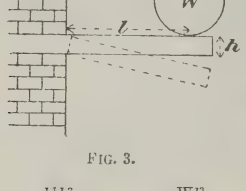


FIG. 3.

$$W = \frac{kbh^2}{l}, \qquad f = \frac{Wl^3}{xbh^3}.$$

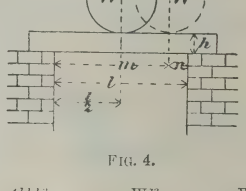


FIG. 4.

$$W = \frac{4kbh^2}{l}, \qquad W' = \frac{Wl^2}{4mn}, \qquad f = \frac{Wl^3}{16xbh^3}.$$

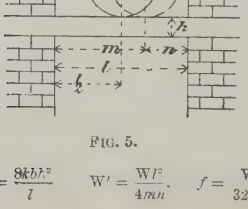


FIG. 5.

$$W = \frac{8kbh^2}{l} \qquad W' = \frac{Wl^2}{4mn}, \qquad f = \frac{Wl^3}{32xbh^3}.$$

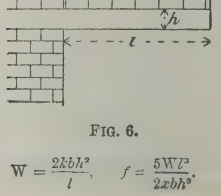


FIG. 6.

$$W = \frac{2kbh^2}{l}, \qquad f = \frac{5Wl^3}{2xbh^3}.$$

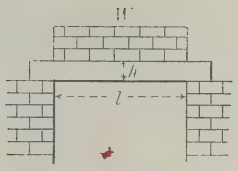


FIG. 7.

$$W = \frac{8kbh^2}{l}, \qquad f = \frac{5Wl^3}{32Ebh^3}.$$

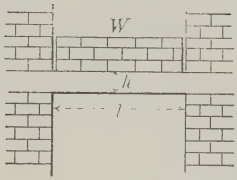


FIG. 8.

$$W = \frac{16kbh^2}{l}, \qquad f = \frac{5Wl^3}{64Ebh^3}.$$

VALUES OF COEFFICIENT k FOR BEAMS OF VARIOUS SECTIONS.
(Factor of safety = 4.)

FIG. 9.

Material.	Coefficient k.
Steel	233
Wrought iron	156
Cast iron	154
English oak	25

FIG. 10.

Material.	Coefficient k.
Steel	233
Wrought iron	156
Cast iron	154
English oak	25

$bh^2 = S^3$.

FIG. 11.

Material.	Coefficient k.
Steel	137
Wrought iron	92
Cast iron	91
English oak	15

$bh^2 = D^3$.

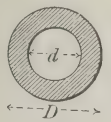


FIG. 12.

Material.	Coefficient k.
Steel	137
Wrought iron	92
Cast iron	91
English oak	15

$bh^2 = D^3 - d^3$.

FIG. 13.

Material.	Coefficient k.
Steel	1360
Wrought iron	910

$bh^2 = ah$.
h = height, a = sectional area in square inches.

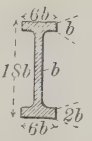


FIG. 14.

Material.	Coefficient k.
Steel	403
Wrought iron	270
Cast iron	266

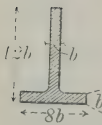


FIG. 15.

Material.	Coefficient k.
Steel	367
Wrought iron	246
Cast iron	243

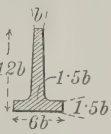


FIG. 16

Material.	Coefficient k.
Steel	388
Wrought iron	260
Cast iron	236

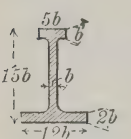


FIG 17.

Material.	Coefficient k.
Steel	1090
Wrought iron	728
Cast iron	720

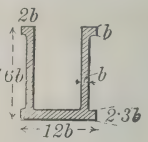


FIG. 18.

Material.	Coefficient k.
Cast iron	923

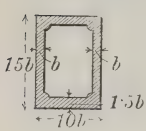


FIG. 19.

Material.	Coefficient k.
Cast iron	821

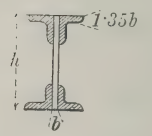


FIG. 20.

Material.	Coefficient k.
Steel	1550
Wrought iron	1040

$bh^2 = ah$.

a = area of all the four angle irons in square inches.

ABSOLUTE AND ULTIMATE STRENGTH OF MATERIALS.

Kind of Material.	Coefficient k (for Rectangular Beams).				Elasticity.
	Safety.	Short Intervals.	Pressing Circumstances.	Ultimate.	x
Wrought iron	120	162	240	488	4,110
Cast iron	150	200	300	600	2,285
Cast steel, soft	385	519	170	1,540	4,300
„ „ hardened	1,050	1,400	2,100	4,200	6,000
Blistered steel, soft	175	233	350	700	4,200
Brass	58	75	113	226	1,280
Copper	53	71	106	212	2,160
Zinc	15	20	30	61	2,360
Tin	17	23	34	69	
Lead	4	6	9	18	100
Ash	45	56	85	170	221
Hickory	67	90	135	270	
Chestnut, sweet	42	56	85	170	
Oak, white	50	66	100	200	300
„ English	25	33	50	100	248
„ Canadian	37	49	73	147	283
Pine, white	34	45	67	135	
„ yellow	38	50	75	150	268
Teak	51	68	102	205	316

The absolute safety weight is here taken as one-quarter of the ultimate breaking weight, but when the weight is acting at short intervals one-third might be relied upon, or in pressing circumstances one-half, when the materials in the beams are known to be of good quality; but the latter value should never be exceeded.

Crushing Strength of Timber.

Approximately the crushing strength (in direction of grain) of timber in lbs. per sq. in. = weight of wood in lbs. per cu. ft. × 165.

(*Percy W. Smith*, exps. made at Northampton Poly. Inst.—*The Engineer*, April 10, 1914.)

Safe Limit of Loading.

Wood, $\frac{1}{10}$: Wrought Iron, $\frac{1}{2}$ to $\frac{1}{3}$: Cast Iron, $\frac{1}{4}$ to $\frac{1}{6}$: of the load which would produce rupture.

Wrought iron will resist about two-thirds as much to crushing as to tension.

A triangular beam which has the same area and depth as a rectangular one is only half as strong as the rectangular one.

A beam of cast iron supported at both ends is six times as strong with the apex up as down; if fixed at one end and loaded at free end it is six times as strong with apex down as with it up.

The strength of a cylindrical beam to that of a square one of the same area is as

$$1 : \tfrac{2}{3}\sqrt{\pi}\text{; or } 1 : 1{\cdot}18 \text{ nearly.}$$

Transverse Strength of Materials.

TABLE SHOWING TRANSVERSE STRENGTH OF MATERIALS.

(Reduced to the uniform measure of one inch square and one foot in length; weight suspended from one end.)

Material.	Breaking Weight.	Value for General Use.
Metals.	Lbs.	
Cast iron, American { means of four divisions of grades	507	125–160
	632	155–210
	733	180–240
	772	192–250
„ mean by Major Wade	681	170–225
„ West Point Foundry, extreme	980	250–325
„ English, Low Moor, cold blast	472	110–140
„ Ponkey, cold	581	145–190
„ hot blast, mean	500	125–165
„ cold „ „	516	130–170
„ Ystalyfera, cold blast	770	195–255
„ mean of 65 kinds	500	125–165
„ mean of 15 kinds, direct from the pig, cold blast	641	160–215
„ planed bar	518	130–170
„ rough bar	531	133–175
Steel, greatest	1,918	350–450
„ puddled (permanent bend)	800	170–225
Woods.		
Ash	168	55
Beech	130	32
Birch	160	40
Chestnut	160	53
Deal, Christiania	137	45
Elm	125	30
Hickory	250	55
Locust	295	80
Maple	202	65
Norway pine	123	40
Oak, African	208	50
„ American white	230	50
„ „ live	245	55
„ Canadian	146	36
„ Dantzic	122	30
„ English	140	35
„ „ superior	188	45
Pitch pine	136	45
„ American	160	50
Riga fir	94	30
Teak	206	60
White pine	92	30
„ American	130	45
Whitewood	116	38
Wrought Iron.		
American	700	160–200
	650	
	600	
Wrought Iron (continued).	Lbs.	
English	400	100–130
„	550	135–180
Swedish *	665	165–210
Mixture of Cast and Wrought Iron, &c.		
Cast iron, Blaenavon	..	145
„ „ 10 per cent. of wrought	..	175
„ „ 30 per cent. of wrought	..	230
„ „ 50 per cent. of wrought	..	185
„ „ and 2½ per cent. of nickel, mean	..	180
„ „ Stirling, second quality	..	154
„ „ Stirling, third quality	..	125
Copper	..	55
Brass	..	58
Stones (American).		
Flagging, blue	31	10
Freestone, Conn.	13	4
„ Dorchester	10·8	3½
„ New Jersey	20·1	6½
	17·8	6
„ New York	24	8
Granite, blue, coarse	18	6
„ Quincy, Mass.	26	8½
Stones (English).		
Adelaide marble	4·5	1½
Arbroath	17	5½
Bangor slate	90	30
Bath	5·2	1¾
Caithness paving, Scotland	68	22
Cornish granite	22	7
Craigleith sandstone	10·7	3½
Darley sandstone, Victoria	1·3	4
Kentish rag	35·8	12
Limestone	11	3½
Llangollen slate	43	14
Park Spring sandstone	4·3	1·4
Portland oolite	21·2	7
Valentia paving, Ireland	68·5	23
Welsh „ „	157	55
Yorkshire, blue	26	8½
„ landing	22·5	7½
„ paving	10·4	3½

* With 340 lbs. the deflection was one inch, and the elasticity of the metal destroyed.

TRANSVERSE STRENGTH OF MATERIALS (*continued*).

Material.	Breaking Weight.	Material.	Breaking Weight.
Concretes.	Lbs.	*Bricks.*	Lbs.
Fire-brick beam, Portland cement .	3·1	Best stock	11·8
„ sand 3 parts, lime 1 part	·7	Fire-brick	14
Cements.		New brick	10·7
Blue clay and chalk	5·4	Old brick	9·1
Portland	37·5 to 10·2	Stock-brick, well burned . . .	5·8
Sheppey	5	„ inferior, burned . .	2·5

(*Haswell.*)

Increase in Strength of Woods by Seasoning.

Wood.	Increase in Strength.	Wood.	Increase in Strength.	Wood.	Increase in Strength.
Ash . .	44·7 per cent.	Elm . .	12·3 per cent.	White pine .	9 per cent.
Beech .	61·9 „	Oak . .	26·1 „		

Spike-Gripping Power of Wood.

Experiments have shown that generally twice as much force is required to extract spikes from oak as from white or yellow pine. The spikes in the experiments were driven across the grain of the wood; when driven with the grain, the holding power is reduced at least one-half The condition of the wood affects the gripping power; experiments have shown that the force required to draw spikes $\frac{9}{16}$ by $\frac{9}{10}$-in., driven 4½ ins. into seasoned oak, is about 4,280 lbs.; the same spikes driven into unseasoned oak require 6,500 lbs. to draw.

Elongation Produced by Stress. Young's Modulus of Elasticity (*e*)

If length, L, in feet, of a bar of section A square inches, increases to L + *l* by stress of P lbs., then $l = \mathrm{LP} \div \mathrm{Ae}$.

(1.)

Woods	*e.*	Woods	*e.*	Metals.	*e.*
Ash . . .	1,600,000	Teak, African .	2,300,000	Manganese bronze, cast	14,000,000
Beech . .	1,350,000	Willow . .	1,400,000	Monel metal .	23,000,000
Birch, common .	1,562,000			Nickel aluminium, forged	21,150,000
„ American	1,258,000	*Metals.*		„ (98·5%), cast	28,900,000
Box . . .	1,856,000	Aluminium (99·3%), cast	9,810,000	„ hard drawn	34,100,000
Cedar . .	488,000	Antimony, hard drawn	11,320,000	Palladium . .	16,300,000 to 18,900,000
Chestnut . .	1,140,000	Brass, cast .	13,000,000 to 15,500,000	Platinum .	13,800,000
Elm . . .	700,000	Bronze, cast		Silver .	22,800,000 to 24,900,000
Fir, Christiania .	1,856,000	Copper, electrolytic	17,400,000	Steel . .	8,530,000 to 11,370,000
„ Riga . .	1,328,000	„ cast .	11,000,000	„ semi . .	30,000,000
„ New England	2,191,000	„ hard drawn	17,600,000	Tantalum . .	13,600,000
„ spruce .	1,800,000	Delta metal .	16,600,000	Tellurium . .	27,000,000
„ larch . .	900,000	Duralumin .	10,000,000	Tin, cast . .	5,950,000
Kawrie or Cowdie	2,357,000	Gun metal. .	8,800,000	Tungsten, drawn wire	3,500,000
Mahogany, Honduras . .	1,255,000	Iron, cast . .	15,000,000	Zinc . . .	514,000
Oak, English and Russian	1,451,000	„ „ malleable	25,000,000	Glass . .	13,400,000
„ Dantzic .	1,191,000	„ electrolytic	25,000,000		8,000,000
„ American, red	2,150,000	Iron, wrought .	25,000,000		
„ African .	2,283,000	Lead, sheet .	720,000		
Pine, pitch .	1,225,000				
Pine, Canada, red	1,840,000				
„ American yellow	1,600,000				
Teak, Indian	2,400,000				

See also Bureau of Standards, Report No. 101.

(2.)

The following were obtained from the flexure of thin strips :—

Material.	*e.*	Material.	*e.*	Material.	*e.*
Celluloid (1) . .	347,500	Drawing - paper (across direction of rolling) .	581,500	Box-wood . .	174,000
,, (2) . .	547,000			Whalebone . .	503,000
Drawing paper (in direction of rolling)	394,000	Thick stiff paper .	852,000		

(*F. H. Humme.*)

Breaking Weight of Wooden Beams.

BREAKING WEIGHT (*w*) IN CWTS. (DEAD LOAD) OF A RECTANGULAR BEAM OF YELLOW PINE 1 IN. BROAD, SUPPORTED AT BOTH ENDS AND LOADED IN THE CENTRE.

Span in Feet (*s*).	Depth in Inches (*d*).											
	3	4	5	6	7	8	9	10	11	12	13	14
4	7·50	13·33	20·83	30·00	40·83	53·33	67·50	83·33	100·84	120·00	140·84	163·33
6	5·00	8·89	13·89	20·00	27·16	35·56	45·00	55·56	67·22	80·15	93·89	108·90
8	3·75	6·67	10·43	15·00	20·42	26·67	33·75	41·67	50·42	60·00	70·42	81·67
10	3·00	5·33	8·33	12·00	16·33	21·32	27·00	33·33	40·33	48·00	56·33	65·33
12	2·50	4·44	6·94	10·00	13·61	17·78	22·50	27·78	33·61	40·00	46·94	54·44
14	2·14	3·81	5·95	8·57	11·67	15·24	19·29	23·81	28·81	34·25	40·24	46·67
16	1·88	3·33	5·21	7·50	10·20	13·33	16·88	20·83	25·21	30·00	35·21	40·83
18	1·67	2·96	4·63	6·67	8·95	11·85	15·00	18·52	22·41	26·67	31·30	36·30
20	1·50	2·67	4·17	6·00	8·17	10·67	13·50	16·67	20·17	24·00	28·17	32·67
22	1·36	2·42	3·79	5·46	7·42	9·70	12·27	15·15	18·33	21·82	25·61	29·70
24	1·25	2·22	3·47	5·00	6·81	8·90	11·25	13·89	16·81	20·00	23·47	27·22
26	1·15	2·05	3·21	4·62	6·28	8·21	10·38	12·82	15·51	18·46	21·67	25·13
28	1·07	1·99	2·98	4·29	5·83	7·62	9·64	11·90	14·40	17·14	20·12	23·33
30	1·00	1·78	2·78	4·00	5·44	7·11	9·00	11·11	13·44	16·00	18·78	21·78

(Formula : $w = \frac{d^2}{s} \times \frac{20}{6}$; see (10), p. 218.)

EXAMPLE.—To find the breaking weight of a beam of Yellow Pine 18 ft. span, 6 ins. broad, and 12 ins. deep. From table, breaking weight for 1 in. broad and 12 ins. deep is 26·67 cwts. ; therefore,

Breaking weight of beam 6 ins. broad = 26·67 × 6 = 160 cwts.

(*H. R. Kempe.*)

BRITISH STANDARD SPECIFICATION FOR STRUCTURAL STEEL FOR BRIDGES AND GENERAL BUILDING CONSTRUCTION.*

(No. 15—1912.) (*Abstract.*)

A steel shall be made by the Open Hearth Process (Acid or Basic) unless either process is required or specified, and must not show on analysis more than ·06 per cent. of sulphur or phosphorus.

B steel (which is not intended for bridges, nor for plates ½ in. in thickness or over, or rivet bars) may be made either by the Open Hearth Process (Acid or Basic) unless any one of the four processes is required or specified, and must not show on analysis more than ·08 per cent. of phosphorus, and not more than ·06 per cent. of sulphur.

Plates, sections, and flat bars must show a tensile breaking strength of 28 to 33 tons per sq. in., with an elongation of not less than 20 per cent. for steel of ·375 in. thickness and upwards, and not less than 16 per cent. for steel below ·375 in. thickness.

* By permission of the British Engineering Standards Association.

Round and square bars (other than rivet bars) must show a tensile breaking strength of 28 to 33 tons per sq. in.; the elongation must be not less than 20 or 24 per cent. (according to the form standard test piece used for making the test).

Rivet Bars, 25 to 30 tons per sq. in. breaking strength, and not less than 25 or 30 per cent. elongation (according to the test piece used).

ADMIRALTY TESTS FOR STEEL AND IRON.

(1925.)

TESTS FOR STEEL.

Tests for Steel Angles, Angle Bulbs, Tees, Tee Bulbs, Zed Bars, Ship Plates, and Ship Sheets.

(*Abstract.*)

Strips cut lengthwise or crosswise to have an ultimate tensile strength of not less than 26 and not more than 30 tons per sq. in. of section, with an elongation of 20 per cent. in a length of 8 ins. For plates of 10 lbs. and sheets elongation may be not less than 18 per cent.

One bar, plate, or sheet is to be selected from every charge, provided the number of bars, plates, or sheets does not exceed 50. If above that number, one for every additional 50 or portion of 50. A sample may be selected from every section of bar, or every thickness of plate or sheet, from each charge, but bars may be received or rejected without a trial of every section, and plates or sheets without a trial of every thickness, on the invoice.

Strips cut crosswise or lengthwise 1½ in. wide must stand bending cold in a press to a curve of which the inner radius is one and a half times the thickness of the steel tested.

The steel shall stand such forge tests (hot and cold for bars and hot for plates and sheets), and other general usage tests as may be considered necessary to prove the soundness of the material and its fitness to stand such treatment and bending as it may be subjected to in the shipyard.

The strips may all be cut in a planing machine, and have the sharp edges taken off.

The ductility of every bar or plate is to be ascertained by the application of such of the above tests as may be considered necessary.

The steel to be free from lamination and injurious surface defects.

Tests for Steel Rivets.

One tensile test shall be taken from each charge, but when the weight of the bars, as rolled from one charge exceeds 5 tons, an additional test shall be made for each further 5 tons or portion thereof.

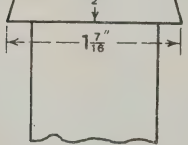


Fig. 21.—Pan Heads, Straight Necks.

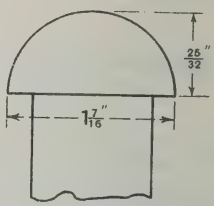


Fig. 22.—Snap Heads, Straight Necks.

The rivets are to be made from British ' acid open-hearth ' steel bars having an ultimate tensile strength, for ' boiler ' rivets, of not less than 24 and not more than 27 tons per square inch, with a minimum elongation of 25 per cent. in a length of 8 inches; for ' ordinary and screw ' rivets, the bars to have an ultimate tensile strength of not less than 26 and not more than 30 tons per square inch, with a minimum elongation of 25 per cent. in a length of 8 diameters of the test piece.

Rivets selected as considered necessary to stand the following tests:—

(*a*) Bending double when cold, and hammered till the two parts of the shank touch, in the manner shown in fig. 26, without fracture.

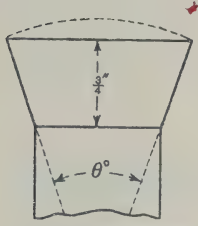


FIG. 23.—Countersunk Heads.

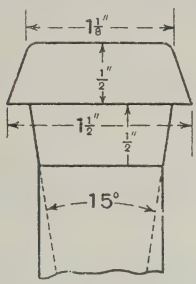


FIG. 24.—Pan Heads, Conical Necks.

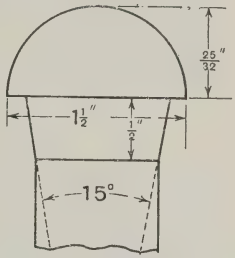


FIG. 25.—Snap Heads, Conical Necks.

FIG. 26.

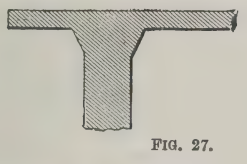


FIG. 27.

(*b*) Flattening of the rivet head, while hot, in the manner shown in fig. 27, without cracking at the edges. The head to be flattened until its diameter is 2½ times the diameter of the shank.

(*c*) The shank of the rivet to be nicked on one side, and bent cold to show the quality of the material.

SCHEDULE OF DIMENSIONS.

Diameter of Rivet.	With Pan or Snap Head and Straight Necks. Fig. 21 or fig. 22.		With Countersunk Heads. Fig. 23.		With Pan or Snap Heads and Conical Necks. Fig. 24 or fig. 25.		
	Diam. of Head.	Depth of Head.	Angle of countersink.	Depth of Head.	Diam. of Head.	Depth of Head.	Depth of Cone.
Inch.	Inches.	Inch.	Degrees.	Inch.	Inches.	Inch.	Inch.
1⅜	2	1⅛	—	—	2	1⅛	
1¼	1¾	1	40	1 1/16	1¾	1	5/8
1⅛	1⅝	29/32	40	1 1/16	1⅝	29/32	9/16
1	1 7/16	25/32	40	⅞	1½	25/32	½
7/8	1 9/32	11/16	45	9/16	1 5/16	11/16	7/16
¾	1⅛	⅝	45	⅜	1 3/16	⅝	⅜
5/8	1 1/16	½	45	¼	1	½	5/16
½	13/16	⅜	45	3/16	⅞	⅜	¼
7/16	11/16	5/16	—	—	—	—	—
⅜	⅝	¼	60	3/16	—	—	—
5/16	½	7/32	—	—	—	—	—
¼	7/16	3/16	60	3/32	—	—	—
5/16	⅜	⅛	—	—	—	—	—

Tests for Steel Bars, Flat, Segmental, Round, Square, and Hexagonal.

Test pieces are to have an ultimate strength of not less than 28 and not exceeding 32 tons per square inch of section, with an elongation of 20 per cent. in a length of 8 ins. The steel to stand such forge tests, both hot and cold, as may be sufficient to prove soundness of material and fitness for service.

Strips not less than 1 in. square or of the full thickness of section of the bar as rolled, must stand bending double in a press to a curve of which the inner radius is one and a half times the thickness of the steel tested. To be tested cold.

Instructions for Treatment of Mild Steel.

All plates or bars which can be bent *cold* are to be so treated, and if the whole length cannot be bent cold, the portion to be bent hot must be of a uniform temperature throughout; the varying temperature from hot to cold portion is to be extended so as to avoid an abrupt termination of the heat.

In cases where plates or bars have to be heated, the greatest care should be taken to prevent any work being done upon the material after it has fallen to the dangerous limit of temperature known as a 'blue heat'—say from 600° to 400° F. Should this limit be reached during working, the plates or bars should be reheated.

Where plates or bars have been heated throughout for bending, flanging, &c., and the work has been completed at one heat, subsequent normalising is unnecessary.

Where simple forge-work has been done, such as the formation of joggles, corners, and easy curves or bends, on portions of plates or bars, and the material has not been much distressed, subsequent normalising may be dispensed with.

Plates or bars which have had a large amount of work put upon them while hot, and have had to be reheated, should be subsequently normalised. It is preferred that this normalising should be done simultaneously over the whole of each plate or bar when this can be done conveniently. If it is inconvenient to perform the operation of normalising at one time for the whole of a plate or bar, portions may be normalised separately, proper care being taken to prevent an abrupt termination of the line of heat. If the severe working has been limited to a comparatively small part of a plate or bar, normalising may be limited to the parts which have been heated, the same care being taken to prevent an abrupt termination of the line of heat.

In important cases where any bar or plate shows signs of failure or fracture in working, the details should be forwarded to the Admiralty, in order that instructions may be given as to the disposal of the bar or plate. Failure need not be reported in every case.

It is not generally necessary to anneal plates or bars after punching as a means of making good damage done in punching. For mild steel plating which forms an important feature in the general structural strength, such as the inner and outer bottom plating, deck plating, deck stringers, &c., the butt straps should have the holes drilled or punched $\frac{1}{8}$-in. small and rimed. In such plating the countersunk holes should be punched about $\frac{1}{8}$ in. less in diameter than the rivets which are used, the enlargement of the holes being made in the countersinking. All countersinking to be carefully done.

All steel and iron work is to be carefully scaled, scraped, and cleaned before being painted, and each portion of it as it is turned out of hand is to have a coat of linseed oil, thin red lead, or other substance, as may be required, as soon as it is sufficiently completed to receive it, in order to prevent as far as possible the work from becoming in any degree injuriously oxidised during the building of the ship. The plates of outer and inner bottom and sides, vertical keel and longitudinals, the lower plates and bulkheads, including those above the inner bottom, all the plates of oil fuel, water ballast and fresh water compartments, and the lower plates of frames, are to be treated as follows, with a view to removing the black oxide or scale:—The plates previous to their being taken in hand for working are to stand a few hours in a liquid consisting of 19 parts of water and one of hydrochloric acid. The plates should be pickled on edge and not laid flat. When plates are removed from the dilute acid bath both the surfaces are to be well brushed and washed to remove any scale which may still adhere to them. They should then be placed in another similar bath filled and kept well supplied with fresh water, or be thoroughly washed with a hose, as may be found necessary; the plates on removal from the fresh water should be placed on edge to dry.

TESTS FOR IRON.

Tests for Bar Iron.

The whole of the samples of every description of iron when tested with the grain are to realise a tensile strength of 22 tons to the square inch.

Forge Test, Hot.

Bars are to be punched with a punch one-third the diameter of the bar, at distances of one and a half and three diameters from the end of the bar, the holes being at right angles to one

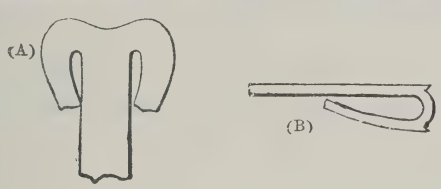


FIG. 28.

another. The holes are then to be drifted out to one and a quarter times the diameter of the bar. The sides of the holes are then to be split, and the ends must admit of turning back without fracture, as shown by (A).

Broad thick flats may be flattened on the edge to admit of being punched.

Thin flats $\frac{3}{4}$ in. thick and below are to be bent to a radius of two and a half times their thickness with the grain, without fracture.

Forge Tests, Cold.

Samples should be notched and bent, as shown by (B), to show the quality of the iron.

Fire Bars. Forge Test, Hot.

The bars are to be subjected to such bending and other usage tests as may be considered necessary.

The standard weight of the bar iron is to be taken at 40 lbs. per sq. ft. for bars of 1 in. thick, and in proportion for bars of all other thicknesses. No bar iron should exceed this prescribed weight, but a latitude of 5 per cent. below this weight will be allowed for thicknesses of $\frac{1}{2}$ an inch and upwards, and of 10 per cent. for less thicknesses. The weight of the iron ordered will be ascertained by weighing the parcel when its weight does not exceed 5 tons, and in lots of 5 tons when its weight exceeds 5 tons.

BRITISH STANDARD SPECIFICATION FOR STRUCTURAL STEEL FOR SHIPBUILDING.*

(No. 13—1910.) (*Abstract.*)

Plates.—The tensile breaking strength of steel plates, determined from Standard test pieces, shall be between the limits of 28 and 32 tons per sq. in. For plates specially intended for cold flanging and marked for identification the tensile breaking strength shall be between the limits of 26 and 30 tons per sq. in. The elongation, measured on a Standard test piece having a gauge length of 8 ins., shall be not less than 20 per cent. for material of ·375 in. in thickness and upwards, and not less than 16 per cent. for material below ·375 in. in thickness.

Angles, Bulb Angles, Channels, etc.—The tensile breaking strength of sectional material, such as angles, bulb angles, channels, etc., shall be between the limits of 28 and 33 tons per sq. in. The elongation, measured on a Standard test piece having a gauge length of 8 ins., shall be not less than 20 per cent. for material of ·375 in. in thickness and upwards, and not less than 16 per cent. for material below ·375 in. in thickness.

Rivet Bars.—The tensile breaking strength of rivet bars shall be between the limits of 25 and 30 tons per sq. in. of section, with an elongation of not less than 25 or 30 per cent. (according to the test piece used).

* By permission of the British Engineering Standards Association.

Bursting Strength of Tubes.

Let,

p = bursting pressure in lbs. per sq. in. inside tube;
D = outside diameter of tube in inches;
d = inside " " "

then,

$$p = \frac{D^2 - d^2}{D^2 + d^2} f.$$

Values of f.

Steel (60° F.) . . . 95,000	Wrought iron (60° F.). 50,000	Copper (rolled) (60° F.) . 35,000
" (800° F.)* . . . 28,800	Cast iron (60° F.) . . 20,000	" " (640° F.)* 14,700
	Brass (rolled) (60° F.) 30,000	

If

t = thickness of tube, and $n = \frac{d}{t}$,

then,

$$p = \frac{2}{n+1+\frac{1}{n+1}} f = \kappa f.$$

TABLE SHOWING VALUES OF κ FOR VARIOUS VALUES OF n.

n	·1	·2	·3	·4	·5	·6	·7	·8	·9	1
κ	·995	·984	·967	·946	·923	·899	·874	·849	·824	·800
n	1·1	1·2	1·3	1·4	1·5	1·6	1·7	1·8	1·9	2
κ	·776	·753	·731	·710	·690	·670	·651	·634	·616	·600
n	2·1	2·2	2·3	2·4	2·5	2·6	2·7	2·8	2·9	3
κ	·584	·569	·555	·541	·528	·516	·504	·492	·481	·471
n	3·1	3·2	3·3	3·4	3·5	3·6	3·7	3·8	3·9	4
κ	·461	·451	·441	·432	·424	·415	·407	·399	·392	·385
n	4·1	4·2	4·3	4·4	4·5	4·6	4·7	4·8	4·9	5
κ	·378	·371	·364	·358	·352	·346	·340	·335	·330	·324
n	5·1	5·2	5·3	5·4	5·5	5·6	5·7	5·8	5·9	6
κ	·319	·314	·310	·305	·301	·296	·292	·288	·284	·280
n	6·1	6·2	6·3	6·4	6·5	6·6	6·7	6·8	6·9	7
κ	·276	·273	·269	·265	·262	·259	·256	·252	·249	·246
n	7·1	7·2	7·3	7·4	7·5	7·6	7·7	7·8	7·9	8
κ	·243	·240	·238	·235	·232	·229	·227	·224	·222	·220
n	8·1	8·2	8·3	8·4	8·5	8·6	8·7	8·8	8·9	9
κ	·217	·215	·213	·210	·208	·206	·204	·202	·200	·198
n	9·1	9·2	9·3	9·4	9·5	9·6	9·7	9·8	9·9	10
κ	·196	·194	·192	·191	·189	·187	·185	·184	·182	·180
n	11	12	13	14	15	16	17	18	19	20
κ	·166	·153	·142	·133	·125	·117	·111	·105	·0998	·0950
n	21	22	23	24	25	26	27	28	29	30
κ	·0907	·0868	·0832	·0799	·0768	·0740	·0713	·0689	·0666	·0645
n	31	32	33	34	35	36	37	38	39	40
κ	·0625	·0606	·0588	·0571	·0555	·0540	·0526	·0513	·0500	·0488
n	41	42	43	44	45	46	47	48	49	50
κ	·0476	·0465	·0454	·0444	·0435	·0425	·0417	·0408	·0400	·0392
n	51	52	53	54	55	56	57	58	59	60
κ	·0384	·0377	·0370	·0364	·0357	·0351	·0345	·0339	·0333	·0328

* *Durston.* *(H. R. Kempe.)*

EXAMPLE (1).—What is the bursting strength of a wrought-iron tube whose bore is 2 ins. (d), and thickness ·1 inch (t)?

$$n = \frac{d}{t} = \frac{2}{\cdot 1} = 20.$$

κ corresponding to 20 is ·095; therefore

$$p = \cdot 095 \times 50{,}000 = 4{,}750 \text{ lbs per sq. in.}$$

If the value of n is not a number whose exact value can be found in the table, then multiply the excess which n has above the next lower value in the table by the difference between the value of κ corresponding to this lower value and the next higher value; then add the result to the lower value of κ, thus:

EXAMPLE (2).—What is the bursting strength of a wrought-iron tube whose bore is 2 ins. (D.), and thickness 1·2 ins. (t)?

$$n = \frac{D}{t} = \frac{2}{1 \cdot 2} = 1 \cdot 67;$$

next lower value of n in the table is 1·6, and

Excess of 1·67 over 1·6 = ·07. Value of κ corresponding to 1·6 = ·670.

Next higher value of κ = ·651;

then

·670 − ·651 = ·019; ·019 × ·07 = ·00133; ·00133 + ·651 = ·65233, the required value of κ;

therefore bursting strength p is

$$p = \cdot 652 \times 50{,}000 = 32{,}600 \text{ lbs. per sq. in.}$$

Values of p for values of $\frac{d}{t}$ exceeding 60 may be calculated very approximately by the formula

$$p = \frac{2}{n+1}, \text{ or } \frac{2t}{d+t}.$$

EXAMPLE (3).—What is the bursting strength of a steel tube 3 ins. in bore and ·045 inch thick?

$$p = \frac{2 \times \cdot 045}{3 + \cdot 045} \times 95{,}000 = \cdot 0296 \times 95{,}000 = 2{,}810 \text{ lbs. per sq. in.}$$

If t is *very* small compared with d, then

$$p = \frac{2t}{d}.$$

Strength of Steel Tubes.

1. The length of tube, between transverse joints tending to hold it to a circular form, has no practical influence upon the collapsing pressure of a commercial lap-welded steel tube so long as this length is not less than about six diameters of tube.

2. The formulæ, as based upon the present research, for the collapsing pressure of modern lap-welded Bessemer steel tubes are as follows:—

$$P = 1{,}000 \left(1 - \sqrt{1 - 1{,}600 \frac{t^2}{d^2}}\right) \quad . \quad . \text{ (A)} \qquad P = 86{,}670 \frac{t}{d} - 1{,}386 \quad . \quad . \text{ (B)}$$

where,

P = collapsing pressure, in pounds per square inch; d = outside diameter of tube in inches; t = thickness of wall in inches.

Formula (A) is for values of P less than 580 lbs., or for values of $\frac{t}{d}$ less than 0·023, while formula (B) is for values greater than these.

These formulæ, while strictly correct for tubes that are 20 ft. in length between transverse joints tending to hold them to a circular form, are at the same time substantially correct for all lengths greater than about six diameters. They have been tested for seven diameters, ranging from 3 ins. to 10 ins., in all obtainable commercial thicknesses of wall, and are known to be correct for this range.

3. For the most favourable practical conditions—viz. when the tube is subjected only to stress due to fluid pressure, and only the most trivial loss could result from its failure—a factor of safety of 3 would appear sufficient.

4. When only a moderate amount of loss could result from failure, use a factor of 4.

5. When considerable damage to property and loss of life might result from a failure of the tube, then use a factor of safety of 6.

6. When the conditions of service are such as to cause the tube to become less capable of resisting collapsing pressure, such as the thinning of wall due to corrosion, the weakening of the material due to overheating, the creating of internal stress in the wall of the tube due to unequal heating, vibration, etc., the above factors of safety should be increased in proportion to the severity of these actions. (*R. T. Stewart.*)

Strength of Lead Pipes.

Internal diameter in ins.	⅛	¼	¾	1	1¼	1½	2
Thickness in ins.	·2	·2	·22	·2	·21	·24	·21
Weight per ft.	2·3	2·6	3·8	4·1	5·3	7·1	9·2
Bursting pressure in lbs. per sq. in.	1,579	1,349	1,191	911	683	734	498

(*Kirkaldy.*)

Allowable Pressure in Thick Hollow Cylinders.

If

P = allowable pressure in lbs. per sq. in.; S_t = allowable working stress of the material in lbs. per sq. in.; K = ratio of the external diameter to the internal diameter; then

$$\frac{P}{S_t} = 0{\cdot}6\,\frac{K^2 - 1}{K^2}.$$

(*Gilbert Cook and Andrew Robertson,* '*Engineering,*' Dec. 15, 1911.)

Chart for Computing Allowable Pressure in Thick Hollow Cylinders.

The chart, fig. 29, prepared by Herman Steiger from the foregoing formula, may be used to determine the allowable pressure for a given cylinder, or to determine the dimensions of a cylinder for a given internal pressure.

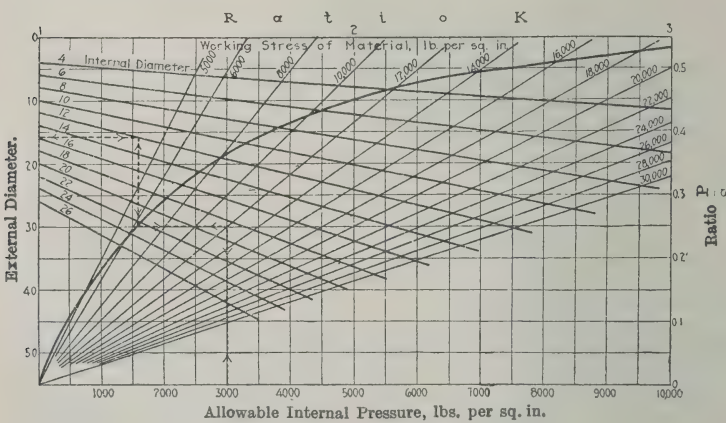


FIG. 29.

Example 1.—Suppose a steel cylinder is 16 ins. outside diameter and 12 ins. inside diameter, and the metal is good for 12,000 lbs. per sq. in. working stress. It is desired to find the allowable

internal pressure.—From the given external diameter, 16 ins., on the vertical scale move horizontally, as shown by the dotted line, to the line marked 12 ins. internal diameter. From this point move vertically to the heavy curved line and then horizontally to the line marked 12,000 lbs. working stress. Directly below this point read off the allowable pressure on the horizontal scale, 3,000 lbs. per sq. in.

Example 2.—Suppose it is desired to know the thickness of a cylinder to stand 3,000 lbs. per sq. in. pressure when the internal diameter is 12 ins. and the allowable stress is 12,000 lbs. per sq. in.—From the 3,000-lb. point on the horizontal scale move upward to the 12,000-lb. stress line and then horizontally to the curved line. From here move vertically to the 12-in. internal diameter line and read off the external diameter. 16 ins., on the left-side scale. (*' Power.'*)

Strength of Iron Chains.

*(Abstract, Safety Pamphlet No. 3 ; ' The use of Chains and other Lifting Gear.' Issued by the Home Office.)**

Ordinary short-link or close-link chains usually have links (fig. 30) with an overall length of four and a half times, and a breadth of three and a quarter times the diameter of the bar. For chains designed to stand severe shocks, links with slightly oval sides, as in fig. 31, instead of with straight sides, as in fig. 30, should be adopted ; they give the chain greater elasticity.

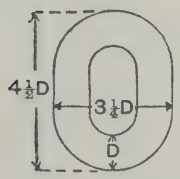


FIG. 30.—Straight-Sided Link.

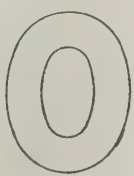


FIG. 31.—Oval Link.

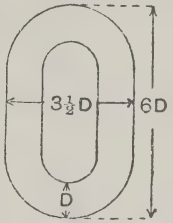


FIG. 32.—Long Link.

A cheaper but weaker form of link (fig. 32) is sometimes used for packing chains and for light foundry chains in cases where it is required to hitch a hook in any link. The links of such chains have an outside length of six times and an outside breadth of three and a half times the diameter of the bar.

The Ultimate Strength of Chain Iron.

The ultimate strength of chain iron varies from 21 to 25 tons per sq. in., but a reduction in strength results from welding and bending action in a link, so that the breaking stress in actual tests is about 15·5 tons per sq. in. for close-link crane chain. Based on this, the breaking strength of this class of chain is obtained by the formula $W = \frac{15 \cdot 5}{2} \pi d^2 = 24d^2$ tons (approximately).

This corresponds very closely with the standard adopted by the Admiralty, War Office, railway and dock companies, and with that specified in the Anchors and Chain Cables Act, 1899, for short-link chain cables, but most authorities agree that *short-link or close-link lifting chains of good quality should have a minimum strength of approximately* $27d^2$ *tons*, for chains up to 1½ in. diameter, but slightly less for chains exceeding this diameter (*see* Table A, p. 194).

Safe Working Loads for Chains.

The tensile test loads, generally known as the Admiralty Proof Loads, are approximately equal to $12d^2$ tons for short-link cable where, d is the diameter in inches of the iron of the chain.

The safe working load generally adopted for close-link welded chains is one-half the proof load, i.e. *the safe working load S.L.* $= 6d^2$ *tons*. With chain having a breaking strength of $27d^2$

* His Majesty's Stationery Office, Adastral House, Kingsway, London, W.C. 2.

tons this allows a factor of safety * of 4·5, which may be regarded as satisfactory if the chain is not subjected to shock or any abnormal use.

TABLE A.

BREAKING, PROOF, AND MAXIMUM WORKING LOADS FOR SHORT-LINK CHAINS.

Size of Chain. Diameter of Iron in ins. (1)	Minimum Breaking Load. Tons. (2)	Proof Load. Tons. (3)	Maximum Working Load. Tons. (4)	Size of Chain. Diameter of Iron in ins. (1)	Minimum Breaking Load. Tons. (2)	Proof Load. Tons. (3)	Maximum Working Load. Tons. (4)
$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{5}$	$\frac{1}{10}$	$1\frac{3}{8}$	51	$22\frac{5}{8}$	$11\frac{3}{10}$
$\frac{9}{16}$	1	$\frac{2}{5}$	$\frac{1}{5}$	$1\frac{7}{16}$	$55\frac{3}{4}$	$24\frac{3}{8}$	$12\frac{1}{8}$
$\frac{5}{8}$	$1\frac{17}{16}$	$\frac{3}{5}$	$\frac{2}{5}$	$1\frac{1}{2}$	$60\frac{3}{4}$	27	$13\frac{1}{2}$
$\frac{11}{16}$	$2\frac{5}{8}$	$1\frac{1}{5}$	$\frac{3}{5}$	$1\frac{9}{16}$	$63\frac{1}{2}$	$29\frac{1}{4}$	$14\frac{3}{5}$
$\frac{3}{4}$	$3\frac{3}{4}$	$1\frac{5}{8}$	$\frac{4}{5}$	$1\frac{5}{8}$	$68\frac{1}{2}$	$31\frac{5}{8}$	$15\frac{4}{5}$
$\frac{13}{16}$	$5\frac{1}{8}$	$2\frac{1}{4}$	$1\frac{1}{8}$	$1\frac{11}{16}$	$73\frac{3}{4}$	$34\frac{1}{16}$	17
$\frac{7}{8}$	$6\frac{3}{4}$	3	$1\frac{1}{2}$	$1\frac{3}{4}$	$79\frac{1}{2}$	$36\frac{3}{4}$	$18\frac{3}{8}$
$\frac{15}{16}$	$8\frac{1}{2}$	$3\frac{3}{4}$	$1\frac{7}{8}$	$1\frac{13}{16}$	$85\frac{3}{4}$	$39\frac{7}{16}$	$19\frac{7}{16}$
1	$10\frac{1}{2}$	$4\frac{5}{8}$	$2\frac{3}{16}$	$1\frac{7}{8}$	$91\frac{1}{2}$	$42\frac{1}{8}$	21
$1\frac{1}{16}$	$12\frac{7}{16}$	$5\frac{5}{8}$	$2\frac{1}{8}$	$1\frac{15}{16}$	$97\frac{5}{8}$	45	$22\frac{1}{2}$
$1\frac{1}{8}$	$15\frac{1}{5}$	$6\frac{3}{4}$	$3\frac{3}{8}$	2	100	48	24
$1\frac{3}{16}$	$17\frac{3}{4}$	$7\frac{9}{10}$	$3\frac{9}{10}$	$2\frac{1}{16}$	106	51	$25\frac{1}{2}$
$1\frac{1}{4}$	$20\frac{3}{5}$	$9\frac{1}{8}$	$4\frac{3}{5}$	$2\frac{1}{8}$	$112\frac{3}{4}$	54	27
$1\frac{5}{16}$	$23\frac{3}{4}$	$10\frac{1}{2}$	$5\frac{1}{4}$	$2\frac{3}{16}$	$119\frac{1}{4}$	$57\frac{1}{2}$	$28\frac{5}{8}$
1	27	12	6	$2\frac{1}{4}$	$126\frac{1}{2}$	$60\frac{3}{4}$	$30\frac{3}{8}$
$1\frac{1}{16}$	$30\frac{2}{5}$	$13\frac{1}{2}$	$6\frac{3}{4}$	$2\frac{5}{16}$	$133\frac{1}{4}$	64	32
$1\frac{1}{8}$	$34\frac{1}{16}$	$15\frac{1}{8}$	$7\frac{9}{10}$	$2\frac{3}{8}$	$140\frac{3}{4}$	$67\frac{1}{4}$	$33\frac{3}{4}$
$1\frac{3}{16}$	38	$19\frac{9}{10}$	$8\frac{2}{5}$	$2\frac{7}{16}$	$149\frac{3}{4}$	$71\frac{1}{2}$	$35\frac{5}{8}$
$1\frac{1}{4}$	$42\frac{1}{8}$	$18\frac{3}{4}$	$9\frac{3}{8}$	$2\frac{1}{2}$	156	75	$37\frac{1}{2}$
$1\frac{5}{16}$	$47\frac{7}{16}$	$20\frac{3}{8}$	$10\frac{3}{10}$				

Where there is any special risk to life or limb a maximum safe working load of $5d^2$ tons should be adopted, and where chains are subjected to shock or heavy wear, or work over pulleys, or are used for lifting vessels containing molten metal or heated liquids, the safe working load should not exceed $3 \cdot 5d^2$ tons.

The maximum safe working load for a long-link chain (fig. 32) *should not exceed two-thirds of that for a short-link chain of the same diameter.*

RINGS.

(1.)

The opening of a ring is determined, to some extent, by the size of the hook or shackle upon which it is to be placed, as this should be sufficient to allow the ring to hang freely and without jamming the end links of any attached chains.

Rings attached to sling chains are usually made with a mean diameter of four to five times the diameter of the iron forming the ring.

A ring must be sufficiently strong to carry safely a load equal to the sum of the safe loads of all the attached chains, and, moreover, must be capable of standing the total proof loads of these chains without deformation. When a proof load is applied to a chain, it causes a slight elonga-

* As a result of experiments on chains carried out at Illinois University, the National Safety Council of the United States of America has recommended that the safe working load for chains should not exceed $0 \cdot 4fd^2$ where f is the safe tensile stress of the material; for good wrought iron this may be taken as 7 tons per sq. in., hence the safe working load would be $2 \cdot 8d^2$ tons, which allows a factor of safety of over 9, i.e. about twice the factor of safety generally adopted in this country.

tion, but this is not objectionable, because in use the load on the links is always applied in the same direction, but a ring is liable to be turned round, and hence deformation must be avoided.

If the safe working load for a ring (fig. 33) is regarded as one-half the proof load; then if D = mean diameter of ring and d_1 = diameter of the iron of the ring, the *maximum safe load can be calculated* by the formula S.L. = $10\ \frac{d_1^3}{D}$ tons. This should also be equal to the maximum safe load of the attached chain, i.e. $6d^2$ where d is the diameter of iron in the chain links.

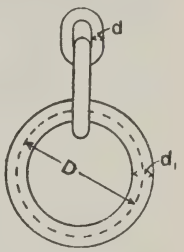


FIG. 33.—Ring.

(2.)

(*Engineering Experiment Station of the University of Illinois, U.S.A.*)

The connecting ring must be designed to resist the same safe load as the chain. The following formula is recommended :—

where $$P = \frac{0{\cdot}617Fd^3}{D}$$

P = safe load; d = diameter of stock in rung;

F = fibre stress of material used; D = main diameter of ring.

SWIVELS.

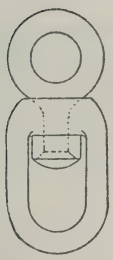


FIG. 34.—Swivel.

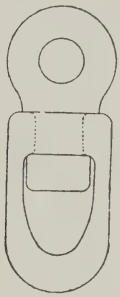


FIG. 35.—Weldless Swivel.

The Board of Trade Regulations * for testing chain cables, etc., require that the various parts of a swivel shall be of the following proportions: 'The diameter of the iron in the common link being regarded as unity, the diameter of the iron in the eye of the swivel must be at least $1\frac{1}{4}$; the diameter of the iron in the pin at the bottom of the thread must be $1\frac{3}{8}$; the diameter of the iron in the crown of the bow piece must be $1\frac{1}{2}$; the depth of the bow piece where bearing on the swivel nut must be $1\frac{1}{2}$; the diameter of the nut of the swivel pin must be $1\frac{5}{8}$. The swivel pin should be screwed to the Whitworth standard and the nut secured thereto by welding it to the end of the pin or by fitting a tapered locking pin.'

The nut or collar on the shank of a swivel must be rigidly secured so that it cannot work loose, and if the shank is riveted at the end, an ample countersink must be allowed in the collar so that the metal cannot be sheared and the shank pulled through.

---

* The Anchors and Chain Cables Act, 1899, with Regulations as to Testing, Scale of Maximum Charges, etc., issued by Board of Trade—to be obtained from H.M. Stationery Office, Adastral House, Kingsway, London, W.C. 2.

Slings

The values of T for different values of W from $\frac{1}{2}$ to 20 tons, calculated for different angles between the legs, and the size of chain necessary to lift these loads safely, without the tension exceeding a safe working load of $6d^2$ tons, is given in Table B. Table C shows the actual loads, calculated from the formula $W = 12d^2 \times \frac{AC}{AB}$ which can be safely lifted by slings of different sizes when used with two legs at various angles.

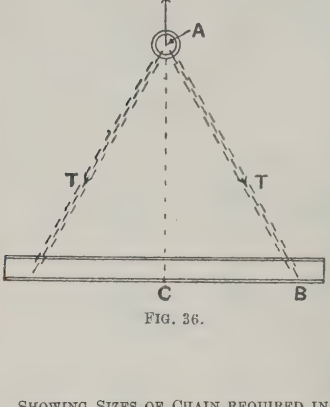
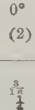
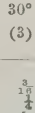
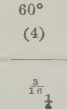
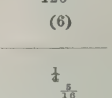
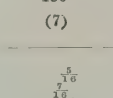


FIG. 36.

Thus it will be seen from Table B that a load of 5 tons can be safely lifted with a two-legged sling of $\frac{11}{16}$ in. chain if the angle between the legs does not exceed 30°, but when the angle is increased to 90° a $\frac{13}{16}$ in. chain is required, whilst with an angle of 150°, i.e. the sling nearly flat, a $1\frac{5}{16}$ in. chain is required if the tension in the chain is not to exceed the safe load $6d^2$ tons. Again, Table C shows that, whereas a two-legged sling made from good quality 1 in. chain can safely lift a load of 12 tons if the sling legs are vertical, the load should not exceed 10·35 tons if the legs are at an angle of 60° nor more than 6 tons with an angle of 120°.

Table B.

Showing Sizes of Chain required in Two-Legged Slings when Tension does not exceed Safe Loads.

Total Load. Tons.	Diameter of Chains in Ins. for given Angles between Legs.					
						
(1)	0° (2)	30° (3)	60° (4)	90° (5)	120° (6)	150° (7)
$\frac{1}{2}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$
1	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{7}{16}$
$1\frac{1}{2}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{1}{2}$
2	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{5}{8}$
$2\frac{1}{2}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{3}{4}$
3	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{13}{16}$
4	$\frac{5}{8}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{15}{16}$
5	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	1
6	$\frac{3}{4}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{13}{16}$	$1\frac{3}{16}$
8	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{13}{16}$	$\frac{7}{8}$	$1\frac{5}{16}$
10	$\frac{15}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	$\frac{7}{8}$	1	$1\frac{7}{16}$
12	1	$\frac{15}{16}$	1	1	$1\frac{3}{16}$	$1\frac{9}{16}$
14	$1\frac{1}{16}$	1	$1\frac{1}{16}$	$1\frac{1}{16}$	$1\frac{5}{16}$	$1\frac{13}{16}$
16	$1\frac{1}{8}$	$1\frac{1}{16}$	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{7}{16}$	2
18	$1\frac{3}{16}$	$1\frac{1}{8}$	$1\frac{3}{16}$	$1\frac{3}{16}$	$1\frac{9}{16}$	$2\frac{1}{16}$
20	$1\frac{1}{4}$	$1\frac{3}{16}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{11}{16}$	$2\frac{1}{8}$
	$1\frac{5}{16}$	$1\frac{1}{4}$	$1\frac{5}{16}$	$1\frac{5}{16}$	$1\frac{13}{16}$	$2\frac{5}{16}$
			$1\frac{7}{16}$	$1\frac{9}{16}$	$1\frac{7}{8}$	$2\frac{7}{16}$
						$2\frac{5}{8}$

Many large users of chains issue to their employees printed rules containing instructions as to the use of multiple slings and the loads which can safely be lifted at various angles of the legs. In some cases a rule is adopted that the horizontal distance between the points of attachment of the load shall not exceed the length of sling leg ; this means, in effect, that the angle between the legs must not exceed 60°.

This is a simple rule which can be readily followed by the slinger, but it makes it necessary always to provide chains which can safely lift any particular load at this opening of the sling legs. Thus, to lift a load of 12 tons, $1\frac{1}{8}$ in. chains should be used.

TABLE C.

SAFE WORKING LOADS FOR TWO-LEGGED SLINGS.

Chain Diameter. Ins.	Safe Working Loads in Tons and Cwts. for given Angles between Legs.					
	0°	30°	60°	90°	120°	150°
(1)	(2)	(3)	(4)	(5)	(6)	(7)
$\frac{3}{16}$	0–8	0–8	0–7	0–5	0–4	0–2
$\frac{1}{2}$	0–15	0–14	0–13	0–10	0–7	0–3
$\frac{5}{16}$	1–3	1–2	1–0	0–16	0–11	0–5
$\frac{3}{8}$	1–13	1–12	1–9	1–3	0–16	0–8
$\frac{1}{2}$	3–0	2–17	2–12	2–2	1–10	0–15
$\frac{5}{8}$	4–13	4–10	4–0	3–6	2–6	1–4
$\frac{3}{4}$	6–15	6–10	5–17	4–15	3–7	1–14
$\frac{7}{8}$	9–3	8–17	7–18	6–9	4–11	2–7
1	12–0	11–11	10–7	8–9	6–0	3–1
$1\frac{1}{8}$	15–3	14–12	13–2	10–14	7–11	3–18
$1\frac{1}{4}$	18–15	18–2	16–5	13–5	9–7	4–16
$1\frac{3}{8}$	22–13	21–17	19–12	16–0	11–7	5–17
$1\frac{1}{2}$	27–0	26–1	23–8	19–1	13–0	6–19
$1\frac{5}{8}$	31–12	30–12	27–8	22–8	15–16	8–3
$1\frac{3}{4}$	36–15	35–9	31–16	25–19	18–7	9–9
$1\frac{7}{8}$	42–3	40–14	36–10	29–16	21–1	10–17
2	48–0	46–8	41–11	33–18	24–0	12–7
$2\frac{1}{8}$	54–2	52–6	46–18	38–5	27–1	13–19
$2\frac{1}{4}$	60–15	58–12	52–12	42–19	30–7	15–13
$2\frac{3}{8}$	67–12	65–6	58–12	47–16	33–16	17–8
$2\frac{1}{2}$	75–0	72–9	65–0	53–0	37–10	19–7

A few firms allow double slings to be used for the normal working load, provided the angle between the legs does not exceed 90°. This is an unsafe practice, as at this angle the tension in each leg is 0·7 times the total load, and unless the normal working load for the sling is taken with a very ample factor of safety the chains may be overstrained.

Shortening Slings.—Chain slings should never be shortened by tying the chain into a knot, a method frequently adopted, but one which causes excessive bending stresses in some of the links, and may result in permanent injury or fracture. If it is necessary to shorten a chain a suitable adjuster should be used, and the links bearing the load will then be subjected only to their normal stresses.

TESTING.

TABLE D.
ORDER BY THE BOARD OF TRADE.

Tests for Short-Link Chain Cables.

Order of the Board of Trade dated June 26, 1900, under s. 8 of the Anchors and Chain Cables Act, 1899. Short-Link Chain Cables.

Breaking and Tensile Strains.

Diameter of Iron Common Links. (1)	Breaking strain which 3 links in each 15 fathoms must withstand previous to the tensile strain being applied. (2)	Tensile strain to be applied to every 15 fathoms separately. (3)	Diameter of Iron Common Links. (1)	Breaking strain which 3 links in each 15 fathoms must withstand previous to the tensile strain being applied. (2)	Tensile strain to be applied to every 15 fathoms separately. (3)
Ins.	Tons.	Tons.	Ins.	Tons.	Tons.
2	96	48	1 3/32	28 6/10	14 3/10
1 31/32	93	46½	1 1/16	27	13½
1 15/16	90	45	1 1/32	25½	12¾
1 29/32	87 2/10	43 6/10	1	24	12
1 7/8	84¼	42⅛	31/32	22½	11¼
1 27/32	81½	40¾	15/16	21	10½
1 13/16	78 8/10	39 4/10	29/32	19 6/10	9 8/10
1 25/32	76	38	7/8	18¼	9⅛
1¾	73½	36¾	27/32	17	8½
1 23/32	70 8/10	35 4/10	13/16	15 8/10	7 9/10
1 11/16	68¼	34⅛	25/32	14 6/10	7 2/10
1 21/32	65 4/10	32 2/10	3/4	13½	6¾
1⅝	63½	31⅝	23/32	12½	6¼
1 19/32	60 8/10	30 4/10	11/16	11½	5⅝
1 9/16	58½	29¼	21/32	10½	5¼
1 17/32	56¼	28⅛	5/8	9½	4¾
1½	54	27	19/32	8 4/10	4 2/10
1 15/32	51¾	25⅞	9/16	7¼	3⅝
1 7/16	49½	24¾	17/32	6½	3⅜
1 13/32	47 4/10	23 7/10	½	6	3
1⅜	45½	22⅝	15/32	5½	2⅝
1 11/32	43½	21⅝	7/16	4½	2¼
1 5/16	41¼	20⅝	13/32	3 8/10	1 9/10
1 9/32	39¼	19⅝	3/8	3¼	1⅝
1¼	37½	18¾	11/32	2 8/10	1 4/10
1 7/32	35 6/10	17 5/10	5/16	2¼	1⅛
1 3/16	33 8/10	16 9/10	9/32	1 8/10	9/10
1 5/32	32	16	¼	1½	¾
1⅛	30½	15½			

NOTES.

(4) The maximum working load for any short or close-link chain should not exceed $6df$ tons, when used under normal conditions without shock; $5d^2$ tons where there is any special danger to life or limb, or $3{\cdot}5d^2$ tons where the chain is subjected to shock or heavy wear, or is used for lifts of molten metal or heated liquid; d being the diameter in inches of the iron of the chain. The maximum working load for a long-link chain should not exceed two-thirds of that for a short-link chain of the same diameter.

(5) When chain slings with two or more legs are used with the legs inclined, care should be taken to ensure that the maximum load in each leg does not exceed that given in recommendation (4) above—see also table.

(8) Slings or other chains should never be shortened by tying knots in them; and wood or other suitable packing should be provided to prevent the links coming in contact with corners of loads of hard material.

(11) All chains and other lifting appliances should—(*a*) If used for lifting in connection with molten metal or molten slag, be examined at least once in every month and be annealed at least once in every six months; or (*b*) if used in connection with lifting machinery or plant operated mechanically or electrically, or used for lifting in metal foundries other than that mentioned in paragraph (*a*) above, be examined at least once in every three months and be annealed at least once in every twelve months; or (*c*) if used for classes of lifts other than those mentioned in (*a*) and (*b*) above, e.g., in connection with hand cranes, be examined at least once in every six months and be annealed at least once in every two years; or (*d*) if used for holding down any crane or other lifting machinery upon a stage or platform should be examined as far as reasonably practicable, *in situ*, at least once in every three months, and be fully examined and annealed at least once in every twelve months, unless the nature of the operations preclude such examinations and annealing.

(12) A chain or other appliance should not be used for lifting its maximum working load if the wear at any part exceeds 10 per cent. of the nominal dimension of that part, or for lifting any load if it has been subjected to a severe shock or is deformed or otherwise injured.

(13) A chain or other appliance which has been altered or repaired should be re-tested before use.

Annealing Wrought Iron Chains.*

Surface hardening, caused by repeated small blows due to inter-link action, chafing, rattling, etc., is the principal cause of brittleness, and low-temperature annealing is the most practical method of correction.

The chain should be placed in a muffle furnace and uniformly heated to a temperature of 650° C. A lower temperature is unsuitable and may be detrimental. The chain should then be cooled slowly either in the furnace or under a layer of non-conducting material. Slow cooling is most important.

After annealing, the chain should be cleaned by brushing, and each link examined for faulty welds, cracks, etc.

Strength of Ropes.†

The following rules are useful for roughly estimating the safe working loads for ropes:

Steel Ropes.

C = Circumference of rope in inches; S = Safe working load in tons.

$$S = C^2 \div 2.$$

Hemp Ropes.

C = Circumference of rope in inches; S = Safe working load in cwts.

$$S = C^2.$$

From a series of 368 tests of Manilla rope in sizes from ¼ in. to 4½ in. diameter carried out at the United States Bureau of Standards during the past three or four years it has been found that the average breaking strength of a three-strand regular lay rope is very closely represented by the formula

$$L = 5000\, d\,(d + 1).$$

---

* 'Notes on the Annealing and Use of Wrought-Iron Chains,' H.M. Stationery Office, Adastral House, Kingsway, W.C. 2, price 1*d.* net.

† See paper read by Mr. Daniel Adamson, M.I.Mech.E., before the Inst. of Mech. Engrs. 1912.

Testing Manilla Rope.

This test has been adopted by the United States Bureau of Standards: Free the rope from oil, soak it for 20 seconds in a solution of bleaching powder acidulated with acetic acid, rinse in water and then in alcohol, and finally expose the fibres of the rope for a minute to the fumes of ammonia. Manilla fibre turns russet brown, while all other rope fibres turn a cherry red.

Preservation of Manilla and Wire Ropes.

Manilla ropes and slings and wire cables through neglect do not last as long as they should because they are allowed to decay and rust, and apart from the waste of not preserving them they are a source of accidents. Ordinary ropes should be painted over with beeswax and resin, about equal parts of each, melted together, and applied with a paint brush. The rusting of wire cables and ropes is arrested by treating them with a mixture of tallow and oil, seven or eight pounds of melted tallow to a gallon of oil, lime being added, when the mixture is cold, until it is thick. The composition is then applied with a brush.

Hemp ropes may be preserved by dipping them when dry into a solution of 1 oz. of sulphate of copper dissolved in $2\frac{1}{2}$ pints of water, and kept in this solution for four days, afterwards being dried; the rope should then be soaked in a solution of 5 ozs. of soap dissolved in $2\frac{1}{2}$ pints of water, and again dried.

Corrosion of Iron Ropes.

Experience has shown that wire ropes of compound construction, subjected to corrosion influences, are likely to deceive engineers as to the strength remaining in them. Where reduction of diameter or circumference of the rope has taken place, not accounted for by the evidence of wear, the part of the rope under examination should first be fully loaded and then relieved of the load. Any noticeable difference in circumference in these circumstances, and the slackening of the outside wires when the load is off, will indicate that internal corrosion has taken place. The extent of corrosion inside the strand can only be estimated by the slackness of the outside wires. The corrosion between the strands can be further examined by untwisting the rope or displaying the strands sufficiently with a marlin-spike.

FATIGUE IN METALS.*

(Contributed by Prof. B. P. Haigh, M.B.E., D.Sc., M.Inst.C.E., and F. W. Thorne, Assoc.M.Inst.C.E., A.M.I.Mech.E.)

It is well known that metal parts, that may be tough and ductile and able to withstand severe test loads, commonly break in a brittle manner after smaller loads have been applied or reversed large numbers of times. This mode of fracture, by which a ductile metal breaks in a brittle manner after repeated changes in a moderate load, is known as 'fatigue.'

Fatigue fractures may be recognised by the shell-like markings that radiate from the origin of the crack, and by the absence of ductile bending, cupping, or reduction of area near the origin. Before fracture, the hair-cracks may often be observed when a bright surface, moistened with blackened oil and cleaned, is vibrated by impact or otherwise. The cracks often extend very slowly in practice, although quickly in laboratory tests in which the full range of test load is applied in each successive cycle.

Methods of Testing.

Metals are immune from fatigue when the applied stress, which is reversed during each cycle of loading, keeps within an ascertainable limit, known as the 'fatigue limit' of the metal. The object of fatigue testing machines is to determine this limiting safe range of stress. This is done by testing several successive test-pieces with different ranges of stress greater than the fatigue

---

* For further information on this subject the reader is referred to *The Fatigue of Metals*, H. J. Gough, M.B.E. (Scott, Greenwood); *The Fatigue of Metals*. Moore and Kommers (McGraw-Hill Book Co.). Bibliographies in the above text-books contain respectively 183 and 448 individual references to original papers.

limit, so that the endurances may be compared. Fig. 37 shows how the results of such tests may be plotted, the base lines of the diagram giving either the endurance N or the logarithm of the endurance as preferred.

A range of stress that gives an endurance of 5–10 million cycles before fracture is usually regarded as only very slightly above the fatigue limit.

In 'Wöhler' testing machines the test-piece is held in a revolving chuck, and is loaded through a ball-bearing on its overhanging end. The profile of the test-piece may be tapered (fig. 38) to give uniform bending stress in sections along a length of about 1 in. To obtain uniform stress near A the length of the cone AO should be three times the leverage AB equal to L.

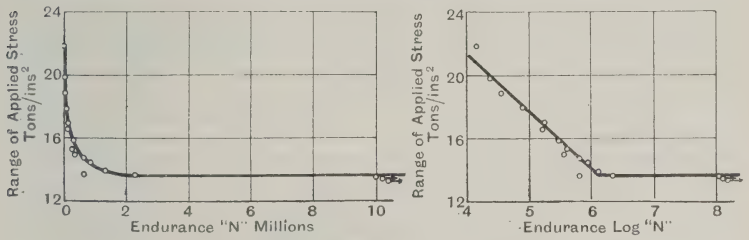


FIG. 37.

In the 'Haigh' testing machine (fig. 39) the test-piece T is loaded in alternate pull and push uniformly distributed across the whole section. The upper end of the test-piece is held by a fixed head H, and the lower end is loaded by an armature A that oscillates in harmonic motion (2000 cycles per minute) between a pair of two-phase magnets, M_1 and M_2. A spring S is employed to compensate the force required to accelerate the armature, and also to apply steady loads to the test-piece in addition to the alternating load applied by the magnets. The loads can be adjusted and measured with accuracy.

Values of Fatigue Limits.

It has been shown conclusively that the fatigue limit may be far below the tensile elastic limit, although, in annealed metals, it may be higher than the yield point. It appears that the ultimate tensile strength affords the most consistent basis of comparison. The following table gives values for ratios between the fatigue limit and the ultimate tensile strength for stresses that reverse between equal values of pull and push.

Metal.	Fatigue limit as percentage of Ultimate tensile strength.	Metal.	Fatigue limit as percentage of Ultimate tensile strength.
Softest annealed iron, of high quality	up to 60%	Alloyed steels, 40 to 70 tons/inch² ult. quenched and tempered	35–50%
Low carbon steels, annealed	up to 60%	Alloyed steels, 70 to 100 tons/inch² ult. quenched and tempered	35–45%
Ditto. As rolled, or quenched and drawn	about 50%	Aluminium alloys of good quality	20–45%
Ditto. Poor qualities	down to 35%	Duralumin, heat-treated	25–40%
Moderate carbon steels, annealed	up to 45%	Rolled brasses, 70–30, 65–35	30–50%
Ditto. As rolled	down to 35%	Monel metal, as rolled or annealed	about 40%
Cast steel and iron	45–35%	Nickel, as rolled or annealed	about 40%
Alloyed steels, 30 to 50 tons/inch² ult. normalised	40–55%	Copper, ,, ,,	30–40%

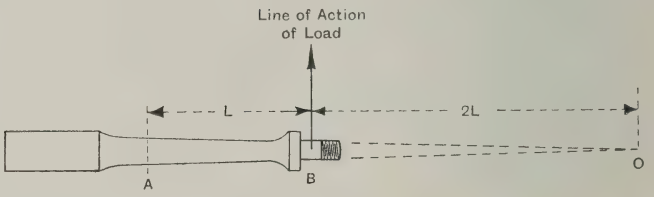


FIG. 38.

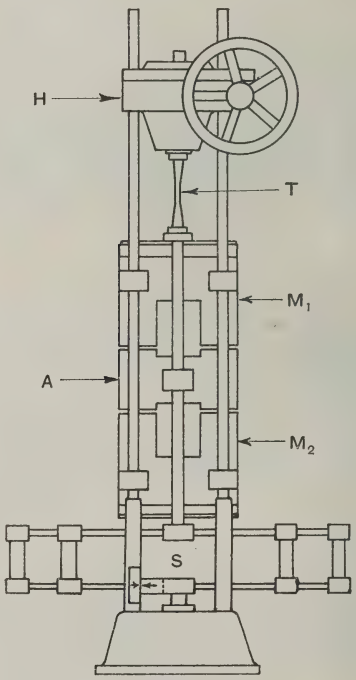


FIG. 39.

In shear-stress the fatigue limits are probably not more than two-thirds of the values indicated above.

When direct stress varies between higher pulls and smaller pushes, the safe range is lower—in different degrees in different metals. If the stress varies between $f_{max.} = s + a$ and $f_{min.} = s - a$, $f_{max.}$ and $f_{min.}$ being the extreme values that are repeated during each cycle of loading, then the fatigue limit a_s, for an alternating stress a acting in combination with a steady tension s, may be expressed

$$a_s = a_o \left(1 - k_1 \left(\frac{s}{u}\right) - k_2 \left(\frac{s}{u}\right)^2\right)$$

where a_o is the ordinary fatigue limit observed when s is zero, and u is the ultimate tensile strength of the metal. The coefficients k_1 and k_2 can be ascertained only by direct experiment, and their sum is found to be usually greater than unity.

Typical values are given in the following table :—

Material.	u ton in.^{2}.	a_o ton in.^{2}.	k_1.	k_2.
0·13 C steel . .	25·1	13·0	0·0	1·0
Naval brass . .	28·7	12·0	1·0	0·0
0·49 C steel . .	43·4	21·5	0·16	2·5
Cast iron . .	14·1	4·5	1·6	— 0·6

Stress Concentrations.

In practice, fatigue cracks almost always start from corners of keyways, sharp fillets, small holes in flanges, or other discontinuities of section where stresses are concentrated locally. As such concentrations are seldom calculable, and are commonly neglected in stress calculations, it follows that higher 'factors of safety' than required for alternating stresses than for steady loads, even when the alternating loads are accurately known.

The ratios in which different discontinuities and surface finishes reduce the apparent fatigue limit are indicated in the table below, in which the strength of a polished cylindrical test-piece is used as a basis of comparison.

Finish or Discontinuity.	Fatigue limit as percentage of Fatigue limit of Polished specimen.
Emery finish, different grades	90% upwards
Filed finish, different roughnesses . . .	80% ,,
Scratches produced by use of needle . . .	84%
Sharp changes of section in shafts . . .	50–75%
Changes of section with fillet	70–90%
Small round holes in flanges	40–60%

The weakening effect of small holes and the like are more evident in hard steels than in mild. The fatigue limits of hardened and tempered springs are much improved when the surface layers are ground away.

Corrosion Fatigue.

The fatigue limits of nearly all metals are much reduced when corrosive reagents, such as salt and fresh water, act on the metal DURING the action of the cyclic stress, although corrosion prior to the action of the cyclic stress has only slight effect.

Steels having widely different ultimate tensile strengths and fatigue limits in air exhibit more nearly uniform fatigue limits in water. In very few steels does the limit in water rise over 10 tons/in.^{2}, although somewhat higher values have been obtained in high-chromium steels.

SECTION VI

BEAMS — GIRDERS — STRESSES — BENDING MOMENTS — INFLUENCE VALUES — MOMENT OF INERTIA — DEFLECTION — COLUMNS — STANCHIONS — RIVETING — STEEL ROOF FRAMING AND COVERINGS — BRITISH STANDARD SECTIONS

(pp. 207—244)

Contributed by Alexander Drew (Consulting Structural Engineer).

SECTION VI

BEAMS — GIRDERS — STRESSES — BENDING MOMENTS — INFLUENCE VALUES—MOMENT OF INERTIA—DEFLECTION — COLUMNS — STANCHIONS — RIVETING — STEEL ROOF FRAMING AND COVERINGS—BRITISH STANDARD SECTIONS.

Contributed by Alexander Drew (Consulting Structural Engineer).

STRESSES ON BEAMS AND GIRDERS.

When designing framed structures, the external forces (or those acting *on* the structure, are always known or assumed, and from these, and the configuration or design of the structure, the internal stresses set up in the several portions of the structure can be determined.

The external forces may be assumed as acting transversely, i.e., at right angles to the beam, or girder, the supporting forces or reactions necessarily acting parallel to and in the opposite direction to the applied forces. Where any force acts other than transversely, it should be resolved into two component forces, one acting in the direction of the length of the beam (and causing compression or tension throughout the complete cross-section), and the other acting transversely : to obtain the total stress at any section of the beam, the tension or compression due to the former component must be added algebraically to the stress from the transverse component.

The beam or girder, whatever its form may be, will tend to shear transversely, and to bend under any system of loading, and the values of these tendencies at any section are termed the 'shearing stress' and the 'bending moment' values at that particular section.

At any vertical section throughout a beam the shearing stress to be resisted is the algebraic sum of all the vertical forces acting to the right or the left of the section chosen ; the forces being reckoned, say, as positive or negative as they act in an upward or downward direction.

For any series of transverse forces W_1, W_2, W_3, etc., acting to one side of the chosen vertical section, the general formula for the shearing stress is :—

$$S = \Sigma W = \pm W_1 \pm W_2 \pm W_3 \text{ etc.} \quad . \quad . \quad . \quad . \quad . \quad (1)$$

At any vertical section throughout a beam, the bending moment to be resisted is the algebraic sum of the moments of the several vertical forces acting to the right or left of the chosen section, these moments being reckoned, say, as positive or negative as they tend to turn the beam about the section line in a clock-way or anti-clock way.

Thus, for any series of transverse forces W_1, W_2, W_3, etc., acting at the several horizontal distances l_1, l_2, l_3, etc., from the chosen vertical section, the general formula for the bending moment is :—

$$M = \Sigma(Wl) = \pm W_1 l_1 \pm W_2 l_2 \pm W_3 l_3, \text{ etc.} \quad . \quad . \quad . \quad . \quad . \quad (2)$$

(In the case of uniformly distributed loads the value W is that for the total load under consideration, the horizontal distance being then measured from the chosen section to the centre of the load.)

Where the supports to a beam are not more than two in number, and the ends of the beam are free, the reactions or supporting loads can readily be found by the law of the lever. These values are then added to the list of external forces, and the shearing forces and bending moment values calculated from formulæ (1) and (2).

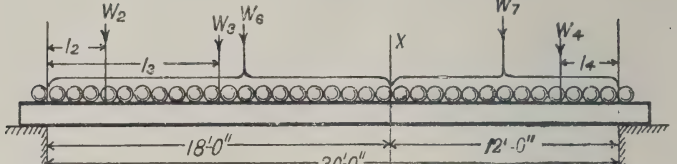


FIG. 1.

EXAMPLE :—A beam 30 ft. clear span between reactions carries three concentrated loads of 150 lbs. each, spaced as shown (fig. 1); weight of beam 60 lbs. per linear foot. l_2, $l_4 = 3'\text{-}0''$, $l_3 = 9'\text{-}0''$.

Reaction, left $= W_1$.	Reaction, right $= W_5$.
Beam $= 30 \times 60 \times \frac{1}{2} = 900$ lbs.	$= 30 \times 60 \times \frac{1}{2} = 900$ lbs.
Load W_2 $= 150 \times \frac{30-3}{30} = 135$,,	$= 150 \times \frac{3}{30} = 15$,,
,, W_3 $= 150 \times \frac{30-9}{30} = 105$,,	$= 150 \times \frac{9}{30} = 45$,,
,, W_4 $= 150 \times \frac{30-27}{30} = 15$,,	$= 150 \times \frac{30-3}{30} = 135$,,
Total $= 1{,}155$ lbs.	Total $= 1{,}095$ lbs.

Shearing stress at Section XX, fig. 1, calculated from left :—

Formula (1) $S = W_1 - (W_2 + W_3 + W_6)$
$= 1{,}155 - [150 + 150 + (60 \times 18)] = 224$ lbs.

Calculated from right $S = W_5 - (W_4 + W_7)$
$= 1{,}095 - [150 + (60 \times 12)] = 224$ lbs.

Bending moment at Section XX, fig. 1, calculated from left :—

Formula (2) $M = W_1 l_1 - (W_2 l_2 + W_3 l_3 + W_6 l_6)$
$= 1{,}155 \times 18 - [(150 \times 15) + (150 \times 9) + (18 \times 60 \times 9)] = +7{,}470$ ft. lbs.

and from right, $M = -W_5 l_5 + (W_4 l_4 + W_7 l_7)$
$= -1{,}095 \times 12 + [(150 \times 9) + (60 \times 12 \times 6)] = -7{,}470$ ft. lbs.

Graphic diagrams for shear and bending moment values can be drawn so as to represent these values at all points throughout the length of a beam or girder, and such diagrams can always be used when setting out some preliminary or trial design, or checking a design ; when carefully and accurately drawn they can be utilised directly in designing all ordinary structures, thus saving a large amount of time and labour which would be necessary if all required values were calculated arithmetically. Since these diagrams show the variations in stress throughout the beam, they can be further utilised to set out the sections or scantlings of the web and flange members, such as the thickness of the web or shear resisting members, and the cut-off in the flange plates or bending resisting members.

The tables of bending moments and shearing stresses given on pages 212 and 213 embrace all simple conditions for concentrated or uniformly distributed loads. Any complex loading occurring in practice can always be resolved into a series of simple loadings; so that diagrams for each simple loading can be drawn on one base line, and these diagrams combined to produce the required diagram for the complex loading, by adding together the several ordinates at any desired section. An example of this is given in fig. 2.

It should be noted that for any series of concentrated loads the bending moment line will form a polygon, the line changing its direction over each of the loaded points ; at each of these points the shearing stress is suddenly changed in value, and the shearing stress diagram takes a stepped outline. For a uniformly distributed load, the bending moment line is always a parabolic curve, while the shearing stress changes by a regular gradation.

When beams or girders are of varying depth, the diagrams on pages 212 and 213, which are based on the condition of uniform depth, must be adjusted, for the following reason :—By formula (3) it will be obvious that the flange stresses are dependent on the depth, and, in fact, *vary inversely as this depth* ; it is, therefore, necessary to alter the bending moment heights in the *inverse ratio to the alteration in depth of the beam.* Reducing the depth of a beam increases the bending moment value, and necessitates the increasing of the height of the bending moment diagram. Fig. 3 and brief explanation accompanying same will indicate how this adjustment can be readily carried out.

Where the shearing stress diagram is intended to be used in setting out the varying thickness of the web plate of a beam, the shear stress diagram must similarly be adjusted by altering its heights in *inverse ratio to the alteration in depth of the beam*; a shallower girder necessitates a thicker web, when the shear value remains unaltered (see fig. 3).

When these graphic diagrams on pages 212 and 213 are used in connection with panel or lattice girders, a modification is necessary, since the flange stress remains constant between the points of connection of flange to lattice bars; the diagram becomes stepped in outline, and may not be the same in both flanges (see fig. 4).

For the majority of the simple panel or lattice girders met with in practice, the various stresses can be most readily obtained by the reciprocal diagram method or by the method of moments.

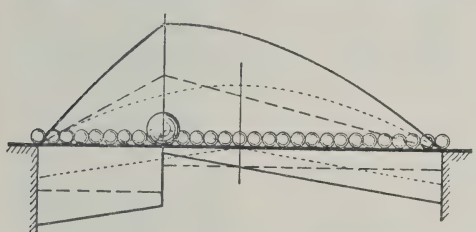


FIG. 2.

This example shows a free-ended beam supporting a uniformly distributed load (see Table 1, type (d)), the bending moment and shearing stress diagrams being indicated by dotted lines; and a single concentrated load (see Table 1, type (e)), the two diagrams being here indicated by broken lines. The combined diagrams for these two loads are shown by full lines, and are got by adding together the two heights at any section, to obtain the new height there.

In some cases, as for girders with fixed ends, negative moments may occur, and these must obviously be added algebraically to the positive moments to obtain the required new combined height.

This figure shows the effect on the bending moment and shearing stress diagrams due to a change in the depth of the girder, and a simple graphic method of indicating this effect on change. The left half of fig. 3 shows diagrams for a free-ended girder of uniform depth carrying a uniformly distributed load (Table 1, type (d)); in the right half of the figure the depth of the girder is reduced as indicated. From the centre to the first section line the depth is uniform, so that the bending moment line remains unaltered, but beyond this section the *height of diagram must be increased in proportion to the reduction in depth of the girder*. This increase can be obtained graphically thus: — Take the vertical section line at point *a*, and draw any line *ab*; on this line mark off *ab* equal to the centre depth of the girder, and *ac* equal to the reduced depth at this section; join *cd* (*d* being at the point of intersection of original diagram line and section line), and draw *be* parallel to *cd* and cutting the section line at *e*. It is obvious that the new vertical heights *ae* and *ad* are proportionate to the original depth and the reduced depth of the girder, or to the lengths *ab* and *ac*, and consequently the adjusted bending moment diagram must pass through the new point *e*. Repeating this operation at a number of other sections will give a series of new points through which the complete adjusted diagram can be drawn.

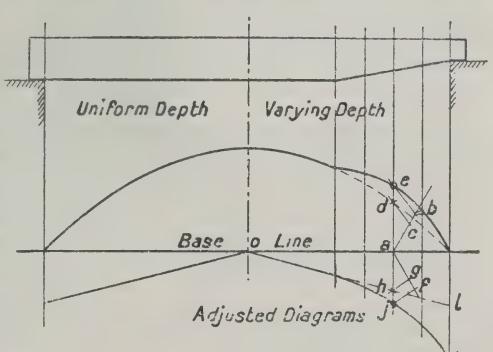


FIG. 3.

Similarly if the shear diagram is to be used for determining the thickness of web plate required, *its depth must first be increased in proportion to the reduction in depth of the girder*, since the sectional area of the web plate is directly proportionate to its depth. Graphically the same operation as above explained can be followed, marking off on the line *af* the lengths *af* and *ag* equal to the centre and reduced depths of the girder, drawing the lines *gh* and *fj* parallel to one another, and deepening the diagram at this section line to the point *j*. The complete adjusted diagram would take the form of the curved line *oik*.

In determining the stresses on the several members of a lattice girder such as that outlined

in fig. 4, it is assumed that at the junctions between the several members the connections will allow of free movement, as if pin jointed: thus the flanges are assumed as hinged at each junction with the lattices, and no allowance is made for the stiffness due to their being actually formed in one piece.

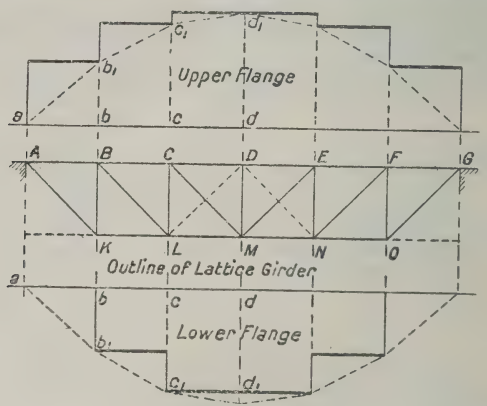


FIG. 4.

This girder rests on abutments at its ends AG, and is loaded at the points BCDEF, say by means of cross girders resting on and fixed to the top flange.

Considering first the load at centre point D, the upper flange does not support it, but the load passes downwards through the vertical strut DM, and is carried by the two inclined ties MC, ME. At the upper ends of these ties the tension due to load at D becomes resolved into compression stress between CE in top flange and compression stress in CL and EN, to which latter stresses must be added a further compression due to loads at points C and E. This transfer of stress is continued to the ends of the girder A and G, the final compression stresses at each end being equal to half the total load, and acting on the bearings at the abutments.

The dotted lines on upper and lower diagrams indicate the bending moment diagrams, and the full lines touching the same show the adjustments necessary owing to the breakage up of flanges and web into a series of members jointed together. The above referred to adjustment is made as follows:—The bending moment diagrams are divided up into sections to correspond with the number of bays in the girder, and as the stress in any section of top or bottom flange must be uniform from joint to joint, the several lines in the diagrams representing the stresses must be parallel to the base line of the diagram throughout the lengths between the joints or junctions between flanges and lattices, all variations in stress taking place suddenly, and being represented by vertical lines in diagram. The stress is the portion AB of upper flange is determined from the moment at point B, and is equal to *bb*; at BC it is equal to *cc*; and at centre portion CDE it is equal to *dd*. Similarly in the lower flange, since the girder is complete without any members extending from K or O to the abutments (as dotted lines), the first stressed bar in lower flange is KL, the amount of stress being determined from height *bb*; the centre portion LMN having its stress determined by the height *cc*; the stress in the two inclined members CM and EM being equal—on the assumption of equal loads at all upper flange joints—must balance one another, and consequently do not affect the lower flange stress at centre.

Influence Value Tables.

It has already been pointed out on page 208 that the several bending moment diagrams given on pages 212 and 213 may be combined to produce one diagram for a series of different loadings.

The same result may be readily obtained, particularly where the value at one particular section is desired, by calculating the bending moment value at that section for each loading and adding the several values together. This process can be much simplified by the use of the tables on pages 214–216, where the value is given of the effect or influence at a series of other points or sections throughout the length of the beam, of a concentrated load applied at certain points. These values are for a load of unit value for a beam of unit length, and for a beam with free ends, with one end fixed, and with both ends fixed; and reactions and inflection points are also noted. The brief notes accompanying the tables will explain their application.

INTERNAL STRESSES.

The tendency of a beam to shear, or to deflect, under any system of loading, is opposed by the resistance of the material of which the beam is composed to shearing, or to tension or compression.

It is the general practice to consider the flanges as resisting the bending moment stresses, the web providing the required shear resistance only.

Considering in the first instance the flange stresses, it is evident, if failure is not to occur, that the moment of resistance at all points in every beam or girder must oppose and balance the bending moment there; this resistance moment is the sum of the moments of all the fibre resistances reckoned about the neutral axis. In a beam of ordinary section the resisting fibres lie at various distances from the neutral axis (owing to the thickness of the flanges); but the simplest case is that of a girder with flanges whose thickness is small in comparison to the depth of the girder, so that without material error all the fibres of one flange may be considered to be practically at one distance from the neutral axis. Thus if d = vertical depth of girder between centres of gravity of the two flanges, M = bending moment due to the external loads, we have the general formula,

$$\pm F = \frac{M}{d} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

F being the horizontal stress in either of the two flanges, being compressive stress in the one and tensile stress in the other.

Where the flanges of a girder are of considerable thickness, in comparison with the depth of the girder, or the beam is of a non-symmetrical or irregular cross-section, the fibres of either flange cannot be considered as practically at one distance from the neutral axis, and it is necessary to take into account the varying distances of the fibres from the neutral axis, in the following manner:—

Under the action of any transverse bending, the intensity of the tensile and compressive stresses varies in different fibres of the beam, being greatest in the extreme fibres at the top and the bottom of the cross-section, while the fibre-stress will be zero at the 'neutral axis.'

Whatever may be the figure of the beam's cross-section, the neutral axis will be a horizontal line drawn through its centre of gravity; and the intensity of the stress in any fibre will be proportional to the vertical distance y of the fibre above or below the neutral axis.

If a denotes the sectional area of any thin layer of fibres lying at the height y above the axis, the total stress in that layer will be proportional to ay, and its moment about the neutral axis will be proportional to $ay \times y = ay^2$. The summation of the quantities ay^2 for the different layers is the so-called 'moment of inertia' of the beam, or

$$I = \Sigma (ay^2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

When the bending moment, M, due to the external forces has been found in the manner already described, the problem will often be to find the maximum stress per square inch, f, in the extreme layer of fibres; and if y denotes the vertical distance of that layer from the neutral axis, we have

$$f = \frac{My}{I} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Where the section of the beam is symmetrical about the neutral axis, as in rolled joists, y = half the vertical depth of the beam between centres of gravity of the two flanges, or $\frac{1}{2}d$, and for such sections the above formula may be written,

$$= \frac{Md}{2I} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

In the case of cast iron, whose strength in compression is much greater than in tension, the tension flange should be made considerably larger than the compression flange. In rolled steel joists the two flanges are made of equal area, and an approximate value for the moment of resistance may be readily found by adding to the actual area of each flange $\frac{1}{6}$th of the area of the web, and then reckoning the depth d as the depth between the centres of gravity of the two flanges; then with A = area of one flange and $\frac{1}{6}$th web, and F = max. safe stress on steel,

$$\text{Approx. resistance moment} = A \times F \times d \quad . \quad . \quad . \quad . \quad . \quad (7)$$

When the thickness of the web is small in comparison with the width of the flanges, the extreme tensile stress, f, under the breaking load, as determined by formula (5), corresponds nearly with the ultimate tensile strength of the material; but should the girder have a very substantial web, or should the section be that of a solid rectangle, the calculated breaking load will generally be found to be lower than the actual breaking load. For such cases see formula on page 218.

TABLE I.—BENDING MOMENTS, SHEARING STRESSES, AND DEFLECTION OF STEEL GIRDERS. Graphic Diagrams and Formulæ.

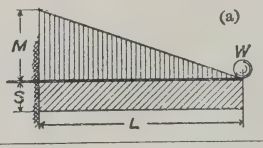
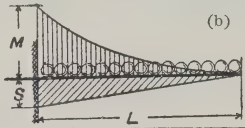
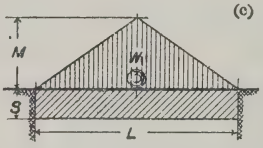
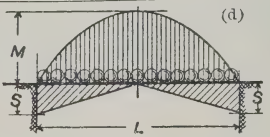
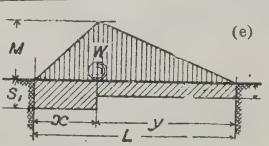
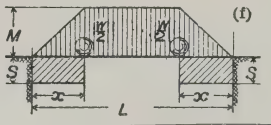
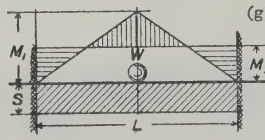
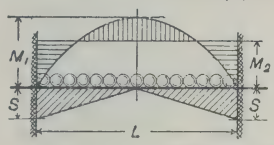
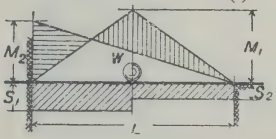
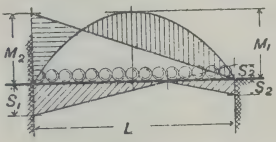
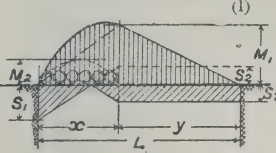
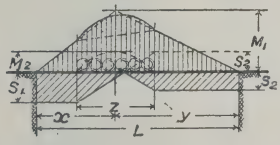
Type.	Formulæ. Bending Moment.	Shearing Stress.	Maximum Deflection.
	$M = WL$	$S = W$	$\frac{WL^3}{3EI}$
	$M = \frac{WL}{2}$	$S = W$	$\frac{WL^3}{8EI}$
	$M = \frac{WL}{4}$	$S = \frac{W}{2}$	$\frac{WL^3}{48EI}$
	$M = \frac{WL}{8}$	$S = \frac{W}{2}$	$\frac{WL^3}{76{\cdot}8EI}$
	$M = \frac{Wxy}{L}$	$S_1 = \frac{Wy}{L}$ $S_2 = \frac{Wx}{L}$	$\frac{Wxy(2L-x)}{27EIL} \times \sqrt{3x(2L-x)}$
	$M = \frac{Wx}{2}$	$S = \frac{W}{2}$	$\frac{Wx(3L^2-4x^2)}{48EI}$
	$M_1 = \frac{WL}{4}$ $M_2 = \frac{WL}{8}$	$S = \frac{W}{2}$	$\frac{WL^3}{192EI}$

TABLE II.—BENDING MOMENTS, SHEARING STRESSES, AND DEFLECTIONS—*contd.*

Type.	Formulæ. Bending Moment.	Shearing Stress.	Maximum Deflection.
(h) 	$M_1 = \frac{WL}{8}$ $M_2 = \frac{WM}{12}$	$S = \frac{W}{2}$	$\frac{WL^3}{384\,EI}$
(i) 	$M_1 = \frac{WL}{4}$ $M_2 = \frac{3}{16}\frac{W}{L}$	$S_1 = \frac{11}{16}W$ $S_2 = \frac{5}{16}W$	$\frac{WL^3}{107{\cdot}3\,EI}$ at a point ·5528 L from Fixed End
(k) 	$M_1 = \frac{WL}{8}$ $M_2 = \frac{WL}{8}$	$S_1 = \frac{5}{8}W$ $S_2 = \frac{3}{8}W$	$\frac{WL^3}{184{\cdot}6\,EI}$ at a point ·5785 L from Fixed End
(l) 	$M_1 = \frac{Wxy}{2L}$ $M_2 = \frac{Wx}{8}$	$S_1 = \frac{W\left(y+\frac{x}{2}\right)}{L}$ $S_2 = \frac{Wx}{2L}$	—
(m) 	$M_1 = \frac{Wxy}{L}$ $M_2 = \frac{Wz}{8}$	$S_1 = \frac{Wy}{L}$ $S_2 = \frac{Wx}{L}$	—

NOTE.—*All curves in these tables are parabolic curves.*

For an example of the combination of bending moments and shearing stress diagrams, and adjustment of same, see page 209.

Table III.—Influence Values.

Beam Free at Both Ends.

Load at Section.	Bending Moment Coefficient at Section noted.																			Left end reaction.	Right end reaction.
	·05	·10	·15	·20	·25	·30	·35	·40	·45	·50	·55	·60	·65	·70	·75	·80	·85	·90	·95		
·05	·047	·045	·042	·040	·037	·035	·032	·030	·027	·025	·022	·020	·017	·015	·013	·010	·007	·005	·002	·95	·05
·10	·045	·090	·085	·080	·075	·070	·065	·060	·055	·050	·045	·040	·035	·030	·025	·020	·015	·010	·005	·90	·10
·15	·042	·085	·127	·120	·112	·105	·097	·090	·082	·075	·067	·060	·052	·045	·037	·030	·022	·015	·007	·85	·15
·20	·040	·080	·120	·160	·150	·140	·130	·120	·110	·100	·090	·080	·070	·060	·050	·040	·030	·020	·010	·80	·20
·25	·037	·075	·112	·150	·187	·175	·162	·150	·137	·125	·112	·100	·087	·075	·062	·050	·037	·025	·112	·75	·25
·30	·035	·070	·105	·140	·175	·210	·195	·180	·165	·150	·135	·120	·105	·090	·075	·060	·045	·030	·015	·70	·30
·35	·032	·065	·097	·130	·162	·195	·227	·210	·192	·175	·157	·140	·122	·105	·087	·070	·052	·035	·017	·65	·35
·40	·030	·060	·090	·120	·150	·180	·210	·240	·220	·200	·180	·160	·140	·120	·100	·080	·060	·040	·020	·60	·40
·45	·027	·055	·087	·110	·137	·165	·192	·220	·247	·225	·202	·180	·157	·135	·112	·090	·067	·045	·022	·55	·45
·50	·025	·050	·075	·100	·125	·150	·175	·200	·225	·250	·225	·200	·175	·150	·125	·100	·075	·050	·025	·50	·50
·55	·022	·045	·067	·090	·112	·135	·157	·180	·202	·225	·247	·220	·192	·165	·137	·110	·082	·055	·027	·45	·55
·60	·020	·040	·060	·080	·100	·120	·140	·160	·180	·200	·220	·240	·210	·180	·150	·120	·090	·060	·030	·40	·60
·65	·017	·035	·052	·070	·087	·105	·122	·140	·157	·175	·192	·210	·227	·195	·162	·130	·097	·065	·032	·35	·65
·70	·015	·030	·045	·060	·075	·090	·105	·120	·135	·150	·165	·180	·195	·210	·175	·140	·105	·070	·035	·30	·70
·75	·012	·025	·037	·050	·062	·075	·087	·100	·112	·125	·137	·150	·162	·175	·187	·150	·112	·075	·037	·25	·75
·80	·010	·020	·030	·040	·050	·060	·070	·080	·090	·100	·110	·120	·130	·140	·150	·160	·120	·080	·040	·20	·80
·85	·007	·015	·022	·030	·037	·045	·052	·060	·067	·075	·082	·090	·097	·105	·112	·120	·127	·085	·042	·15	·85
·90	·005	·010	·015	·020	·025	·030	·035	·040	·045	·050	·055	·060	·065	·070	·075	·080	·085	·090	·045	·10	·90
·95	·002	·005	·007	·010	·013	·015	·017	·020	·022	·025	·027	·030	·032	·035	·037	·040	·042	·045	·047	·05	·95

These tables are calculated for a unit load, and for a girder of unit length ; the length of the girder is divided into 20 equal sections and the bending moment values are given at each section point for unit load placed on each section line in turn. For any weight and any length of span it is necessary only to multiply the proper tabular coefficient by the number of units in the load and in the span to obtain the actual

(Continued on p. 215.)

TABLE IV.—INFLUENCE VALUES.

BEAM FIXED AT RIGHT END.

Load at Section.	Bending Moment Coefficient at Sections noted—(To left of heavy broken line—positive moments; to right—negative).																				Left end reaction.	Right end reaction.	Inflection point.
	·05	·10	·15	·20	·25	·30	·35	·40	·45	·50	·55	·60	·65	·70	·75	·80	·85	·90	·95	1·00			
·05	·046	·042	·039	·035	·031	·027	·024	·020	·016	·012	·009	·005	·001	·002	·006	·010	·014	·017	·021	·025	·925	·075	·333
·10	·042	·085	·078	·070	·063	·055	·048	·040	·033	·025	·018	·010	·003	·005	·012	·020	·027	·034	·042	·049	·851	·149	·331
·15	·039	·078	·116	·105	·094	·083	·072	·061	·050	·038	·027	·016	·005	·007	018	·030	·041	·054	·066	·073	·777	·223	·328
·20	·035	·070	·106	·141	·126	·111	·096	·082	·067	·052	·037	·023	·008	·007	·022	·037	·051	·066	·081	·096	·704	·296	·324
·25	·032	·063	·095	·127	·158	·140	·122	·103	·085	·066	·048	·030	·011	·007	·025	·044	·062	·081	·099	·117	·633	·367	·319
·30	·028	·056	·085	·113	·141	·169	·147	·125	·103	·082	·060	·038	·016	·006	·027	·049	·071	·093	·114	·136	·564	·436	·313
·35	·025	·050	·075	·099	·124	·149	·174	·148	·123	·098	·073	·048	·025	·002	·028	·053	·078	·103	·128	·153	·497	·503	·305
·40	·022	·043	·065	·086	·108	·129	·151	·173	·144	·116	·087	·059	·031	·002	·026	·054	·083	·111	·140	·168	·432	·568	·296
·45	·018	·037	·055	·074	·092	·111	·129	·148	·167	·135	·104	·072	·041	·009	·020	·053	·085	·116	·148	·179	·371	·629	·285
·50	·016	·031	·047	·063	·078	·094	·109	·125	·141	·156	·122	·087	·053	·019	·016	·050	·084	·119	·153	·187	·313	·687	·273
·55	·013	·026	·039	·052	·065	·077	·090	·103	·116	·129	·142	·105	·068	·031	·006	·043	·080	·118	·155	·192	·258	·742	·258
·60	·010	·021	·031	·042	·052	·062	·073	·083	·094	·104	·114	·125	·085	·045	·006	·034	·074	·113	·152	·192	·208	·792	·243
·65	·008	·016	·024	·033	·041	·049	·057	·065	·073	·082	·090	·098	·106	·064	·022	·020	·062	·104	·146	·188	·162	·838	·224
·70	·006	·012	·018	·024	·030	·036	·042	·049	·055	·061	·067	·073	·079	·085	·041	·003	·047	·090	·135	·178	·122	·878	·203
·75	·004	·009	·013	·017	·021	·026	·030	·034	·039	·043	·047	·052	·056	·060	·064	·019	·027	·072	·118	·164	·086	·914	·179
·80	·003	·006	·008	·011	·014	·017	·020	·022	·025	·028	·031	·034	·036	·039	·042	·045	·003	·050	·097	·144	·056	·944	·153
·85	·002	·002	·003	·005	·006	·008	·010	·011	·013	·014	·016	·018	·019	·021	·022	·024	·026	·027	·021	·070	·118	·032	·968	·122
·90	·001	·001	·002	·003	·004	·004	·005	·006	·006	·007	·008	·009	·009	·010	·011	·012	·012	·013	·036	·085	·015	·985	·087
·95	·000	·000	·001	·001	·001	·01	·001	·001	·002	·002	·002	·002	·002	·003	·003	·003	·003	·003	·003	·046	·004	·996	·046

bending moment value desired. Thus for a load of 2 tons on a girder of 20 feet span the tabular values multiplied by 2×20 will give the actual values in foot tons.

Taking the above load and span, and a free-ended girders, with load at ¼th span (or line ·25) the bending moment under the load and at centre of span will be respectively (·187×2×20) 7·48 and (·125×2×20) 5 foot-tons respectively. For a girder fixed at one end the corresponding values will be (·158×2×20) **6·32 and** (·066×2×20) 2·64 foot-tons respectively.

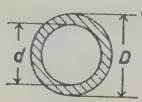
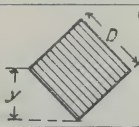
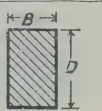
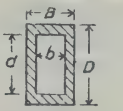
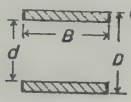
TABLE V.—INFLUENCE VALUES.

BEAM FIXED AT BOTH ENDS.

Load at Section.	Bending Moment Coefficient at Sections noted (outside heavy broken lines—negative moment; inside—positive).																					Left end reaction.	Right end reaction.	Left Infl. pt.	Right Infl. pt.
	0	·05	·10	·15	·20	·25	·30	·35	·40	·45	·50	·55	·60	·65	·70	·75	·80	·85	·90	·95	1·00				
·05	·045	·004	·004	·004	·003	·003	·003	·002	·002	·002	·001	·001	·000	·000	·000	·001	·001	·001	·002	·002	·002	·993	·007	·045	·333
·10	·081	·032	·016	·015	·013	·012	·011	·009	·008	·006	·005	·004	·002	·001	·001	·002	·003	·005	·006	·007	·009	·972	·028	·083	·321
·15	·108	·061	·014	·032	·029	·026	·023	·020	·017	·014	·011	·008	·005	·002	·001	·004	·007	·010	·013	·016	·019	·940	·060	·115	·315
·20	·128	·083	·038	·006	·051	·046	·041	·036	·030	·025	·020	·015	·010	·004	·001	·006	·011	·016	·022	·027	·032	·896	·104	·143	·308
·25	·141	·098	·056	·013	·028	·070	·063	·055	·047	·039	·031	·023	·016	·008	·000	·008	·016	·023	·031	·039	·047	·845	·155	·166	·300
·30	·147	·108	·069	·030	·009	·049	·088	·077	·066	·056	·045	·034	·023	·013	·002	·009	·020	·031	·041	·052	·063	·784	·216	·186	·292
·35	·148	·112	·076	·040	·004	·032	·068	·103	·090	·075	·061	·047	·033	·019	·005	·009	·023	·037	·051	·066	·080	·718	·282	·206	·282
·40	·144	·111	·079	·047	·014	·018	·050	·083	·115	·098	·080	·063	·045	·027	·010	·008	·025	·043	·061	·078	·096	·648	·352	·222	·272
·45	·136	·107	·079	·050	·021	·007	·036	·065	·094	·122	·101	·080	·059	·037	·016	·005	·026	·048	·069	·090	·111	·575	·425	·237	·262
·50	·125	·100	·075	·050	·025	·000	·025	·050	·075	·100	·125	·100	·075	·050	·025	·000	·025	·050	·075	·100	·125	·500	·500	·250	·250
·55	·111	·090	·069	·048	·026	·005	·016	·037	·059	·080	·101	·122	·094	·065	·036	·007	·021	·050	·079	·107	·136	·425	·575	·362	·237
·60	·096	·078	·061	·043	·025	·008	·010	·027	·045	·063	·080	·098	·115	·083	·050	·018	·014	·047	·079	·111	·144	·352	·648	·272	·222
·65	·080	·066	·051	·037	·023	·009	·005	·019	·033	·047	·061	·075	·090	·103	·068	·032	·004	·040	·076	·112	·148	·282	·718	·282	·206
·70	·063	·053	·041	·031	·020	·009	·002	·013	·023	·034	·045	·056	·066	·077	·088	·049	·009	·030	·069	·108	·147	·216	·784	·292	·188
·75	·047	·039	·031	·023	·016	·008	·000	·008	·016	·023	·031	·039	·047	·055	·063	·070	·028	·013	·056	·098	·141	·155	·845	·300	·166
·80	·032	·027	·022	·016	·011	·006	·001	·004	·010	·015	·020	·025	·030	·035	·041	·046	·051	·006	·038	·083	·128	·104	·896	·308	·143
·85	·019	·016	·013	·010	·007	·004	·001	·002	·005	·008	·011	·014	·017	·020	·023	·026	·029	·032	·014	·061	·108	·060	·940	·315	·115
·90	·009	·007	·006	·005	·003	·002	·001	·001	·002	·004	·005	·006	·008	·009	·011	·012	·013	·015	·016	·032	·081	·028	·972	·321	·083
·95	·002	·002	·002	·001	·001	·001	·000	·000	·000	·001	·001	·002	·002	·002	·003	·003	·003	·004	·004	·004	·045	·007	·993	·333	·045

In some cases the values are only approximate, since the tabular values give 3 decimal figures only; for example, in the first calculation the full tabular value would be ·1875, and the accurate result 7·5 foot-tons. The values for a load distributed over a part or the whole of the span may be obtained approximately by assuming the total load as formed of a number of equal loads placed at adjoining section points; the result here would generally be near enough to the true value for all practical purposes.

TABLE VI.—MOMENTS OF INERTIA, SECTIONAL AREA, ETC., FOR VARIOUS SIMPLE SECTIONS.

Section.	Area.	Neutral Axis.	Moment of Inertia.
(a)	$\frac{\pi D^2}{4}$ $\pi = 3{\cdot}1416$	Through centre	$\frac{\pi D^4}{64}$
 (b)	$\frac{\pi(D^2 - d^2)}{4}$	Through centre	$\frac{\pi (D^4 - d^4)}{64}$
(c)	D^2	Through centre	$\frac{D^4}{12}$
 (d)	D^2	Through centre $y = \frac{D}{1{\cdot}4142}$	$\frac{D^4}{12}$
 (e)	BD	Through centre	$\frac{BD^3}{12}$
 (f)	$BD - bd$	Through centre, parallel to B Through centre, parallel to D	$\frac{BD^3 - bd^3}{12}$ $\frac{DB^3 - db^3}{12}$
(g)	$\frac{BD}{2}$	$y = \frac{2}{3}D$ Through Base Line, through Apex, and parallel to Base	$\frac{BD^3}{36}$ $\frac{BD^3}{12}$ $\frac{BD^3}{4}$
 (h)	$B(D - d)$	Through centre and parallel to B	$\frac{B(D^3 - d^3)}{12}$

The value for the moment of inertia for the standard steel sections of rolled joists, angles, tees, etc., can be got from the section books issued by many of the rolling-mill companies and constructional engineering contracting firms.

The value for the moment of inertia about any other axis parallel to and at a distance y, from the neutral axis = I for neutral axis + (area $\times$ y^2) . . . . . (8)

The value for the radius of gyration is $R = \sqrt{\frac{I}{Are}}$ . . . . . (9)

Solid Rectangular Beams.

Where the strength can be shown to vary directly as the breadth of the beam and as the square of the depth, it is better, for the reason given above, to calculate the strength by means of coefficients derived from direct experiments in cross-breaking; thus

$$\text{Breaking load in tons} = \frac{\text{Breadth in inches} \times \text{depth in inches}^2}{\text{Span in inches}} \times c \times k \quad . \quad . \quad . \quad . \quad . \quad (10)$$

TABLE VII.

Material	Coeff. c.	Ends of Beams	Coeff. k.
Mild Steel	3·50	One fixed, other end loaded	1
Cast Iron	2·30	,, ,, load distributed	2
Fir (Spruce)	0·60	Both supported, load in centre	4
English Oak	0·75	,, ,, ,, distributed	8
Red Pine	0·65	Both fixed, ,, in centre	6
Yellow Pine	0·50	,, ,, ,, distributed	12
Pitch Pine	0·75		

The factor of safety for steel may be $\frac{1}{4}$th to $\frac{1}{3}$th breaking load; for the other materials it may vary between $\frac{1}{3}$th and $\frac{1}{4}$th breaking load. If the beam carries a heavy live load the safe load should be further reduced.

Safe Working Stress.

The safe working stress is usually taken as some fraction of the ultimate strength of the material; it requires careful consideration and much experience to decide on the correct value for this fraction. This consideration is further complicated when any portion of the total load is such that it is applied and removed, since these conditions may produce a considerable dynamic effect. Another point which must be kept in view is that the maximum stress, however produced, must be well within the elastic limit of the material, so as to avoid any permanent distortion or set. For mild steel such as is used in general structural work, the ultimate strength will vary from 27 to 32 tons per square inch, with an elastic limit of 14 tons per square inch and upwards; the maximum stress should therefore not exceed 9 to 10 tons per square inch, or say a factor of safety of 3 with respect to the ultimate strength.

Repeated or fluctuating loads.—These can be classed under two heads, first where the load is such as will produce stresses in the structure varying from zero to a maximum; and second where the load will produce stresses varying from compression to an equal tension.

Wöhler's law of stresses for repeated loading and reversed loading may be briefly indicated thus:—

Class 1.—Static or dead loading . . . . . ratio 3
,, 2.—Loads producing stresses varying from zero to maximum . ,, 2
,, 3.— ,, ,, ,, ,, ,, comp. to equal tension ,, 1

As the load carried by any structure is always composed partly of dead and partly of live or fluctuating loads, difficulties arise in determining the stresses in the several members of the structure, for the following reasons. The weight of the structure itself provides a necessary dead load, irrespective of any other additional dead load (for example, water flowing into a tank, although actually a live load, is so steadily added to the structure as to be *practically* a dead load), and it is obvious that the dynamic resistance of this dead load must more or less reduce the actual dynamic effect on the structure of any live load. A simple formula has been suggested by Mr. E. H. Stone, M.Inst.C.E., which he calls his 'Range' formula, and which takes into account the *immediate effect* (due to sudden loading, as when a train passes over a bridge), and the *cumulative effect* (that produced in time by repeated loading and unloading). The former effect, as compared with that due to the same amount of dead load, was deduced from experiments, while the second effect was deduced from Wöhler's experiments: these two effects were then combined in a rational manner to give the total effect of the moving load, as compared with that of an equivalent amount of dead load. Starting with a maximum safe stress for *dead loads alone* of 9 tons per square in. for mild steel, Mr. Stone deduced the following formula for obtaining the equivalent dead load stresses for varying percentages of life to dead loads, which would produce all actual maximum stress of 9 tons per sq. in. (p = percentage of live to dead load).

$$\text{Safe working stress} = 9 - (5 \times p) \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

The following table is based on this formula '—

TABLE VIII.—DEAD LOAD EQUIVALENTS FOR COMBINED LIVE AND DEAD LOADS.

p	E.D.L.	p	E.D.L.	p	E.D.L.
100	4·00	65	6·89	30	8·55
95	4·49	60	7·20	25	8·69
90	4·95	55	7·49	20	8·80
85	5·39	50	7·75	15	8·89
80	5·80	45	7·99	10	8·95
75	6.19	40	8·20	5	8·99
70	6·55	35	8·39	0	9·00

p = percentage of live load in total load. E.D.L. = equivalent dead load stress.

Deflection of Beams.

Deflection formulæ for various arrangements of loading are given in Tables I. and II., pp. 212 and 213. For structural steel the value of E (the modulus of elasticity) may be taken at 30,000,000 lbs. and the value of W must then be expressed in lbs. also. For the value of E for other materials see *e*, p. 184.

CONTINUOUS BEAMS OR GIRDERS.

When a beam is carried upon more than two supports the reactions of the several supports cannot be directly determined, and, in consequence, the bending stresses can only be found by reference to the changes of form which must always accompany those stresses, and which are fixed or limited at certain points by the conditions of the structure—conditions which vary in each case with the width and the number of spans, as well as with the disposition of the load.

When the beams are of uniform section, such as rolled steel joists or rectangular timber beams, the strength or load which may be safely carried will depend on the greatest bending moment of *either kind* which occurs throughout the whole length of the bridge, and the advantage of a continuous beam to cover a number of spans will depend on the number and spacing of the supports. In cases where all the spans are of uniform length, and the load is uniformly distributed throughout, the maximum bending moment values for varying numbers of spans can be found in many text-books.

For the General Problem, however, where the spans are of different lengths and the loads vary, the determination of the stresses in continuous beams is a very complex one. The easiest solution is to be found in the Graphic Method, generally known as the 'Method of Characteristic Points,' described by Professor Fidler in a paper read before the Inst. C.E. (*Proc.*, vol. lxxiv.), and treated in detail in Fidler's 'Bridge Construction.'

Before proceeding to graphically determine the stresses in a continuous beam, it is necessary to find the Characteristic Points, and the following examples will illustrate how these points can be fixed for any bending moment diagram for a free-ended single span beam.

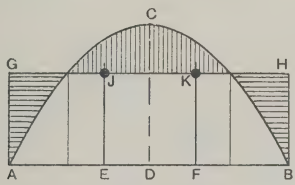


FIG. 5.

In fig. 5, AB represents the span, and the parabola ACB the bending moment diagram due to a uniformly distributed load. Divide the span into three equal parts, AE, EF, and FB, and at points E and F erect perpendiculars EJ and FK. The height of the points J and K (which lie on the line GH) above the base line is fixed by the condition that *the moment of the area ACB*

about one end of the span shall equal the moment of the rectangle AGHB about the same point. This diagram being symmetrical about the vertical centre line, the centre of gravity of these two areas will fall on this centre line, and the moment arm for each area will be equal to ½ AB, or one-half span.

In this case of the *parabolic B.M. diagram.*

$$\text{Area ACB} = \tfrac{2}{3}\ \text{CD} \times \text{AB}, \text{and area AGHB} = \text{AG} \times \text{AB}$$

$$\therefore\ \text{EJ} = \text{FK} = \tfrac{2}{3}\ \text{CD}.$$

The Characteristic Points for a uniformly distributed load, therefore, occur at a height equa to ⅔ the central height of the parabola.

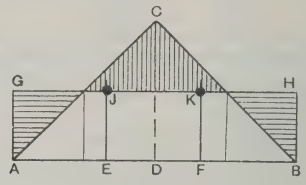


FIG. 6.

Fig. 6 shows the diagram for a *concentrated central load*, represented by the B.M. diagram ACB, and by the same reasoning as for a distributed load, the height EJ and FK will now be equal to ½ the central height CD.

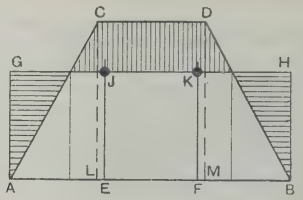


FIG. 7.

Fig. 7 shows another symmetrical case where two equal concentrated loads are placed at the same distance from the centre line; here again the same reasoning as before is adopted. Assuming a span AB = 20 ft., with the two equal loads of 6 tons at 4 ft. from ends A and B. BM = 6 × 4 = 24 ft.-tons = scale height for CL and DM. Then moment of area ACDB about A = AM × MD = 16 × 24 = 384, and height of an equal rectangle AGBH = $\frac{384}{20}$ = 19·2 = EJ and FK.

Where the *B.M. diagram is not symmetrical about the centre vertical line*, the two Characteristic Points, although still situated on the two perpendicular lines at ⅔ AB from each end, *are not now*

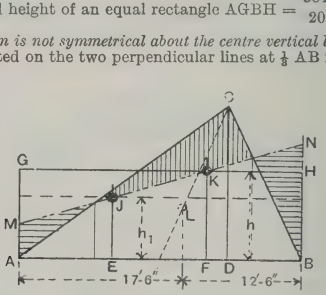


FIG. 8.

at the same height above the base line, since the moment of a non-symmetrical B.M. diagram will not be the same about the ends A and B.

For example, fig. 8 shows a beam, say, 30 ft. long with a load at D of, say, 10 tons placed at a distance of 8 ft. from B. The upward reaction at end A $= \frac{8}{30}$ of 8 tons $= 2 \cdot 13$ tons, and the height DC of B.M. diagram ACB $= 2 \cdot 13 \times 22 = 46 \cdot 9$ ft.-tons. Draw a line from the centre point of AB to C, and set off CL $= \frac{2}{3}$ of the total length, then L is the centre of gravity of the B.M. diagram ACB. The moment of the area ACB will be greater about end A, thus h (= height of rectangle to balance the greater moment) will be found thus :—

$$\text{Area ACB} \times 17 \cdot 5 = h \times 30 \times 15.$$

or,
$$\frac{30 \times 46 \cdot 9}{2} \times 17 \cdot 5 = h \times 450;$$

$$\therefore \quad h = \frac{704 \times 17 \cdot 5}{450} = 27 \cdot 4,$$

and similarly about B,

$$\text{Area ACB} \times 12 \cdot 5 = h \times 30 \times 15;$$

$$\therefore \quad h = 19 \cdot 6.$$

These values, therefore, give the relative heights of the Characteristic Points K and J.

This method of locating the Characteristic Point is applicable for any condition of loading, as it is only necessary to set up the combined B.M. diagram for any desired combination of loads, calculate the area, and the positive of the centre of gravity of this combined diagram, and proceed as indicated to calculate the height of the Characteristic Points, noting that two calculations will be required if the B.M. diagram is not symmetrical about the vertical centre line.

In the case of any Bending Moment Diagram for a simple free-ended beam, the bending stress at any point in its length is proportionate to the vertical height at the required point, measured from the original base or datum line AB, *and Characteristic Points are unnecessary.* Should the beam be rigidly fixed at the ends, use must be made of the new base or datum line found by drawing a line through the Characteristic Points and scaling the required bending moment values from this new line (marked GH in figs. 5–7, and MN in fig. 8). The positive B.M. stresses (in the middle portions of the beam) are measured *upwards* from the new base line to that of the B.M. diagram, while the *negative* stresses are measured downwards. The B.M. is zero at the contra-flexure points, where the bending stresses change from positive to negative. Thus theoretically each span consists of two end cantilevers supporting a hinged or free-ended beam, it being obvious that at the points where the bending *directions* change, the bending stress must be zero. (*Note.*— The bending stress in the end portions are generally termed negative stresses and represented thus −, while the centre portions are positive stresses and represented thus +. In the diagram, negative stressed portions are indicated by horizontal line shading, and positive stressed portions by vertical shading.)

Bending Moment for Continuous Girders.

If the rigid fixing at either end of a beam is altered or interfered with, the alteration in bending stresses which then follows throughout the length of the beam can be represented by an alteration in the position of the straight base line, so that it would pass above or below the Characteristic Points. When a continuous length of beam is carried over two or more spans, this interference occurs at the points of support between the several spans, and Fidler's method enables the effect of this interference to be graphically represented in the following manner : see fig. 9, which represents a four-span continuous bridge carrying a distributed load; the B.M. diagrams in this case take the form of a parabola. For any other form of loading the suitable diagram must be used.

Calculate the bending moment for each span separately as for a free-ended beam, draw the corresponding diagram for each span on the base line AE, and set off the Characteristic Points, as previously explained, marking these points 1 to 8 as shown. If the outer ends of end spans, at A and E, rest freely on the supports, the two end Characteristic Points 1 and 8 are disregarded. The problem now is to draw a series of straight lines, AO, OP, PQ, and QE, but *these lines must be so placed that they pass above and below, or below and above the Characteristic Points on either side of the vertical lines over the supports B, C, and D, at distances inversely proportionate to the adjoining spans*; for instance, in fig. 9 the distance of the line AO above point 2 should be greater than that of the line OP under point 3 in the proportion of the lengths BC to AB. These conditions are carried to the end, points 4 and 5 being at proportionate distances above and below, and points 6 and 7 below and above the broken line, which finally must end at E. This broken line, AOPQE, will now represent the new base line from which the positive or negative bending moment values for any point in the bridge can be scaled off. As in the simpler diagrams previously given,

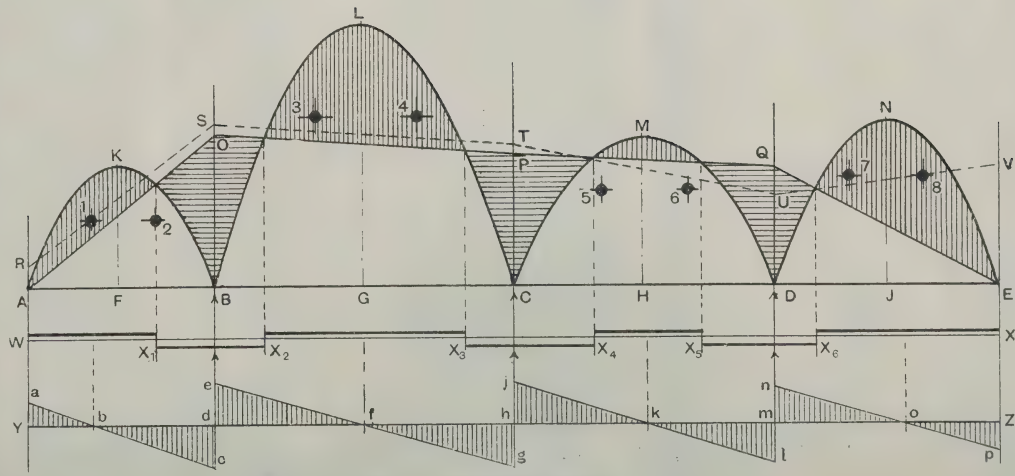


FIG. 9.—Four-Span Continuous Bridge carrying a Distributed Load.

If this bridge is designed so that the outer ends of the end spans are rigidly fixed, the new base line must pass through the two extreme Characteristic Points 1 and 8, and the new line will take the form indicated by the line RSTUV, the heights AR and EV indicating the maximum amount of negative bending moment at the extreme end points.

These diagrams should be drawn out to as large a scale as possible, and it will facilitate the fixing of the irregular base or datum line if tracing paper is first placed over the bending moment layout, and the necessary trial lines pencilled on same, the correct setting out, when found, being pricked through to the diagram below.

THERE IS ONLY ONE SERIES OF BROKEN LINES WHICH WILL FULFIL THE CONDITIONS REQUIRED IN ANY ONE CASE, and some difficulty may at first be experienced in drawing this line.

the portions subjected to positive bending are shown by vertical shading, and negative bending by horizontal shading.

The sketch diagram WX (p. 222) indicates graphically the system or combination of simple beams and cantilevers into which a continuous beam over several supports may be divided.

The sketch diagram YZ gives a graphic representation of the shears and reactions: the reaction is always equal to the shear. The shear at the points of maximum bending becomes zero, or passes from positive (+) above the base line, to negative (−) below the base line at these points.

The shear and reaction can be readily calculated from a consideration of diagram WX.

Shear at A (+) = reaction as from a free-ended beam WX; this value is set up on line YZ, at Ya, shear left of B (−) = total load on span less left reaction (set off *dc* on line YZ). Shear on right of B (+) = load on cantilever X_1, X_2, plus half-load on suspended span X2, X3 (set up *de* on line YZ). Reaction on support B = sum of above (−) and (+) shears. By the same reasoning, the remaining shears and reactions can be determined. Draw lines connecting the points *ac*, *eg*, *jl*, and *np*, to complete the shear diagram; and note that the shear zero points *b*, *f*, *k*, and *o*, should occur at the centres of the imaginary beams in diagram WX.

COLUMNS AND STRUTS.

The strength of columns or struts depends not only on the cross-sectional area of the member, but also on its height or length, and on the condition of fixture of the ends. As the ratio of the length to the least diameter increases, the column becomes more *slim* or liable to fail by bending under its load; such a ratio might thus be called the *ratio of stiffness or slenderness* of a column, and so that this ratio may be used as a means of comparing the strength or stiffness for columns of varying cross-section, the value for the radius of gyration (see p. 217) of the cross-section should be adopted instead of the diameter, since it is obvious, for example, that the diameters of a square and a round column may be the same, but the stiffness would be quite different.

The ends of columns are divided into three classes or conditions of fixture:—

1. Both ends hinged, or free to move.
2. One end hinged and one end fixed.
3. Both ends fixed.

The limiting values for $\frac{L}{R}$ in the tables on pp. 224, 225 have been taken as 200, and this should very rarely be exceeded, owing to the practical difficulties in producing a perfectly straight long column, and one free from internal stresses; and also of erecting such a column on rigid foundations, and ensuring a truly central loading.

The formulæ adopted in calculating tables are given under the same.

Eccentric Loads.

Where the load is not applied centrally, as where a beam rests on a bracket attached to a column, considerably increased stresses may arise through the bending tendency introduced.

The equivalent concentric load which would produce the same maximum stress as that due to the actual eccentric load may be found as follows:—

$$\text{EQUIVALENT CONCENTRIC LOAD} = W\left(1 + \frac{xy}{R^2}\right) \quad . \quad . \quad (12)$$

where

W = Eccentric load;

x = Distance from neutral axis of stanchion to point of application of load = eccentricity;

y = Distance from neutral axis to edge of stanchion nearest load;

R = Radius of gyration of cross section of stanchion about neutral axis at right angles to line of eccentricity.

TABLE IX.—MILD STEEL STANCHIONS.

Maximum Safe Stresses per sq. in. of Section. (London Building Act Amendment, 1909.)

$\frac{L}{R}$	A	B	C	$\frac{L}{R}$	A	B	C	$\frac{L}{R}$	A	B	C
2	4·50	5·50	6·50	72	2·70	3·70	4·70	142	—	*1·90*	2·95
4	4·40	5·40	6·40	74	2·65	3·65	4·65	144	—	*1·80*	2·90
6	4·35	5·35	6·35	76	2·60	3·60	4·60	146	—	*1·70*	2·85
8	4·30	5·30	6·30	78	2·55	3·55	4·55	148	—	*1·60*	2·80
10	4·25	5·25	6·25	80	2·50	3·50	4·50	150	—	*1·50*	2·75
12	4·20	5·20	6·20	82	2·45	3·45	4·45	152	—	*1·40*	2·70
14	4·15	5·15	6·15	84	2·40	3·40	4·40	154	—	*1·30*	2·65
16	4·10	5·10	6·10	86	2·35	3·35	4·35	156	—	*1·20*	2·60
18	4·05	5·05	6·05	88	2·30	3·30	4·30	158	—	*1·10*	2·55
20	4·00	5·00	6·00	90	2·25	3·25	4·25	160	—	*1·00*	2·50
22	3·95	4·95	5·95	92	2·20	3·20	4·20	162	—	*·90*	*2·40*
24	3·90	4·90	5·90	94	2·15	3·15	4·15	164	—	*·80*	*2·30*
26	3·85	4·85	5·85	96	2·10	3·10	4·10	166	—	*·70*	*2·20*
28	3·80	4·80	5·80	98	2·05	3·05	4·05	168	—	*·60*	*2·10*
30	3·75	4·75	5·75	100	2·00	3·00	4·00	170	—	*·50*	*2·00*
32	3·70	4·70	5·70	102	*1·90*	2·95	3·95	172	—	*·40*	*1·90*
34	3·65	4·65	5·65	104	*1·80*	2·90	3·90	174	—	*·30*	*1·80*
36	3·60	4·60	5·60	106	*1·70*	2·85	3·85	176	—	*·20*	*1·70*
38	3·55	4·55	5·55	108	*1·60*	2·80	3·80	178	—	*·10*	*1·60*
40	3·50	4·50	5·50	110	*1·50*	2·75	3·75	180	—	*·00*	*1·50*
42	3·45	4·45	5·45	112	*1·40*	2·70	3·70	182	—	—	*1·40*
44	3·40	4·40	5·40	114	*1·30*	2·65	3·65	184	—	—	*1·30*
46	3·35	4·35	5·35	116	*1·20*	2·60	3·60	186	—	—	*1·20*
48	3·30	4·30	5·30	118	*1·10*	2·55	3·55	188	—	—	*1·10*
50	3·25	4·25	5·25	120	*1·00*	2·50	3·50	190	—	—	*1·00*
52	3·20	4·20	5·20	122	*·90*	2·45	3·45	192	—	—	*·90*
54	3·15	4·15	5·15	124	*·80*	2·40	3·40	194	—	—	*·80*
56	3·10	4·10	5·10	126	*·70*	2·35	3·35	196	—	—	*·70*
58	3·05	4·05	5·05	128	*·60*	2·30	3·30	198	—	—	*·60*
60	3·00	4·00	5·00	130	*·50*	2·25	3·25	200	—	—	*·50*
62	2·95	3·95	4·95	132	*·40*	2·20	3·20				
64	2·90	3·90	4·90	134	*·30*	2·15	3·15				
66	2·85	3·85	4·85	136	*·20*	2·10	3·10				
68	2·80	3·80	4·80	138	*·10*	2·05	3·05				
70	2·75	3·75	4·75	140	*·00*	2·00	3·00				

A = Both ends hinged.
B = One end hinged, one end fixed.
C = Both ends fixed.

The above table gives an extension of the empirical working or safe stress values adopted by the London County Council, and embodied in their (General Powers) Act 1909. This table should be used for all districts where the L.C.C. Building Regulations are enforced.

The limit of height of a stanchion should, as a rule, not exceed the lesser of the two following values :—

1. 160 times the least radius of gyration.

2. 40 times the least overall or breadth of the section.

The economical ratio of slenderness for the three types of stanchions, namely, the point beyond which the section increases at a greater rate than below this point, occurs at 100, 140, and 160. Below these values the rate varies one ton per sq. in. for each variation of 40 in the $\frac{L}{R}$ values; above these values the rate of change is 2 tons, being double.

TABLE X.—MILD STEEL STANCHIONS.

Maximum Safe Stresses per sq. in. of Section. (Prof. Claxton Fidler's Formulæ.)

L/R	A	B	C	L/R	A	B	C	L/R	A	B	C
2	6·50	6·50	6·50	72	3·72	4·55	5·39	142	1·33	2·17	3·02
4	6·49	6·49	6·50	74	3·61	4·47	5·33	144	1·30	2·13	2·97
6	6·48	6·48	6·49	76	3·50	4·38	5·26	146	1·26	2·08	2·91
8	6·47	6·48	6·49	78	3·39	4·30	5·20	148	1·23	2·04	2·86
10	6·44	6·46	6·48	80	3·28	4·20	5·13	150	1·20	2·00	2·80
12	6·41	6·44	6·47	82	3·18	4·12	5·07	152	1·17	1·96	2·75
14	6·39	6·42	6·46	84	3·08	4·04	5·00	154	1·15	1·92	2·70
16	6·35	6·40	6·45	86	2·99	3·91	4·93	156	1·12	1·88	2·65
18	6·31	6·37	6·43	88	2·89	3·87	4·86	158	1·10	1·85	2·60
20	6·27	6·35	6·42	90	2·80	3·79	4·79	160	1·07	1·81	2·55
22	6·22	6·31	6·40	92	2·72	3·72	4·72	162	1·05	1·77	2·50
24	6·16	6·27	6·38	94	2·63	3·63	4·65	164	1·02	1·74	2·46
26	6·10	6·23	6·36	96	2·55	3·56	4·57	166	1·00	1·70	2·41
28	6·04	6·19	6·34	98	2·47	3·48	4·50	168	·98	1·67	2·37
30	5·97	6·14	6·31	100	2·40	3·42	4·43	170	·96	1·64	2·33
32	5·90	6·09	6·29	102	2·32	3·34	4·36	172	·94	1·61	2·28
34	5·81	6·03	6·26	104	2·26	3·27	4·28	174	·92	1·58	2·24
36	5·73	5·98	6·23	106	2·19	3·20	4·21	176	·90	1·55	2·20
38	5·64	5·92	6·20	108	2·12	3·13	4·14	178	·88	1·52	2·16
40	5·55	5·85	6·16	110	2·06	3·06	4·07	180	·86	1·49	2·12
42	5·45	5·79	6·13	112	2·00	3·00	4·00	182	·84	1·46	2·08
44	5·35	5·72	6·09	114	1·94	2·93	3·93	184	·82	1·43	2·05
46	5·24	5·64	6·05	116	1·89	2·87	3·86	186	·81	1·41	2·01
48	5·13	5·57	6·01	118	1·83	2·80	3·79	188	·79	1·38	1·98
50	5·02	5·50	5·97	120	1·78	2·75	3·72	190	·78	1·36	1·94
52	4·91	5·41	5·92	122	1·73	2·69	3·65	192	·76	1·33	1·91
54	4·79	5·33	5·88	124	1·69	2·63	3·58	194	·75	1·31	1·88
56	4·67	5·25	5·83	126	1·64	2·58	3·52	196	·73	1·28	1·84
58	4·55	5·16	5·78	128	1·60	2·52	3·45	198	·72	1·26	1·81
60	4·43	5·08	5·73	130	1·56	2·47	3·39	200	·70	1·24	1·78
62	4·31	5·00	5·68	132	1·51	2·42	3·32				
64	4·19	4·90	5·62	134	1·47	2·35	3·26				
66	4·07	4·82	5·57	136	1·44	2·32	3·20				
68	3·95	4·73	5·51	138	1·40	2·27	3·14				
70	3·83	4·64	5·45	140	1·36	2·22	3·08				

A = Both ends hinged.
B = One end hinged, one end fixed.
C = Both ends fixed.

The following are the formulæ used in calculating the above table:—

$$\left.\begin{matrix}\text{Breaking strength in}\\ \text{tons per square inch}\end{matrix}\right\} = \frac{F + f - \sqrt{(F+f)^2 - 2{\cdot}4\,Ff}}{1{\cdot}2} \quad . \quad . \quad (13)$$

= Ultimate compressive strength = 26 tons per square inch in this table.

$$F = \begin{cases} E\pi^2 \times \left(\frac{R}{\cdot 6L}\right)^2 \text{ for both ends fixed.} \\ E\pi^2 \times \left(\frac{R}{\cdot 8L}\right)^2 \text{ for one end hinged, one end fixed.} \\ E\pi^2 \times \left(\frac{R}{L}\right)^2 \text{ for both ends hinged.} \end{cases}$$

E = Modulus of elasticity = 12,000 tons.
π = 3·1416.
R = Minimum radius of gyration of cross section.
L = Height of stanchion in inches.

TABLE XI.—CAST-IRON COLUMNS.

Maximum Safe Stresses per sq. in. of Section. (London Building Act Amendment, 1909.)

$\frac{L}{R}$	Hinged ends.	Hinged and fixed.	Fixed ends.	$\frac{L}{R}$	Hinged ends.	Hinged and fixed.	Fixed ends.	$\frac{L}{R}$	Hinged ends.	Hinged and fixed.	Fixed ends
2	4·4	4·9	5·4	32	2·9	3·4	3·9	62	1·4	1·9	2·4
4	4·3	4·8	5·3	34	2·8	3·3	3·8	64	1·3	1·8	2·3
6	4·2	4·7	5·2	36	2·7	3·2	3·7	66	1·2	1·7	2·2
8	4·1	4·6	5·1	38	2·6	3·1	3·6	68	1·1	1·6	2·1
10	4·0	4·5	5·0	40	2·5	3·0	3·5	70	1·0	1·5	2·0
12	3·9	4·4	4·9	42	2·4	2·9	3·4	72	0·9	1·4	1·9
14	3·8	4·3	4·8	44	2·3	2·8	3·3	74	0·8	1·3	1·8
16	3·7	4·2	4·7	46	2·2	2·7	3·2	76	0·7	1·2	1·7
18	3·6	4·1	4·6	48	2·1	2·6	3·1	78	0·6	1·1	1·6
20	3·5	4·0	4·5	50	2·0	2·5	3·0	80	0·5	1·0	1·5
22	3·4	3·9	4·4	52	1·9	2·4	2·9	82	0·4	0·9	1·4
24	3·3	3·8	4·3	54	1·8	2·3	2·8	84	0·3	0·8	1·3
26	3·2	3·7	4·2	56	1·7	2·2	2·7	86	0·2	0·7	1·2
28	3·1	3·6	4·1	58	1·6	2·1	2·6	88	0·1	0·6	1·1
30	3·0	3·5	4·0	60	1·5	2·0	2·5	90	0·0	0·5	1·0

Riveted Joints in Girderwork.

The riveted joint of a tension-member may fail either by the tearing of the plate, or the shearing of the rivets, or in some special cases by the crippling of the bearing area between plate and shank of rivet.

The tearing of the plates is resisted by the tensile strength of the plate per square inch of area taken over the *net effective section* between rivet holes. When the assumed line of fracture is at right angles to the direction of the pull, the sectional area measured on this line between the rivet-holes is the net effective section. But when the line of fracture may follow a zigzag or inclined direction, the effective sectional area should be taken at about three-fourths of the area measured on such a line. The fraction is not universally applicable, but will generally meet the case of ordinary zigzag riveting.

The shearing of the rivets is resisted by the shearing strength of the rivet-material per square inch of area taken over the effective section of the rivets engaged. Assuming all the rivets to take an equal share in the work, the number of rivets engaged in resisting the total pull upon the tie will be the whole number of rivets in the case of a lap-joint such as that shown in figs. 13 and 14; while in the case of a butt joint, such as No. 2 of fig. 15, it will consist only of the rivets employed on either side of the joint. But in any large assemblage of rivets allowance should be made for unequal distribution of the work when calculating the number of rivets required for a joint.

For rivets in 'single shear,' as in the case of lap joints, and in butt joints covered by a single cover-plate, the effective section of each rivet is simply the area of its cross-section. But wherever practicable double cover-plates should be employed, placing the rivets in 'double shear,' and in this case the effective section of each rivet should be reckoned at $1\frac{3}{4}$ times the sectional area of the rivet.

In wrought-iron work the shearing strength of good rivet iron per square inch is fully equal to the tensile strength of the plates; and the effective rivet section is often made equal to the effective plate section, an allowance of 10 per cent. being sometimes added in the case of large joints.

The shearing strength of steel rivets may be taken at 24 tons per square inch, while the plates have often a tensile strength of 26 to 32 tons; and the rivet section will generally require to be somewhat greater than the plate section.

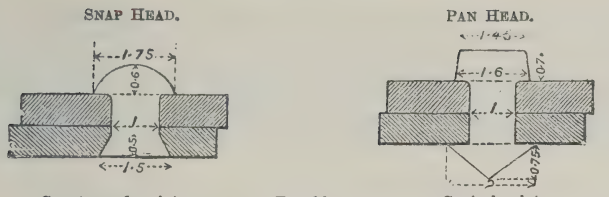
The 'efficiency' of a riveted joint is the ratio between its strength at the weakest part and the strength of the unperforated plate.

RIVETS AND RIVETING.

Rivets.

The heads (fig. 10) are usually either snap or pan. Snap heads are used generally in machine riveting. The points are countersunk, conical, or snap; the former are largely employed in iron shipbuilding, where a flush surface is required, the latter in ordinary boiler and bridge work. Conical, also called *staff* or *hammered*, points are chiefly used where the restricted space only allows of hammering.

The following figures give ordinary proportions of rivet heads in terms of diameter of shank.



Countersunk point. FIG. 10. Conical point.

If d = diameter of shank of rivet,

Length allowed for forming	pan head	= 2 to $2\frac{1}{2}d$.
" " "	snap head	= $1\frac{1}{2}d$.
" " "	countersunk head	= $1d$.
" " "	conical head	= $1\frac{1}{4}d$.

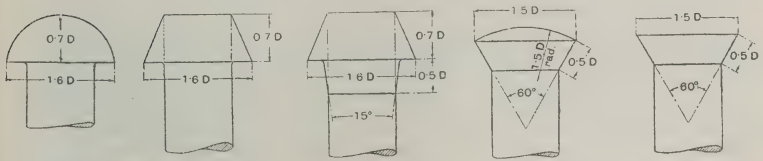
When a conical form is given under the rivet head an additional length of about $\frac{1}{4}''$ is usually allowed in making the rivet.

BRITISH STANDARD SPECIFICATION FOR DIMENSIONS OF RIVETS.*

($\frac{1}{2}$ INCH TO $1\frac{3}{4}$ INCH DIAMETER.)

(No. 275—1927.) (*Abstract.*)

(This Specification does not apply to Boiler Rivets.)



Snap Head. Pan Head. Pan Head with Tapered Neck. Rounded Countersunk Head. Flat Countersunk Head.

FIG. 11.

d = Nominal diameter of rivet.

Diameter of Rivets or Rivet Holes.

d = diameter of rivet hole; t = thickness of thickest plate.

Fairbairn's Rule, commonly used | When t is less than $\frac{1}{2}''$, $d = 2t$. When t is equal to, or greater than, $\frac{1}{2}''$, $d = 1\frac{1}{2}t$.

With rivets so proportioned the total number in single shear, such as lap, single cover and grouped, joints will, when the resistance to shearing is taken at 4 tons and bearing at 5 tons, depend on the number required for shearing, in $\frac{1}{2}''$ plates and over $\frac{1}{2}''$, and for bearing with plates under $\frac{1}{2}''$ thick.

When more than 2 plates, besides the covers, are riveted together, as in flanges of girders, some engineers add $\frac{1}{16}''$ to d for each extra plate.

Unwin's Rule for Rivets: $d = 1{\cdot}2\sqrt{t}$.

Rivets of more than $\frac{3}{8}''$ diameter are rarely used.

The rivet holes are made about $\frac{1}{16}''$ larger than the rivets, to allow for expansion when heated. For ordinary work the proportions of diameter of punch to that of boister hole in the die varies from 1 : 1·15 to 1 : 1·2; the greater the difference, the less cleanly cut will the burr be, but the metal round the hole will be stronger, being less reduced and more compressed. The hole when punched is slightly conical.

It is hardly practicable to temper a punch so as to punch a great number of holes if it is of less diameter than the thickness of the plate, or even with a diameter of equal thickness. Punched

* By permission of the British Engineering Standards Association.

holes should never be less than $1\frac{1}{2}$ diameter apart or from edge to plate. For drilled holes the limit may be 1 diameter.

Pitch of rivets = distance from centre to centre of rivet. *Longitudinal pitch* = pitch in direction of stress. *Transverse pitch* = pitch at right angles to stress. *Zigzag pitch* = minimum pitch when placed zig-zag; this should slightly exceed the transverse pitch. *A pitch line* is a straight line running through the centres of a longitudinal or transverse row.

Minimum pitch = $2\frac{1}{2}$ diameters for punched or 2 for drilled holes.

Maximum longitudinal pitch in compression plates = 12 times thickness of weakest outside plate.

Strength of Rivets and Plates.

Mean tensile breaking strength of best iron, such as BBB Staffordshire, best Yorkshire or Lowmoor (and the best only should be used), is from 26·33 tons per square inch (Fairbairn) to 25·98 tons per square inch (Kirkaldy); whilst the contraction of fractured area should not be less than 40 per cent.

Forge Tests for Rivets.—When cold they should stand being hammered double, or being upset endways, a $\frac{3}{4}$″ rivet $2\frac{1}{2}$″ long being hammered down to a cylinder 1″ long, without any sign of cracking. A rivet ready heated for use should now and then be taken, allowed to cool, and treated as above, to see if they are being overheated. Much overheating is detected by the burnt, scaly appearance of the surface of shank; $\frac{3}{4}$″ or even 1″ rivets should not be over 20 minutes in the fire. When *hot* the heads should stand hammering down to less than $\frac{1}{8}$″ thick without cracking at edges; and the stalks should stand having a punch, even more than their own diameter, driven through them without cracking round the hole.

The foregoing tests are equally applicable to steel rivets.

Careful punching does not injure soft plates, but weakens hard or steely iron plates from 15 to 30 per cent.

Sharp edges of drilled holes reduce the shearing resistance of rivets, but rounding the edges of the holes increases resistance of rivet.

Drilled rivet holes reduce the strength of a plate by $1\frac{1}{2}$ per cent., and punched holes about 9 per cent. Punching and counter-sinking reduce the strength 4 per cent., but if the holes are punched, reamered, and countersunk, there is little reduction in the strength of the plate. Comparative tests of high tensile plates, connected by iron and by high tensile rivets, show that $\frac{1}{2}$-in. iron rivets worked cold in single shear have a shearing strength of 3·6 tons compared with 6·6 in the case of high-tensile steel worked cold, and 9·6 in case of the same steel worked hot. For double shear the same figures are 6·6, 13·1, and 18·4. Experiments show that the bearing pressure with high tensile steel rivets does not exceed 60 lbs. per square inch.

(Sir P. Watts.)

Riveting.

Riveting gangs consist of a holder-up, two riveters, and one or two boys for heating and supplying rivets. Rivets are heated in a portable hearth or a reverberatory furnace. Before inserting the rivet the plates round the hole should be hammered into positive contact, and the holes ascertained to be concentric.

Riveting hammers vary from 2 to 7 lbs., the holding-up hammer or 'dolly' from 10 to 40 lbs. 130 to 150 rivets is a day's work of 13 hours of a riveting gang. 360 to 720 can be put in by a riveting machine.

Riveted Joints.

The loss of strength is ordinarily, for chain-riveted joints (fig. 12), 15 per cent.; for double-riveted, 30 per cent.; for single-riveted joints, 44 per cent., as compared with the strength of the plate unpunched.

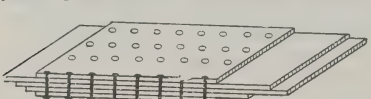


FIG. 12. Grouped Butt Joint. Chain Riveting.

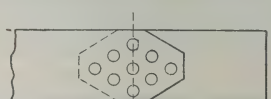


FIG. 13. Lap Joint. Lozenge Riveting.

Zigzag Riveting.—In this (fig. 14) the distance between the transverse pitch lines should be at least two-thirds the transverse pitch of the rivets.

Lozenge Riveting.—Arranging the rivets in this way, as illustrated by the lap-joint of fig. 13, and calculating the plate strength and rivet strength at each of the possible lines of fracture, it is possible to obtain a strength of joint equal to the strength of the whole plate less one rivet

hole. With the same efficiency a still better joint for a plate-tie may be made with two cover-plates of lozenge form, the maximum row of rivets being repeated on each side of the butt joint.

Cover-Plates.—In the special case last mentioned the united thickness of the cover-plates will have to be considerably greater than that of the main tie, and will be determined by the net section of cover taken through the maximum row of rivets. In chain riveting generally it would be theoretically sufficient to make the united cover thickness equal to the plate thickness, but in practice it is often made somewhat greater, both in single and double covers. If there is

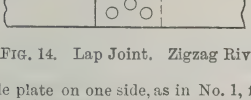


FIG. 14. Lap Joint. Zigzag Riveting.

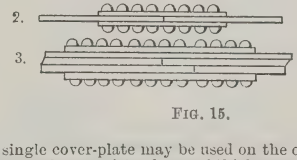


FIG. 15.

a whole plate on one side, as in No. 1, fig. 15, a single cover-plate may be used on the other side equal in thickness to the jointed plate. When the member consists of several thicknesses of plate, as in No. 3, they may with advantage be all jointed between one pair of cover-plates, arranging the several layers to break joint one with another, and leaving between every two joints a space containing a sufficient number of rivets to take up the pull of one layer of plates.

STEEL ROOF FRAMING AND COVERING.

Some points to be considered in designing steel roofing :—

(a) *Nature of Covering.*—If of **corrugated sheeting,** either ridged or curved type may be adopted ; in the latter the joints near crown of roof lie nearly horizontal, and leakage through rain being blown in must not be overlooked. A continuous covering, such as tarred felt or asphalt laid on timber boarding, is applicable to ridged or curved types.

With slates or tiles, the ridged type (except in very large spans) is necessary.

(b) *Rise of Roof.*—Where this is not controlled by architectural requirements, a rise of about $\frac{1}{5}$ span will generally prove most economical. Where water-tightness is essential, $\frac{1}{4}$ rise may be adopted ; a rise of even $\frac{1}{6}$ may be used with slates or tiles. In very exposed positions a low rise of $\frac{1}{8}$ has been provided, but here leakage is likely to prove troublesome.

(c) *Lighting and Ventilation in Roof.*—Skylights can be arranged in any position on the slope of a ridged roof, or set low down on curved roofs. Lights are often combined with ventilators at the centre or ridge of the roof. Ventilation at the ridge may consist merely in slightly raising the ridge cap above the sheets ; by the use of special ridge sheeting (often curved) supported on raised purlins ; or where greater ventilation is desired, the cover may be raised to a considerable height and louvres provided for weather-tightness. See sketch examples.

(d) *Temporary Structures.*—Here the framing and covering can be designed as light as possible, and factor of safety reduced to about 3, since allowance for rusting and general deterioration need not be considered. The self-supporting type of curved roof should be considered in this case.

(e) *Special Headroom* generally requires a special type, as the main tie cannot economically be raised much over $\frac{1}{10}$ rise for ridged or curved trusses of ordinary design. Two suggestions are given in sketch examples to meet this requirement.

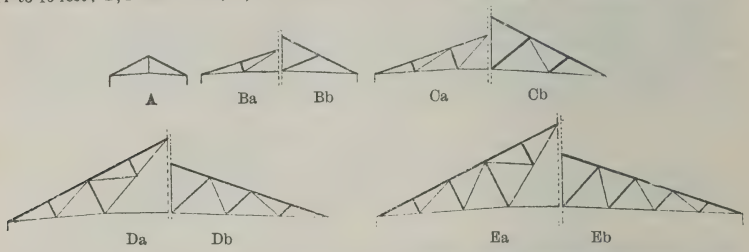
Struts, or members in compression, should be kept as short in length as possible. The strength of *ties* to resist tension is independent of their length, and no consideration on this point is necessary, except that relating to the carrying of their own weight.

The Purlins (or horizontal bearers which carry the weight of the roof covering, and the wind and snow load on the same, and transmit these loads to the roof trussing) should rest, wherever possible, on the rafters close to the points where the struts and ties meet, and stiffen the same ; loads applied between these junction points (usually termed 'joints') may induce considerable *bending stresses* which must be carefully allowed for, generally at considerable expense, owing to the large increase in section which may be necessary.

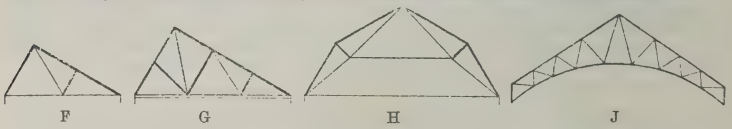
The unsupported length of the Rafter, that is to say the length between the joints, should not exceed 7 to 8 feet where the purlins are fixed close to the joints ; where purlins occur between the joints, this length should be considerably reduced, or the section of the rafter increased to resist the bending stresses.

In the brief notes accompanying the outline types of roofs shown here, the application of some of the preceding hints will be evident.

For ordinary conditions the types shown below are suitable up to the following spans :—A, 14 to 16 feet; B, 25 to 30 feet; C, 45 to 50 feet; D, 55 to 60 feet; and E, 70 to 80 feet.

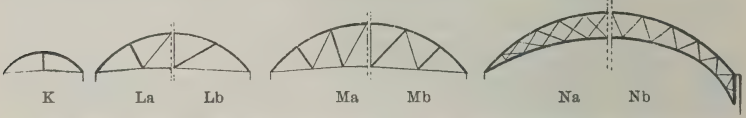


The two rises indicated in B to E are $\frac{1}{4}$ span and $\frac{1}{5}$ span. Where special conditions do not arise, the form of bracing should be decided by considerations of length and inclination of the several struts, and convenience for designing efficient junctions between the struts and ties.



Types of ridged roofs designed to provide special lighting, such as is required in spinning and weaving factories, are shown in F and G ; the long slope is covered with sheets or slates, and the short, upright slope glazed and turned towards the north, thereby ensuring efficient light free from direct sunshine. These types are frequently called 'Saw Tooth' or 'Northern Light' roofs, and are suitable for spans of 18 to 20 ft., and 25 to 30 ft. respectively.

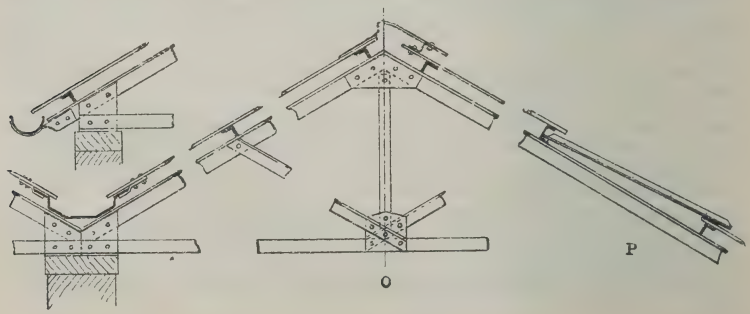
Types H and J show a high rise, with main tie raised to give increased headroom ; the former may frequently be adopted for factories, the lower slopes being glazed and the upper slopes sheeted or slated.



The curved types K, L, and M, are similar to A, B, and C, but arranged for a curved sheet covering, and may be adopted for similar spans.

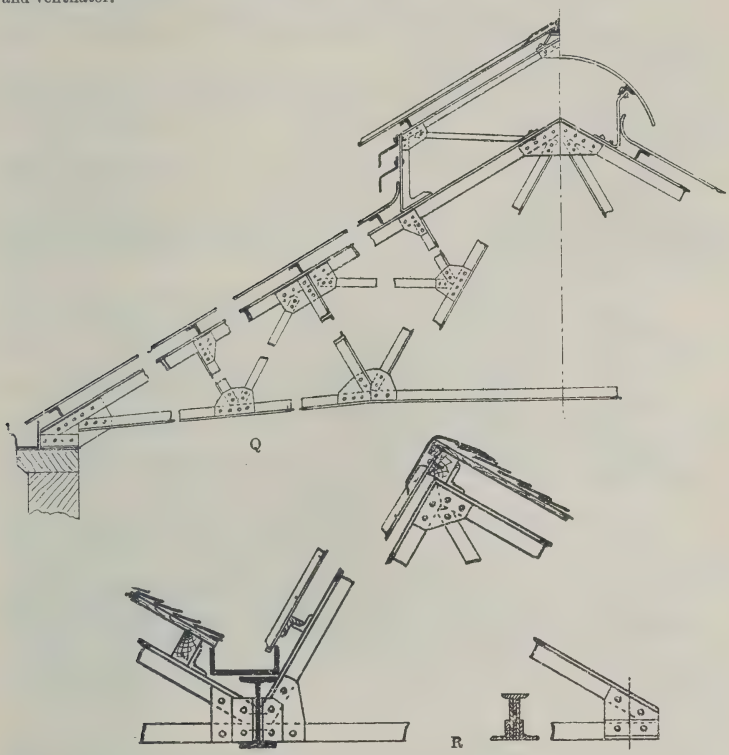
Types N are in reality curved lattice girders, and may be designed for large spans such as are required for markets, railway stations, public halls, or the like.

Sketch Details.—Sketch O suggests details for a ridged roof, type Bb ; on left two spans meet and a valley gutter is provided, and above this a sketch of an eaves gutter. At the apex, or



ridge an ordinary ridge cap is shown on the left, and on the right a plain ridge cap is slightly raised on light brackets to provide a small amount of ventilation.

Sketch P indicates a skylight in the slope of a roof. Sketch Q gives suggestions for type Da. At apex a simple ventilator is shown on the right, and a larger louvred ventilator on the left. In the latter arrangement the straight cover may be glazed, thus forming a combined skylight and ventilator.



Sketch R is that for a 'Northern Light' roof, types F or G. A valley detail (showing this part supported on a rolled steel beam) with valley gutter is given, also an eaves shoe detail. The long slope is covered with timber and slates, and the short slope is glazed.

Corrugated Sheets and Fixings.

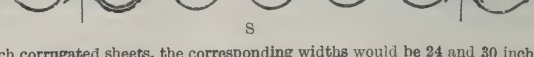
Sheets can be procured in lengths from 5 to 10 feet, and in thicknesses from No. 16 to No. 26 B.W.G.; for special or very temporary purposes, No. 28 B.W.G. might be adopted.

Corrugations from 2 inches width in light sheets to 6 inches in heavy sheets may be got but for usual conditions the following sizes will meet all requirements:—

5–5	inch	corrugated	sheets	(depth of corrugations	1¼	in.)
6–5	"	"	"	("	"	") See Sketch S.
8–3	"	"	"	("	"	¾ ")
10–3	"	"	"	("	"	")

For ordinary conditions, a side lap of one corrugation (see right-hand of Sketch S) would be adopted, but for special water-tightness or stiffness, two corrugations lap could be given (see left

hand of Sketch S). The effective width of the above sheets would be 25 inches or 30 inches, for single lap, and 20 inches or 25 inches for double lap.



S

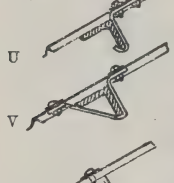
In the 3-inch corrugated sheets, the corresponding widths would be 24 and 30 inches for single lap, and 21 and 27 inches for double lap.

The end lap is generally given at 6 inches.

Sheets should be connected together by means of galvanised rivets or bolts and washers: these would be placed about 9 inches apart along the side laps, and at every second corrugation in the end laps, being in all cases placed on *top* of the corrugations to check leakage. For *double* corrugation side lap, place rivets or bolts in edge of outer sheets 6 inches apart, and 12 inches apart in inner corrugation.

Five-inch corrugations should always be adopted for No. 16 B.W.G. sheets, and may be used for No. 18 or 20 B.W.G.: 3-inch corrugations may be used for 18 or 20 B.W.G., and always for No. 22 B.W.G. and upwards.

In very exposed positions, and particularly with light gauge sheets, Wind Ties of flat or light angle section are sometimes carried along the length of the roof, and held in position by means of the purlin bolts. A line of these wind ties near each eaves level will considerably strengthen the roofing; and it is sometimes also desirable to add ties over some of the intermediate horizontal joints in the roof sheets, to further resist excessive winds.



T

U

V

W

X

In addition to connecting the roof sheets together, they must be securely fixed to the roof purlins, and suggestions for doing this are here indicated :—

Sketch T, simple hook bolt, with nut and washer outside, connecting sheets to steel angle purlin.

Sketch U, similar to above, but formed of flat clip bolted to sheets.

Sketch V, a more secure form of flat clip carried round purlin and connected to sheets by bolts and washers at ends.

Sketch W, galvanised nail or wood screw and washer, for simple connection to timber purlin.

Sketch X, flat stirrup carried round timber purlin; generally made with slight clearance in width, to admit of adjustment due to change of temperature in the sheet covering.

The washers used with the bolts are sometimes formed of lead, or of some special form, to check leakage at the bolt holes. Red lead putty, or rope yarn soaked in red lead paint, is sometimes specified to be placed under the nut or washer, but such special treatment is seldom justifiable, or really necessary.

Corrugated Sheets are generally galvanised, but in localities near the sea, or in manufacturing districts near chemical works where acid fumes are likely to be given off, the galvanising quickly disappears, and the sheets become eaten through and spoiled. In such conditions the galvanised sheets should receive two good coats of metallic oxide paint shortly after erection, or plain corrugated sheets used, these being well tarred and sanded after erection, and at such intervals as may be necessary to efficiently protect the sheets from corrosion.

Self-Supporting Roofing.

This term is generally applied to a type of curved roofing in which the curved rafters of the trussing are dispensed with, their place being supplied by the corrugated sheets themselves.

In the smaller spans, say up to 15 to 20 feet, the roof sheets are connected to timber or steel eaves beams, and the bracing (generally as types K or L) bolted direct to the curved roof sheets.

In the larger spans angle purlins are securely clipped to the curved sheets and extend for the whole length of the roof, and the bracing is bolted to these purlins.

By this method of construction, and the use of corrugated roof sheets of sufficient strength to serve the double purpose of acting as a curved rafter, and as a weather protection, spans over 55 feet have been erected. It will not, however, as a rule be found economical to exceed about 35 feet span with this type of roofing; and as the life of this structure depends on the strength of the covering, it is seldom justifiable to provide self-supporting roofs for permanent use, as this would entail great watchfulness and cost in protecting the sheeting against corrosion or decay.

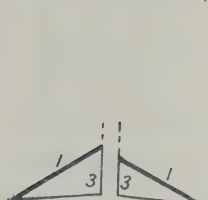
The curved form of covering must be used for this type of roof, and for the larger spans the sheets may be arranged to *break joint* in the longitudinal joints; greater stiffness may be obtained in the complete sheeting by this arrangement of joints. The lower sheets of the roof, which have to resist a much greater strain than the upper ones, are sometimes made of a heavier gauge. Special attention is also necessary to ensure sufficient strength when bolting or riveting the several sheets together.

Ridged Roof Truss Multipliers.

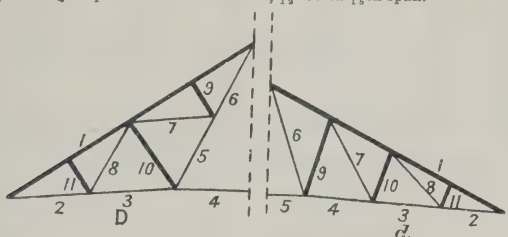
The determination of the stresses in the several members of any roof truss is usually a comparatively simple matter, although a considerable amount of time may sometimes be required to carry out this operation. Where any of the following types of trusses can be adopted, the above calculations can be much simplified and shortened.

For special water-tightness with slate or tile covering, one-third rise (shown by capital letters) should be adopted; one-fourth rise is usual for corrugated sheet covering.

Rise of rafters, $\frac{1}{3}$rd and $\frac{1}{4}$th span. Rise of main ties, $\frac{1}{12}$th and $\frac{1}{15}$th span.

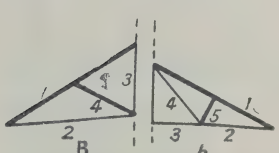


Spans to 15 ft.

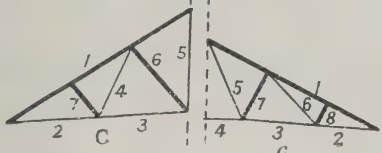


Spans, 46 to 60 ft.

Compression shown by heavy lines. Tension shown by light lines.



Spans, 16 to 30 ft.



Spans, 31 to 45 ft.

In spans up to 60 feet the rafter is usually kept the same section throughout the full length; the table gives only the max. stress multiplier for rafters.

The max. practical span for any type of truss is indicated by multiplying the numbers of sections formed in the rafters by the joints in the same by $7\frac{1}{2}$ feet. Thus Type B gives $4 \times 7\frac{1}{2} = 30$ feet max. span, and Type D $= 8 \times 7\frac{1}{2} = 60$ feet max. span.

Purlins are assumed to be set at the joints; if not, allowance must be made for the extra stress due to the bending tendency produced.

It is also assumed that the trusses are fixed at both ends; if this is not so, the stress in some members may be increased by about 10 per cent.

The multipliers in the following table are calculated as for *unit load and unit span*. Thus for dead-load stress the actual dead load of truss covering, etc., is multiplied by the particular dead-load multiplier for the required member; for live-load stress, the live load from the actual normal wind pressure on one side of the truss is used along with the particular live-load multiplier. Both stresses added together give the max. stress on the member.

Member.	Type A.		Type *a*.		Type B.		Type *b*.		Type C.		Type *c*.		Type D.		Type *d*.	
	D.L.	L.L.	D.L.	L.L.	D.L.	L.L.	D.L.	L.L.	D.L.	L.L.	D.L.	L.L.	D.L.	L.L.	D.L.	L.L.
1	·51	·58	·63	·69	·76	·70	·97	1·04	·86	·85	1·08	1·25	·90	93	1·13	1·37
2	·42	·29	·57	·48	·63	·81	·87	1·12	·72	·98	·96	1·35	·76	1·06	1·12	1·46
3	·07	·05	·07	·06	·31	·34	·55	·45	·58	·63	·77	·92	·65	·81	·87	1·12
4	—	—	—	—	·24	·58	·33	·69	·12	·30	·56	·46	·42	·27	·72	·81
5	—	—	—	—	—	—	·22	·50	·41	·45	·28	·56	·26	·54	·57	·47
6	—	—	—	—	—	—	—	—	·21	·50	·19	·43	·36	·81	·27	·56
7	—	—	—	—	—	—	—	—	·14	·34	·22	·50	·10	·26	·19	42
8	—	—	—	—	—	—	—	—	—	—	·15	·31	·10	·26	·16	·36
9	—	—	—	—	—	—	—	—	—	—	—	—	·10	·25	·22	·50
10	—	—	—	—	—	—	—	—	—	—	—	—	·21	·50	·17	·37
11	—	—	—	—	—	—	—	—	—	—	—	—	·10	·25	·11	·25

D.L. = Dead Load. L.L. = Live Load.

GALVANISED CORRUGATED IRON SHEETS. Approximate Weight.

(Sheets 26 inches in width; 8–3 inch corrugations.)

Length of Sheet in Feet.						
5	6	7	8	9	10	B. W. G.
Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	
33½	40	46½	53½	60	66½	16
26¾	32	37¼	42¾	48	53¼	18
21¼	25½	29¾	34	38¼	42½	20
16¾	20	23¼	26½	30	33½	22
13¼	16	18¾	21¼	24	26¾	24
10½	12½	14½	16½	18¾	20¾	26
9¼	11¼	13	15	16¾	18¾	28

Approximate Number of Sheets per Ton.

Length of Sheet in Feet.						
5	6	7	8	9	10	B. W. G.
67	56	48	42	37	34	16
84	70	60	52	46	42	18
105	88	75	66	58	53	20
134	112	96	84	74	67	22
168	140	120	105	93	84	24
216	180	154	135	120	108	26
240	200	171	150	133	120	28

Approximate weight per 100 superficial feet of Galvanised Corrugated Straight Iron Sheets when fixed on roof.

18	20	22	24	26 W.G.
cwts. qr. lbs.	cwt. qrs. lbs.	cwt. qrs. lbs.	cwt. qr. lbs.	cwt. qr. lbs.
2 1 21	1 3 21	1 2 7	1 1 7	1 0 7

One ton of Galvanised Corrugated Iron Sheets will cover when fixed, and after allowing for laps, the undermentioned areas :—

16	18	20	22	24	26	28	B.W.G.
600	800	1020	1280	1530	2100	2250	Square Feet.

It is a bad practice to fasten zinc-iron sheets together with iron or steel rivets. These soon rust if not painted, and the rust penetrates through the edges of the iron encircling the rivets, and thus proves disastrous. Only zinc, or zinc-covered rivets, should be so used.

The great objection to iron as a roofing material is the drip from the moisture which condenses on the underneath side of it.

PAINTING FOR GALVANISED IRON ROOFS.

Paint adheres badly to new galvanised iron, but after a few years it will do so readily and will add greatly to its life. In order to make paint adhere to new galvanised iron, brush it over with the following mixture : Chloride of Copper, 1 part ; Nitrate of Copper, 1 part ; Salammoniac, 1 part ; Water, 64 parts, to which is added 1 part of Hydrochloric Acid. This mixture turns the galvanised iron black ; it should be left for at least 12 hours before being painted.

It is stated that paint will adhere to galvanised iron if the surface is washed with vinegar before painting.

(See also ' Painting,' Section XLII.)

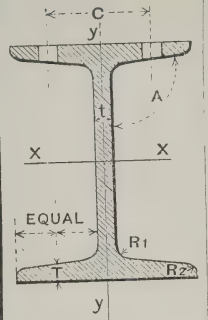
VARNISH FOR GALVANISED IRON ROOFS.

Heat 100 gallons of tar to a low boiling point, and add 100 lbs. of fresh slaked lime, sifted over the top and then worked down. Boil this mixture until it becomes pasty. Let it settle for a few minutes and then add 20 lbs. of tallow and 5 lbs. of powdered resin. Stir until thoroughly mixed and all ingredients dissolved ; then allow to cool. The mixture should not be raised to a higher temperature than 100 degs. F. If too thick, it can be thinned down with paraffin or naphtha.

TRANSVERSE STRENGTH OF CORRUGATED IRON.

If W = breaking weight in tons (distributed) ; L = unsupported length of sheet in inches ; t = thickness of sheet in inches ; b = breadth of sheet in inches ; d = depth of corrugations in inches:

$$W = \frac{44{\cdot}6\ t\ b\ d}{L}$$



For	Angle A
B.S.B.'s	98°
D.L.B.'s	98°
A.S.B.'s	99° 27′ 44″

FIG. 16.

Reference Mark.	Size Ins.	Weight per Foot Lbs.	DIAGRAM. Web t	Flange T	Radius R1	Radius R2	Area Sq. Ins.	Moments of Inertia. About X–X	Moments of Inertia. About Y–Y	Radii of Gyration Inches. About X–X	Radii of Gyration Inches. About Y–Y	Section Modulus. About X–X	Centres of Holes. c Ins.	Reference Mark.
B S B 30	24 × 7½	100	·6	1·07	·7	·35	29·4	2654	6·92	9·5	1·5	221·1	4·5	B S B 30
,, 29	20 × 7½	89	·6	1·01	·7	·35	26·17	1670	62·63	7·99	1·54	167·0	4·5	,, 29
A S B 29A	20 × 6½	65	·498	·79	·6	·3	19·12	1178	27·40	7·85	1·20	117·8	3·5	A S B 29A
B S B 28	18 × 7	75	·55	·928	·65	·325	22·06	1149	47·04	7·21	1·46	127·6	4·0	B S B 28
A S B 28A	18 × 6	55	·466	·691	·56	·276	16·18	806·3	20·84	7·06	1·13	89 59	3·5	A S B 28A
B S B 27	16 × 6	62	·55	·847	·65	·325	18·23	725·7	27·08	6·31	1·21	90·71	3·5	B S B 27
,, 26	15 × 6	59	·5	·88	·6	·3	17·35	628·9	28·22	6·02	1·27	83·85	3·5	,, 26
,, 25	15 × 5	42	·42	·647	·52	26	12·35	428·	11·81	5·88	·978	57·06	2·75	,, 25
,, 24	14 × 6	57	·5	·873	·6	3	16·76	532·9	27·96	5·63	1·29	76·12	3·5	,, 24
,, 23	14 × 6	46	·4	·698	·5	·25	13·53	440·5	21·6	5·7	1·26	62·92	3·5	,, 23
,, 22	12 × 6	54	·5	·883	·6	·3	15·88	375·5	28·3	4·86	1·33	62·58	3·5	,, 22
,, 21	12 × 6	44	·4	·717	·5	·25	12·94	315·3	22·27	4·93	1·31	52·55	3·5	,, 21
D L B 20A	12 × 5	39	·44	·664	·54	·37	11·47	260·9	12·16	4·77	1·03	43·48	2·75	D L B 20A
B S B	12 × 5	32	·35	·55	·45	·225	9·41	220·	9·753	4·83	1·01	36 66	2·75	B S B 20
,, 19	10 × 8	70	·6	·97	·7	·35	20·6	344·9	71·67	4·09	1·86	68·98	4·75	,, 19
,, 18	10 × 6	42	·4	·736	·5	·25	12·35	211·5	22·95	4·13	1·36	42·3	3·5	,, 18
D L B 17A	10 × 5	35	·42	·652	·52	·26	10·29	167·2	11·89	4·03	1·07	33·45	2·75	D L B 17A
B S B 17	10 × 5	30	·36	·552	·46	·23	8·82	145·6	9·79	4·06	1·05	29·12	2·75	B S B 17
A S B 17B	10 × 4·66	25	·3	·491	·41	·19	7·353	122·7	6·778	4·08	·960	24·53	2·5	A S B 17B
B S B 16	9 × 7	58	·55	·924	·65	·325	17·06	229·5	46·3	3·66	1·64	51·0	4·0	B S B 16
D L B A	9½ × 3½	21·5	·34	·453	·44	·22	6·324	83·41	3·446	3·63	·738	18·03	2·0	D L B 15A
B S B 15	9 × 4	21	·3	·46	·4	·2	6·176	81·1	4·2	3·62	·824	18·02	2·25	B S B 15
,, 14	8 × 6	35	·44	·597	·54	·27	10·29	110·5	17·95	3·27	1·32	27·62	3·5	,, 14
,, 13	8 × 5	28	·3	·575	·45	·225	8·24	89·32	10·26	3·29	1·11	22·33	2·75	,, 13
,, 12	8 × 4	18	·28	·402	·38	·19	5·294	55·69	3·578	3·24	·822	13·92	2·25	,, 12
,, 11	7 × 4	16	·25	·387	·35	·175	4·706	39·21	3·414	2·88	·851	11·2	2·25	,, 11
,, 10	6 × 5	25	·41	·52	·51	·255	7·35	43·61	9·116	2·43	1·11	14·53	2·75	,, 10
,, 9	6 × 4½	20	·37	·431	·47	·235	5·88	34·62	5·415	2·42	·959	11·54	2·5	,, 9
,, 8	6 × 3	12	·26	·348	·3	·18	3·53	20·21	1·339	2·39	·616	6·736	1·5	,, 8
,, 7	5 × 4½	18	·29	·448	·39	·195	5·29	22·69	5·664	2·07	1·03	9·076	2·5	,, 7
,, 6	5 × 3	11	·22	·376	·32	·16	3·235	13·61	1·462	2·05	·672	5·444	1·5	,, 6
D L 5A	4½ × 1½	10	·32	·48	·32	·16	2·941	9·275	·413	1·78	·375	3·905	—	D L B 5A
B S B 5	4½ × 1½	6·5	·18	·325	·28	·14	1·912	6·73	·263	1·87	·37	2·833	—	B S B 5
,, 4	4 × 3	9·5	·22	·336	·32	·16	2·794	7·52	1·281	1·64	·677	3·76	1·5	,, 4
D L B 3A	4 × 1½	8	·3	·388	·3	·15	2·353	5·328	·324	1·50	·371	2·664	—	D L B 3A
B S B 3	4 × 1½	5	·17	·24	·27	·135	1·47	3·668	·186	1·58	·355	1·834	—	B S B 3
,, 2	3 × 3	8·5	·2	·332	·3	·15	2·5	3·787	1·262	1·23	71	2·524	1·5	,,
D L B 1A	3½ × 1½	6	·25	·344	·25	·125	1·765	3·086	·183	1·32	·322	1·763	—	D L B 1A
B S B 1	3 × 1½	4	·16	·248	·26	·13	1·176	1·659	·124	1·18	·324	1·106	—	B S B 1

The properties of British Standard Sections in above table are published by permission of the Engineering Standards Association. (*Dorman, Long & Co. Ltd.*)

Safe Loads in Tons Uniformly Distributed.

Size.	Weight per Foot.	Span in Feet.																		Deflection Coefficient.	Weight per Foot.	Size.
		2	4	6	8	10	12	14	16	18	20	22	24	26	28	30	32	36	40			
Ins.	Lbs.																				Lbs.	Ins.
24 ×7½	100					102	92	79	69	61	55	50	46	42	9	36	34	30	27	·00078	100	24 ×7½
20 ×7½	89				94	83	69	58	52	46	41	38	34	32	29	27	26	23	20	·000937	89	20 ×7½
20 ×6½	65				69	59	49	42	37	33	29	27	25	23	21	20	18	16	15	·000937	65	20 ×6½
18 ×7	75				78	64	58	45	40	35	32	29	26	24	22	21	20	17		·00104	75	18 ×7
18 ×6	55				56	45	37	32	28	25	22	20	19	17	16	15	14	12·5	11	·00104	55	18 ×6
16 ×6	62			73	56	45	38	32	28	25	22	20	19	17	16	15	14	12		·00117	62	16 ×6
15 ×6	59			62	52	42	35	30	26		21	19	17	16	15	14	13	11		·00125	59	15 ×6
15 ×5	42			47	35	28	24	20	18	16	14	13	12	11	10	9·5	9			·00125	42	15 ×5
14 ×6	57			59	47	38	31	27	24	21	19	17	16	14	13	12				·00133	57	14 ×6
14 ×6	46			43	39	31	26	22	19	17	15	14	13	12	11	10				·00133	46	14 ×6
12 ×6	54		53	52	39	31	26	22	19	17	15	14	13	12	11	10				·00156	54	12 ×6
12 ×6	44			40	33	26	22	19	16	14	13	12	11	10	9					·00156	44	12 ×6
12 ×5	39		45	36	27	22	18	15	13	12	11	9·9	9·0	8·3	7·7	7·2				·001562	39	12 ×5
12 ×5	32		32	30	23	18	15	13	11	10		8·3	7·6	7	6·5					·00156	32	12 ×5
10 ×8	70			53	43	34	28	24	21	19	17	15	14	13						·001875	70	10 ×8
10 ×6	42			35	26	21	17	15	13	11	10	9·6	8·8	8						·001875	42	10 ×6
10 ×5	35		37	28	21	17	14	12	10	9·3	8·4	7·6	7·0	6·4						·001875	35	10 ×5
10 ×5	30		30	24	18	14	12	10	9	8	7·2	6·6	6	5·6						·001875	30	10 ×5
10 ×4·66	25		24	20	15	12	10	9	8	6·8	6·1	5·6	5·1	4·7	4·4	4·1				·001875	25	10 ×4·66
9 ×7	58		44	42	32	25	21	18	16	14	12	11	10	9·8						·00208	58	9 ×7
9½×3½	21·5	26	22	15	11	9·0	7·5	6·4	5·6	5·	4·5	4·1	3·7	3·5						·002027	21·5	9½×3½
9 ×4	21		22	15	11		7·5	4	5·6	5	4·5	4	3·7							·00208	21	9 ×4
8 ×6	35		31	28	17	14	11	9·8	8·6	7·7	7	6·3								·00234	35	8 ×6
8 ×5	28		25	18	14	11	9	8	7	6·2	5·5	5								·00234	28	8 ×5
8 ×4	18	19	17		8·7		5·8	5	4·3	3·8	3·5	3·2								·00234	18	8 ×4
7 ×4	16	15	14	9·4	7	5·6	4·7	4	3·5	3·1	·8	2·5								·00268	16	7 ×4
6 ×5	25	22	18	12		7·3	6	5·2	4·5	4										·003125	25	6 ×5
6 ×4½	20	20	14	9·6	7·2	5·8	4·8	4·1	3·6	3·2										·003125	20	6 ×4½
6 ×3	12	14	8·4	5·6	4·2	3·4	2·8	2·4	2·1	1·9										·003125	12	6 ×3
5 ×4½	18	18	11·3	7·6	5·6	4·5	3·8	3·2	2·8											·00375	18	5 ×4½
5 ×3	11	9·8	6·8	4·5	3·4	2·7	2·3	1·9	1·7											·00375	11	5 ×3
4½×1½	10		4·9	3·3	2·4	2·0	1·6	1·4												·00395	10	4½×1½
4½×1½	6·5	7	3·5	2·4	1·8	1·4	1·2	1												·00395	6·5	4½×1½
4 ×3	9·5	7·8	4·7	3·1	2·3	1·9	1·6	1·3												·00469	9·5	4 ×3
4 ×1¾	8	6·6	3·8	2·2	1·6	1·3	1·1	·95												·00469	8	4 ×1¾
4 ×1½	5	4·6	2·3	1·5	1·1	·91	·76	·65												·00469	5	4 ×1½
3 ×3	8·5	5·3	3·2	2	1·6	1·2		·9												·00625	8·5	3 ×3
3½×1½	6	4·4	2·2	1·5	1·1	·88	·73	·63												·005357	6	3½×1½
3 ×1½	4	2·8	1·4	·92	·6	·55	·46	·39												·00625	4	3 ×1½

(*Dorman, Long & Co. Ltd.*)

BROAD FLANGE BEAMS AS STANCHIONS. Safe Loads in Tons on Stanchions for Heights of 8 to 32 Feet. (L.C.C. Formula. Ends Fixed.)										SECTION.		BROAD FLANGE BEAMS AS GIRDERS. Safe distributed Loads in Tons for Spans of 8 to 64 Feet.															
8′	10′	12′	14′	16′	18′	20′	24′	28′	32′	Approx. Size. Ins.	Wt. per Foot. Lbs.	8′	10′	12′	14′	16′	18′	20′	24′	28′	32′	36′	40′	44′	48′	56′	64′
33	30	27	24	21	18	13	—	—	—	5½ × 5½	23	8	7	5	5	4	—	—	—	—	—	—	—	—	—	—	—
36	33	30	27	24	21	18	—	—	—	6 × 6	25	10	8	6	5	5	—	—	—	—	—	—	—	—	—	—	—
45	42	38	35	32	28	25	14	—	—	6½ × 6½	31	12	10	8	7	6	6	—	—	—	—	—	—	—	—	—	—
53	49	46	42	39	35	32	25	11	—	7 × 7	35	16	13	11	9	8	7	6	—	—	—	—	*Figures in italics produce a deflection greater than span ÷ 360.*				
68	64	60	56	53	49	45	37	27	12	8 × 8	44	22	18	15	13	11	10	9	—	—	—	—					
76	73	69	65	61	57	53	46	38	25	8½ × 8½	48	—	22	18	16	14	12	11	9	—	—	—					
81	77	73	69	65	61	58	50	42	32	9 × 9	50	—	24	20	17	15	13	12	10	—	—	—					
95	91	86	82	78	73	69	60	52	43	9½ × 9½	59	—	29	25	21	18	16	15	12	—	—	—					
100	95	91	87	82	78	74	65	56	48	10 × 10	61	—	32	27	23	20	18	16	13	11	—	—	—	—	—	—	—
104	100	96	91	87	83	78	70	61	53	10½ × 10½	64	—	—	29	25	22	19	17	14	12	—	—	—	—	—	—	—
126	121	116	111	107	102	97	88	78	69	11 × 11	76	—	—	37	32	28	25	22	19	16	14	—	—	—	—	—	—
136	131	127	122	117	112	108	98	89	79	12 × 12	80	—	—	43	37	32	29	26	22	18	16	14	—	—	—	—	—
151	146	141	135	130	125	119	109	98	87	12½ × 12	90	—	—	51	44	38	34	31	25	22	19	17	15	—	—	—	—
153	148	143	137	132	126	121	110	99	88	13½ × 12	92	—	—	—	47	41	37	33	27	23	20	18	16	15	—	—	—
169	163	157	151	145	139	133	121	109	97	14 × 12	101	—	—	—	54	48	42	38	32	27	24	21	19	17	—	—	—
171	165	159	153	147	140	134	122	110	97	15 × 12	102	—	—	—	58	51	45	41	34	29	25	22	20	18	17	—	—
184	177	171	164	157	151	144	131	118	105	16 × 12	110	—	—	—	66	58	51	46	38	33	29	25	23	21	19	—	—
187	180	173	166	160	153	146	133	119	106	17 × 12	112	—	—	—	71	62	55	50	41	35	31	27	25	22	21	18	—
204	196	189	182	174	167	159	145	130	115	18 × 12	122	—	—	—	81	71	63	57	47	41	35	31	28	26	24	20	—
207	199	192	184	176	169	161	146	131	116	19 × 12	124	—	—	—	87	76	68	61	51	43	38	34	30	28	25	22	19
224	216	208	199	191	183	175	158	141	125	20 × 12	135	—	—	—	98	86	76	69	57	49	43	38	34	31	29	24	21
231	222	213	205	196	187	178	161	144	126	22 × 12	139	—	—	—	—	97	86	78	65	55	48	43	39	35	32	28	24
252	243	233	223	214	204	194	175	156	136	24 × 12	152	—	—	—	—	115	102	92	76	65	57	51	46	42	38	33	28
259	249	239	229	219	209	199	179	158	138	26 × 12	157	—	—	—	—	127	113	102	85	72	63	56	51	46	42	36	32
282	271	260	249	237	226	215	193	171	149	28 × 12	171	—	—	—	—	147	131	118	98	84	73	65	59	53	49	42	37
289	278	266	254	243	231	220	196	173	150	30 × 12	176	—	—	—	—	161	143	129	107	92	80	71	64	58	53	46	40
296	284	272	260	248	236	224	200	176	152	32 × 12	180	—	—	—	—	175	155	140	116	100	87	77	70	63	58	50	43
321	308	295	282	268	255	242	215	189	163	34 × 12	195	—	—	—	—	199	177	159	133	114	99	88	79	72	66	57	50
329	315	301	287	274	260	246	218	191	163	36 × 12	200	—	—	—	—	—	191	172	143	123	107	95	86	78	71	61	53
336	322	308	293	279	265	250	222	193	165	38 × 12	205	—	—	—	—	—	204	184	153	131	115	102	92	83	76	65	57
344	329	314	299	284	270	255	225	196	166	40 × 12	210	—	—	—	—	—	219	197	164	141	123	109	98	89	82	70	61

The above tables are as published by Messrs. R. A. Skelton & Co. (Steel & Engineering), Ltd.

TABLE SHOWING THE WEIGHTS AND DIMENSIONS OF, AND ESTIMATED SAFE PERMANENT DISTRIBUTED LOADS, IN TONS, FOR, VARIOUS COMPOUNDED ROLLED STEEL GIRDERS SUPPORTED AT BOTH ENDS.

No of. Section.	Weight per Ft. in Lbs.	Size in Inches.	Compounded of	Distributed Load, in Tons that 1 Ft. will carry, being 1-4th of the Breaking Strain.	Distributed Load, in Tons that 1 Ft. will carry, being 1-5th of the Breaking Strain.
G1 C1	144	21¼ × 12	1 G 1 and 2 5/8-in. plates	1,533	1,226
G1 C2	260	21¼ × 18	2 G 1 „ 2 5/8 „	2,654	2,123
G1 C3	377	21¼ × 24	3 G 1 „ 2 5/8 „	3,775	3,020
G1 C4	195	22½ × 12	1 G 1 „ 4 5/8 „	2,335	1,868
G1 C5	335	22½ × 18	2 G 1 „ 4 5/8 „	3,821	3,056
G1 C6	480	22½ × 24	3 G 1 „ 4 5/8 „	5,306	4,245
G1 C7	248	23¾ × 12	1 G 1 „ 6 5/8 „	3,183	2,546
G1 C8	414	23¾ × 18	2 G 1 „ 6 5/8 „	5,054	4,043
G1 C9	582	23¾ × 24	3 G 1 „ 6 5/8 „	6,925	5,540
G2 C1	130	19¼ × 12	1 G 2 „ 2 5/8 „	1,279	1,023
G2 C2	227	19¼ × 16	2 G 2 „ 2 5/8 „	2,061	1,649
G2 C3	340	19¼ × 24	3 G 2 „ 2 5/8 „	3,092	2,474
G2 C4	180	20½ × 12	1 G 2 „ 4 5/8 „	2,007	1,606
G2 C5	294	20½ × 16	2 G 2 „ 4 5/8 „	2,989	2,391
G2 C6	440	20½ × 24	3 G 2 „ 4 5/8 „	4,483	3,587
G2 C7	360	21¾ × 16	2 G 2 „ 6 5/8 „	3,975	3,180
G2 C8	540	21¾ × 24	3 G 2 „ 6 5/8 „	5,962	4,770
G3 C1	98	17 × 10	1 G 3 „ 2 ½ „	808	647
G3 C2	178·5	17 × 14	2 G 3 „ 2 ½ „	1,353	1,083
G3 C3	266	17 × 20	3 G 3 „ 2 ½ „	1,986	1,589
G3 C4	132	18 × 10	1 G 3 „ 4 ½ „	1,228	983
G3 C5	271	19 × 14	2 G 3 „ 6 ½ „	2,505	2,004
G3 C6	394	19 × 20	3 G 3 „ 6 ½ „	3,618	2,894
G5 C1	77	16 × 10	1 G 5 „ 2 ½ „	618	495
G5 C2	127	16 × 12	2 G 5 „ 2 ½ „	907	725
G5 C3	191	16 × 18	3 G 5 „ 2 ½ „	1,360	1,088
G5 C4	111·25	17 × 10	1 G 5 „ 4 ½ „	1,014	811
G5 C5	210	18 × 12	2 G 5 „ 6 ½ „	1,813	1,450
G5 C6	315	18 × 18	3 G 5 „ 6 ½ „	2,720	2,176
G6 C1	142	14½ × 15	2 G 6 „ 1 ½ „	898	718
G6 C2	93	15 × 10	1 G 6 „ 2 ½ „	684	547
G6 C3	164	15 × 14	2 G 6 „ 2 ½ „	1,137	910
G6 C4	245	15 × 20	3 G 6 „ 2 ½ „	1,667	1,334
G6 C5	127	16 × 10	1 G 6 „ 4 ½ „	1,056	845
G6 C6	212	16 × 14	2 G 6 „ 4 ½ „	1,633	1,306
G6 C7	326	16 × 20	3 G 6 „ 4 ½ „	2,370	1,896

LIMITING DIMENSIONS OF ROLLED STEEL.

Dimensions. O, Ordinary. M, Maximum.	Flat.		Round.		Square.		Angle.		Tee.		Joists.		Channel.	
	O	M	O	M	O	M	O	M	O	M	O	M	O	M
Length in feet	40	60	24	60	24	60	50	60	50	60	36	50	36	60
Width in inches	18	18	4	4	4	4	6×6	8×8	5×3	6×3	24×7½	24×7½	15×4	15×4
Thickness in ins.	1	1	—	—	—	—	7/8	1	3/8	½	½	5/8	5/8	5/8

BRITISH STANDARD SECTIONS.*

(No. 6—1924.) ***(Abstract.)***

The British Engineering Standards Association list of Rolled Steel Sections for structural purposes comprises seven different sections, viz. (1) Equal Angles, (2) Unequal Angles, (3) Bulb Angles, (4) Bulb Plates, (5) Channels, (6) Beams, and (7) T Bars.

The following are abbreviated tables showing some of the standard dimensions recommended.

Attention is directed to the footnotes on pp 241, 242.

*By permission of the British Engineering Standards Association.

BRITISH STANDARD SECTIONS.

EQUAL ANGLES.

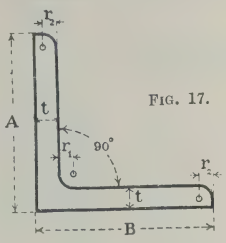


FIG. 17.

BULB PLATES.

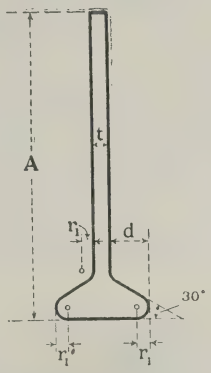


FIG. 21.

BULB ANGLES.

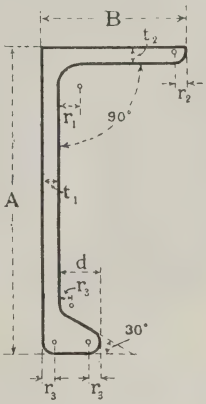


FIG. 20.

UNEQUAL ANGLES

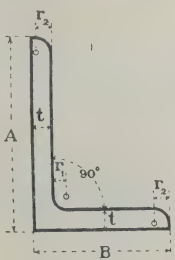


FIG. 18.

T BARS.

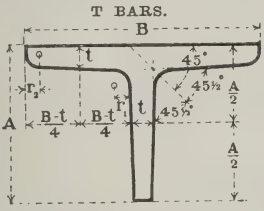


FIG. 19.

BEAMS.

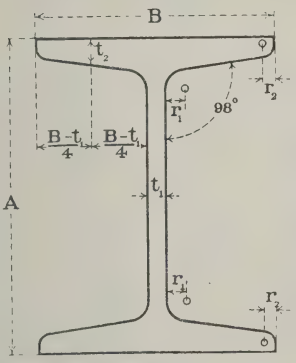


FIG. 23.

CHANNELS.

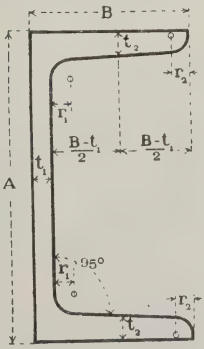


FIG. 22.

EQUAL ANGLES (Fig. 17).

Size. A × B	Standard Thickness. Minimum. t	Mean. t	Maximum t	Maximum Thickness rolled.	Minimum Thickness rolled.	Radii. Root. r_1	Toe. r_2
1 × 1	0·125	—	0·250	0·300	0·125	0·18	0·13
1½ × 1½	0·125	—	0·250	0·350	0·125	0·21	0·15
2 × 2	0·175	—	0·300	0·400	0·175	0·24	0·17
2½ × 2½	0·250	0·375	0·500	0·500	0·200	0·27	0·19
3 × 3	0·250	0·375	0·500	0·525	0·250	0·30	0·21
3½ × 3½	0·300	0·425	0·500	0·575	0·275	0·33	0·23
4 × 4	0·300	0·425	0·500	0·625	0·300	0·36	0·25
6 × 6	0·450	—	0·625	0·775	0·425	0·48	0·34
8 × 8	0·550	—	0·750	0·950	0·550	0·60	0·42

In the complete table sixteen sizes are given. All sizes are in inches.

UNEQUAL ANGLES (Fig. 18).

Size. A × B	Standard Thickness. Minimum t	Mean. t	Maximum t	Maximum Thickness rolled.	Minimum Thickness rolled.	Radii. Root. r_1	Toe. r_2
2 × 1½	0·175	—	0·300	0·375	0·150	0·23	0·16
3 × 2	0·250	0·375	0·500	0·500	0·200	0·27	0·19
3½ × 2½	0·250	0·375	0·500	0·525	0·250	0·30	0·21
4 × 3	0·300	0·425	0·500	0·575	0·275	0·33	0·23
5 × 4	0·375	—	0·500	0·650	0·325	0·39	0·27
6 × 4	0·375	—	0·500	0·700	0·350	0·42	0·29
8 × 4	—	—	0·625	0·775	0·425	0·48	0·34
9 × 4	—	—	0·650	0·825	0·450	0·51	0·36
10 × 4	—	—	0·675	0·850	0·475	0·54	0·38

In the complete table twenty sizes are given. All sizes are in inches.

T BARS (Fig. 19).

Size. Flange × Web. B × A	Standard Thickness. Minimum. t	Mean t	Maximum. t	Radii. Root. r_1	Toe. r_2
1 × 1	0·125	0·187	—	0·18	0·13
1½ × 1½	0·187	0·250	—	0·21	0·15
2 × 2	0·250	0·312	0·375	0·24	0·17
2½ × 2½	0·250	0·312	0·375	0·27	0·19
3 × 3	0·312	0·375	0·437	0·30	0·21
4 × 3	0·375	0·500	—	0·30	0·21
5 × 4	0·500	0·625	—	0·39	0·27
6 × 4	0·500	0·625	—	0·42	0·29
6 × 6	0·500	0·625	—	0·48	0·34

In the complete table ten sizes are given. All sizes are in inches.

BULB ANGLES (Fig. 20).

Size	Standard Thickness.		Max. Web Thickness rolled.	Min. Web Thickness rolled.	Radii.			Bulb.
	Web.	Flange						
A × B	t_1	t_2	t_1	t_1	r_1	r_2	r_3	d
4×2½	0·26		0·46	0·21	0·42	0·21	0·16	0·52
5×2½	0·30		0·50	0·25	0·42	0·21	0·20	0·65
6×3	0·33		0·54	0·28	0·48	0·24	0·24	0·78
7×3	0·38		0·59	0·33	0·48	0·24	0·28	0·91
8×3	0·40		0·61	0·35	0·48	0·24	0·32	1·04
9×3½	0·43		0·65	0·38	0·54	0·27	0·36	1·17
10×3½	0·45		0·67	0·40	0·54	0·27	0·40	1·30
11×3½	0·48		0·70	0·43	0·54	0·27	0·44	1·43
12×4	0·50		0·73	0·45	0·60	0·30	0·48	1·56
15×4	0·57		0·82	0·52	0·60	0·30	0·60	1·95

In the complete table eighteen sizes are given. All sizes are in inches.

BULB PLATES (Fig. 21).

Size.	Standard Thickness.	Maximum Thickness rolled.	Minimum Thickness rolled.	Radius	Bulb.
A	t			r_1	d
7	0·35	0·450	0·275	0·28	0·91
10	0·50	0·625	0·375	0·40	1·30

The above table is complete. All sizes are in inches.

CHANNELS (Fig. 22).

Size.		Calculated Weight per Foot in Lbs.	Standard Minimum Thickness.		Radii.	
			Web.	Flange.	Root.	Toe.
A	B		t_1	t_2	r_1	r_2
3	1½	4·60	0·20	0·28	0·30	0·15
4	2	7·09	0·24	0·31	0·36	0·18
5	2½	10·22	0·25	0·38	0·42	0·21
6	3	12·41	0·25	0·38	0·48	0·24
7	3	14·22	0·26	0·42	0·48	0·24
8	3½	20·21	0·32	0·52	0·54	0·27
9	3½	22·27	0·34	0·54	0·54	0·27
10	3½	24·46	0·36	0·56	0·54	0·27
12	3½	25·25	0·35	0·50	0·54	0·27
12	4	31·33	0·40	0·60	0·60	0·30
15	4	36·37	0·41	0·62	0·60	0·30
17	4	44·34	0·48	0·68	0·60	0·30

In the complete table eighteen sizes are given. All sizes are in inches.

NOTE.—The alterations from the original to the revised sections given in this and the preceding page will take place, particularly in the larger sizes, only as the rolls wear out, or break, and thus require replacing. For some time, therefore, it will be necessary to enquire as to sections available.

A. GIRDER SECTIONS (Fig. 23).

Reference Number.	Size.		Approx. Weight per Foot in Lbs.	Standard Thickness.		Radii.	
				Web.	Flange.	Root.	Toe.
	A	B		t_1	t_2	r_1	r_2
NBSB 1	3	1½	4	0·16	0·249	0·25	0·12
„ 2	4	1½	5	0·17	0·239	0·27	0·13
„ 3	4½	2	7	0·19	0·322	0·29	0·14
„ 4	5	2½	9	0·20	0·347	0·33	0·16
„ 5	6	3	12	0·23	0·377	0·37	0·18
„ 6	7	3½	15	0·25	0·398	0·41	0·20
„ 7	8	4	18	0·28	0·398	0·45	0·22
„ 8	9	4	21	0·30	0·457	0·45	0·22
„ 9	10	4½	25	0·30	0·505	0·49	0·24
„ 10	12	5	30	0·33	0·507	0·53	0·26
„ 11	13	5	35	0·35	0·604	0·53	0·26
„ 12	14	5½	40	0·37	0·627	0·57	0·28
„ 13	15	6	45	0·38	0·655	0·61	0·30
„ 14	16	6	50	0·40	0·726	0·61	0·30
„ 15	18	6	55	0·42	0·757	0·61	0·30
„ 16	20	6½	65	0·45	0·820	0·65	0·32
„ 17	22	7	75	0·50	0·834	0·69	0·34
„ 18	24	7½	90	0·52	0·984	0·73	0·36

B. HEAVY BEAMS AND PILLARS (Fig. 23).

NBSHB 1	4	3	10	0·24	0·347	0·37	0·18
„ 2	5	4½	20	0·29	0·513	0·49	0·24
„ 3	6	5	25	0·33	0·561	0·53	0·26
„ 4	8	6	35	0·35	0·648	0·61	0·30
„ 5	9	7	50	0·40	0·825	0·69	0·34
„ 6	10	6	40	0·36	0·709	0·61	0·30
„ 7	10	8	55	0·40	0·783	0·77	0·38
„ 8	12	8	65	0·43	0·904	0·77	0·38
„ 9	14	8	70	0·46	0·920	0·77	0·38
„ 10	16	8	75	0·48	0·938	0·77	0·38
„ 11	18	8	80	0·50	0·950	0·77	0·38

The above tables are complete.

All sizes are in inches.

NOTE.—For a considerable time the majority of the above sections will not be available, owing to the cost of providing the sets of rolls required for each size, and it will be necessary to utilise the present sections. For particulars of these present available sections of beams, a reference should be made to the table on p. 235, where all present British Standard Beams are marked BSB 1 to 30.

Strengths of Different Sections as Beams and as Struts.

The tables given on pp. 243 and 244 are calculated by Fidler's formula for struts with fixed ends, assuming an ultimate strength of 60,000 lbs. per sq. in. and allowing a factor of safety of 5.

BRITISH STANDARD EQUAL-SIZED ANGLES.

SAFE LOADS AS STRUTS; FIXED ENDS.

Size in Inches.	Thickness in Inches.	Weight per Foot in Pounds.	Area of Cross-Section. Sq. Ins.	Safe Loads in Tons (Struts) for least axis of Gyration. Length. 4 ft.	6 ft.	8 ft.	10 ft.	12 ft.	14 ft.
2 × 2	·300	3·77	1·11	3.0	1·6	—	—	—	—
2¼ × 2¼	·300	4·28	1·26	3·9	2·2	1·5	—	—	—
2½ × 2½	⅜	5·89	1·73	6·0	3·5	2·3	1·6	—	—
2½ × 2½	½	7·65	2·25	7·7	4·6	3·0	2·1	1·4	—
2¾ × 2¾	⅜	6·53	1·92	7·3	4·6	3·0	2·1	1·5	—
2¾ × 2¾	½	8·50	2·50	9·5	6·0	3·9	2·8	2·0	1·4
3 × 3	⅜	7·18	2·11	8·5	5·7	3·8	2·7	2·0	1·4
3 × 3	½	9·36	2·75	11·1	7·5	4·9	3·6	2·6	1·9
3½ × 3½	·425	9·50	2·80	12	9·0	6·4	4·6	3·5	2·7
3½ × 3½	½	11·1	3·25	15	11	8·0	5·7	4·4	3·4
4 × 4	·425	10·9	3·22	15	12	8·6	6·3	4·7	3·8
4 × 4	½	12·8	3·75	17	14	10	7·3	5·5	4·5
4½ × 4½	½	14·5	4·25	20	17	13	10	7·7	6·2
5 × 5	½	16·2	4·75	23	20	16	13	9·6	7·7
6 × 6	½	24·2	7·11	36	32	28	23	19	15

(R. A. Skelton & Co.)

BRITISH STANDARD CHANNELS AS STRUTS.

Size in Inches. Width. Depth.	Weight per Foot run. Lbs.	Area of Cross Section. Sq. Ins.	Safe Loads in Tons (Struts). If liable to failure in either direction. Lengths. 6 ft.	8 ft.	10 ft.	12 ft.	If braced against failure in weaker direction. Lengths. 6 ft.	8 ft.	10 ft.	12 ft.	14 ft.	16 ft.
3 × 1½	5·27	1·55	—	—	—	—	7·02	6·14	5·01	4·08	—	—
3½ × 2	6·75	1·99	5·55	—	—	—	9·43	8·62	7·66	6·43	—	—
4 × 2	7·96	2·34	6·53	4·33	—	—	11·4	10·7	9·69	8·73	7·35	—
5 × 2½	11·0	3·23	11·3	8·14	—	—	16·4	15·6	14·7	13·6	12·5	—
6 × 2½	12·0	3·54	12·1	8·74	6·23	—	18·2	17·7	16·9	16·0	15.0	—
6 × 3	14·0	4·26	17·5	13·6	10·5	—	22·0	21·4	20·4	19·6	18·3	—
6 × 3	16·3	4·79	19·4	15·0	11·5	8·77	24·7	24·1	22·8	21·8	20·5	19·3
6 × 3½	17·9	5·27	23·1	19·8	15·7	12·4	27·3	26·5	25·2	24·2	22·7	21·3
7 × 3½	20·2	5·95	25·9	22·4	17·5	13·9	31·0	30·4	29·4	28·2	27·3	25·8
8 × 3½	22·7	6·68	28·9	24·7	19·1	15·2	35·0	34·4	33·8	32·5	31·5	30·5
8 × 4	25·7	7·59	34·7	30·7	25·4	21·0	39·7	39·0	38·5	37·0	35·5	34·7
9 × 4	28·6	8·40	38·6	34·0	27·9	23·0	44·2	43·6	43·0	42·2	40·7	39·5
10 × 4	30·2	8·87	—	35·4	28·9	23·8	46·7	46·3	45·7	45·1	43·8	42·6
11 × 4	33·2	9·77	—	39·0	31·9	25·9	51·5	51·2	50·6	50·0	49·3	47·8
12 × 4	36·5	10·7	—	42·1	34·2	27·6	56·4	56·3	55·6	55·0	54·5	53·2

(R. A. Skelton & Co.)

The assumed ultimate strength of 60,000 lbs. per sq. in. is approximately 26·8 tons per sq. in. For an assumed ultimate strength of 70,000 lbs. per sq. in., or nearly 31·2 tons per sq. in., the values in the tables multiplied by ⅞ would be approximately correct. Similarly, for a factor of safety of 4, multiply the values by ¼.

British Standard Tees.

Safe Loads as Struts; fixed ends.

Size in Ins. Breadth. Height.	Thickness in Ins.	Weight per Ft. run. Lbs.	Area of Cross Section. Sq. Ins.	Safe Loads in Tons (Struts.) Lengths. 4 ft.	6 ft.	8 ft.	10 ft.	12 ft.	14 ft.
2½ × 2½	5/16	5·00	1·47	5·2	3·2	2·1	1·5	—	—
2½ × 2½	3/8	5·92	1·74	6·2	3·8	2·5	1·7	—	—
3 × 2	3/8	5·93	1·74	7·5	5·5	3·8	2·8	2·1	—
3 × 2½	3/8	6·56	1·93	8·1	5·7	3·9	2·8	2·1	—
3 × 3	3/8	7·21	2·12	8·7	6·0	4·1	2·9	2·2	—
3 × 3	7/16	8·30	2·44	10	6·9	4·7	3·4	2·5	—
3½ × 3½	3/8	8·49	2·50	11	8·5	6·0	4·2	3·3	—
3½ × 3½	7/16	9·78	2·88	13	9·8	6·9	5·0	3·8	3·0
3½ × 3½	½	11·1	3·26	14	11	7 9	5·6	4·3	3·4
4 × 3	½	11·1	3·26	15	13	10	7·6	5·7	4·6
4 × 4	½	12·8	3·76	17	14	11	7·9	6·0	4·9
5 × 3	½	12·8	3·76	19	17	15	12	9·8	7·8
5½ × 3½	½	13·7	4·02	20	18	16	13	10	8·2
5 × 4	½	14·5	4·27	21	19	16	13	10	8·3

British Standard Joists as Struts.

Size in Inches.	Weight per Foot run. Lbs.	Area of Cross Section. Sq. Ins.	Safe Load in Tons. Lengths. 6 ft.	8 ft.	10 ft.	12 ft.
3 × 3	8½	2·50	8·4	6·0	4·3	—
4 × 3	9½	2·80	9·0	6·2	4·8	—
5 × 3	11	3·24	10·0	7·1	5·1	—
7 × 4	16	4·7	18	14	11	8·1
9 × 4	21	6·18	24	18	13	10
6 × 5	25	7·35	33	29	23	19.
10 × 5	30	8·82	39	33	26	21
12 × 5	32	9·41	40	34	27	21
15 × 5	42	12·4	52	42	34	26
12 × 6	44	12·9	61	55	48	40
15·6	59	17·3	81	73	63	52

British Standard Unequal Angles as Beams.

Safe Distributed Loads: Long Side Vertical.

Maximum working stress 7½ tons per square inch.

Size in Ins.	Thickness. Ins.	Weight per Ft. run. Lbs.	Distributed Load in Tons. Maximum Stress 7½ tons per square inch. 4 ft.	5 ft.	6 ft.	7 ft.	8 ft.	9 ft.
2 × 1½	·300	3·27	·34	·27	·22	—	—	—
2½ × 2	·300	4·28	·54	·44	·36	—	—	—
3 × 2	¼	4·04	·65	·52	·44	—	—	—
3 × 2	5/16	5·89	·95	·76	·63	·54	—	—
3½ × 2½	5/16	7·18	1·4	1·1	·91	·78	·68	—
3½ × 2½	¼	9·36	1·8	1·4	1·2	1·0	·89	·79
4 × 2½	5/16	7·81	1·7	1·4	1·1	·99	·86	·77
4 × 2½	½	10·2	2·2	1·8	1·5	1·3	1·1	1·0
4 × 3	½	11·1	2·3	1·9	1·6	1·3	1·2	1·0
4½ × 3	½	11·9	2·9	2·3	1·9	1·7	1·5	1·3

(R. A. Skelton & Co.)

SECTION VII

BRIDGES AND BRIDGEWORK
(METAL)

Contributed by Henry F. Cornick, M.C., Assoc.M.Inst.C.E. (Chartered Civil Engineer), late Major, Royal Engineers

PART I
STEEL BRIDGEWORK
(pp. 247 to 275)

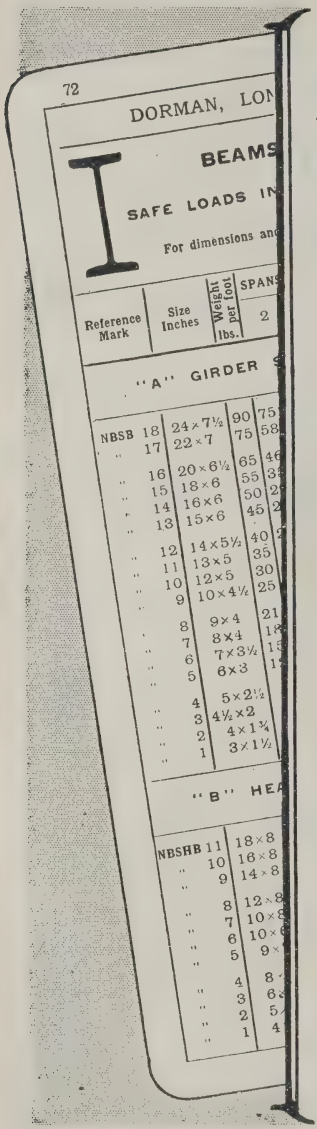
PART II
PLATE GIRDER BRIDGES
(pp. 277 to 283)

PART III
LATTICE GIRDER BRIDGES
(pp. 285 to 291)

PART IV
SWING AND BASCULE BRIDGES
(pp. 293 to 299)

PART V
ARCH BRIDGES
(pp. 301 to 303)

PART VI
SUSPENSION BRIDGES
(pp. 305 to 307)


72

DORMAN, LON

I BEAMS

SAFE LOADS IN

For dimensions and

Reference Mark	Size Inches	Weight per foot lbs.	SPANS
			2
"A" GIRDER S			
NBSB 18	24×7½	90	75
" 17	22×7	75	58
" 16	20×6½	65	46
" 15	18×6	55	3
" 14	16×6	50	2
" 13	15×6	45	2
" 12	14×5½	40	
" 11	13×5	35	
" 10	12×5	30	
" 9	10×4½	25	
" 8	9×4	21	
" 7	8×4	18	
" 6	7×3½	15	
" 5	6×3	12	
" 4	5×2½		
" 3	4½×2		
" 2	4×1¾		
" 1	3×1½		
"B" HEA			
NBSHB 11	18×8		
" 10	16×8		
" 9	14×8		
" 8	12×8		
" 7	10×8		
" 6	10×6		
" 5	9×		
" 4	8×		
" 3	6×		
" 2	5×		
" 1	4		

The Dorman Long handbook for constructional engineers

Gives full information as to the dimensions and properties of the new and old British standard sections ; tables of safe loading of beams and stanchions are given, also standard designs of connections and other data of value.

The Book will be sent upon application to the London Office to Architects, Engineers and General Contractors.

DORMAN LONG & CO., LTD.

LONDON AND MIDDLESBROUGH

LONDON OFFICE: 55 BROADWAY, WESTMINSTER, S.W.1. ALSO ASSOCIATED COMPANIES, BRANCHES AND AGENTS, AT MELBOURNE, SYDNEY, JOHANNESBURG, CAPE TOWN, DURBAN, BUENOS AIRES, CALCUTTA, WELLINGTON, CAIRO, KHARTOUM, ALEXANDRIA, KARACHI, SHANGHAI, PARIS.

Output capacity 1,750,000 *tons of steel per annum, British from raw material to finished product.*

DORMAN LONG

& Company Limited

Incorporating Bolckow, Vaughan & Company Limited

F. 247

SECTION VII

PART I

STEEL BRIDGEWORK

TYPES — ECONOMICAL SPAN — PROPORTIONS — STANDARD CLEARANCES — LIVE LOADS — BRIDGE FOUNDATIONS — PERMISSIBLE STRESSES — WIND PRESSURE — CENTRIFUGAL FORCE—MOMENTUM — IMPACT—TEMPERATURE—BEARINGS—FLOORING—CAMBER, RIVETING, ETC.

TYPES.

Bridges consist of two parts:

(*a*) The substructure, comprising abutments, piers, caissons, and other forms of foundations.

(*b*) The superstructure, which comprises the bridge structure itself.

There are three known methods of carrying weight over space—*i.e.* by means of:

(1) The beam; (2) the arch; (3) the rope.

All bridges can be reduced to one, or a combination, of these types.

(1) THE BEAM.—The strength of the beam depends on transverse bending. The theoretic stress in the extreme fibres is inconsistent with experiments, but the so-called discrepancy diminishes with the thickness of the web and the ratio of depth to span of the beam.

The principal groups of girder bridges are:

(A) Girders of uniform depth; (B) girders of uniform horizontal stress.

Group A: Parallel Girders.—In this case the shearing force is resisted by the web, and the diagram of moments forms the stress diagram.

Group B: Parabolic Girders—In this case, for uniform loading the shearing force is resisted by the flanges and the diagram of moments forms a diagram of depth.

All these types can be used in their inverted form, and it is from these forms that the Arch and Rope are evolved.

(2) THE ARCH.—For any given system of loading a theoretical arch can be designed so as to be in compression only, the neutral axis of the arch coinciding with the line of pressure; but unless it can resist bending, the arch will be in a state of unstable equilibrium. The horizontal stress varies with the rise.

(3) THE ROPE.—The shape which the loaded cable takes up is the same as the link polygon for the given load system drawn from one support to the other with a polar distance equal to the horizontal component of the pull in the cable. Stable equilibrium is obtained under changes of load. The horizontal stress varies with the dip.

Economical Span.

In bridging a large opening by a number of small spans, the economical span may be approximated in the following manner :—

Let

P = cost of one pier ; G = cost of main girders for one span erected ; n = number of small spans ; l = length of small spans ; L = total span

then

$$n = \frac{L}{l}.$$

$$\text{Cost of piers} = (n-1)\,P.$$

Cost of main girders = nG, and G may be taken as proportional to the square of the span so that,

$$G = al^2,$$

therefore

$$\text{Total cost} = C = (n-1)\,P + nal^2,$$

where a is a constant.

To get the minimum value of C, differentiate with respect to l and put it equal to zero. Therefore

$$\frac{dC}{dl} = 0.$$

But

$$C = \left(\frac{L-l}{l}\right) P + \frac{Lal^2}{l}.$$

$$= \frac{LP}{l} - P + Lal,$$

therefore

$$\frac{dC}{dl} = \frac{-LP}{l^2} + La = 0,$$

therefore

$$\frac{P}{l^2} = a,$$

or

$$P = al^2 = G.$$

From this it follows that the most economical condition is when the cost of one pier equals the cost of one main girder for one span.

If G equals cost of 100 ft. span, the result may be expressed:

$$\text{Economical span} = \frac{100\sqrt{P}}{\sqrt{G}}. \qquad (\textit{Ency. Brit.})$$

Another formula is as follows :—

Let

H = total height of each pier measured from bottom of foundation ; L = total length of bridge;

r = { price per cubic yard of masonry (or) price per ton of steel } ; n = number of piers;

then for a double line

$$n = 0{\cdot}063L\sqrt{\frac{L-10rH}{rHL}} - 1;$$

for a single line

$$n = 0{\cdot}04L\sqrt{\frac{L-10rH}{rHL}} - 1.$$

These formulæ do not apply when H is more than 98 ft.

Economical Proportions of Girders.

Plate Girders.

Let

x = depth of girder in ins.; W = weight of girder in lbs.; f = allowable working stress on gross area; t = thickness of web; L = extreme length of girder; M = total bending moment at centre (inch-pounds).

Then the weight of the web is $\frac{10}{3}$ Ltx, and that of the flanges, assuming the flange plates are the required theoretical lengths, $= \frac{10}{3}\left(1{\cdot}6\,\frac{\mathrm{ML}}{fx}\right)$,

therefore

$$\mathrm{W} = \frac{10}{3}\,\mathrm{L}tx - \frac{10}{3}\left(1{\cdot}6\,\frac{\mathrm{ML}}{fx}\right),$$

and W is a minimum when $x = 1{\cdot}27\sqrt{\frac{\mathrm{M}}{ft}}$.

This formula is based upon the assumption that no portion of the web is included in flange area.

If the resistance of web is considered, then

$$x = 1{\cdot}46\sqrt{\frac{\mathrm{M}}{ft}}.$$

If the flanges are of constant section throughout, the foregoing formulæ become

$$x = 1{\cdot}41\sqrt{\frac{\mathrm{M}}{t}}\ \text{(web neglected)};\quad x = 1{\cdot}63\sqrt{\frac{\mathrm{M}}{ft}}\ \text{(web considered)}.$$

Lattice Girders.

In trusses of the N type the maximum stiffness or minimum deflection is obtained when

$$x = \frac{d}{2}\sqrt{\frac{y+2}{y-2}} \times y,$$

where

d = length in feet between verticals; y = number of bays.

(*Sir Wm. Arrol & Co., Ltd.*)

For plate girders, the economic depth may be taken roughly as from $\frac{1}{15}$ to $\frac{1}{12}$ of the span, $\frac{1}{12}$ often being adopted; and for lattice girders should be not less than $\frac{1}{10}$ of the span, preferably $\frac{1}{8}$.

If shallower girders are necessary, the sections must be increased, so that the maximum deflection will not be greater than if the limiting ratio had not been exceeded.

Breadth of Flanges.

For plate girders this may be taken as about $\frac{1}{3}$ of the depth, or $\frac{1}{30}$ to $\frac{1}{10}$ of the span.

The width of booms in lattice girders should not be less than $\frac{1}{12}$ of the unsupported distance.

Cross Girders.—Depth should not be less than $\frac{1}{10}$ of the span, preferably $\frac{1}{8}$.

Rail Bearers should have a depth of not less than $\frac{1}{15}$ of the span.

Spans, Standard Clearances, etc.

Over-bridges carrying roads over railways are usually of the following spans (see fig. 1):

Over single line, 16 to 18 ft. Over double line, 28 to 30 ft.

Widths of over-bridges, between inside of parapets, are regulated by Act of Parliament, as follows, the roads being of the widths given for 50 yards each way:

Turnpike roads, 35 ft. Other public carriage roads, 25 ft. Private roads, 12 ft.

Under-bridges carrying roads under railway, span and headroom fixed by Act of Parliament.

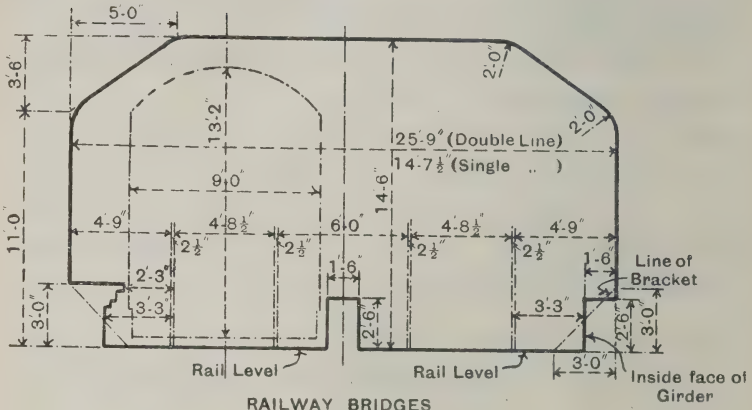
Turnpike road, clear headroom, 12 ft. at springing, 16 ft. for centre 12 ft. of arch. Where tramways cross the bridge a minimum headroom of 18 ft. should be provided.

Other public and carriage roads, clear headroom 12 ft. at springing, 15 ft. for centre, 13 ft. of arch.

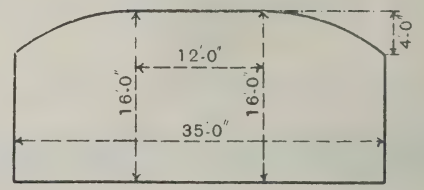
Private roads, clear headroom 14 ft. for centre breadth of 9 ft. of arch.

In all cases, under-bridges must have a parapet at least 4 ft. 6 ins. above rail level.

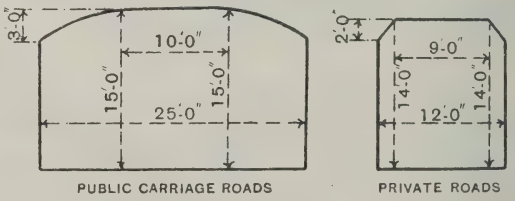
STANDARD CLEARANCES FOR BRIDGES.



RAILWAY BRIDGES



TURNPIKE ROADS



PUBLIC CARRIAGE ROADS PRIVATE ROADS

ROAD BRIDGES

FIG. 1.

Live Loads on Bridges.

Rolling Load on Railway Bridges.

(1)

The effective weight of the rolling load, per foot lineal of each line of way, will depend upon the width of span. The heaviest train would consist of locomotives coupled closely together, but while a bridge of short span might frequently be covered by such a train, or by the two engines that often head the train, a bridge of very long span would never be subjected to such a load per foot lineal over its entire length. But for spans shorter than the length of one engine, the load may be very much greater than the weight of the engine per foot of its extreme length. Taking the worst position of the wheel-loads, and the heaviest class of engines now in use upon English lines of railway, the equivalent distributed load per foot lineal may be estimated as follows :—

For spans of 10 feet, a load of 4·40 tons per foot.						For spans of 100 feet, a load of 2·07 tons per foot.					
„	20	„	„	3·25	„	„	„	150	„	„	2·00	„	„
„	30	„	„	2·88	„	„	„	200	„	„	1·90	„	„
„	60	„	„	2·30	„	„	„	300	„	„	1·75	„	„

In designing cross girders the maximum axle load is taken at 22 tons.

These loads are exclusive of any allowance for impact which may be allowed for by varying the unit stress per sq. in. to suit the application of the load :—From 4·5 tons per sq. in. for cross girders to 6·5 tons per sq. in. for main girders 100 feet span and over. The better plan, however, is to calculate the impact allowance from one of the impact formulæ given on p. 268.

The loads given above are considered to be the maximum that can be obtained from the present loading gauge of British railways.

(2)

Bridge to be designed for a moving load of two engines coupled and a uniformly loaded train for each track. Greatest stresses in any possible circumstances to be taken.

Where not specified take the following :—

For Main Girders.—Distributed rolling load per track = 1·5 tons per foot run, and an excess rolling load to occupy any position on the bridge at the same time equal to $15 \text{ tons} + \left(\frac{\text{span in feet} - 10}{10}\right)$ but not greater than 25 tons.

For Cross Girders.—The load on each cross girder from each track :

For all girders up to 7 ft. centres = 19 tons,
„ „ beyond 7 ft. „ = 1·5 tons per ft. run of track plus an excess load of

$$10 \text{ tons} - \left(\frac{10 - \text{centres of cross girders in ft.}}{5}\right),$$

but not greater than 10 tons. *(Sir Wm. Arrol & Co., Ltd.)*

Locomotive Weight Distribution on Bridges.

In the case of the four-cylinder 'Baltic' tank engines—London Midland and Scottish Railway—these engines are of the 4–6–4 type, 40 ft. 4 ins. total wheel base, 49 ft. 10½ ins. long over buffers. The total weight in working order is 99 tons 17 cwt., the adhesion weight being 55 tons 15 cwt.

The general weight distribution is :

	T.	C.	Q.	lbs.
Weight per ft. on coupled wheels . . .	4	2	0	12
„ „ engine wheel base . . .	2	9	2	2
„ „ over buffers	2	0	0	5

For further details of locomotives refer to 'Weights of Locomotives and Rolling Stock, Section XIII, Part II, pp. 572, 628.

British Standard Unit Loadings for Railway Girder Bridges and Highway Girder Bridges.

TABLE I.

British Standard Unit Loading for Railway Bridges for a Single Line of Way.

For Gauges of 4 ft. 8½ ins. and over.

FIG. 2.

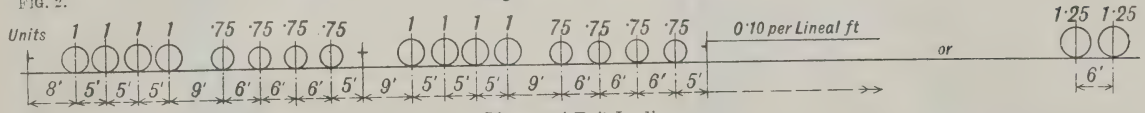


Diagram of Unit Loading.

In this table the unit of axle load is taken as 1 ton (2,240 lbs.).

For convenience of calculation the train load has been taken as equivalent to a train of vehicles with axles spaced 10 ft. apart, each axle carrying a load of one unit, the first axle being spaced 5 ft. from the end.

For British railways the loading recommended in the Requirements for Passenger Lines and Recommendations for Goods Lines of the Minister of Transport in regard to Railway Construction and Operation (1925) is 20 units for heavy main lines, with lower multiples for lines constructed for lighter loading.

1	2	3	4	5	6	7	8
Span.	Main Girders and Rail Bearers.						Cross Girders.
	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum shear.		Maximum cross girder or pier reaction.
	At centre of span.		At ¼ point of span.		At centre of span.	At abutments.	
Ft.	Ft. Tons.	Tons.	Ft. Tons.	Tons.	Tons.	Tons.	Tons.
4	1·250	2·500	0·938	2·500	0·625	1·250	1·250
5	1·563	2·500	1·172	2·500	0·625	1·250	1·250
6	1·875	2·500	1·406	2·500	0·625	1·250	1·333
7	2·188	2·500	1·641	2·500	0·625	1·429	1·571
8	2·500	2·500	1·875	2·500	0·625	1·563	1·750
9	2·813	2·500	2·344	2·778	0·625	1·667	1·889

1	2	3	4	5	6	7	8
Span.	Main Girders and Rail Bearers.						Cross Girders.
	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum shear.		Maximum cross girder or pier reaction.
	At centre of span.		At ¼ point of span.		At centre of span.	At abutments.	
Ft.	Ft. Tons.	Tons.	Ft. Tons.	Tons.	Tons.	Tons.	Tons.
10	3·125	2·500	2·813	3·001	0·625	1·750	2·000
11	3·636	2·645	3·281	3·182	0·625	1·818	2·182
12	4·219	2·813	3·750	3·333	0·625	1·875	2·333
13	4·808	2·959	4·219	3·462	0·673	1·923	2·462
14	5·500	3·143	4·688	3·572	0·714	1·964	2·571
15	6·250	3·333	5·156	3·667	0·750	2·000	2·717

16	7·000	3·500	5·625	3·750	0·781	2·125	2·844	49	45·190	7·378	35·859	7·806	1·395	4·367	6·367
17	7·750	3·647	6·094	3·824	0·809	2·235	2·956	50	46·752	7·480	37·125	7·920	1·415	4·420	6·460
18	8·500	3·778	6·563	3·889	0·833	2·333	3·056	52	50·091	7·706	39·750	8·154	1·452	4·519	6·712
19	9·329	3·928	7·031	3·947	0·855	2·421	3·145	54	53·583	7·938	42·375	8·370	1·486	4·648	6·944
20	10·313	4·125	7·500	4·000	0·875	2·500	3·225	56	57·076	8·154	45·000	8·571	1·518	4·768	7·196
21	11·298	4·304	8·250	4·190	0·893	2·571	3·333	58	60·569	8·354	47·625	8·759	1·547	4·879	7·466
22	12·284	4·467	9·000	4·364	0·909	2·636	3·432	60	64·259	8·568	50·250	8·933	1·575	4·983	7·717
23	13·272	4·616	9·750	4·522	0·924	2·696	3·522	62	68·133	8·791	52·875	9·097	1·613	5·081	7·952
24	14·260	4·753	10·500	4·667	0·938	2·750	3·635	64	72·184	9·023	55·750	9·292	1·648	5·203	8·203
25	15·250	4·880	11·250	4·800	0·950	2·830	3·740	66	76·428	9·264	58·750	9·495	1·682	5·318	8·439
26	16·240	4·997	12·094	4·962	0·962	2·904	3·837	68	80·673	9·491	61·750	9·686	1·713	5·456	8·676
27	17·231	5·105	12·984	5·129	0·972	2·972	3·954	70	84·918	9·705	64·875	9·886	1·743	5·586	8·914
28	18·223	5·207	13·875	5·286	0·982	3·036	4·063	72	89·390	9·932	68·250	10·111	1·771	5·736	9·139
29	19·216	5·301	14·766	5·431	0·991	3·095	4·164	74	94·014	10·164	71·625	10·324	1·807	5·878	9·351
30	20·375	5·433	15·656	5·567	1·000	3·150	4·283	76	98·639	10·383	75·250	10·561	1·842	6·013	9·553
31	21·563	5·565	16·547	5·694	1·032	3·226	4·395	78	103·264	10·591	79·000	10·803	1·875	6·141	9·756
32	22·750	5·688	17·438	5·813	1·063	3·297	4·500	80	107·928	10·793	82·750	11·033	1·906	6·272	9·963
33	23·938	5·803	18·328	5·924	1·091	3·364	4·621	82	112·917	11·016	86·875	11·301	1·936	6·405	10·159
34	25·125	5·912	19·313	6·059	1·118	3·426	4·735	84	117·907	11·229	91·000	11·566	1·964	6·533	10·369
35	26·313	6·014	20·344	6·200	1·143	3·486	4·843	86	123·007	11·442	95·125	11·798	2·000	6·663	10·570
36	27·500	6·111	21·375	6·333	1·167	3·542	4·965	88	128·506	11·682	99·250	12·030	2·034	6·795	10·761
37	28·688	6·203	22·406	6·459	1·189	3·615	5·081	90	134·006	11·912	103·375	12·252	2·067	6·922	10·944
38	29·875	6·289	23·438	6·579	1·211	3·684	5·191	92	139·506	12·131	107·500	12·464	2·098	7·052	11·120
39	31·155	6·391	24·469	6·692	1·231	3·750	5·295	94	145·006	12·341	112·188	12·731	2·128	7·184	11·309
40	32·523	6·505	25·500	6·800	1·250	3·813	5·394	96	150·506	12·542	116·875	12·986	2·156	7·310	11·490
41	33·891	6·613	26·531	6·902	1·268	3·872	5·488	98	156·006	12·735	121·563	13·231	2·184	7·439	11·673
42	35·260	6·716	27·656	7·024	1·286	3·929	5·595	100	161·922	12·954	126·250	13·467	2·210	7·570	11·860
43	36·629	6·815	28·828	7·151	1·302	4·000	5·721	105	177·974	13·560	138·109	14·030	2·276	7·876	12·295
44	37·998	6·909	30·000	7·273	1·318	4·068	5·841	110	195·315	14·205	150·531	14·597	2·355	8·182	12·736
45	39·368	6·999	31·172	7·389	1·333	4·133	5·956	115	213·565	14·857	163·625	15·177	2·426	8·478	13·183
46	40·737	7·085	32·344	7·500	1·348	4·196	6·065	120	232·281	15·485	176·750	15·711	2·492	8·775	13·633
47	42·107	7·167	33·516	7·606	1·362	4·255	6·170	125	251·581	16·101	190·313	16·240	2·556	9·064	14·072
48	43·628	7·271	34·688	7·708	1·375	4·313	6·271	130	271·540	16·710	204·375	16·769	2·631	9·354	14·523

Impact must be added, see p. 269 (British Engineering Standard Specification—extract).

TABLE I.—*continued.*

1	2	3	4	5	6	7	8
	Main Girders and Rail Bearers.						Cross Girders.
Span.	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum shear.		Maximum cross girder or pier reaction.
	At centre of span.		At ¼ point of span.		At centre of span.	At abutments.	
Ft.	Ft. Tons.	Tons.	Ft. Tons.	Tons.	Tons.	Tons.	Tons.
135	291·501	17·274	218·438	17·259	2·711	9·637	14·963
140	312·071	17·833	233·750	17·810	2·793	9·921	15·393
145	333·319	18·390	249·688	18·368	2·886	10·200	15·862
150	354·566	18·910	265·625	18·889	2·973	10·480	16·300
155	376·233	19·418	282·375	19·432	3·055	10·755	16·774
160	398·726	19·936	299·250	19·950	3·136	11·031	17·219
165	421·397	20·431	316·313	20·449	3·219	11·303	17·697
170	445·143	20·948	334·250	20·973	3·297	11·576	18·147
175	468·889	21·435	353·000	21·516	3·381	11·846	18·629
180	492·766	21·901	371·750	22·030	3·461	12·117	19·083
185	517·731	22·388	390·938	22·541	3·543	12·384	19·568
190	542·699	22·850	410·625	23·053	3·624	12·653	20·026
195	567·814	23·295	430·313	23·538	3·703	12·918	20·513
200	594·037	23·761	450·750	24·040	3·785	13·185	20·975
205	620·262	24·205	472·063	24·563	3·863	13·449	21·463
210	647·591	24·670	493·625	25·073	3·938	13·714	21·929
215	675·019	25·117	515·250	25·563	4·012	13·977	22·419
220	702·450	25·544	537·750	26·073	4·091	14·241	22·886

1	2	3	4	5	6	7	8
	Main Girders and Rail Bearers.						Cross Girders.
Span.	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum shear.		Maximum cross girder or pier reaction.
	At centre of span.		At ¼ point of span.		At centre of span.	At abutments.	
Ft.	Ft. Tons.	Tons.	Ft. Tons.	Tons.	Tons.	Tons.	Tons.
225	730·373	25·987	560·250	26·560	4·167	14·502	23·378
230	759·729	26·425	583·125	27·043	4·239	14·765	23·848
235	789·660	26·882	606·875	27·546	4·311	15·026	24·340
240	819·594	27·320	631·250	28·056	4·388	15·288	24·813
245	850·700	27·778	655·625	28·544	4·461	15·547	25·306
250	881·901	28·221	680·625	29·040	4·532	15·808	25·780
255	913·859	28·670	705·938	29·529	4·602	16·067	26·275
260	946·256	29·116	732·000	30·031	4·677	16·327	26·750
265	979·203	29·561	759·188	30·559	4·749	16·585	27·245
270	1012·871	30·011	786·500	31·072	4·819	16·844	27·722
275	1046·541	30·445	814·625	31·598	4·887	17·102	28·218
280	1081·079	30·888	842·750	32·105	4·961	17·361	28·696
285	1116·016	31·327	871·313	32·611	5·032	17·618	29·193
290	1151·992	31·779	900·375	33·117	5·100	17·876	29·672
295	1188·362	32·227	929·438	33·607	5·168	18·132	30·169
300	1225·806	32·688	959·250	34·107	5·240	18·390	30·650

Impact must be added, see p. 269 (British Engineering Standard Specification—extract).

TABLE IV.

BRITISH STANDARD UNIT LOADING FOR HIGHWAY BRIDGES.

FIG. 3. (Each 10 ft. width of carriageway of the bridge is to be assumed as completely occupied by a succession of Standard trains.)

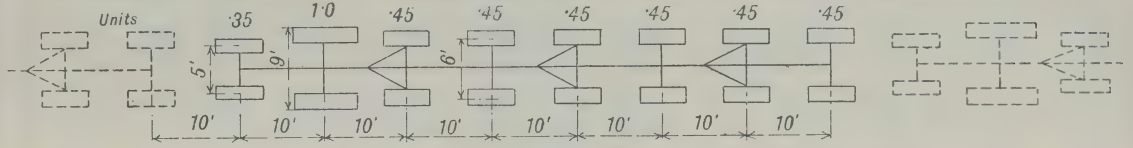


Diagram of Unit Loading.

In this table the unit of axle load is taken as 1 ton (2,240 lbs.).

When the passage of exceptional loads over a bridge is to be anticipated, a special multiple of the unit loading should be specified as applicable to road bearers, cross girders, and short panel lengths.

The multiple recommended by the Minister of Transport as a minimum for all road bridges in Great Britain is 15 units.

1	2	3	4	5	6	7	8
Span.	Main Girders and Longitudinal Rail Bearers.						Cross Girders.
	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum shear.		Maximum cross girder or pier reaction.
	At centre of span.		At ¼ point of span.		At centre of span.	At abutments.	
Ft.	Ft. Tons.	Tons.	Ft. Tons.	Tons.	Tons.	Tons.	Tons.
4	1·000	2·000	0·750	2·000	0·500	1·000	1·000
5	1·250	2·000	0·938	2·000	0·500	1·000	1·000
6	1·500	2·000	1·125	2·000	0·500	1·000	1·000
7	1·750	2·000	1·313	2·000	0·500	1·000	1·000
8	2·000	2·000	1·500	2·000	0·500	1·000	1·000
9	2·250	2·000	1·688	2·000	0·500	1·000	1·000

1	2	3	4	5	6	7	8
Span.	Main Girders and Longitudinal Rail Bearers.						Cross Girders.
	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum shear.		Maximum cross girder or pier reaction.
	At centre of span.		At ¼ point of span.		At centre of span.	At abutments.	
Ft.	Ft. Tons.	Tons.	Ft. Tons.	Tons.	Tons.	Tons.	Tons.
10	2·500	2·000	1·875	2·000	0·500	1·000	1·000
11	2·750	2·000	2·063	2·000	0·500	1·041	1·073
12	3·000	2·000	2·250	2·000	0·500	1·075	1·133
13	3·250	2·000	2·438	2·000	0·500	1·104	1·185
14	3·500	2·000	2·681	2·043	0·500	1·129	1·229
15	3·750	2·000	2·953	2·100	0·500	1·150	1·267

Impact must be added, see p. 269 (British Engineering Standard Specification—extract).

TABLE IV.—*continued.*

1	2	3	4	5	6	7	8
Span.	Main Girders and Longitudinal Road Bearers.						Cross Girders.
	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum shear.		Maximum cross girder or pier reaction.
	At centre of span.		At ¼ point of span.		At centre of span.	At abutments.	
Ft.	Ft. Tons.	Tons.	Ft. Tons.	Tons.	Tons.	Tons.	Tons.
16	4·000	2·000	3·225	2·150	0·500	1·169	1·300
17	4·250	2·000	3·497	2·194	0·500	1·185	1·329
18	4·500	2·000	3·769	2·233	0·500	1·200	1·356
19	4·821	2·030	4·041	2·269	0·500	1·213	1·379
20	5·175	2·070	4·313	2·300	0·500	1·225	1·400
21	5·529	2·106	4·584	2·328	0·502	1·257	1·462
22	5·906	2·148	4·856	2·354	0·505	1·286	1·518
23	6·356	2·211	5·128	2·378	0·507	1·313	1·570
24	6·806	2·269	5·400	2·400	0·508	1·338	1·617
25	7·256	2·322	5·672	2·420	0·510	1·360	1·660
26	7·705	2·371	5·944	2·439	0·512	1·381	1·700
27	8·155	2·416	6·244	2·467	0·513	1·400	1·737
28	8·605	2·459	6·600	2·514	0·514	1·418	1·771
29	9·055	2·498	6·956	2·559	0·516	1·434	1·803
30	9·505	2·535	7·313	2·600	0·517	1·450	1·833
31	9·954	2·569	7·669	2·639	0·518	1·479	1·890
32	10·404	2·601	8·025	2·675	0·519	1·506	1·944
33	10·854	2·631	8·381	2·709	0·520	1·532	1·994

1	2	3	4	5	6	7	8
Span.	Main Girders and Longitudinal Road Bearers.						Cross Girders.
	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum shear.		Maximum cross girder or pier reaction.
	At centre of span.		At ¼ point of span.		At centre of span.	At abutments.	
Ft.	Ft. Tons.	Tons.	Ft. Tons.	Tons.	Tons.	Tons.	Tons.
34	11·304	2·660	8·738	2·741	0·521	1·556	2·041
35	11·754	2·687	9·094	2·772	0·521	1·579	2·086
36	12·204	2·712	9·450	2·800	0·522	1·600	2·128
37	12·654	2·736	9·806	2·827	0·523	1·620	2·168
38	13·167	2·772	10·163	2·853	0·524	1·639	2·205
39	13·722	2·815	10·519	2·877	0·524	1·658	2·241
40	14·278	2·856	10·875	2·900	0·525	1·675	2·275
41	14·834	2·894	11·381	2·961	0·531	1·702	2·329
42	15·390	2·931	11·888	3·019	0·537	1·729	2·381
43	16·027	2·982	12·394	3·074	0·542	1·753	2·430
44	16·702	3·037	12·900	3·127	0·548	1·777	2·477
45	17·377	3·089	13·406	3·178	0·553	1·800	2·522
46	18·052	3·139	13·913	3·226	0·558	1·822	2·565
47	18·727	3·188	14·419	3·272	0·562	1·843	2·606
48	19·402	3·234	14·925	3·317	0·567	1·863	2·646
49	20·077	3·278	15·431	3·359	0·571	1·882	2·684
50	20·752	3·320	15·938	3·400	0·575	1·900	2·720
52	22·102	3·400	16·950	3·477	0·583	1·952	2·823

54	23·452	3·474	18·019	3·559	0·590	2·000	2·919	150	144·258	7·694	112·156	7·976	1·148	4·353	7·627
56	24·802	3·543	19·200	3·657	0·596	2·045	3·007	155	153·348	7·915	119·328	8·212	1·177	4·474	7·871
58	26·213	3·616	20·381	3·748	0·603	2·086	3·090	160	163·427	8·171	126·500	8·433	1·203	4·588	8·100
60	27·778	3·704	21·563	3·834	0·608	2·125	3·167	165	173·508	8·413	134·516	8·696	1·233	4·724	8·376
62	29·344	3·786	22·744	3·913	0·621	2·176	3·268	170	184·756	8·694	142·531	8·943	1·262	4·853	8·635
64	31·101	3·888	23·925	3·988	0·633	2·223	3·363	175	196·131	8·966	150·688	9·185	1·289	4·987	8·903
66	32·901	3·988	25·106	4·058	0·644	2·268	3·452	180	207·673	9·230	159·125	9·430	1·314	5·114	9·156
68	34·701	4·082	26·400	4·141	0·654	2·310	3·535	185	219·876	9·508	167·563	9·661	1·351	5·246	9·419
70	36·501	4·172	27·750	4·229	0·664	2·350	3·614	190	232·251	9·779	176·281	9·896	1·387	5·371	9·668
72	38·301	4·256	29·100	4·311	0·674	2·397	3·711	195	244·626	10·036	185·141	10·127	1·421	5·501	9·928
74	40·101	4·335	30·450	4·389	0·686	2·442	3·803	200	257·278	10·291	194·000	10·347	1·453	5·625	10·175
76	41·901	4·411	31·800	4·463	0·697	2·484	3·889	205	270·501	10·556	203·609	10·594	1·488	5·754	10·432
78	43·761	4·488	33·150	4·533	0·708	2·524	3·972	210	284·001	10·819	213·219	10·830	1·523	5·876	10·676
80	45·779	4·578	34·500	4·600	0·719	2·563	4·050	215	297·501	11·070	223·141	11·071	1·555	6·003	10·930
82	47·797	4·663	36·188	4·707	0·734	2·623	4·173	220	311·278	11·319	233·688	11·330	1·586	6·125	11·173
84	50·001	4·762	37·875	4·810	0·749	2·681	4·290	225	325·626	11·578	244·234	11·579	1·621	6·251	11·424
86	52·251	4·861	39·563	4·907	0·763	2·736	4·402	230	340·251	11·835	255·063	11·829	1·654	6·372	11·665
88	54·501	4·955	41·250	5·000	0·776	2·789	4·509	235	354·876	12·081	266·031	12·075	1·686	6·495	11·913
90	56·751	5·045	42·938	5·089	0·789	2·839	4·611	240	369·779	12·326	277·000	12·311	1·717	6·613	12·150
92	59·001	5·131	44·625	5·174	0·801	2·897	4·726	245	385·251	12·580	288·813	12·574	1·751	6·746	12·418
94	61·251	5·213	46·356	5·260	0·813	2·952	4·836	250	401·001	12·832	300·625	12·827	1·783	6·874	12·676
96	63·501	5·292	48·175	5·353	0·824	3·005	4·942	255	416·751	13·075	312·578	13·075	1·814	7·006	12·939
98	65·810	5·372	49·994	5·442	0·835	3·056	5·043	260	432·779	13·316	324·813	13·326	1·844	7·133	13·192
100	68·279	5·462	51·813	5·527	0·845	3·105	5·140	265	449·376	13·566	337·047	13·567	1·877	7·263	13·453
105	74·750	5·695	56·359	5·725	0·880	3·240	5·410	270	466·251	13·815	349·563	13·810	1·909	7·389	13·704
110	81·500	5·927	61·531	5·967	0·911	3·364	5·655	275	483·126	14·055	362·219	14·050	1·940	7·518	13·962
115	88·250	6·139	67·016	6·216	0·940	3·496	5·917	280	500·279	14·294	374·875	14·281	1·970	7·643	14·211
120	95·279	6·352	72·500	6·444	0·967	3·617	6·158	285	518·001	14·540	388·375	14·536	2·002	7·771	14·467
125	102·875	6·584	78·828	6·727	1·000	3·746	6·416	290	536·001	14·786	401·875	14·782	2·034	7·895	14·714
130	110·750	6·815	85·156	6·987	1·031	3·865	6·654	295	554·001	15·024	415·516	15·024	2·064	8·022	14·968
135	118·625	7·030	91·625	7·240	1·059	3·993	6·907	300	572·233	15·260	429·438	15·269	2·093	8·145	15·213
140	126·746	7·243	98·375	7·495	1·086	4·111	7·143								
145	135·384	7·469	105·125	7·733	1·118	4·236	7·393								

Impact must be added, see p. 269 (British Engineering Standard Specification—extract).

Bridge Foundations.

Loads on Bridge Foundations.

Cylinder Bridge Piers. Approximate Safe Loads per Square Foot (*J. Newman*).

	Tons per Sq. Ft.
Firm sand in estuaries and bays	5·0 to 5·6
Considered safe on firm sand (in Holland)	6·0
Very firm, compact sand, foundations at depths not less than 20 ft.	6·7 to 7·8
Firm shale and clean gravel	6·7 to 8·9
Compact gravel	7·8 to 10·0

Calculated loadings (neglecting friction), tons per square foot: Benares bridge, 11·19; Jubilee bridge, 9; Gorai bridge, 8·5 (all deep foundations); Blackfriars bridge, 4·75; Rupnarayan bridge, 6·7 (or, assuming skin friction as 5 cwts. per square foot, the loading on the base of the pier is about 4·5 tons per square foot).

Clyde Bridge, Caledonian Railway.

Caissons 48 to 89 ft. long and 20 to 23 ft. wide:

Land Piers.—Distances from river-bed about 50 and 60 ft.; depths below level of river-bed about 20 and 30 ft. Depths below quay level and loadings, 50 ft., 4·12 tons per sq. ft.; 61 ft., 4·73 tons; 57 ft., 4·75 tons.

River Piers.—Depths below river bed and loadings: 44 ft., 5·66 tons per sq. ft.; 44 ft., 5·96 tons; 48 ft., 6·04 tons; 44 ft., 6·12 tons.

Skin Friction on Bridge Foundations.

(Caissons and 'Wells' or 'Cylinders.')

Foundations of the Empress Bridge over the Sutlej.

Brick Cylinders, 19 Ft. in diameter, sunk on an average 110 Ft.	Skin Friction. Cwts. per Sq. Ft.		
	Mean.	Max.	Min.
Average of 36 cylinders sunk to depths varying from 10 to 27 ft., average 19 ft., under their own weight only, 25½ to 33 tons, average 31 tons.	0·85	1·33	0·63
Average of 100 observations in sinking 36 cylinders, at depths from 17 to 64 ft., loads 31 to 249 tons, average 132 tons besides the weight of the cylinders.	2·13	4·08	1·29
Average of 9 observations in sinking 3 cylinders by pneumatic process at depths varying from 44 ft. to 63 ft.	2·71	3·52	2·32

Papaghni Bridge.—Cast-iron cylinders 12 ft. in diameter and brick cylinders of the same diameter. The following frictional resistances were noted when sinking the piers; cwts. per sq. ft. In the upper sand 2·08 to 2·20; in black clay, 3·50 to 5·60; in silt below the clay, 2·72 to 4·28; in the lower sand, 2·58 to 3·16.

Chittravati Bridge.—Friction through 33 ft. of sand, 10 ft. of clay, and 7 ft. of clay, sand, and boulders—2·32 to 3·77 cwts. per sq. ft. Friction through 33 ft. of sand, 10 ft. of clay, and 3 ft. of sand and clay—2·93 to 3·62 cwts. per sq. ft.

Tavy and Laira Bridges, Devonshire.—2·0 to 2·8 cwts. in mud.

Concrete Cylinders at Haulbowline Dockyard (H. E. Oakley).—Twenty-three cylinders, 8 ft. in diameter, were sunk through silt. The average skin friction for a cylinder varied from 4·2 to 8·6, the mean being 5·9. In only three cases was the average for a cylinder less than 5·2 cwt. per sq. ft. In seven cases the maximum resistance recorded exceeded 9 cwts. per sq. ft.

Caissons of Barnes Railway Bridge.—The caissons are 28 ft. by 18 ft. The skin friction in London clay was 2·03 cwts. for one, and 2·95 cwts. for the other, per sq. ft.

A Shaft sunk at Oxford Circus, London.—Skin friction only 1·0 cwt. per sq. ft. The low value being attributed to (1) the cylinder keeping its shape very well; (2) being kept very upright; and (3) a pocket being formed below the cutting edge and pugged clay inserted.

A Shaft sunk at Greenwich.—In sand and ballast, skin friction 4 to 4½ cwts.; in shelly clay, 6 cwts. per sq. ft.

Waterway of Bridges.

The simplest cases of waterway calculation are those in which the conditions are such that the amount of heading up is not restricted. In these cases the limiting velocities which may be allowed decide the area of waterway which is necessary. Above the bridge the mean velocity of the stream will be reduced by the heading up.

It may be assumed that the water surface will fall to the original water level before it passes the piers, unless the channel of the river below the bridge has been altered. Taking the water level as the same, the new depth will be the same as the original depth, *plus* any lowering of the bed of the stream at the bridge site. The new width may be taken as the length of waterway. These two dimensions give the new area from which the new velocity can be calculated directly from the discharge, or by

$$V_2 = V_1 \times \frac{d_1 l_1}{d_2 l_2},$$

where

d_1 = original depth; d_2 = new depth; l_1 = mean width of stream cross-section; l_2 = length of waterway.

The new mean velocity is that to which the piers of the bridge are subjected, as well as the bed of the stream for a short distance below the bridge. This distance depends on the velocity and depth of the current. The limiting velocities can best be decided by a study of records of the behaviour of local strata and local masonry under similar conditions.

The following figures provide a rough guide to the velocities usually allowed.

Limiting Velocities in Feet per Second.

On clay, 0·70 to 1·0; on sand which must not be disturbed (shallow foundations) 1·2 to 1·5; on firm, pebbly beds, 3·0 to 3·5; on stony or shingly beds, 4·0; on stratified rock, 6·0; on hard rock, 10·0.

For inferior brickwork, with better masonry in the cutwaters, and on the floor of the waterway at the feet of the cutwaters, about 5·0; better masonry, up to about 8 ft. per second. For ordinary pitching (floor) about 5·0. For very well-shaped bridge openings with thick piers allowing of easy curves, higher values may be allowed, and very high values may be permitted when the masonry is exceptionally strong, such as ashlar of hard stone with fine joints.

Waterway Calculations.

In the case of a bridge or culvert intended for the passage of flood waters which pool up on the upstream side, the discharge may usually be calculated without velocity of approach, especially when the water level on the downstream side is not much lower. This is an ordinary case where a railway embankment crosses land liable to flooding. The head is the difference between the water level below the opening and that above it, when the latter has reached its maximum. This may be a natural maximum, or one fixed by the discharge capacities of the bridges and culverts through the embankment. The amount of headway above this level, under the bridge floor, will depend upon what flotsam is carried by the water in floods.

The formula for discharge is:—

$$D = c\,A\,\sqrt{2gh},$$

where

D = discharge in cubic feet per second (cusecs); A = area of opening in square feet up to the lower water level; h = head, or difference of water levels (in ft.); g = 32·2 approx.; c is a coefficient depending partly upon the size of the opening, and partly upon its form, especially the curves of the cutwaters and wing walls.

Ordinary values for c are in many cases 0·90; for large and well-designed openings, 0·92 to 0·95; for smaller openings, 0·85 to 0·90.

In the case of culverts through an embankment, or for bridges, for the maximum discharge before the embankment is overtopped, it may be necessary to know the discharge when the opening is drowned. If it is drowned on both sides the discharge is:—

$$D = c\,A\,\sqrt{2gh},$$

where

h = difference of water levels, and c = about 0·75 to 0·85, according to the form and size of the opening.

Where the floor of the bridge presents a sharp edge a fairly low value of c should be taken and a fairly high value for well-shaped culverts.

In British measure, $\sqrt{2g} = 8{\cdot}02$; in metric measure, $\sqrt{2g} = 4{\cdot}419$.

When the upstream end is drowned and the downstream end is not drowned, the discharge is found by:—

$$D = cl\sqrt{2g}\left[\frac{2}{3}\left(h^{\frac{3}{2}} - h_2^{\frac{3}{2}}\right) + \left(h_1 - h\right)^{\frac{1}{2}}\right],$$

where

l = the width of the opening; h = the difference in water levels; h_1 = the head above the bottom of the opening; h_2 = the head above the top of the opening; c may be taken as 0·75 to 0·85.

When the span is a long one and the waterway deep, a fairly high value may be given to c, but for a culvert the value may be taken as lower than that for the same culvert when wholly drowned.

In a considerable number of cases the waterway of the bridge is nearly as great as that of the stream, or quite as great when a single span clears the flood channel. Even with a considerable number of spans the bridge may obstruct the waterway very little if the piers are thin and are provided with good cutwaters and joinwaters. The velocity of approach may then be taken as v, the mean velocity of the stream, and the following relation will serve for preliminary calculations :

$$D = c\,A\,\sqrt{2g\,(h + h_a)};$$

where

$h_a = \frac{v^2}{2g}$; $A = dl$, l being the waterway of the bridge, and d the depth to the floor under the bridge.

For sluggish streams h may be neglected, and for a stream with a sharp fall it may be taken as the fall in a length measured about $\frac{2}{3}$ that of the piers. The equation, so used, may be useful in preliminary calculations ; where, for instance, the mean velocity of the stream is measured, but not its cross-sectional area, or where the discharge has been separately found. In the case of a sluggish stream it may be possible to fix the value of d, approximately.

Calculations involving Afflux.

Where the actual fall is small, as in most large rivers, and where the bridge considerably impedes the river, the total head driving the water through the opening is usually calculated as : $(x + h_a)$, where x = the afflux and h_a = head equivalent to the velocity of approach. The discharge is then :

$$D = c\,A\,\sqrt{2g\,(x + h_a)},$$

where

A = the smallest cross-sectional area of the waterway under the bridge, usually taken as that given by a water surface at the original water level.

$$x = \frac{v^2}{2g}\left(\frac{l^2}{c^2 l_1^2} - \frac{d^2}{(d + x)^2}\right),$$

where

l = the mean width of the stream ; d = mean depth of the stream below the bridge ; l_1 = the total width of waterway in the bridge openings ; c = coefficient of contraction, say 0·9 to 0·95 for large bridge openings and about 0·85 for small openings with only moderately good cutwaters and wing walls.

First insert the following provisional value of x in the right-hand side of the above equation.

$$x\ (\text{provisional}) = \frac{v^2}{2g}\left(\frac{1{\cdot}1\,l^2}{l_1^2} - 1\right).$$

This gives a new value of x.

Repeat the solution of the longer equation with the new value of x on the right-hand side.

Next, to find h_a.

The head is calculated as that at one place, due to afflux and velocity of approach, and the velocity of approach is, therefore, the new reduced velocity of the deepened stream above the bridges, thus :

$$v_a = \frac{d}{d + x}\,v, \text{ and } h_a = \frac{v_a^2}{2g};$$

we can then use the equation :

$$D = c\,A\,\sqrt{2\,g\,(x + h_a)}.$$

If the length of the waterway is only estimated at first, within limits, take a probable value, and having found x, find l_1 by the equation :

$$l_1 = c\,\frac{vl\,(d + x)}{\sqrt{[2gx\,(d + x)^2 + v^2 d^2]}}.$$

In some cases x will be decided by the maximum water level which may be allowed, and the value so found will be used in the equations.

Discharge through Long Culverts.

The foregoing formulæ for discharge are suitable for short culverts, and for fairly long culverts when the velocities are low. For long culverts and for high velocities the head necessary to carry the water through the barrel regarded as a pipe must be added. The heads may be found by a pipe discharge formula, or, for small diameter culverts, may be taken from the tables (see 'Waterworks,' Section XXVI., Part III.).

(Reginald Ryves, M.Cons.E., Assoc.M.Inst.C.E.)

Weights of Bridgework.

Dead Loads.—The following are weights of various types of bridge floors, etc.:—

Jack arch flooring .	250 lbs. per sq. ft.	Jarrah wood-block paving 2½ ins. thick, including mastic, pegs, and screws	13½ lbs. per sq. ft.
Steel trough flooring .	13–56 ,,	Tar macadam . .	120 lbs. per cu. ft.
Roadway setts in cement	120 ,,	Concrete filling over buckled plates . .	140 lbs. per cu. ft.
Timber sleepers . .	125 lbs. each.	Fish plates . . .	40 lbs. per pair.
Ballast (average depth 12 ins.) . . .	120 lbs. per sq. ft. of floor.	,, bolts . . .	5½ lbs. per set (4).
Sleepers and ballast .	112 lbs. per cu. ft.	Oak keys . . .	2½ lbs. per pair.
95-lb. rails, chairs, cross sleepers and all fastenings	1½ cwt. per lin. ft. of single track.	Chairs . . .	42½ lbs. each.

WEIGHTS OF GIRDERS, ETC.

Unwin's Formula.

W_1 = weight of girder per ft. run, exclusive of cross-girders or flooring

$$= \frac{Wr}{cs - lr},$$

where

W = load to be carried in tons; l = span in ft.; r = ratio of span to depth; s = working stress in tons, sq. ins.; c = a constant, which may be taken from 1,200 to 1,400 for small plate girders, and 1,700 to 1,900 for truss bridges.

Johnson, Bryan & Turneaure Formula.

Deck plate girders	$w = 9l + 120$
,, lattice girders	$w = 7l + 200$
Through pin bridge	$w = 5l + 350$

(These figures are for single tracks)

where

l = span in ft.; w = weight of each girder in lbs. per ft. run.

American Formulæ.

(1)

The following empirical formulæ, although roughly approximate, are useful as guides in assuming dead loads of ordinary truss road bridges (exclusive of flooring, cross-girders, and joints):—

For a capacity of 100 lbs. per sq. ft.,

$$W = s + 5b + \frac{(b-6)}{8} + \frac{l^2}{300} \quad . \quad . \quad . \quad . \quad . \quad . \quad (1).$$

For a capacity of 80 lbs. per sq. ft.

$$N = s + 5b + \frac{bl}{16} + \frac{l^2}{300} \quad . \quad . \quad . \quad . \quad . \quad . \quad (2).$$

For any other capacity of live load s_1, find W by formula (1), remembering $s = 100$; call required dead load W_1, then $W_1 = W - \left[1 + \frac{(b-12)l}{320}\right](100 - s_1)$,

where

W = dead load per lin. ft. exclusive of floor and joists (lbs.); s = live load per sq. ft. (lbs.); b = clear width of roadway (ft.); l = length of span (ft.).

(2)

For weight of plate and angle girders,

$$P = \frac{Wl}{20d}$$

where

P = weight of beam (lbs.); W = total live load supported (lbs.); l = distance in feet between supports; d = total depth of beam in inches.

Working Stresses.

The following figures are those of a leading structural engineering company, in which a reduced working stress is used.

Permissible Maximum Stresses.

(A) *Railway Bridges.*

The working stresses shown below have been proportioned to allow for dynamic action of the live load on lightly loaded girders or members of girders.

(B) *Road Bridges.*

To provide for the effects of the dynamic action of the live load on lightly loaded girders or members of girders, the live load stresses to be increased by 25 per cent. in the case of main girders and 33⅓ per cent. in the case of floor girders.

(C) *Railway and Road Bridges.*

(1) The combined stresses resulting from the rolling load, dead load, wind, momentum, and centrifugal force, shall not produce a greater tensile stress than half of the elastic limit, or equal to $\frac{3}{10}$ of the minimum ultimate tensile strength of the material, nor more than the corresponding compressive shearing, bearing, and bending stresses as hereinafter set forth, but

(2) The combined stresses resulting from the rolling load and dead load alone, exclusive of wind, momentum, and centrifugal force, shall not produce greater tensile stresses than those given below.

Tensile Stresses.

Railway Bridges.

For main girders, cross-girders, and rail bearers of plate construction:

Under 20ft. span	4½ tons per sq. in.
20 ft. to 25 ft.	4⅝ ,, ,,
25 ft. to 30 ft.	5 ,, ,,
30 ft. to 50 ft.	5¼ ,, ,,
50 ft. to 80 ft.	5½ ,, ,,

For trusses and lattice girders:

80 ft. and under 160 ft. span—	
Bottom chords	5½ tons per sq. in.
Diagonals	4½ to 5½ tons per sq. in.
160 ft. and under 200 ft. span—	
Bottom chords	5⅝ tons per sq. in.
Diagonals	4½ to 5¾ tons per sq. in.
200 ft. to 400 ft. span—	
Bottom chords	6 to 7 tons per sq. in.
Diagonals	4½ to 7 tons per sq. in.
All spans—for wind bracing	8½ tons per sq. in.
Floor suspenders	2⅔ ,, ,,

Road Bridges.

For main girders, cross-girders, and stringers:

Bottom chords	7 tons per sq. in.
Diagonals	5–7 tons per. sq. in.
Wind bracing	8½ ,, ,,
Floor suspenders	3½ ,, ,,

Note.—The 4½ tons stress on diagonals of railway bridges, and the 5 tons stress on road bridges, will apply to those at the centre portion of the span and to the counterbracing at the same point. The higher stresses will apply at the end portions where variations of stress are not so great. Intermediate diagonals will be subject to stresses lying between these limits.

Compression Stresses.

Railway and Road Bridges.

For plate girders, the gross area of the compression flange shall not be less than that of the tensile flange, nor shall the compression stresses per sq. in. be more than 85 per cent. of the corresponding tensile stresses.

For truss and lattice girders the compression stress per sq. in. shall, in the case of riveted members, not exceed the fraction '0·95—0·003R' of the corresponding specified tensile stress nor in the case of pin joints the fraction '0·95—0·0045R,' where R = the ratio of the length of the unbraced portion of the member to its least radius of gyration; nor in any case shall it exceed 85 per cent. of the said tensile stress.

No compression member in railway bridges shall have a greater length than 100 times its least radius of gyration, or 45 times its least width, except for wind bracing, which may have a length not exceeding 120 times its least radius of gyration.

In the case of road bridges no compression member shall have a length exceeding 120 times its least radius of gyration, except wind bracing, which may have a length not exceeding 140 times its least radius of gyration.

ALTERNATING STRESSES.

Railway and Road Bridges.

Members subject to alternate tension and compression stresses shall be proportioned as struts to resist the greater stress plus half the lesser, except for wind bracing, which shall be proportioned for the greater stress only. The sum of the stresses to be used for connections.

PERMISSIBLE UNIT STRESSES IN SWING BRIDGES.

The unit working stresses in main girders shall be 10 per cent. less than those for simply supported spans. *(Sir W. Arrol & Co., Ltd.)*

PERMISSIBLE UNIT STRESSES, IN POUNDS PER SQUARE INCH.

Kind of Strain, and Loading.		Structural Steel.	Axle Steel.	Steel Castings.	Cast Iron.	Rolled Copper.	Brass.
Tension	(*A*)	16,000	18,000	12,000	3,000	6,000	3,000
	(*B*)	10,600	12,000	8,000	2,000	4,000	2,000
	(*C*)	5,300	6,000	4,000	1,000	2,000	1,000
Compression	(*A*)	16,000	18,000	16,000	12,000	6,000	3,000
	(*B*)	10,000	12,000	10,000	8,000	4,000	2,000
Bending	(*A*)	16,000	18,000	15,000	6,000	—	—
	(*B*)	10,600	12,000	10,000	4,000	—	—
	(*C*)	5,300	6,000	5,000	3,000	—	—
Shear	(*A*)	12,000	14,000	9,000	3,000	3,600	—
	(*B*)	8,000	9,600	6,000	2,000	2,400	—
	(*C*)	4,000	4,800	3,000	1,000	1,200	—
Torsion	(*A*)	10,000	12,000	7,500	3,000	—	—
	(*B*)	6,600	8,000	5,000	2,000	—	—
	(*C*)	3,300	4,000	2,500	1,000	—	—

A. For a static load; *B*. For a varying load producing stresses of tension or compression only; *C*. For a varying load producing equal maximum strains in opposite directions, accompanied by shocks and vibrations.

The permissible unit stresses for bending (given opposite *B*)—10,000 for cast steel and 4,000 for cast iron—should be used for determining the strength of toothed gearing. The teeth in cut gearing should conform to the following proportions :—

$$P = s\,p\,f\,y.$$

where,

P = pressure on tooth in pounds ; s = permissible unit stress ; p = pitch, in inches ; f = face of tooth, in inches ; y = a factor.

For wheels moving at slow speed (up to about 100 ft. per minute), where strength only is to be considered, $y = 0{\cdot}05$.

The strength of uncut teeth should be computed for a face of one and one-half times the pitch, or

$$P = \frac{3\,s\,p^2\,y}{2};$$

which conforms approximately with the assumption that the pressure is carried on one corner of the tooth.

For higher velocities, which tend to increase the shock and wear, this value should not exceed :—

$$y = 0{\cdot}05\,\frac{10}{\sqrt{v}} = \frac{0{\cdot}5}{\sqrt{v}}$$

where

v = velocity in pitch circle, in feet per minute.

For bronze wheels, the same values should be used as for cast iron.

FIXED BEARINGS.

For the permissible unit pressure for fixed bearings of different metals, those for compression, in the table, should be used.

Working Stresses in Structural Steel.

(BRITISH STANDARD SPECIFICATION FOR GIRDER BRIDGES.)

Except as hereinbefore modified, structures shall be so designed that the calculated working stresses in structural steel of the 'A' quality specified in Clause 1, Part 1, of this Specification shall not exceed the following:—

For Parts in Tension.

On the net section for axial stress,

8 tons (17,920 lbs.) per sq. in. (12·60 kg. per mm.²).

For Parts in Compression.

On the gross section of the compression flanges of plate girders and I beams with outside edges stiffened with angles or channels,

$$8\left(1-0{\cdot}0075\frac{l}{b}\right) \text{ tons per sq. in.}$$

Ditto, with stiffened edges,

$$8\left(1-0{\cdot}01\frac{l}{b}\right) \text{ tons per sq. in.}$$

where,

l = the greatest unsupported length as defined in Clause 11, Part 4, of this Specification,

and

b = the breadth of the flange,

provided that the gross area of the compression flange shall be not less than the gross area of the tension flange.

On the gross section of compression members of truss and lattice girders with riveted connections for axial stress,

$$8\left(1-0{\cdot}0033\frac{l}{r}\right) \text{ tons per sq. in.}$$

On the gross section of compression members of truss and lattice girders with pin connections for axial stress,

$$8\left(1-0{\cdot}005\frac{l}{r}\right) \text{ tons per sq. in.}$$

Where

l = the greatest length of the unbraced portion of the member

and

r = the least radius of gyration,

provided that, in truss and lattice girders, the working compressive stress on the gross section shall in no case exceed 6·8 tons (15,232 lbs.) per sq. in. (10·71 kg. per mm.²).

For Parts in Shear.

On the gross section of web plates,

5 tons (11,200 lbs.) per sq. in. (7·87 kg. per mm.²).

On shop rivets, turned tight-fitting bolts and pins,

6 tons (13,440 lbs.) per sq. in. (9·45 kg. per mm.²).

For Bearing Areas.

On shop rivets, turned tight-fitting bolts and pins,

12 tons (26,880 lbs.) per sq. in. (18·90 kg. per mm.²).

NOTE.—For connections which are to be made in the field or where black bolts are used instead of rivets, an excess of 15 per cent. in the case of field rivets and 20 per cent. in the case of black bolts over the number required according to the above working stresses, both for shear and bearing, shall be provided.

For Pins subjected to Bending.

On the extreme outer fibres,

12 tons (26,880 lbs.) per sq. in. (18·90 kg. per mm.²).

Where structural steel of the milder quality specified in Clause 1, Part 1, of this Specification for material to be pressed cold is used, the foregoing working stresses where applicable shall be reduced by 10 per cent.

FIRST-CLASS BRITISH PRACTICE IN BRIDGEWORK.

Brief Specification.

1. Materials to conform to the British Standard Specifications.

2. All steel to be by open-hearth acid or basic process.

3. No material to be used until tested and passed.

4. Steel castings to be Grade 'C.'

5. Bolts and nuts, coach screws, clips, etc., to be Grade 'C' wrought iron.

6. All oak to be English or Scotch.

7. Edges of all flange, end, and web plates and ends of all bars to be planed, or where planing not possible, to be dressed fair and true by hand.

8. Wherever shearing is adopted a $\frac{1}{8}$ in. strip to be left for planing, plates and bars showing less than $\frac{1}{8}$ in. rejected.

9. All plates, bars, angles, tees, etc., carefully levelled, straightened, joggled, or upset by pressure and not by hammering, both before and after they are drilled.

10. Templates and gauges to be made to correspond exactly with each other in order that all holes may be straight through and through. Steel gauge $\frac{1}{16}$ in. less in diameter than hole shall pass through easily, irrespective of thickness of plates or bars.

11. Holes for turned bolts to be a driving fit.

12. All holes to be drilled and burrs carefully removed. Holes for rivets to be $\frac{1}{16}$ in. larger than nominal size of rivet.

13. Wherever practicable, all riveting to be done by machine. All loose rivets and such as may be cracked or have burnt or badly formed heads or with heads eccentric with the shaft or where the shaft does not fully fill the hole, cut out and replaced by others.

14. No edge of rivet or bolt hole to be nearer edge of plate or bar, etc., than a distance equal to one and a half times the diameter of rivet.

15. All work riveted or bolted together to be absolutely in contact over whole surface. Immediately before being assembled together whole of surfaces to be thoroughly cleaned and painted with a thick coat of genuine red lead paint.

16. All bolts to be forged and not welded. Screwed to Whitworth's thread. Those taking a shear stress to be so screwed that no part of the thread is in contact with bearing surface.

17. The barrel of all turned bolts to be $\frac{1}{16}$ in. larger in diameter than the screwed portion. Machined washers to be provided.

18. All bolts to be dipped in hot boiled linseed oil.

19. Cover plates connecting different thicknesses of plates to be planed off, the step to be sloped off.

20. All work to be temporarily erected and put together with service bolts.

21. All steelwork to be scraped and cleaned of all rust, mill-scale, and dirt and brushed with wire brushes.

22. All steelwork not to be galvanised to be given two coats of Dixon's Silica Graphite Paint No. 2, or natural colour, and a further two coats after erection, the last a finishing coat of approved colour. No oils or thinners to be used.

Girder Bridge Specification.

A specification issued by the British Engineering Standards Association (No. 153—1922) deals with railway and highway bridges. The first part is concerned with materials of construction (structural and rivet steel, cast steel, steel for pins and rollers, wrought iron and cast iron); these are required to be in accordance with existing B.S. specifications, and the latter are printed as appendices, making the present document complete in itself. The second part gives requirements as regards workmanship, and includes clauses on straightening, planing and shearing, drilling, punching and reaming, rivets and riveting, smithed work, bolts and nuts, temporary erection at contractors' works, painting, measurement, and packing and marking for export.

WIND PRESSURE.

During a period of fourteen years, from 1890 to 1904, observations were made at fourteen different stations in the United Kingdom, and it was found that the maximum mean wind velocity only reached the following values:—

77–80	miles per hour	on one	occasion.
73–77	,,	,,	,,
70–73	,,	,,	five occasions.
65–69	,,	,,	,,

The records of many stations in the British Isles from 1906 to 1913 show that the maximum gusts recorded were:—

99 miles per hour, once.
90 ,, ,, twice.
85–90 ,, ,, nine times.

Velocities very much in excess of these values have been officially recorded by the Robinson anemometer, the velocity of the cups being multiplied by 3, to give the actual velocity of the wind. This figure is now recognised to be too high, and a value of 2·2 is substituted as a multiplier.

The velocity of wind, and therefore the pressure, increases with the height above ground.

The following observations are due to Mr. Adam Hunter, M.Inst.C.E., and were taken on the Forth Bridge by means of small gauges 1·5 sq. ft. in area placed at different heights.

Year.	Date.	Pressure in Pounds per sq. ft. at Various Heights.				
		50 ft.	163 ft.	214 ft.	214 ft.	378 ft.
1901	Jan. 26	—	15	25	—	65
,,	Nov. 23	—	50	55	55	60
1902	Dec. 13	—	27·5	31	34	18
1903	Jan. 10	15	20	25	27·5	60
,,	,, 31	—	19·5	29	26	65
,,	Mar. 18	20	20	25	29	31
,,	,, 21	10	20	20	22·5	54
1904	,, 26	—	20	32	27	52
,,	Dec. 29	—	22·5	22·5	32·5	—
1905	Jan. 21	—	21	30	23	—
,,	Mar. 18	—	32·5	32·5	42	60
,,	Feb. 28	10	22	20	20	38
1906	Jan. 26	15	—	—	—	59
,,	,, 11	10	20	23·5	25	30
,,	Feb. 8	10	15	25	25	55
Average . . .		13·0	23·0	28·0	30·0	50·0

The two readings at 214 ft. are at opposite ends of the bridge.

Relation between Velocity and Pressure.

(See pp. 582, 1183.)

Notes on Wind Pressure on Bridges.

There is considerable shielding effect of the leeward members of a framed structure by those on the windward side, dependent mainly upon their distance apart.

The determination of effective wind pressure on bridges presents many difficulties, arising chiefly out of the irregular shape and construction of those other than plate girder spans.

Plate girders are not, strictly speaking, flat plates, and up to 10 per cent. increase in area is often allowed for the effect of their cup-shape form.

The following remarks are taken from the British Engineering Standard Specification:—

'Where wind pressure has to be taken into consideration, it shall be treated as a moving load not subject to any impact effect and shall be assumed to act horizontally at a slight angle to the transverse axis of the structure so as to take effect on the exposed area of the flooring and the leeward parts in the case of openwork structures, except where any portion may be temporarily screened by a moving load. Where the leeward girder is at a distance from the windward girder not exceeding twice its depth, the effective area of the former shall be taken at half the exposed surface, but where the distance apart exceeds twice the depth, the whole exposed surface shall be taken into account. When the structure is unoccupied by a moving load, the maximum pressure shall be assumed to be 50 lbs. per sq. ft. (244·12 kg. per m.^{2}), but when there is a moving load thereon a pressure of 30 lbs. per sq. ft. (146·47 kg. per m.^{2}) in the case of railway bridges, and 20 lbs. per sq. ft. (97·65 kg. per m.^{2}) in the case of road bridges, shall be assumed as acting on the exposed surfaces both of the structure and the moving load, the pressure on the latter being taken as acting through the centre of gravity of the exposed area. The maximum results from the wind blowing in either direction, and with the structure loaded or unloaded, shall be taken. Where, owing to the position of the bridge or local conditions, these pressures cannot be reached or may be exceeded, the engineer may, at his discretion, alter any of them, as required.'

In providing the necessary anchorage for the bridge a train of empty passenger carriages to be assumed to be on the bridge, weighing 10 cwt. per lin. ft., and in the case of a double track the leeward track only to be loaded.

It is customary to specify a reduced wind pressure for a loaded span for the following reasons :—

(*a*) Because the total area exposed to the wind is increased, and hence the allowance necessary for the rise in pressure due to gusts above that corresponding with the maximum mean wind velocity is not so great as is necessary with a smaller area.

(*b*) Because the resistance offered by a body like a train is not so high as that offered by a surface like a bridge.

During a storm of great intensity the speed of trains crossing bridges will probably be reduced, owing to a feeling of insecurity on the part of drivers, but it cannot be assumed that the impact of trains, except for spans less than 150 ft., will be reduced ; for this depends more upon critical speeds than upon their maximum velocities.

It is therefore necessary, in such cases, where a reduced wind pressure is assumed for calculation, to carefully consider the effects of wind in all its aspects, treating it particularly as a moving load on the train.

The shielding effect of double bars will be to a large extent eliminated if the wind is treated as acting at a slight angle as mentioned.

In bridges of large dimensions it is necessary to make a somewhat different and more exact computation of wind pressures, and, where the spaces are large, separate calculations for each individual member based upon their size, shape, and form, may be essential.

Centrifugal Force.

South American Railway Practice.

Centrifugal force on bridges on curves taken at 0·02 of the live load for each degree of curvature up to 5°, and reduced by 0·001 for every degree of curvature above 5°—the force considered as acting at a height of 5 ft. above rail level. In short spans where impact is high, a percentage for impact is added.

British Practice.

Centrifugal force for each degree of curvature, assumed to be 1 per cent. of the maximum rolling load on all tracks for a speed of 30 miles and under, and 1 per cent. added for each increase in speed of 10 miles per hour. Considered as acting at 5 ft. above rail level.

The radius in feet (R) is reduced to degrees of curvature (D) by means of the following formula:—

$$D = \frac{5730}{R}.$$

Theoretically, centrifugal force varies *directly* as the weight of the train, and as the square of the velocity at which it travels, and *inversely* as the radius of the curve, or

$$P = \frac{W}{g} \frac{V^2}{r},$$

where

W = weight of train in tons ; V = its velocity in ft. per sec. ; r = radius of curvature of rails in ft. ; g = 32, the force of gravity ; P = centrifugal force in tons.

The whole of the centrifugal force is not usually neutralised by the superelevation of the rails, hence the allowances are made as above.

British Standard Specification for Centrifugal Force.

Where a structure carrying a railway is situated on a curve, provision shall be made in designing the members for the stresses due to the centrifugal action of the moving loads, each track on the structure being considered as occupied. The allowance for the centrifugal effect shall be calculated from the following formula :—

$$C = \frac{WV^2}{15\ R}$$

where,

C = the centrifugal effect per lineal foot considered as a moving load, acting at a height of 6 ft. (1·83 m.) above the level of the rails, unless otherwise specified by the engineer ;

W = the equivalent distributed live load per lineal foot;

V = the allowable maximum speed of the train in miles per hour, as specified by the engineer; and

R = the radius of the curve in feet.

No increase for impact effect shall be made on the stresses due to the centrifugal action.

Momentum of Trains.

Tractive Force and Braking Stresses.

In bridges of large dimensions special attention should be given to the details of the structure, to provide for the longitudinal stresses resulting from sudden application of continuous brakes to the train while on the bridge. The horizontal force resulting from such action can be taken as one-fifth of the weight of the train.

The adhesion weight of locomotives is usually about 450 lbs. per ton of load on the coupled axles.

(Refer also to Power, Tractive Force, and Adhesion of Locomotives, Train Acceleration, etc., Section XIII., Part II., 'Railways.')

LONGITUDINAL FORCES.

Where a structure carries a railway, provision shall be made for the stresses due to the tractive effort of the live load and the braking effect resulting from the application of the brakes to such load while passing thereover, these forces being considered as acting at rail level.

(*a*) For railways worked by steam or electric locomotives, the amount of the tractive effort on one track shall be ascertained by multiplying one and three-quarter times the maximum end shear due to the live load on that track by a factor equal to :—

$$\frac{20}{L + 75}$$

where,

L = the span in feet. The load on the track shall be calculated from the shear table for end shear as set out in Appendix I., or such other loading as may be specified by the engineer. The factor shall not exceed 0·15 as a maximum.

The braking effect shall be similarly determined by using a factor equal to :—

$$\frac{12}{L + 90} + 0{\cdot}075$$

also limited to a maximum of 0·15.

(*b*) In the case of lines worked solely on the electrical multiple unit system, the amount of the tractive effort on one track shall be ascertained by multiplying the sum of the actual when loads on the span by a factor equal to :—

$$\frac{5}{L - 10} + 0{\cdot}10$$

where,

L = the span in feet. The factor shall not exceed 0·20 as a maximum.

The braking effect shall be similarly determined by using a factor of 0·20 for all spans.

(*c*) Where the structure carries more than one track, all the tracks shall be considered as being occupied simultaneously by the live loads, tractive and braking forces being applied to alternate tracks. The maximum effect on any girder with two tracks so occupied shall be allowed for, but where there are more than two tracks, a suitable reduction may be made on these forces for the additional tracks at the discretion of the engineer.

No increase for impact effect shall be made on the stresses due to the longitudinal forces.

(*B.E.S.A.*)

Impact.

If a load is applied suddenly to a structure, vibration will ensue and the strain—and thus the stress—will reach twice the value which would occur if the load were gradually applied. There is a great similarity between the results of experiments on variation of stress and sudden loading. The French Government obtain working stresses by means of the Launhardt-Weyrauch formula, but in America and this country various impact formulæ have been devised, but which are by no means final.

Theory prescribes that the following circumstances at least should be taken into account in any impact formula :—

(1) The length of the bridge ;
(2) The ratio of live to dead load ;
(3) The natural frequency of the bridge ;
(4) The fluctuation in rail pressure due to balance weights, hammering, etc.;
(5) The size of the driving wheels.

To supply the necessary information required under (3), a systematic research is required dealing with bridges of various sizes and types.

Waddel's Formula.

$$I = \frac{400}{L + 500},$$

where

L = length of that portion of the span which is covered by the live load when the maximum stress under consideration is produced, and I is the percentage by which the maximum static stress is to be increased.

Allowance for Impact in Bridges

If

S = span in ft.; V = velocity in ft. per sec.; T = time in secs. required by a train travelling at velocity V to travel from the point where a given member commences to receive the stress to the point where the stress is a maximum (i.e. in a plain girder, T is the time taken to pass half the span); D = maximum allowable deflection; t = time taken for a free falling body to fall through a height D, then

$$t^2 = \frac{D}{16}, \text{ or } t = \tfrac{1}{4}\sqrt{D}.$$

If

P = percentage allowance for impact, then

$$P = \frac{50V\sqrt{D}}{S}.$$

As the formula is correct for a single concentrated load only, it gives excessive allowances for long spans, and P may be reduced $\frac{1}{30}$th of its found value for each 25 ft. span in spans of over 25 ft.

In all cases where V is such that the load traverses the half span in less time than t, the percentage for impact must be 100, the maximum V being fixed at 100 ft. per sec.

For a deflection of $\frac{1}{2500}$ of the span, $P = \frac{V}{\sqrt{S}}$, less the allowance for spans above 25 ft.

These formulæ are based on the idea that impact must vary with the time of application of the load. On a long bridge even a high speed train only reaches its position of maximum bending moment after several seconds, so that the bridge deflects slowly to its maximum deflection and goes very little beyond what the train at rest would produce. This applies to main members. Floor beams and their attachment, however, receive maximum stress very rapidly, for the train advances to full effect in the brief space of one panel length. On floor stringer end-connections the first end is loaded absolutely instantaneously; the stringer itself takes its maximum load in about half a panel length, and so on; each member is to be allowed for impact according to the time required from its first feeling the load to the time of maximum moment. No simple load can produce an impactive stress more than double its static stress, but a swaying locomotive may have more than half its load on the wheels of one side, so that a wheel which may cause a maximum load on, say, the first end of a floor stringer, may happen to be loaded at the moment perhaps 30 per cent. more than its half share of the load on its axle, and at the same time it may happen that the balance weight is just delivering its downward hammer blow. Thus care is needed to give ample strength to details even of larger bridges, but the main members of a large bridge are but little affected by the trivial details of locomotive swaying, hammer blow, &c.

British Engineering Specification for Impact Effect.

An appropriate addition shall be made to the live load as specified to allow for impact effect which shall be held to include the stresses set up by the vibrations in the structure as a whole and in the individual members.

(*a*) In the case of a railway worked by steam locomotives, the addition shall be ascertained by multiplying the live load by a factor derived from the following provisional formula:—

$$I = \frac{120}{90 + \frac{n+1}{2}L}$$

where,

I = the factor;

L = the loaded length in feet of the track or tracks producing the maximum stress in such girder or member;

and

n = the number of tracks which the girder or member is designed to support or assist in supporting.

The loaded length of a rail bearer or trough flooring supported by the cross girders shall be taken as the effective span of such rail bearer or trough flooring. The loaded length to be taken for a cross girder shall be the distance between the centres of the cross girders on either side thereof, or between the centre of the cross girder on one side and the centre of the bearing on the abutment on the other.

For floor members, where the live load is effectively distributed, the maximum value of I need not exceed 1·15.

The factor I may also be reduced at the discretion of the engineer in cases where, owing to physical or other reasons, the following speeds will never be attained or exceeded:—

For lines of 4 ft. 8½ in. (1·44 m.) gauge and upwards:—

Bridges with spans up to 50 ft. (15·24 m.), 60 miles (96·56 km.) per hour.

Bridges with spans above 50 ft. (15·24 m.) and up to 150 ft. (45·72 m.), 45 miles (72·42 km.) per hour.

Bridges with spans above 150 ft. (45·72 m.) and up to 300 ft. (91·44 m.), 30 miles (48·28 km.) per hour.

Bridges with spans above 300 ft. (91·44 m.) and up to 400 ft. (121·92 m.), 15 miles (24·14 km.) per hour.

For lines of smaller gauge than 4 ft. 8½ in. (1·44 m.): The assumed speeds should not exceed two-thirds of the above.

(*b*) In the case of a railway worked wholly electrically, the factor may be reduced by such an amount as shall be specified by the engineer.

(*c*) In the case of a road bridge the additon shall be ascertained by multiplying the rolling live load by a factor equal to two-thirds the value of I as determined from the provisional formula in (*a*) with a maximum of 0·70, L being the loaded length as defined in (*a*), and *n* the number of lines of rolling load traffic which the girder or member is designed to support or assist in supporting.

The allowance for impact on road bridges is based on the assumption that the carriageway on and immediately adjacent to the bridge is solidly constructed and maintained in good surface.

No addition for impact shall be made to live load due to pedestrian or equivalent light traffic.

Temperature Variation and Stresses.

Variation of Temperature and Expansion.

Records which have been made at Greenwich show the average range for twenty years to be 11° below and 13° above the mean temperature. The maximum annual range of temperature in the shade may be 85°, and in the sun as much as 137°. The theoretical expansion which these temperatures would cause in steel would be $\frac{5}{8}$ in. and 1 in. respectively in a length of 100 ft.

The observed movement in the Forth Bridge is $\frac{7}{16}$ in. per 100 ft., while the rise in the crown of one of the cast-iron arched ribs of the Old Southwark Bridge was 1¼ ins., the length of the chord of the intrados of which was 246 ft. and the versed sine, 23 ft. 1 in.

In tropical and sub-tropical countries the variation between the highest day temperature and the lowest night temperature may be as much as 150° F. or more.

Provision for Expansion.

For spans less than 40 ft. it is not customary to provide bearings with means of expansion, but for spans in excess of this, provision for expansion to the extent of 1 in. in every 100 ft. should be allowed. No allowance is usually made for expansion and contraction in the direction of the width of the structure, but in large and wide bridges it will be necessary to consider this.

Temperature Stresses.

The forces due to unequal temperatures on different parts of a lattice girder bridge act very slowly and, due to the slow yielding property of ductile metals such as steel, cause the parts of the frame to adjust themselves to the gradual stresses to which they are subjected. Nevertheless temperature stresses in large bridges must be carefully considered.

If the ends of a bar or portion of a bridge are rigidly fixed, a difference of temperature of 100° F. will cause a stress of 8·54 tons per sq. in.

Let

l = length of a bar at 0° F. (in feet); l_1 = its length at t° F. (in ft.); ϕ = coefficient of linear expansion for 1° F.; E = Young's modulus,

then

Elongation for $t° = \phi . t . l$; $l_1 = l(1 + \phi t)$;

Stress per ° F. = $\phi \times$ E tons per sq. in.

Bedplates and Bearings.

Types of Bearings.

Bridges of spans below 100 ft. are not usually fitted with any special means for allowing free expansion, but are simply fitted with a sole plate resting on a bearing block, the holes in the sole plate at one end of the bridge being slotted to allow of movement. The bearing block should rest upon hair-felt or sheet lead.

Bridges over 100 ft. span should rest at one end on roller bearings of not less than 3 ins. diameter, or some other form of expansion bearing.

The average type of roller bearing is usually very liable to rust up and become filled with dirt, etc., that falls from the bridge, and so becomes practically useless, unless very carefully designed with a view to eliminating this source of trouble.

For bearings of large span bridges to be really satisfactory, they should have some means of providing for the deflection of the bridge under load. This is now arranged for by having some sort of knuckle at both ends of the bridge; a bearing of this type is shown in fig. 4, which, however, is not to be regarded as satisfactory as regards the roller portion, which should be protected from rust and dirt.

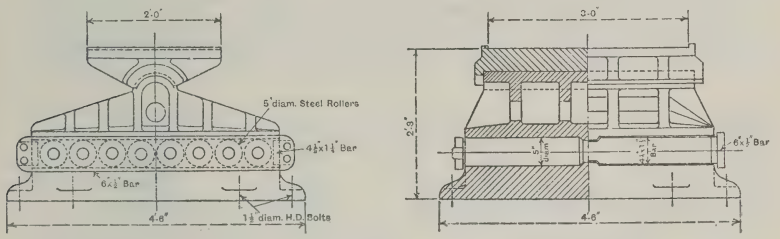


FIG. 4.—Combined Knuckle and Roller Bearing.

The pressure in pounds per lineal in. on rollers of mild steel should not exceed 300 d, where d equals the diameter of the roller in ins.

The bedplates should be of sufficient area to distribute the load over the masonry abutments, the limiting or working pressures being

Granite	. . . . .	20	tons per sq. ft.
Hard stone	. . . . .	16	,, ,,
P.C. concrete (1-2-4)	. . . . .	10	,, ,,

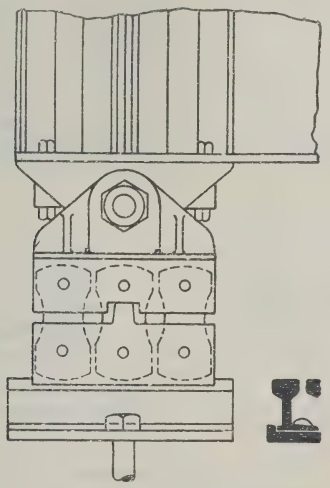


FIG. 5.—Combined Rocker and Roller Bearing.

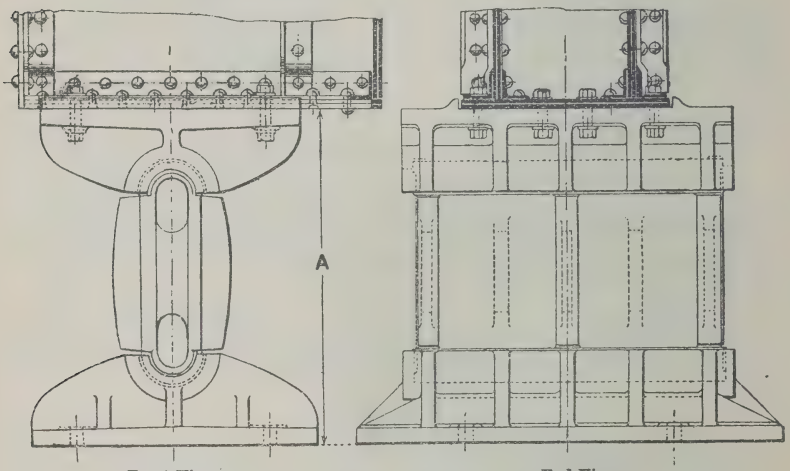
A form of end bearing is shown in fig. 5, which is a combined rocker and expansion bearing for the free end of the bridge. The pin in this case takes the place of the knuckle in providing for deflection.

Rigby's Patent Expansion Rocker Bearing.

A very excellent bridge bearing for spans of 100 ft. and over, is the one designed and patented by the late Mr. H. Rigby, M.Inst.C.E.

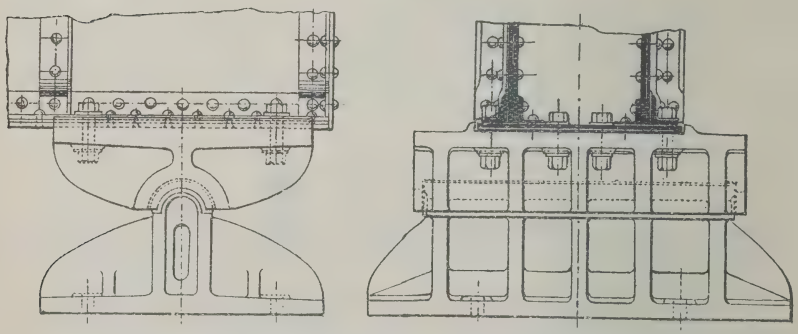
The bearing for both fixed and free ends is shown in figs. 6 and 7.

In this type the knuckles provide for deflection and the rocker for expansion. These bearings are considerably cheaper than most other forms, much simpler in construction, and are quite immune from the effects of rust and dirt in operation.



Front View. End View.

FIG. 6.—Rigby's Patent Expansion Rocker Bearing.



Front View. End View.

FIG. 7.—Rigby's Patent Bearing for Fixed End.

Flooring.

Buckled Plate Flooring.

The resistance of square buckled plates, bolted or riveted down all round, is double the resistance of the same plate merely supported all round, and if the two opposite sides be wholly unsupported, its resistance is reduced in the proportion of 8 to 5.

The stiffness of buckled plates is as the square of the thickness, and inversely as the curvature.

Two inches curvature suffices for 4 ft. square and $\frac{1}{4}$ in. thick.

Ordinary plates are made 3 ft. and 4 ft. square.

Weight and Safe Load of Buckled Plates, 3 ft. square.

No.	Thickness of Metal.	Weight in lbs.	Safe Load distributed in Tons.			
			Passive.		Impulsive.	
			Per Plate.	Per Sq. Ft.	Per Plate.	Per Sq. Ft.
1	18 B.W.G. .	17·3	0·27	0·03	0·20	0·022
2	16 ,, .	23·6	0·43	0·048	0·32	0·036
3	12 ,, .	38·7	0·64	0·071	0·48	0·053
4	$\frac{1}{8}$ in. . .	45·0	1·00	0·112	0·75	0·083
5	$\frac{3}{16}$ in. . .	67·5	2·5	0·278	1·7	0·180
6	$\frac{1}{4}$ in. . .	90·0	4·5	0·5	3·0	0·333
7	$\frac{5}{16}$ in. . .	112·5	6·2	0·689	4·7	0·522
8	$\frac{3}{8}$ in. . .	135·0	9·0	1·0	1·8	0·755

The safe loads may be doubled for buckled plates of puddled steel.

(*The Practical Engineer.*)

Buckled plates are usually made 3 ft. to 6 ft. square and $\frac{1}{4}$ in. to $\frac{3}{8}$ in. thick. They can also be obtained in long lengths having several buckles to the plate.

To calculate the load uniformly distributed over a buckled plate, which will crush it—the plate being square and fastened all round the edges—multiply the depth to which the plate is buckled by the square of the thickness, both in inches, and by 165; the product will be the crushing load in tons (nearly). Central load which will crush a buckled plate, about one-third of uniformly distributed load. (*Rankine.*)

Westwood's Corrugated Plates and Troughs.

These plates and troughs are made of ordinary steel plate and can be obtained in sizes from $2\frac{1}{2}$ ins. deep to 16 ins. deep, and in lengths up to 30 ft., without joint in the smaller sizes and up to 40 ft. in the larger sizes and thicknesses. The advantages claimed are :—

(1) Ordinary plate is used and pressed in moulds in such manner that no deformation of the plate occurs.

(2) The sections can be made of any thickness to suit requirements and are therefore very economical from the point of view of weight.

The smaller sizes are manufactured with several corrugations in one plate and the larger sizes with only one trough or corrugation.

The makers publish a catalogue in which all the sizes manufactured are given, with the moments of resistance, weight, thickness, and safe load per sq. ft. of area covered for various spans.

Distribution of Load.

It may be assumed that the wheel loads are distributed over 5 ft. run of the floor if the sleepers are parallel with the troughs and 10 ft. if at right angles to the troughs.

The strength and stiffness of trough or corrugated flooring, when riveted up, is considerable.

Each trough may be treated as a girder, but then each girder is connected to its fellow; thus when the load is applied to, say, one trough it cannot deflect without dragging down its adjoining troughs for some distance on each side.

Floor Troughing.

Lindsay's troughs, fig. 8, are composed of special rolled sections of the form of splayed channels and can be obtained in many sizes. The advantages claimed are that the webs are thin and the bottoms and tops thick, and so approach what is required from the girder point of view, also that the riveting is situated at the neutral axis, where the bending stresses are a minimum.

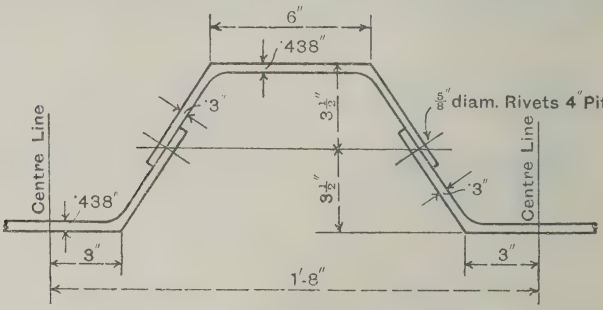
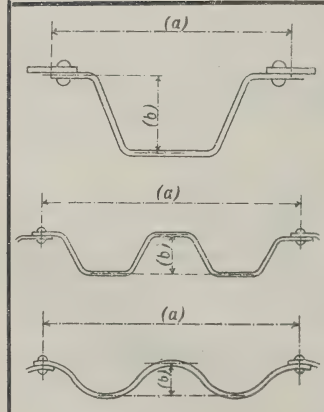


Fig. 8.—Lindsay's Steel Trough C Minimum.

Weight over sq. ft. of area covered = 24·52 lbs. Section Modulus = 21·62.

Westwood's Steel Troughing.



Section.	Depth Centres. (a)	Normal Width Centres. (b)	Thicknesses.	Max. Length Pressed.
	Ins.	Ft. Ins.	Ins.	Ft. Ins.
A	15	2 8	$\frac{3}{8}$–$\frac{3}{4}$	40 0
B	12	2 8	$\frac{3}{8}$–$\frac{3}{4}$	40 0
C	10	2 6	$\frac{3}{8}$–$\frac{3}{4}$	40 0
D	9	2 0	$\frac{3}{8}$–$\frac{3}{4}$	40 0
F	7½	2 0	$\frac{5}{16}$–$\frac{11}{16}$	40 0
G	6½	1 8	$\frac{1}{4}$–$\frac{5}{8}$	35 0
H	6	1 8	$\frac{1}{4}$–$\frac{5}{8}$	35 0
	5	2 8	$\frac{3}{16}$–$\frac{7}{16}$	30 0
K	4	2 8	$\frac{3}{16}$–$\frac{7}{16}$	30 0
L	3	2 0	$\frac{3}{16}$–$\frac{3}{8}$	30 0
O	4	2 8	$\frac{3}{16}$–$\frac{7}{16}$	20 0
P	3	2 0	$\frac{3}{16}$–$\frac{7}{16}$	20 0
M	3	1 6	$\frac{1}{8}$–$\frac{3}{16}$	10 0
N	2½	1 0	$\frac{1}{8}$–$\frac{3}{16}$	10 0

Camber.

The provision of an upward camber in bridge girders gives a better appearance to the bridge, and, moreover, obviates the additional stress, due to the centrifugal force of the train running round the curve, which would otherwise occur due to deflection.

Bridges of 100 ft. span and upwards are usually given a camber of 1 in. for every 40 ft. in length. Plate girders of small dimensions are not usually cambered.

For parallel or bow-topped lattice girders the increase in the length of the top boom and bays therein can be found as follows:—

If

c = camber in ins.; N = number of bays; D = depth between centres of flanges; l = length of one bay in ft.; L = horizontal length of bottom boom in feet; x = increase in length of one bay in top boom in inches; y = total increase in length of top boom,

then $$x = \frac{8cD}{N^2l}, \quad y = \frac{8cD}{Nl} \text{ when } c = 1 \text{ in. in } 40 \text{ ft.}; \quad x = \frac{LD}{5N^2l}, y = \frac{LD}{Nl}.$$

Riveting.

Rivet Pitches.

The pitch of rivets in the direction of the stress should not exceed 6 ins., or 16 times the thickness of the thinnest outside plate or bar, nor less than 3 times the diameter of the rivet.

At the ends of compression members the pitch should not exceed 4 times the diameter of the rivets for a distance equal to twice the width of the member.

In the flanges of girders the pitch should not exceed 4 ins. In webs composed of two or more plates, the rivets, used solely for causing the several thicknesses to act as one plate, should not be spaced more than 12 ins. apart in any direction.

The distance from the centre of the rivet (or bolt hole) to the edge of a plate or bar should not be less than $1\frac{1}{2}$ times the diameter of the rivets in the case of planed or rolled edges. When practicable, the distance should be at least 2 diameters of the rivet, but not to exceed 8 times the thickness of the plate.

In chain riveting the distance between centre lines of adjacent rows should preferably not be less than 3 diameters of rivet, and in no case less than $2\frac{1}{2}$ diameters.

In zigzag or staggered riveting the distance between centre lines of adjacent rows should preferably be not less than $2\frac{1}{2}$ diameters, and never less than 2 diameters.

Diameters of Rivets.

There are various rules for obtaining the most suitable diameter of rivets, but they are not often used in structural steelwork.

In practice, a $\frac{3}{4}$ in. or $\frac{7}{8}$ in. diameter rivet is adopted whenever possible, and it is best not to use any formula to obtain the diameter in terms of the thickness of the plate.

The following is an exceedingly good rule and is almost universally followed:—

$\frac{3}{4}$ in. rivets for $\frac{3}{8}$ in. plates.
$\frac{7}{8}$ in. ,, ,, $\frac{1}{2}$ in. ,,
1 in. ,, ,, $\frac{5}{8}$ in. ,, and over.

It is difficult to get rivets of larger diameter than 1 in. closed by hand.

SECTION VII.

PART II

PLATE GIRDER BRIDGES.

LIVE LOAD — WEBS AND STIFFENERS — JOINTS — CROSS GIRDERS—RAIL AND ROAD BEARERS—CURTAILMENT OF FLANGE PLATES—DEFLECTION.

LIVE LOAD.

Criterion for Maximum Bending Moment.

For a maximum bending moment under any load in a system of wheel loads on a girder, the load considered must be as far from one end of the girder as is the centre of gravity of all the wheel loads from the other end of the girder. Or, in other words, the load considered and the centre of gravity of all the loads in the girder must be equidistant from the ends of the girder. It is often possible to determine by inspection which load will produce the absolute maximum bending moment due to the passage of the live load.

Criterion for Maximum Shear.

The maximum end shear for a system of concentrated moving loads will occur when the reaction is a maximum.

The maximum positive shear in any section of a girder occurs when the foremost load is at the section, provided W is not greater than $\frac{P_1 L}{a}$. If W is greater than $\frac{P_1 L}{a}$, the greatest shear will occur when some succeeding load is at the point.

In this case—

W = the sum of the loads; P_1 = the foremost load; L = the length of the span; a = the distance apart of the foremost and next load in succession.

In plate girders with cross girders, to determine the position of wheels to give maximum shear in panel to left of any cross girder, one of the large wheels is always placed over the cross girder; the wheel so placed should satisfy the following conditions:—

(1) The load to left of the cross girder, and including the load over the cross girder, must be equal to or greater than the total load on bridge divided by number of panels.

(2) The load to left of the cross girder, and not including the load over the cross girder, must be less than the total load on bridge divided by number of panels.

It will sometimes be found that more than one arrangement of the loads will satisfy the above conditions. In such cases it is necessary to figure the shear for the different loadings and use the greater. (*American Bridge Co.*)

WEBS AND STIFFENERS

Distance apart of Stiffeners.

The distance apart which stiffeners should be placed can be found by the following formula:—

$$L = \sqrt{\left(\frac{5}{K} - 1\right) 1500 t^2},$$

where

L = distance apart in ins.; t = thickness of web in ins.; K = stress per sq. in. to be provided for—*i.e.* $\frac{\text{total shear at a section}}{\text{net area of web}}$.

The position of stiffeners should be so arranged as to cause no irregularity in rivet pitches, *i.e.* the pitch should alter if it is necessary at a stiffener. All stiffeners should bear tightly at the top and bottom against the flange angles, and wherever possible, stiffeners should be placed at all web joints.

In British practice, stiffeners are seldom placed at distances wider apart than the depth of the girder, with 6 ft. as a maximum when the thickness of the web is less than $\frac{5}{16}$ of the unsupported distance between the flange angles.

Stiffener angles over the bearings should be designed to carry the entire shear at that point without exceeding the safe stress (packings should not be included in the area); they should be proportioned as struts with fixed ends, having a length equal to $\frac{3}{4}$ of the depth of the girder.

Compound Web Plates.

Web plates composed of two or more plates should not be used where the total thickness is less than 1 in. (For Riveting, see p. 226.)

Reduction of Web Thickness.

The following is a graphical method of obtaining the necessary lengths of web plate of given thickness or the thickness of web plates of given length. The thickness of web plate required at the abutments must first be calculated in each case.

Distances such as 'a' measured to $\frac{3}{4}$ in. scale in fig. 9, give the reduction in thickness permissible.

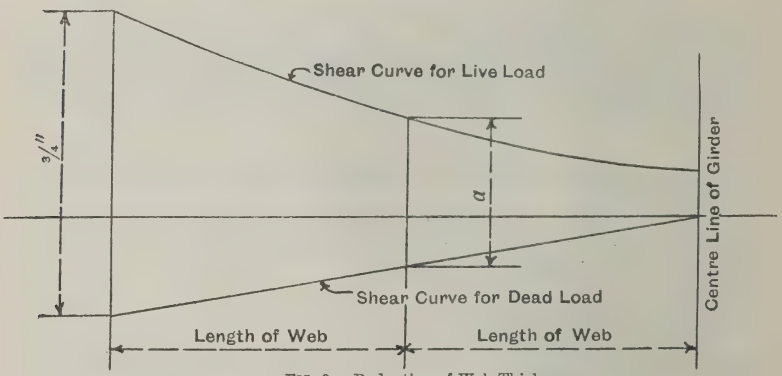


FIG. 9.—Reduction of Web Thickness.

JOINTS.

Thickness of Cover Plates.

The thickness of cover plates may be determined as follows:—

$$\text{Single covers} = t + \frac{t}{8}:$$

$$\text{Sum of thickness of double covers} = t + \frac{1}{4};$$

where

t = thickness of plates joined.

Flange Angle Splices.

If a girder is so long that the angles cannot be obtained in one length, or if for any other reason it is necessary to splice the angles and the girder is to be shipped in one length, the splices in the angles should be as far apart as possible and on opposite sides of the web plate. Where the area of the flange angle is too great to be conveniently spliced by one angle, a splice angle with area equal to about $\frac{3}{4}$ of the area of the flange angle should be placed on each side of the girder.

Web Splices.

When part of the web is counted as flange area and the bending moment at the web joint actually requires this area of web to bring the flange stress down to the maximum allowed, there should be, in addition to the number of rivets required to take shear, sufficient rivets as near each flange angle as practicable to take the proportional part of the flange stress taken by the piece of web.

The net area of these splice plates must be made equal to area of web used as flange area multiplied by the depth of girder divided by the distance centre to centre of splice plates.

The number of rivets required to equal the net area of two splice plates, multiplied by the unit stress in flanges and divided by least value of one unit (shear or bearing) in web plates.

It is desirable to arrange the rivets so as not to cut out more than two rivet holes in the line next the joint, or at the ends of the plates; see fig. 10.

The vertical splice plates should have rivets enough on each side of splice to take the vertical shear at the position of the splice.

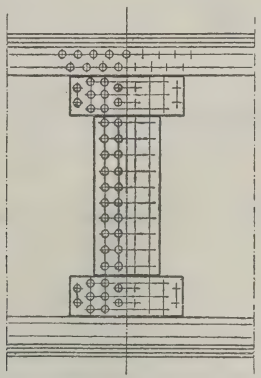


FIG. 10.—Web Joint.

Flange Splices.

If for any reason the flange plates have to be joined, a cover must be provided at each splice; the rivets connecting the cover must be such that their strength is equal to the plate spliced.

It is often possible to arrange the splices in a stepped manner with only one cover plate; such a joint is shown in fig. 11, and is suitable for a shipping joint.

The final cover must be at least equal in thickness to the thickest plate spliced, and, in the example, serves as a cover for the four plates.

The distance apart of the 'steps' must be such that the strength of the rivets between each is not less than will suffice to develop the full strength of each plate.

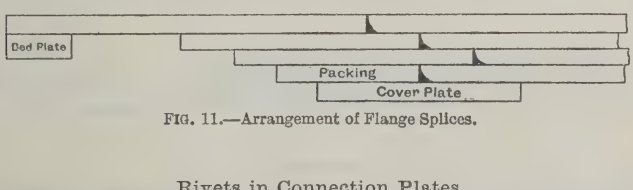


FIG. 11.—Arrangement of Flange Splices.

Rivets in Connection Plates.

When a single system of lateral bracing is used, the number of rivets connecting the plates to the girder should be sufficient to take the sum of components (in direction of girder) of both laterals connecting on the plate. (See fig. 12.) If AB requires 5 rivets and BC requires 4 rivets

lay off in the direction of *ab* to any convenient scale a distance *by* of 5 and in the direction *bc* a distance *bw* of 4. Drop perpendiculars *wv* and *yx* and scale the distances *vb* and *bx*. The sum of these two distances represents the number of rivets required to connect the plate to the girder.

If a double system of bracing is used, they are proportioned for tension only, and the number of rivets required to connect the plate to the girder will be the number required to take the component of one lateral in the direction of the centre line of the girder.

There should be rivets enough connecting the plate to the cross girder or strut (if any) to take the component of one lateral in the direction of the cross girder or strut.

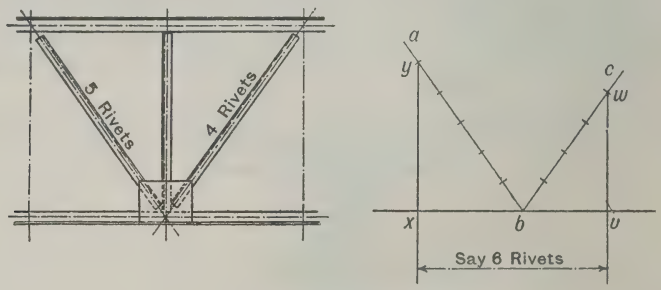


FIG. 12.—Rivets in Connection Plates.

CROSS GIRDERS.

Distance apart of Cross Girders.

Generally speaking, it is not economical to place the cross girders in railway bridges closer together than 5 ft.; at intervals shorter than 5 ft. the load will usually consist of the heaviest axle load of the live load, while for intervals of 5 ft. and upwards it is necessary to consider the several positions that may be occupied by the axles in the heaviest portion of the live load.

The distance apart should be considered in relation to the centres of the heaviest wheel loads which will come upon them.

It is often found that increasing the spacing of the cross girders adds but little to the load which they will have to sustain, and it is usually economical to adopt a wide spacing, say from 6–10 ft.

In lattice girders the spacing of cross girders, depth of truss, and panel length must all be considered in relation to each other, for the cross girders will usually be placed at panel points to avoid secondary stresses in the booms.

The economy of placing cross girders far apart only applies in a modified degree to highway bridges, as in such cases the dead weight of the roadway forms a considerable part of the total load.

PROPORTIONS.

The depth span ratio of cross girders should be not less than 1: 10, preferably 1: 8, otherwise in bridges with buckled plate flooring considerable trouble is likely to occur with the rivets holding the plates down, working loose, or the plates fracturing at the seatings.

CONNECTIONS.

The connections of the main girders should be rigid, and 50 per cent. should be added to the number of rivets required to take the shear to resist the bending action.

In plate girders the web stiffeners may be made to considerably strengthen the end connections of cross girders, and in lattice girders a bracket plate of ample proportions should be fitted.

RAIL AND ROAD BEARERS.

Rail Bearers.

These should be in general placed 5 ft. apart, and must be proportioned to take the greatest wheel loads, imposed by one, two, or more axles, as the case may be. Although they act as continuous girders, much depends upon the rigidity of the end connections and the deflection of the cross girders. Where they rest upon the cross girders they have a greater chance of acting continuously; but usually for the purpose of designing their section they will be treated as separate spans.

Road Bearers.

The distance apart of road bearers will be decided in relation to the width of the bridge and the centres of the wheels composing the maximum live load, and will be greatly decided by the type of bridge floor adopted. They should be designed as supported at the ends.

PROPORTIONS OF RAIL AND ROAD BEARERS.

The depth should not be less than $\frac{1}{12}$ of their span.

Curtailment of Flange Plates.

Unless this is done the girders will be stronger at the abutments than they need be. This can be remedied by either of the following methods:—

(1) Keeping the depth constant and varying the thickness of the flanges along the length.

(2) Varying the depth of the girder and keeping the flange thickness constant. By this method the girder theoretically becomes parabolic.

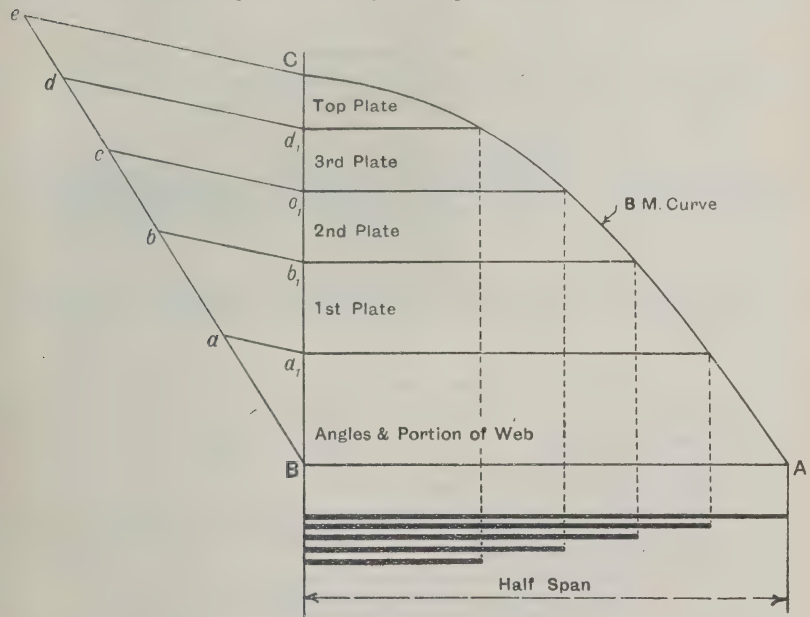


FIG. 13.—Curtailment of Flange Plates.

(1) and (2) Varying both the thickness and depth of the girder. By this method we get the hog-backed and fish-bellied girder according as the top or bottom flange is curved.

There are variations of this, for both top and bottom flanges are sometimes curved.

In most cases (1) is the most economical on account of there being less workmanship required in the manufacture.

The curtailment of flange plates in method (1) is carried out as follows :—
Referring to fig. 13, ABC is the bending moment curve on half the girder AB.

On any inclined line *Be* set out points *a*, *b*, *c*, *d*, as follows :—

Be = Modulus of section of total flange at centre of girder (including portion of web if allowed for in calculations).
de = Modulus of section of top plate (net).
cd = ,, ,, 3rd ,, ,,
bc = ,, ,, 2nd ,, ,,
ab = ,, ,, 1st ,, ,,
Ba = ,, ,, bottom plate and angles (including portion of web if allowed for in calculations).

Join *e*C and draw parallels dividing the vertical *Be* into similar spaces.

Lines drawn horizontally to meet the bending moment curve will now give the theoretical necessary lengths of the flange plates.

In practice the plates are continued for from two to four pitches of the rivets and the first plate over the main angles is always continued to the ends of the girder.

Deflection.

Elastic Deformation.

Is one which disappears entirely on the removal of the external forces causing the stress.

The amount of deflection in girders depends mainly upon the following :—

(1) The length of the girder ;

(2) The unit stress in the flanges.

The deflection arises from the top flange being compressed and the bottom flange being extended, and the amount of deflection is not materially affected by the kind of web in deep girders.

Mohr's Theorem.

A loaded beam takes up the same form as an imaginary cable of the same span, which is loaded with the bending moment curve on the beam and subjected to a horizontal pull equal to the flexural rigidity (EI). The deflected form of the beam is called the *elastic line* of the beam.

Graphical Construction for any Loading.

(Flanges of beam of uniform section.)

(1) Draw the bending moment diagram for the given loading of the beam.

(2) Treat the bending moment diagram as a load diagram and divide it up into narrow vertical strips ; set down the mid-ordinates on a vector line.

(3) Draw the vector diagram with a polar distance equal to the flexural rigidity (EI).

(4) Draw the funicular polygon for this vector diagram and reduce it to a horizontal base, then the link polygon gives the elastic line.

The scale to which the deflections can be scaled is determined as follows :—
1 in. on the length of the bending moment diagram = x ft.
1 in. on the height of the bending moment diagram = y ft. tons.

Therefore in fig. 14 1 in. in height of any mid-ordinate of the bending moment diagram 1, 2, represents $d \times x \times y$ sq. ft. tons. Calculate the length of EI in sq. ft. tons to this scale $= \frac{EI}{144 \times d \times x \times y}$ inches), where E = modulus of elasticity in tons per sq. in., and I = moment of inertia in inch units) ; then the deflections will be to the scale of 1 in. = x ft.

It will be found convenient to reduce the length of EI for practical use, so the polar distance can be taken as $\frac{EI}{n}$, where n is some convenient divisor. The deflection scale will then be 1 in. $= \frac{x}{n}$ ft.

(E. S. Andrews, B.Sc., Assoc.M.Inst.C.E.)

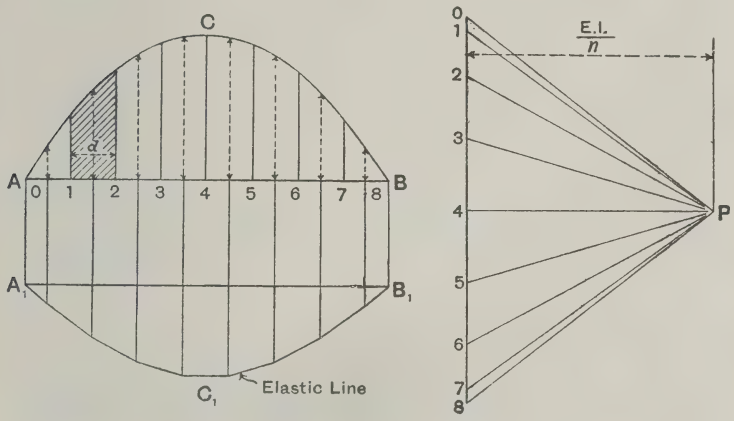


FIG. 14.—Graphical Determination of Deflection.

ACTUAL AND THEORETICAL DEFLECTION.

The difference between the actual and calculated deflection is due to various causes; in riveted girders the stiffness of riveted joints and covers, and the position and form of the floor, all tend to reduce the actual deflection, while poor quality of workmanship in the riveting and fitting together of parts tend to cause the actual deflection to exceed that calculated.

SECTION VII.

PART III.

LATTICE GIRDER BRIDGES.

LIVE AND DEAD LOAD STRESSES—PROPORTIONS OF BOOMS AND STRUTS—PIN-JOINTED MEMBERS—VARIATION OF STRESSES—PORTAL BRACING—DEFORMATION AND DEFLECTION.

LIVE AND DEAD LOAD STRESSES.

Criterion for Maximum Bending Moment and Maximum Shear.

The maximum bending moment at any joint in a bridge loaded with wheel loads will occur when the average load on the left of the section is the same as the average load on the whole span.

The maximum shear in any panel in a bridge loaded with wheel loads will occur when the load on the panel is equal to the load on the bridge divided by the number of panels.

(*Ketchum.*)

Maximum Loads on Inclined Braces.

The maximum tension in any inclined brace occurs when the leading axle is at the foot of the brace, the live load covering the larger segment of the bridge from the foot of the brace to the farthest support.

The maximum compression in any brace occurs when the leading axle is under the head of the brace; the live load covering the shorter segment of the girder to the nearest support.

(*R. J. Woods.*)

Shears and Counterbracing.

(*a*) The maximum live load shear (positive) in any panel of a truss is equal to the left reaction when all the joints at the right of the panel are loaded and all joints at the left of it are unloaded.

(*b*) The dead load negative shear in any panel at the right of the centre is equal to the amount of dead load between the panel and the centre of the span (including a half panel load at the centre if the truss has an even number of panels).

(*c*) No counter is required in any panel beyond the centre of the truss in which the dead load negative shear exceeds the maximum live load positive shear; but in each panel beyond the centre in which the live load positive shear can exceed the dead load negative shear, a counter is required.

Stresses in Parallel-Flanged Girders with Uniform Rolling Load.

Divide span into a number of equal parts, one less than the number of bays (see fig. 15), obtaining points such as a, a_1. On the base A_1B_1 set up length $A_1C = \frac{PL}{2}\left(\frac{b-1}{b}\right)$, where P = intensity of load, L = span of girder, b = the number of bays.

Through C draw a parabola with vertex at B, and project points such as a, a_1, down to meet the parabola in points such as e, e_1, and project horizontally across bays as shown.

Then the maximum stresses in the diagonals for passage of the live load from right to left are obtained from this curve. Negative shears in a similar manner.

The position of the load to give maximum stresses in diagonals is obtained from

$$x = \left(\frac{n.y}{b-1}\right),$$

where

x = distance travelled by the load beyond end of nth bay, y = length of bay, and b = number of panels.

(E. S. Andrews, B.Sc., Assoc.M.Inst.C.E.)

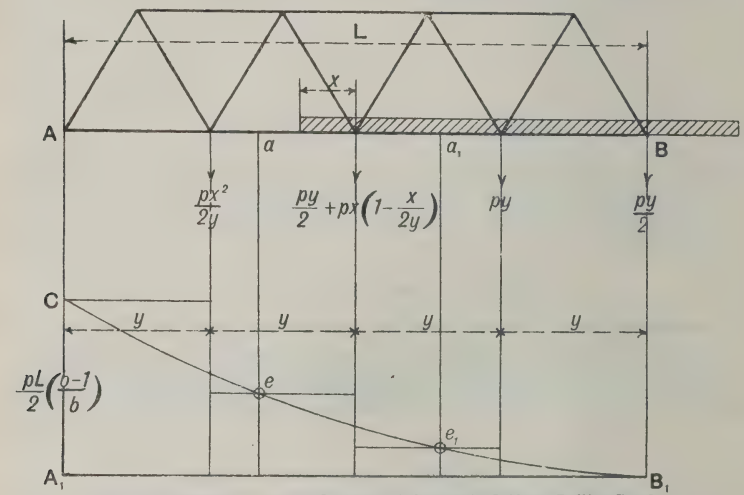


FIG. 15.—Stresses in Parallel-Flanged Girders with Uniform Rolling Load.

Distribution of Dead Load.

For spans under 200 ft. the total dead load shall be assumed to act at the loaded chord. For 200 ft. and over the total dead load will be distributed at top and bottom chords as follows:—

On Loaded Chords.

(*a*) Half the load resulting from trusses;
(*b*) Weight of horizontal wind bracing in plane of chords:
(*c*) Weight of floor system, permanent way, etc.;
(*d*) Half load resulting from weight of cross bracing in the case of deck bridges.

On Unloaded Chords.

(*a*) Half load resulting from trusses;
(*b*) Weight of horizontal wind bracing in plane of chords;
(*c*) Half the load resulting from weight of cross bracing in the case of deck bridges and the whole of it in the case of a through bridge. *(Sir Wm. Arrol & Co., Ltd.)*

Stress in Members.

In parallel flanged lattice girders the stresses in the diagonals

$$= \frac{\text{load on diagonal} \times \text{length of diagonal}}{\text{depth of girder}},$$

i.e. load × sec. ϕ, ϕ being the angle which the diagonal makes with a vertical line.

Ritter's 'Method of Sections.'

This consists of drawing 'lines of section' cutting the bridge truss in not more than three of its members. These lines divide the truss into two parts, one of which is supposed to be removed, the external forces acting on the other portion alone being considered.

The stress in any one of the three members cut by the 'line of section' can be found by taking moments round the point of intersection of the other two.

This method is not applicable to finding the stresses in the flanges of parallel-flanged girders; for the flanges if produced would not meet.

In fig. 16 draw the sectional line *ab*, cutting FE, FG, GA; then,

$$\text{Stress in FG} \times x = 60 \times y,$$ J being the point of intersection of FE and AG.

It will be seen that the moments of the stresses on the members which meet at the point indicated become zero; and it is only necessary to equate the moment of the stress on the third member to the algebraic sum of the moments of all the external forces acting on the portion of the truss considered.

If it is required to find the stress in the member HE, with the live load in the position shown, then,

$$\text{Stress in HE} \times x' = (60 \times 10) - (20 \times 10).$$

Stresses in the vertical struts can be found in a similar manner.

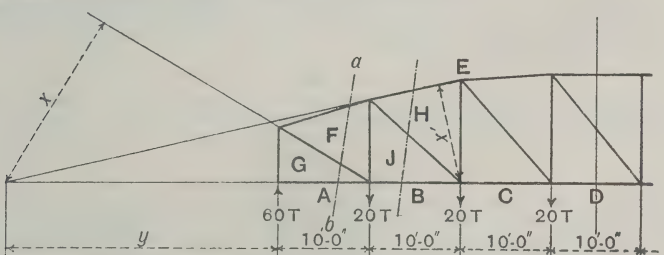


Fig. 16.—Ritter's Method of Sections.

Braced or Lattice Struts.

The least number of panels

$$= \frac{\text{Least radius of gyration of whole strut}}{\text{Least radius of gyration of unbraced portion}}.$$

In the case of a strut composed of an angle at each corner, the least radius of gyration of unbraced portion must be taken as one of the angles.

Batten plates should be spaced from 2–3 times the overall width of the side braced, while the inclination of diagonals to the axis of the member should not be less than 45°.

The thickness of bars with single lacing should not be less than $\frac{1}{40}$ of their length; with double bars, not less than $\frac{1}{60}$ of their length.

Ties.

Ties should usually be designed to be of rigid construction, but may be of rolled flat bars except near the centre of the span, where they must be composed of rigid sections.

Counter bracing must be of similar construction to the centre ties.

Rolled bars forming long ties should be provided with distance pieces to reduce vibration.

Pin-jointed Members.

Pin Connections.

The tearing, shearing, and bearing strengths of the joint should be made as nearly as possible equal to each other, and to the tensile or compressive strength of the bar in which the joint occurs.

Usually the bearing stress on the pin is taken at somewhat less than for rivets, to allow for imperfect fitting of the pin.

The bending strength of pins is very important, for if the pin becomes distorted the distribution of the loads and the calculation of the stresses is very uncertain.

Secondary bending in the top booms and strut members of a bridge merit greater attention when they are connected with pin joints. In this country pin joints will not often occur anywhere other than in suspension bridges, but in America pin-jointed bridges are very common.

The position of pins may be so fixed that when the maximum stress (direct compression) comes upon the chord through the pins, it will produce an amount of bending moment equal to that produced by the weight of the member, but in the opposite direction.

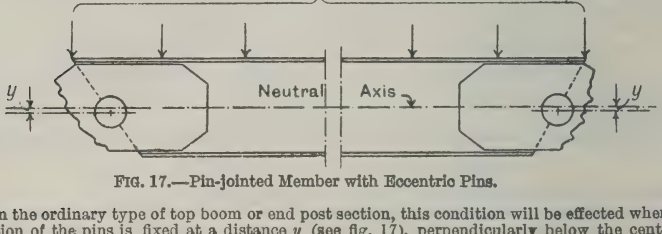


FIG. 17.—Pin-jointed Member with Eccentric Pins.

In the ordinary type of top boom or end post section, this condition will be effected when the position of the pins is fixed at a distance y (see fig. 17), perpendicularly below the centre of gravity of the section, given by the formula

$$y = \frac{M}{Sc},$$

where M is the bending moment in inch pounds due to the weight of the member, and Sc is the total maximum direct compressive stress in the member in pounds.

Variation of Stresses.

WÖHLER'S EXPERIMENTS.

When the elastic limit is not overpassed it is possible, under certain conditions, to fracture steel by repeating the stress a sufficient number of times. Wöhler showed that it was the range of stress, and not the maximum stress, that determined the number of applications before rupture.

The general conclusions are that:

(*a*) The greatest tensile stress a bar will bear for an indefinite number of times is for iron and steel about half its ultimate static breaking stress.

(*b*) The number of repetitions required to produce fracture is increased if the range through which the stress is varied is reduced.

(*c*) A bar will be strongest when exposed to varying stresses of the same kind, and weakest when exposed to stresses of opposite kinds, *i.e.* tension and compression.

The stress caused by a load suddenly reversed is three times the static stress, and a load suddenly applied causes a stress of twice the static stress. (Refer to 'Impact,' p. 268.)

RELATION BETWEEN SUDDEN LOADING AND REPETITION OF STRESS.

The similarity between the results of experiments on variation of stress and sudden loading has given rise to many assumptions that Wöhler's results were really due to the effects of sudden loading, and that the two are really aspects of the same question.

Although there are many points which require to be decided in the consideration of these experiments, for practical reasons it seems best to allow for either one or the other, and not for both simultaneously in one member.

In the case of cross girders and vertical suspenders, allowance should be made for the full value of the elastic vibrations due to sudden loading; while rail bearers and main girders, both plated and latticed, of spans up to 100 ft., should be designed with an allowance for sudden loading or impact, using one of the impact formulæ evolved for the purpose.

In the flanges of girders and the principal members of arched and suspension bridges of large dimensions the stress variation takes place more gradually, and allowance for variation of stress can be made by adopting a reduced stress, based on the amount of variation of stress.

LAUNHARDT-WEYRAUCH METHOD.

The working stress is varied according to the relative amounts of live and dead loads.

MEMBERS SUBJECTED TO TENSION ONLY.

$$\text{Working stress} = \frac{f}{1{\cdot}5}\left(1 + \frac{\text{minimum load}}{2 \times \text{maximum load}}\right) \text{ tons per sq. in.},$$

being the static or dead load working stress.

DYNAMIC METHOD.

$$\text{Working stress} = \frac{f}{1 + \frac{\text{live load.}}{\text{total load}}};$$

$$\text{The sectional area of a bar in tension} = \frac{\text{maximum load}}{\text{working stress}}.$$

MEMBERS SUBJECTED TO ALTERNATE STRESSES.

Members of bridges subjected to alternate tension and compression should be proportioned as struts, to resist the greater stress added to ½ of the lesser stress, except in the case of wind bracing, where the members should be proportioned to resist the greater stress. The sum of the stresses should be used in designing the connections.

Portal Bracing.

Design.

There are many types of portal bracing which may be used; A to F, fig. 18, show some common forms.

The fixing of the lower ends may be assumed as either hinged or fixed.

They may be considered as fixed by virtue of the direct stress due to vertical loading, when $V \times \frac{1}{2}s$ is $>$ M max. at A or B with fixed ends; M max. $= \frac{Rd}{4}$, where V = total stress in the column.

The following treatment for the calculation of the stresses is due to Mr. Milo S. Ketchum in 'Steel Mill Buildings,' in which book he deals with the stresses in many types, with both hinged and fixed ends.

Stresses in Portal (a). Columns Hinged.

The deflections in the columns are assumed to be equal, then,

$$H = H_1 = \frac{R}{2}.$$

Taking moments about the foot of the windward column,

$$V_1 = -V = \frac{Rh}{s},$$

Having found the external forces, the stresses in the members may be found either by algebraic or graphical methods.

To obtain the stress in member GC, pass a section (Ritter's method of sections), cutting GF, EF, and GC, and take moments of the external forces to the right of the section about point F as centre.

$$GC = -\frac{Hh}{(h-d)\sin\phi}.$$

But $$H = \frac{R}{2}, \text{ and } (h-d)\sin\phi = \frac{s}{2} \times \cos\phi.$$

Substituting these values in the above expressions,

$$GC = -\frac{Rh}{s\cos\phi} = -V\sec\phi.$$

Resolving at C and F we have: stress in EF = 0, and also stresses in EE_1 and $HE_1 = 0$.

To obtain stress in GD, pass a section cutting HG, HE_1 and GD, and take moments of the external forces to the left of the section about point H as centre.

$$GD = \frac{Hh}{(h-d)\sin\phi} = +V\sec\phi.$$

To obtain stress in GF, pass a section cutting GF, EF and GC, and take moments of the external forces to the right of the section about point C as centre.

$$GF = +\frac{R(h-d)+Hd}{h-d}.$$

To obtain stress in HG, pass a section cutting HG, HE_1 and GD, and take moments of the external forces to the left of the section about point D as centre.

$$HG = -\frac{Hd}{h-d}.$$

The stress in the windward post AF is zero above and V below the foot of the knee brace C; the stress in the leeward post is zero above and V, below the foot of the knee brace D.

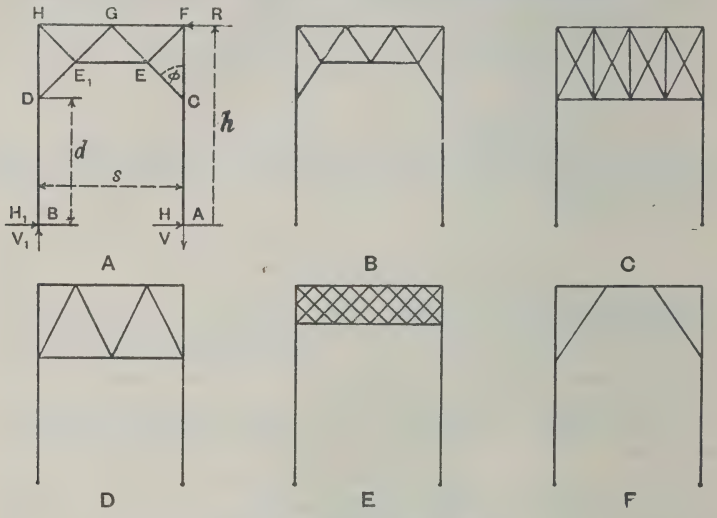


FIG. 18.—Types of Portal Bracing.

The shear in the posts is H below the foot of the knee brace, and above the foot of the knee brace is given by the formula—

$$S = \frac{Hd}{h-d} = \text{stress in HG.}$$

The maximum moment in the posts occurs at the foot of the knee braces C and D, and is

$$M = Hd.$$

The stresses in portal (*b*) are found in a similar manner, but it will be found that all members are stressed in portals (*b*) and (*d*).

Deformation and Deflection.

ACTUAL AND THEORETICAL DEFLECTION.

There is usually a difference between the actual and calculated deflection of framed structures. This is due to various causes, principally the rigidity and stiffness of the joints or otherwise, and the actual deflection is usually in excess of that calculated.

Contrary to statements that the amount of deflection is independent of any change of form which may take place in the web, and is not affected by the form of web, it is found that the deformation of the web members contributes largely to the total deflection.

DEFLECTION OF A LATTICE GIRDER.

The theoretical deflection of a framed structure can be found both analytically and graphically when the elastic deformation of each member caused by the stress in it is known.

The following method is that developed by Maxwell and Mohr, and is taken from Sir W. Arrol & Co.'s Handbook :—

Example.—Find the deflection at the centre of a lattice girder 100 ft. span, 10 ft. deep, and loaded with a distributed load of 1 ton per ft. run (see fig. 19).

Let

L = length of any member in ft.; *l* = length of any member in ins.; S = total stress in any member due to 100 tons distributed load; P = stress per sq. in. in any member due to 100 tons distributed load; ϕ = total stress in any member due to 1 ton placed at the point at which

It is required to find the deflection ; A = gross area of any member in sq. ins. plus 5 to 10 per cent. to allow for the effect of covers, joints, etc. ; E = modulus of elasticity of steel which may be taken as 12,000 tons per sq. in.

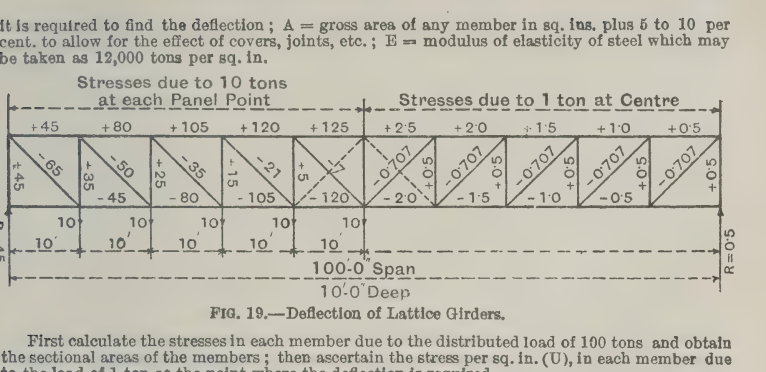


FIG. 19.—Deflection of Lattice Girders.

First calculate the stresses in each member due to the distributed load of 100 tons and obtain the sectional areas of the members ; then ascertain the stress per sq. in. (U), in each member due to the load of 1 ton at the point where the deflection is required.

The elastic deformation of any member is $\frac{S}{EA}$ (or $\frac{LS}{1000\,A}$, if L is in ft.).

The results are then placed in tabular form as follows :—

Member.	L. Length	S. Total Stress	A. Area.	P. Stress per Square Inch.	$\frac{P\,l}{E}$.	U.	Contribution to Deflection $\frac{P\,l\,U}{E}$	Summation.
Top Boom	Ft.	Tons.	Sq. Ins.	Tons.	In.	Tons	In.	In.
Bay 1 . .	10	+ 45	9	+5	+0·05	+0·5	+0·025	
,, 2 . .	10	+ 80	16	+5	+0·05	+1·0	+0·050	
,, 3 . .	10	+105	21	+5	+0·05	+1·5	+0·075	
,, 4 . .	10	+120	24	+5	+0·05	+2·0	+0·100	
,, 5 . .	10	+125	25	+5	+0·05	+2·5	+0·125	
								+0·375
Bottom Boom								
Bay 1 . .	10	Nil.	Nil.	Nil.	Nil.	Nil.	Nil.	
,, 2 . .	10	− 45	7·5	−6	−0·06	−0·5	+0·03	
,, 3 . .	10	− 80	13·4	−6	−0·06	−1·0	+0·06	
,, 4 . .	10	−105	17·5	−6	−0·06	−1·5	+0·09	
,, 5 . .	10	−120	20·0	−6	−0·06	−2·0	+0·12	
								+0·300
Web.								
Tie 1 . .	14·14	− 65	13	−5	−0·0707	−0·707	0·05	
,, 2 . .	14·14	− 50	10	−5	−0·0707	−0·707	0·05	
,, 3 . .	14·14	− 35	7	−5	−0·0707	−0·707	0·05	
,, 4 . .	14·14	− 21	4·5	−5	−0·0707	−0·707	0·05	
,, 5 . .	14·14	− 7	1·4	−5	−0·0707	−0·707	0·05	
								+0·250
End Post .	10	+ 45	11·25	+4	+0·04	+0·5	+0·02	
Strut 1 . .	10	+ 35	8·75	+4	+0·04	+0·5	+0·02	
,, 2 . .	10	+ 25	6·25	+4	+0·04	+0·5	+0·02	
,, 3 . .	10	+ 15	3·75	+4	+0·04	+0·5	+0·02	
,, 4 . .	10	+ 5	1·25	+4	+0·04	+0·5	+0·02	+0·100
							Half sum	+1·025
								2
						Total	deflection	+2·050

The required deflection of the girder is thus found to be 2·05 in., the flanges contributing 1·35 in., and the web 0·70 in.

SECTION VII.

PART IV.

SWING AND BASCULE BRIDGES.

PARTICULARS OF BRIDGES—WORKING STRESSES—FRICTION AND LUBRICATION—HYDRAULIC LIFTING AND SLEWING MACHINERY—CIRCULAR GIRDERS.

SWING BRIDGES.

The most usual types are either of the 'centre bearing' or 'rim bearing' type, and may be either 'single leaf' or 'double.'

The depth of a single-leaf swing bridge is four times greater than that required for a double-leaf bridge of the same span, when the bridge is treated while in motion as a cantilever.

On the other hand, the length of a single-leaf swing bridge is less than the combined lengths of the two leaves for the same opening, but the double leaf type requires two sets of operating machinery.

The full advantages of swing bridges are only to be obtained when they span two openings; in such cases the necessity for counterpoises or kentledge does not exist and the effects of wind pressure are to a great extent neutralised.

A further classification of types is as follows:—

(*a*) Those which turn entirely upon rollers, *i.e.* the rim bearing type.

(*b*) Those where the weight is proportioned to be borne partly by rollers and in part by a centre pivot.

(*c*) Those swung entirely upon a pivot.

(*d*) Those which are lifted on a water centre by hydraulic power.

(*e*) Those which are lifted on a water centre by hydraulic power, but not sufficient to lift the whole weight.

In the latter type the tail is heavier than the toe, and the excess weight is borne by rollers running on a roller path, and adds stability to the bridge when swinging.

In type (*e*) of the above classification should also be included those swing bridges which are carried on a circular pontoon, the effect of which is practically a low-pressure hydraulic ram.

The general principle of these water-borne swing bridges is as follows:—A sealed circular pontoon or buoy is placed under the centre of gravity of the superstructure, and suitably and rigidly connected thereto in such manner as to be always submerged with a constant buoyancy and capable of being turned with the superstructure.

The upward pressure due to the buoyancy of the pontoon is less than the total weight of the bridge, the remainder of the weight resting upon roller paths which are carried on a gridiron or circular girder supported upon piles in the bed of the river, as in several bridges over the river Weaver, Cheshire, designed by Mr. John A. Saner, M.Inst.C.E.

One of these bridges consists of two braced girders 112 ft. long, 17 ft. high over the centre of gravity, with a 19 ft. 6 ins. roadway, and weighs 303 tons, of which 255 tons is water born e.

In this case the consumption of electric power to turn the bridge is 0·24 of a unit for each swing, a swing including the withdrawal of the wedges opening and closing of bridge and replacement of wedges.

It is stated that this type of swing bridge is more economical in operating expenses than any other swing bridge of equal size.

A swing bridge at the Spencer Dock, Dublin, has the pontoon working in an outer caisson, the 5 per cent. excess weight of the bridge being carried upon a centre pivot placed at the bottom of the pontoon. Lateral rollers with the roller path fixed to the underside of the bridge are provided in this design.

Lattice girders are preferable for all spans in excess of 100 ft. on account of their much less resistance to wind pressure.

In all cases where rollers at the tail end are provided, an excess of counter weight should be provided, so that the rollers are loaded when the bridge is swinging with not less than 10 tons when the bridge is covered with snow and allowance has been made for the increase in weight of the roadway and footpaths due to wet.

Hydraulic presses or wedges should be provided at heel and toe to relieve the rollers and pivot of weight when the bridge is in the fixed position for sustaining road traffic.

BASCULE BRIDGES.

Bascule bridges may be either single leaf or double, but are usually double-leafed.

The most modern types are as follows :—

(*a*) Those in which the pivot is overhead and the balance weight either overhead or below the level of the bridge floor.

(*b*) Those where the pivot is situated below the level of the bridge floor and the counter-weight likewise.

(*c*) Those in which there is no fixed axis, but the tail of the bridge is in the form of a circular segment, upon which the bridge rolls backward and forward.

PARTICULARS OF BRIDGES.

SWING BRIDGES.

Name of Bridge.	Span (Opening).	Width of Road.	Length of Long Arm.	Length of Short Arm.	Depth of Main Girders at Centre over Angles.	No. of Rollers.	Diam. of Roller Circle.	Weight (less Kentledge).
	Ft. Ins.	Ft. Ins.	Ft. Ins.	Ft. Ins.	Ft. Ins.		Ft. Ins.	Tons.
Manchester Ship Canal, Old Quay, Runcorn .	120 0	20 0	139 0	96 10	28 0	60	22 11	650
Moore Lane .	120 0	25 0	140 0	98 0	27 8½	64	27 10½	790
Stagg Inn. . .	120 0	25 0	140 0	98 0	27 8	64	27 10½	790
Northwich Road .	120 0	36 0	148 0	100 0	30 0	60	38 9	1350
Knutsford Road .	120 0	36 0	148 0	100 0	30 0	60	38 9	1350
Barton Bridge .	90 0	25 0	111 0	81 0	26 0	64	27 10½	640
East India Docks, London (Road and Railway) .	80 0	11 6	95 0	30 0	16 8	Centre	Pivot.	480
King George V. Dock, London, (Skew Bridge, 22°) . .	108 0	20 8 and 2 footpaths.	150 0	50 0	25 0	Centre	Pivot.	950
Kilbowie (Railway) (Skew Bridge, 24°) . .	22 3	—	152 7 133 8	36 4	17 6	—	19 4	416
Dry Docks, Marseilles, France (one road, one rail-track, 3 main girders) .	92 0	52 6 and 1 footway	126 0	77 5	11 6	Centre	Pivot	575

BASCULE BRIDGES.

Name of Bridge.	Span (Opening.)	Width of Road.	Depth of Main Girder at Cr. over Angles.	Type.	Weight.
	Ft.	Ft. Ins.	Ft. Ins.		Tons.
Southern Railway, over River Swale, Kent .	62	33 0 centre of girders.	—	Rolling lift.	520 (plus counter-weight.)
Cart Navigation Trust, Inchinnan . . .	90	20 0	20 5	Rolling lift	207
King George V. Dock, London (double leaf) .	50	17 6 (and 2 footways)	29 0	Pivot.	381
Taylor Street, Chicago (Scherzer double leaf) .	295	28 0	40 0	Rolling lift.	—

Working Stresses.

PERMISSIBLE WORKING STRESSES.

The maximum permissible unit stresses per sq. in. should be less for the main girders, cross girders, and rail bearers than those allowed for simply supported spans; about 10 per cent. is a usual figure. (Refer to ' Working Stresses,' p. 264.)

ROLLERS.

Rollers supporting swing bridges should generally be of solid forged steel or cast steel, with suitable gun-metal liners and collars and provision for adjustment at the ends of the rollers.

The rollers and surfaces of both upper and lower paths should form parts of conical surfaces with a common vertex at the centre.

The size of the rollers should bear some relation to the diameter of the roller circle, and in general will not exceed 2 to 3 ft. diameter.

BEARING PRESSURE ON ROLLERS.

The following safe bearing pressures in pounds per lineal inch on rollers are taken from the *Transactions of the American Society of Civil Engineers*, vol. xxxiii.

Description.	Rollers at rest.	Rollers in Motion.
Cast Iron	400 d	200 d
Rolled or Cast Steel (28–32 tons) . . .	800 d	400 d
Axle Steel (35–38 tons)	—	500 d
Tool Steel (60 tons)	—	800 d
Tool Steel hardened	—	1000 d

where d is the diameter in inches.

The values given are for rollers and bearing surfaces of the same material; if they are of different materials, lower values should be used.

Sir W. Arrol & Co., in their Handbook, give the safe load in pounds per lineal inch on rollers, as 300 d, where d = the diameter of the roller, the mean diameter being taken for conical rollers, the rollers and plates being of cast iron, wrought iron, or steel; for rollers in motion one-half the above value to be taken. It is stated that the working stress is one-fifth the elastic limit for cast iron, and one-third the limit for wrought iron or steel, the elastic limit being reached in steel plates and steel rollers of 1 in. to 16 ins. diameter at 880 d.

Bearing Pressures on Rotating and Sliding Surfaces.

MAXIMUM BEARING VALUES FOR ROTATING AND SLIDING SURFACES IN POUNDS PER SQUARE INCH.

Bearings on which the Speed is slow and intermittent.	
Pivots for swing bridges, hardened tool steel on special phosphor-bronze	3,500
Trunnion bearings for bascule bridges, axle steel on phosphor-bronze	2,000
Wedges, cast-iron on bronze	600
Wedges, cast-iron on cast-iron or structural steel	500
Screws which transmit motion on projected area of thread	200
Ordinary Cases, Parts moving at Moderate Speeds.	
Hardened steel on hardened steel	2,000
Hardened steel on bronze	1,500
Tool steel (not hardened) on bronze	900
Structural steel on bronze	600
Cast-iron on structural steel	400
Cast-iron on cast-iron	400
On crosshead slides, speed not exceeding 600 ft. per minute	50

In order to prevent heating and seizing at higher speeds, the pressure on pivots or foot-step bearings for vertical shafts and journals should not exceed :—

$$\text{On pivots, } p = \frac{160000}{n\,d}\text{; on journals, } p = \frac{300000}{n\,d},$$

where

n = number of revolutions per minute; d = diameter of journal or pivot in inches.

For crank pins and similar joints with alternating motion, the limit bearing values given in the above formula may be doubled.

Ball and Roller Bearings.

(See Section XX., p. 961.)

Centre Pivots.

The construction of centre pivots for swing bridges requires careful consideration. They should be of ample proportions, and the pressure per square inch should not exceed 2 to 3 tons per square inch, or excessive friction will ensue.

The pivots should preferably consist of four parts; the lower casting and upper castings of steel, with an upper revolving pin of tool steel with a convex surface, and the lower fixed cup of phosphor bronze with a concave surface of slightly larger radius than the upper revolving pin, to assist lubrication.

Pivots have been designed in which a 'Tabloid' was placed between the upper revolving pin and the lower stationary cup. The 'Tabloid' was of phosphor bronze with spherical upper and lower surfaces, the upper pin on the bridge and the lower cup also having spherical surfaces. This type of pivot is, however, only suitable for bridges in which the slewing power is applied at two or more diametrically opposite points, for where the slewing force is applied at only one point, a slight displacement of the bridge on the seating has been found to occur, resulting in considerable eccentricity.

Pressures on Quadrants and Tracks of Rolling Lift Bascule Bridges.

The following data refer to four bridges constructed during recent years :—

Clear Span.	Radius of Quadrant.	Rolling Load.	Width of Bearing.	Pressure on Track.	Materials in Quadrants and Tracks
Ft. Ins.	Ft. Ins.	Tons.	Inches.	Tons per Lineal Inch.	
58 3	18 3	414	26·0	15·92	Mild steel 28–32 tons.
100 0	21 0	631	38·0	16·61	Quad. plate mild steel.
100 0	21 0	516	22·0	23·45	Track plate cast steel
150 0	28 0	2,922	50·25	58·15 average.	Track plate cast steel 35 tons. Quad. plate mild steel 36-40 tons.

(*Selected Paper No.* 8, *Inst.C.E.*)

Friction and Lubrication.

Rolling Friction.

Let

P = force required to propel roller (lbs.); W = load on roller (lbs.); R = radius of roller (ins.); h = depth of depression caused in track (ins.); K = a constant; then,

$$P = \frac{WK}{R - h},$$

or, as h is usually very small compared with R,

$$P = \frac{WK}{R} \text{ (nearly).}$$

The value of K varies from 0·007 to 0·015.

Some Experiments on Friction.

The following experiment, undertaken by Sir Wm. Arrol & Co., Ltd., is particularly applicable to the friction of roller rings in turntable swing bridges :—

Two beams, each 175¼ ins. long by 5¾ ins. broad, each weighing 1570 lbs., were planed to a smooth surface and accurately levelled. One beam was laid dry upon the other and afterwards two rollers, 2¼ in. diameter, were placed 7 ft. apart between them. The force required to start motion was obtained for different loads.

In experiments with eleven American turntable swing bridges, the friction showed a maximum of 7·94 lbs. per 1,000 lbs. weight of moving structure; but one bridge showed as low as 3·53 lbs.

Further experiments in America on new swing bridges, before the gearing, etc., had had an opportunity to wear smooth, were as follows: for bridges which have been in operation for some time, the coefficients given will be less:—

Load.	Force to Start Motion.		Coefficient of Friction.	Percentage of Load.	Remarks.
	Total.	Pounds Per Ton.			
lbs.	lbs.	lbs.			
1570	450–500	650–716	0·29–0·32	29–32	Sliding Friction.
2130	475–560	500–590	0·224–0·263	22–26	,,
1570	2·73	3·9	0·00174	0·17	Rolling friction.
2240	5·44	5·44	0·00243	0·24	,,

For rim-bearing swing bridges, the late C. Shaler Smith found the total frictional resistance in the livering to vary from 0·004 to 0·008 of the load on the rollers. Messrs. Boller & Schumacher found the coefficient to be 0·0035 for the Thames River Bridge, and Theodore Cooper found this to be 0·0038 for the Second Avenue Bridge.

For centre bearing bridges, Mr. C. Shaler Smith found the frictional resistance at the circumference of the pivot to be 0·09 of the weight turned. Mr. Schneider found the coefficient of frictional resistance at the circumference of the pivot, on hardened steel and phosphor bronze discs, with the usual working pressure of about 3,000 lbs. per sq. in., to be 0·067 at the start and 0·015 to keep the bridge moving at uniform speed. For the total frictional resistance, including that of the shafts and gearing required for hand operations, the highest coefficients observed were 0·115 for starting and 0·08 for keeping the bridge in motion.

(*Proceedings American Society of Civil Engineers.*)

Experiments by Beauchamp Tower show that a steel shaft in a lubricated gun-metal bearing seizes at about 600 lbs. per sq. in. under steady running, while in a dry bearing they seized at about 80 lbs. per sq. in.

Professor Goodman found that the seizing pressure increases as the viscosity of the oil increases.

Horse-Power Absorbed in Revolving Bearings.

Let

W = total load on bearing in lbs.; D = diameter of bearing in ins.; N = number of revolutions per minute; ϕ = coefficient of friction.

For Cylindrical Bearings:

$$\text{Work done per minute in ft.-lbs.} = \frac{\phi W \pi D N}{12};$$

$$\text{Horse-power absorbed} = \frac{W \pi D N \phi}{12 \times 33{,}000} = \frac{\phi W D N}{126{,}000}.$$

For Flat Pivots:

$$\text{Horse-power absorbed} = \frac{\phi W D N}{132{,}000}.$$

Lubrication.

Oil should always be led to the point of least pressure in a bearing.

Oils for heavy pressure should be thick, that they may resist being forced out.

Lubricants for Various Cases.

Nature of Pressure, etc.	Type of Lubricant.
Heavy pressure with slow speed . . .	Graphite, soapstone, tallow, and other greases.
Heavy pressure with high speed . . .	Sperm oil, castor oil, and heavy mineral oils.
Light pressure and high speed . . .	Sperm oil, refined petroleum, olive, rape, and cotton-seed oils.
Ordinary machines	Lard oil, heavy mineral and vegetable oils.

Oils act chemically in a varying degree on most metals, and care must be exercised in selecting the most suitable kind.

HYDRAULIC LIFTING AND SLEWING MACHINERY.

Power required for Lifting Swing Bridges.

This is a simple calculation based on the total weight of the bridge lifted and the friction due to moving parts, leathers, and so on.

Power required for Slewing Swing Bridges.

$P = 0{\cdot}015$, for rim-bearing; $P = 0{\cdot}15\ W \frac{r}{R}$, for centre-bearing; H.P. $= \frac{P\,v}{550}$; $v = \frac{R\,\pi}{2t}$,

where

P = Force required to turn the bridge, acting in centre line of track; W = total moving weight, in lbs.; H.P. = horse power; r = radius of pivot; R = radius of centre line of track; v = maximum linear velocity in centre line of track in feet per second; t = time, in secs., in which maximum velocity must be obtained, or one-half the number of seconds required to open the bridge.

The foregoing formulæ generally give results in excess of the power actually needed under ordinary conditions, and practically cover all contingencies.

If the bridge has to be operated by hand-power, the gearing has to be arranged either for the number of men available or for the time required to turn and operate the bridge. In calculating the strength of the gearing, etc., the power of one man should be taken at 125 lbs., as this is about the force a strong man, with a foothold, can exert for a short time.

Setting and Wedging Blocks.

It is usually not advisable to allow a bridge to rest upon the pivot when undergoing stress due to moving loads.

In some bridges the centre pivot is also the lifting press, bearing blocks being provided at suitable points to sustain the bridge when in position. In this case, however, there is difficulty in preventing the ram from turning in the cylinder.

With a solid pivot, the hydraulic lifting rams are placed at the tail of the bridge, where, when the rams have lifted the bridge off the pivot, a pair of sliding bearing blocks are inserted and the lifting power then withdrawn. In this type four or more rollers are provided to take a portion of the weight of the bridge while slewing, and to keep the bridge 'upon an even keel.'

The two leaves of a double swing bridge are often linked together at the meeting point, the object being to prevent any perceptible difference in level between the extremities of the bridge leaves when a heavy load is transferred from one to the other

CIRCULAR GIRDERS.

Bending Moments in Circular Girders.

(Equally spaced Supports.)

No. of Points of Support.	Reaction at Point of Support (lbs.).	Maximum Shear (lbs.).	B.M. Over Point of support (in.-lbs.).	B.M. Midway between Supports (in.-lbs.).	Angular Distance Point of support to Point of Maximum Torsion.	Maximum Torsional Moments (in.-lbs.).
					° ′	
4	$\frac{W}{4}$	$\frac{W}{8}$	$-0{\cdot}03415\ Wr.$	$+0{\cdot}01762\ Wr.$	19 12	$0{\cdot}0053\ Wr.$
6	$\frac{W}{6}$	$\frac{W}{12}$	$-0{\cdot}01482\ Wr.$	$+0{\cdot}00751\ Wr.$	12 44	$0{\cdot}00151\ Wr.$
8	$\frac{W}{8}$	$\frac{W}{16}$	$-0{\cdot}00827\ Wr.$	$+0{\cdot}00416\ Wr.$	9 33	$0{\cdot}00063\ Wr.$
12	$\frac{W}{12}$	$\frac{W}{24}$	$-0{\cdot}00365\ Wr.$	$+0{\cdot}00190\ Wr.$	6 21	$0{\cdot}000185\ Wr.$

W = Total uniform load supported by girder in lbs.; r = radius of girder in ins.

SECTION VII.

PART V.

ARCH BRIDGES.

Types and Stresses.

Arched Ribs.

The arched rib or roadway girder may be in either of the following forms:—

(*a*) Rigidly fixed at each end.
(*b*) Hinged at each end.
(*c*) Hinged at centre and at each end.
(*d*) Hinged near one abutment.

Stresses in Arches.

The arch may be looked upon as an inverted suspension bridge, the links of the suspension bridge being in tension and those in the arch being in compression, but the latter would be in unstable equilibrium under any loading, and so in practice the arch is made capable of resisting bending stresses in addition to direct compression.

Horizontal Thrust.

When once the horizontal thrust for any given arch has been found, it is a simple matter to obtain the stresses.

The horizontal thrust can be determined accurately in three cases only, viz.:—

(1) A parabolic arch uniformly loaded.
(2) A parabolic arch with uniform load over half the span.
(3) A three-pinned arch.

Linear Arch and Line of Pressure.

A linear arch is one which is theoretically supposed to be subject to direct compressive stress only and incapable of resisting flexure.

The line of pressure is the theoretical line, the ordinates of which vary as the bending moments for the load which the arch supports. It is synonymous with the curve of equilibrium.

Eddy's Theorem is thus:

'The Bending Moment at any point in an arch is equal to the product of the Horizontal Thrust into the vertical intercept between the centre line or neutral axis of the arch and the line of pressure.'

It is almost universally the practice with steel arches to hinge them at the abutments, and often at the crown also. By this means the alterations in form of the arch due to the passage of the load and variation of temperature more readily adapt themselves to the varying conditions. Moreover, the resulting thrusts pass through these points and the axis of the arch, or approximately so, and the bending stresses are thus reduced and can be determined with greater accuracy.

Stresses in Parabolic Arch, Hinged at the Abutments.

Let

l = length of span of arch; v = versine or rise; w = load per unit of length, T = horizontal thrust at the crown and abutments; C = compressive stress at any point x from the centre, then

$$T = \frac{w l^2}{8v}$$

This expression represents the horizontal component of the pressure at any point in the arch

$$C = \sqrt{\left(\frac{w l^2}{8v}\right)^2 + (wx)^2}.$$

When loaded with a single weight W at the centre of the arch

$$T = \frac{Wl}{4v}.$$

If W rests at a point x from centre of arch, then,

$$T = \frac{W(l - 2x)}{4v}.$$

When loaded with a uniform load over half the span,

$$T = \frac{wl^2}{16v}.$$

STRESSES IN THREE-PINNED ARCH.

When loaded with a single weight W (see fig. 20),

$$T = \frac{W l}{4v}.$$

If the lines of pressure be inclined at an angle ϕ to the horizontal, the direct compressive stress, s, at any section in the rib is

$$s = \frac{W l}{4v} \sec \phi \cos \delta,$$

where

δ = angle which the tangent at the section under consideration makes with the line of pressure.

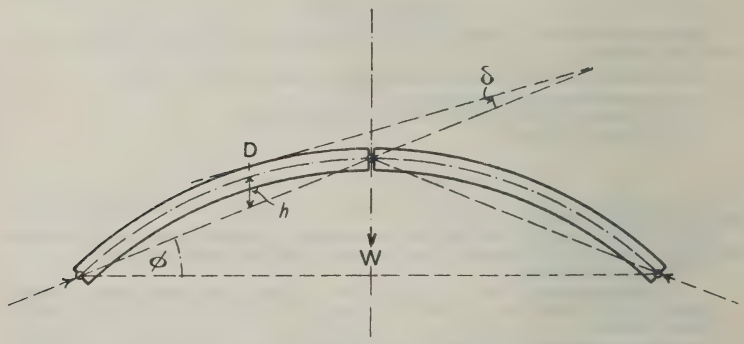


FIG. 20.—Three-Pinned Arch.

The bending moment at section D

$$= \frac{W l}{4v} \times h,$$

(h being the vertical distance between the line of pressure and the neutral axis of the rib at the given section).

For other loadings the same rules apply, the line of pressure always passing through the pin joints.

Combined Stresses in Arched Ribs.

In any arch where the line of pressure does not coincide with the neutral axis of the rib, there will be the following forces at any cross section :—

(1) A maximum compressive stress

$$= \frac{Q}{A} + \frac{M}{Z_c}.$$

(2) A maximum tensile stress

$$= \frac{M}{Z_t} - \frac{Q}{A}.$$

(3) A mean shear stress over the section

$$= \frac{S}{A},$$

where

Q = the thrust; S = the shear (obtained by resolving the horizontal thrust along (or tangential) and perpendicular to the neutral axis of the section under consideration); A = the area of the cross section of the rib; M = the bending moment obtained by applying Eddy's Theorem; Z_c and Z_t = the moduli of section in compression and tension respectively.

Stresses in Braced Arches.

Having found the direction and magnitude of the abutment reactions for any loading, the stresses can be found in a similar manner, those in the bracing being conveniently determined by Ritter's Method of Moments, described under 'Lattice Girders,' page 287.

Effects of Temperature Variation.

A rise in temperature causes an increased thrust in two-pinned and rigid arches, but theoretically causes no such increase in three-pinned arches, although practically some slight difference may be occasioned by any stiffness of the pin joints.

For a short research on Temperature Stress, see *The Theory and Design of Structures*, by Ewart S. Andrews, B.Sc., A.M.Inst.C.E.

Ratio of Rise to Span.

This is usually taken as from 1 : 5 to 1 : 10, but varies greatly according to local conditions of headroom.

The three-pinned arch over the river Exe—a road-bridge carrying tram-cars, erected in 1905, has a ratio of rise to span of 1 : 13·2, and is probably the flattest steel arch in existence in this country.

SECTION VII

PART VI.

SUSPENSION BRIDGES.

Types and Stresses.

Advantages and Disadvantages of Suspension Bridges.

The principal advantage is their cheapness, and in large spans the advantage of the suspension bridge is so great that bridges of this type of 800 or 900 ft. span can be constructed at much less cost per foot run than girder bridges of half this span.

There are many disadvantages attending suspension bridges, the chief being lack of stiffness and rigidity, both vertically and laterally, which makes them unsuitable for carrying railways unless elaborate means are taken for stiffening them.

A suspension bridge constructed on 'Bridge's System,' in which the suspension rods are inclined, is much stiffer vertically than one in which the rods are vertical.

Types of Stiffened Suspension Bridges.

In order to lessen the changes of shape and oscillations set up by a moving load, the roadway girders may be constructed of rigid form and braced laterally. The principal varieties of this and other types of stiffened suspension bridges are:

(*a*) Roadway stiffening girders hinged at each end and at centre.
(*b*) Roadway stiffening girders hinged at each end.
(*c*) Roadway stiffening girders hinged in either of the above ways and bridge consisting of two cables at each side stiffened by bracing.

Examples of some of the above types are as follows:—

Types (*a*) *and* (*b*).—Williamsburg Bridge, over East River, New York, built 1903, span 1,600 ft. Double line of railway, four lines of tramways, two roadways and two footways and cycle tracks.

Manhattan Bridge over East River, New York, built 1909, span 1,470 ft., four lines of tramway, one roadway, and two footways.

Type (*c*).—Proposed Hudson River Bridge, New York, clear span 2,880 ft., four lines of railway, four lines of tramway, two roadways, and two footways. A further example of this type is the Tower Bridge, London, in which the side spans are of this type.

An example of a bridge constructed with inclined and vertical suspenders is the Brooklyn Bridge, New York, built in 1884, span 1,595 feet, which carries a double line of tramway, two roadways, and one footway.

Stresses in Cable Uniformly Loaded.

The shape which a loaded cable takes up is the same as the link polygon for the given load system, drawn from one support to the other, with a polar distance equal to the horizontal component H of the pull or tension in the cable.

If

d = the dip of the cable; L = span; W = weight carried by the cable, then,

If

$$H = \frac{WL}{8d}.$$

A = area of the cable; f = safe tensile stress,

$$fA = \frac{W}{2}\sqrt{\left(1 + \frac{L^2}{16d^2}\right)}.$$

Stresses in Anchor Cables.

There are two chief methods by which the cables of suspension bridges are carried over the piers —

(1) Main and anchor cables passing continuously over rollers fixed to the top of the piers.

(2) Main and anchor cables passing continuously over saddles mounted on rollers on the tops of the piers.

Let

ϕ = angle which bridge cable makes with a vertical at point of support; δ = angle which anchor cable makes with a vertical at point of support; T = tension in bridge cable; T_1 = tension in bridge anchor cable.

Case (1).—Tension in anchor cable is equal to tension in bridge cable, whatever the inclinations,

$$= T_1 = T;$$

Horizontal force acting on piers $$= H\left(1 - \frac{\sin \delta}{\sin \phi}\right);$$

Vertical pressure on piers $$= \frac{W}{2}\left(1 + \frac{\cos \delta}{\cos \phi}\right).$$

Case (2).—Tensions in anchor and bridge cables not always the same, but there is no horizontal force on the piers,

$$T_1 = \frac{T \sin \phi}{\sin \delta};$$

Vertical pressure on piers $$= \frac{W}{2}(1 + \tan \phi \cot \delta).$$

Stresses in Suspension Bridges with Pin-Jointed Stiffening Girders. Type (*a*).

Uniform Rolling Load.

With a uniform load over the whole span there is no bending moment on the girders, the whole behaving as an unstiffened suspension bridge.

When the load covers half the span there will be a reverse or negative bending moment on the unloaded girder equal to the positive bending moment on the loaded girder.

$$\text{Maximum negative B.M.} = \frac{wl^2}{64};$$

and occurs at $\frac{1}{4}$-span.

The maximum positive bending moment at any point distant z from the centre pin, the head of the load being distant $x = \frac{l^2}{2l + z}$ from ditto

$$= \frac{wz\,(l^2 - z^2)}{2\,(l + z)}.$$

The absolute maximum positive bending moment = $0{\cdot}0753\ wl^2$ (or $\frac{wL^2}{53}$ approximately), and occurs when $z = 0{\cdot}53l$, the lead of the load being distant x from the centre pin.

Isolated Rolling Load.

With a concentrated rolling load passing over the girders :—

$$\text{Maximum negative B.M.} = \frac{WL}{16},$$

and occurs at $\frac{1}{4}$ span.

When the load is on one of the girders the bending moment is a maximum at any point when the load just reaches it.

Maximum B.M. at any point distant y from the centre pin

$$= \frac{Wy}{2L^3}(L^2 - 4y^2).$$

Absolute maximum positive B.M. = 0·096 WL, and occurs when load is 0·289L from the centre pin.

STRESSES WITH STIFFENING GIRDERS HINGED AT ENDS ONLY. TYPE (*b*).

In this case the stiffening girders will have to bear considerable stresses due to the increase in dip due to increase in temperature, which may amount to as much as half the safe stresses.

With a uniform rolling load passing over the span—

$$\text{Absolute maximum bending moment} = \frac{wL^2}{54},$$

and occurs at $\frac{1}{2}$-span, when the load covers $\frac{2}{3}$ of span.

The head of the load is always a point of contraflexure of the girder.

Assuming that the positive bending moment is a maximum at the mid point, then:

$$\text{Maximum bending moment in the length covered by the load} = \frac{wx^2(L-x)}{8L};$$

when head of load is distant x from the nearest pin.

in the foregoing:—

w = load in tons per foot run; W = concentrated rolling load in tons; L = length of span in feet; l = length of half span in feet.

CHANGES OF TEMPERATURE.

In England the observed expansion of steel structures appears to be about $\frac{1}{4000}$ of their original length from their coldest temperatures. This corresponds to a range of temperatures of 60° F. In suspension bridges, with movable saddles at the tops of the piers, the augmented dip of the central cable is due to increase in length of the anchor cables and consequent movement of the saddles as well as the increase in length of the central cable itself.

Let

l = the half span of central cable; D = central dip; D_1 = augmented dip due to rise in temperature of 60° F.; L = length of semi-parabola measured along curve; then by the properties of the parabola,

$$L = \sqrt{l^2 + \frac{4}{3}D^2};$$

and

$$D = 0{\cdot}866\sqrt{L^2 - l^2};$$

therefore

$$D_1 = 0{\cdot}866\sqrt{\left(L + \frac{L}{2400}\right)^2 - l^2}.$$

To this must be added the increased dip due to inward movement of the saddles.

ECONOMIC PROPORTIONS.

Economy in the weight of a cable can be effected by adopting a liberal depth or dip, but it is impossible in practice to increase the depth beyond a certain amount, because the greater the dip the greater will be the distortion and oscillation caused by a moving load.

In most of the existing examples of suspension bridges of various types, the dip does not exceed $\frac{1}{10}$ of the span, and in many cases $\frac{1}{12}$ to $\frac{1}{10}$ has been adopted.

SECTION VIII

JOINTING BEAMS—TENONS—STRUTTING AND PUGGING—STRENGTH AND STIFFNESS OF BEAMS AND POLES—ROOFS AND ROOFING—FLOORS—JOISTS—FIRE-RESISTING FLOORS—EXPANDED METAL—PARTITIONS—STAIRS.

(pp. 311-344.)

(With Contributions by Alexander Drew and Reginald Ryves, Assoc.M.Inst.C.E.)

SECTION VIII

JOINTING BEAMS—TENONS—STRUTTING AND PUGGING—STRENGTH AND STIFFNESS OF BEAMS AND POLES—ROOFS AND ROOFING — FLOORS — JOISTS — FIRE-RESISTING FLOORS — EXPANDED METAL — PARTITIONS —STAIRS.

JOINTING BEAMS.

Fishing.

To resist Tension.—For lengthening ties a plain fished joint is most economical for labour and material. The two ends of the beams are butted together, and secured on each side by means of bolts, as shown by fig. 1, which shows the fishing of a 6-in. × 4-in. beam with an iron plate on one side and a wooden plate on the other, the bolts being 1 in. in diameter with 1½-in. heads, and the washers 2½ ins. × ¼ in.

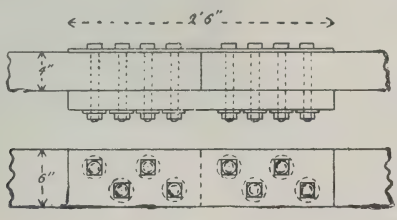


FIG. 1.

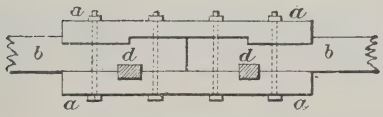


FIG. 2.

Another method is shown by fig. 2.

In this method, in order to relieve the strain on the bolts, the beam is indented or tabled as shown on upper side, or joggles or keys, *d*, *d*, are inserted. In either case the effective area is reduced by the indents or keys. The joint sectional area of the fish-pieces, *a*, *a*, should equal that of the beam, *b*. The joint sectional area of bolts should be at least one-fifth that of timber minus bolt-holes. Bolts are better square than round. Large washers should be under bolt-heads and nuts; for oak the washers should be 2½ times diameter of bolt, and for fir 3½ times.

Scarfing.

To resist Compression (fig. 3).—The bearing surfaces, *e*, *e*, should always be perpendicular to compressing force. Fig. 4 is a more elaborate form.

FIG. 3.

FIG. 4.

The lengths of scarfs should bear the following proportions to the depth :—

Wood.	Without Bolts.	With Bolts.	Bolts and Indents Combined.
Hard wood (oak, ash, elm) . .	6	3	2
Fir and straight-grained woods .	12	8	4

The sum of the depths of the indents should equal $1\frac{1}{3}$ depth of beam.

If assisted by fish-plates, it may be equal to the strength of a simple fished joint, which is as strong as the tie itself, less one rivet hole ; and no advantage is gained by making the joint complicated, but it should be formed similar to that shown by fig. 3.

To resist Tension.—Fig. 5 is often used. It will hold without bolts or straps. At *c* a key or joggle of hard wood is inserted, wedged in so as to tighten the joint moderately ; depth of key

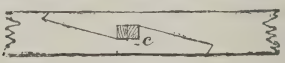


FIG. 5.

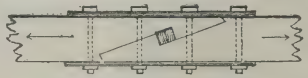


FIG. 6.

one-third depth of beam. This joint has only one-third the strength of solid timber tie ; it may be increased by bolting an iron fish-plate on each side, as in fig. 6.

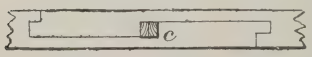


FIG. 7.

To resist Compression and Tension.—Fig. 7 is suitable. The wedges at *c* are only required when bolts are to be added, in which case they are used to bring the parts together before the bolts are inserted, in order to prevent violent strains on the latter.

To resist Cross Strains.—Fig. 8 is the best. The joint at *a*, being in compression, should be kept square ; the lower, having to resist tension, may be oblique. There should be as great a thickness as possible at *b c*. To resist cross-strain, the scarf is strongest when made vertical, *i.e.*, the pressure in the figure being at right angles to the paper.

FIG. 8.

FIG. 9.

To resist Cross Strain and Tension.—The resistance to tension is given by indenting the beam and inserting wedges, as shown in fig. 9.

T Joint.

A very effective method of securing two pieces of wood, A and B, at right angles to each other, is to insert in a mortice hole in B a piece of aluminium *a* which has been drilled with a hole into which an ordinary wood screw *s* is forcibly screwed. The aluminium being comparatively soft enables the wood screw to readily work its way through the hole, whilst it forms a nut which enables the screw to be very tightly screwed up as shown in fig. 10.

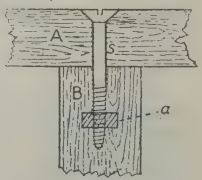


FIG. 10.

Building-up.

For building-up or joggling beams the method shown by fig. 11 may be adopted. This figure shows a portion of a 9 × 18-inch beam, 22 feet long, formed of two 9 × 9-inch beams secured

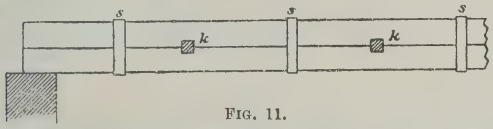


FIG. 11.

by wrought-iron straps, *s*, *s*, *s*, each 2 × $\frac{5}{16}$ inches; keys, *k*, each 2 × 2 inches, prevent the two pieces of which the beam is formed from sliding on one another.

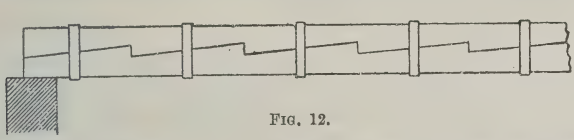


FIG. 12.

Fig. 12 shows a similar beam with the two halves indented the one into the other so as to prevent sliding.

Mortise and Tenon.

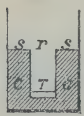


FIG. 13.

Mortising Upright into Horizontal.—The tenon, T (fig. 13), is usually one-third breadth of vertical. The pin-hole, if any, is generally placed at one-third length of tenon from shoulder, *s s*, and its diameter=one-fourth thickness of tenon. The mortise is made a little deeper, so that the shoulder may bear well; *r* is the *root*; C, C are the *abutment cheeks*.

Mortising Horizontal Beams.—The tenon is a projection from the end of the cross-beam, fitting closely into a mortise cut in the side of the main beam. It should be at the middle of its depth, so that the centre of the mortise may be at the neutral axis. Practically its lower edge is generally *on* the neutral axis.

Shouldered or Tusk Tenon.

To keep a cross-beam steady in its position a tenon requires length; to bear its share of load it requires depth; but a tenon long and deep would weaken the main beam too much. To avoid this a shouldered tenon is used. G (fig. 14) is the girder of a floor, B the binding joist, *a* the shoulder, *b* the tenon. The shoulder penetrates the girder about one-sixth depth of joist. Depth of tenon about one-sixth depth of joist; length about twice its depth, or it may be carried right through a thin girder and pinned, as in the case of a trimmer running into a trimming joist. Care should be taken that both tenon and shoulder bear exactly. Tredgold says the tenon should be one-third depth of joist above its lower edge. It should be, however, in the neutral axis.

FIG. 14.

Post and Beam Joints.—To frame a horizontal beam into the side of post the 'shouldered tenon' is used, but the long tenon should be on edge or have its narrowest dimensions horizontal.

When the beam rests on the top of the post, a small tenon in centre of post may fit into mortise on the under side of the beam. Or the post may fit into a shallow rectangular notch in beam, the notch being divided into two parts by a *bridle* left uncut in the middle of the notch, about one-fifth breadth of beam.

Wedges.

When the end of the tenon is seen, it is kept tight by driving wedges of hard wood dipped in glue or white-lead between tenons and sides of mortise. When end of tenon is not seen, small wedges, called *fox wedges* (fig. 15), are inserted in saw cuts in end of tenon; when tenon is driven home, they enter and expand it. The mortise is dovetailed or made slightly wider at bottom. This joint, when properly made, has the advantage that the post or style shrinks towards the shoulder and not away from it, and for this reason is sometimes employed for first-class work.

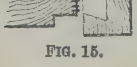


FIG. 15.

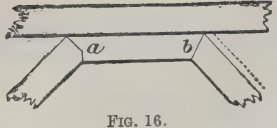


FIG. 16.

Joint of Straining-Beam.

In oblique joints under compression the joint should be either perpendicular to one piece, as at *a* (fig. 16), or bisect the angle, as at *b* ; the latter is the best in the case given in the figure. With a thicker strut at *b* it would be as shown by the dotted line.

Joints in carpentry should have white-lead applied with a brush over the surface forming the joint.

Strutting.

Joists laid with boards on top, and lath and plaster underneath, are not found to be sufficient to prevent bending and vibration under pressure. To remedy this, which is injurious to walls and to timber, strutting is placed in straight, continuous rows, at intervals not more than 4 feet apart, which reduces to a great extent the defect.

The following are three methods of strutting :—

Herring-bone Strutting.—This consists of pieces about 2 inches square fixed as shown in fig. 17.

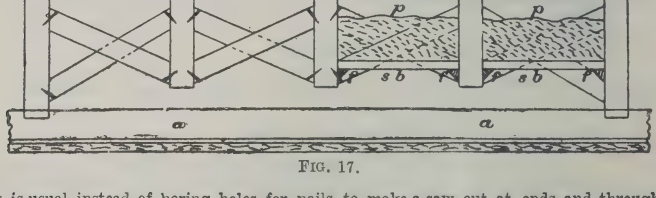


FIG. 17.

It is usual, instead of boring holes for nails, to make a saw cut at ends, and through this channel to drive the nail. By doing this any danger of splitting is avoided. This method has the recommendation that it acts where its leverage to prevent bending is greatest, viz., at the edges of joists, and is used largely in dwelling-houses.

FIG. 18.

Solid Strutting.—1 or 1¼-inch boards, nearly the same depth as joists, are cut in tightly, and nailed between joists, as in fig. 18. This method is largely used for floors of factories and heavy buildings ; and if cut so that the entire shoulder butts against the joists, is undoubtedly the firmest method; but unless care is exercised, the shoulders are apt to bite on the centre only. Sometimes long bolts, either of round bar or gaspipe, are passed through all the joists, and the strutting squeezed together with the joists and held tightly.

Key Strutting.—In this method struts are tenoned at their ends, and pass through mortises cut in the joists, and are secured with wooden keys. The method is expensive, weakens the joist, and altogether is a defective and unsatisfactory arrangement.

Pugging.

To prevent sound, sound-boarding, *s b* (fig. 17), is supported by fillets, *f*, *f*, cut diagonally out of 2-inch × 1¼-inch stuff nailed to bridging joists half-way between floor-boards and ceiling joist, *a*, *a* ; over these pugging, *p*, *p*, of coarse plaster about 1½ inch thick is put. Dry moss or a mixture of lime mortar, earth, and smith's ashes, or chopped straw, is sometimes used instead of plaster. Strips of felt are sometimes placed between the joists and boarding to deaden sound.

Strength and Stiffness of Beams and Poles.

b = breadth, *d* = depth, L = span.

Strength of Rectangular Beams.—For fir, safe distributed dead load, in cwts.

$$= \frac{bd^2 \text{ in inches}}{\text{L in feet}}.$$

For cedar and mahogany about the same load as for fir.

For larch	¾
For beech, Dantzic oak	1¼
For English oak, ash	1½
For teak	1¾
For greenheart	2

} Above load for fir.

This formula gives a theoretical factor of safety of about 8; the practical factor of safety for large scantlings is about 5.

Strength of Circular Poles is $\frac{6}{10}$ strength of square beam, diameter of pole being equal to side of square.

Stiffness of Beams and Poles.—When the beam or pole is loaded with the safe load given above the deflection will be about $\frac{1}{50} \cdot \frac{\text{L}^2 \text{ in feet}}{d \text{ in inches}}$ for fir, oak, mahogany; for teak, about half that for fir.

Deflection of Beams.

In timber beams for permanent purposes, supported or fixed at both ends, the deflection usually allowed is $\frac{1}{40}$ inch per foot run; $\frac{1}{80}$ for those fixed at one end.

$$\text{V} = \frac{n\text{W}c^3}{e\text{I}},$$

where

V = deflection in inches, W = load in pounds, c = length in inches for beams fixed at one end, half length if supported or fixed at both ends; I = moment of inertia, e = modulus of elasticity (page 184)

Description.	Value of n.		
	Uniform Cross Section.	Uniform Strength and Depth.	Uniform Strength and Breadth.
Fixed at one end, loaded at other	$\frac{1}{3}$	$\frac{1}{2}$	$\frac{4}{9}$
" " loaded uniformly	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{1}{2}$
Supported at both ends, loaded in middle	$\frac{1}{6}$	$\frac{1}{4}$	$\frac{1}{3}$
" " loaded uniformly	$\frac{5}{48}$	$\frac{1}{8}$	·1427
Fixed at both ends, loaded in middle	$\frac{1}{24}$		
" " loaded uniformly	$\frac{1}{48}$		

In the case of a beam of uniform strength, fixed at both ends and loaded at a point distant a_1 from one end and a_2 from the other, the deflection at point of application of load is—

$$\text{V} = \frac{(a_1 a_2)^2 \text{W}}{6eIc}.$$

The greatest deflection is at a point distant from one end $= a_2 - \sqrt{\frac{1}{3}a_2(a_2 - 2a_1)}$, a_2 being the larger division.

Table of Moments of Inertia.

Section of Beam.	I =	Section of Beam.	I =
Rectangular and square	$\frac{bd^3}{12}$	Ellipse, vertical semiaxis a, horizontal c	$\frac{\pi}{4}ca^3$
" hollow, $bd - b_1d_1$, also I-shaped section, where d_1 = depth of hollow parts, b_1 = sum of breadths of hollow parts	$\frac{bd^3 - b_1d_1^3}{12}$	Ellipse, hollow	$\frac{\pi}{4}(ca^3 - c_1a_1^3)$
		Semicircle standing with semicircular part up or down	·11r^4
Circle, radius r	$\frac{\pi}{4}r^4$	Isosceles triangle, base = b, height = d, standing on either base or apex	$\frac{bd^3}{36}$
" hollow	$\frac{\pi}{4} \cdot (r^4 - r_1^4)$		

All dimensions in inches; $\pi = 3{\cdot}1416$.

ROOFS.

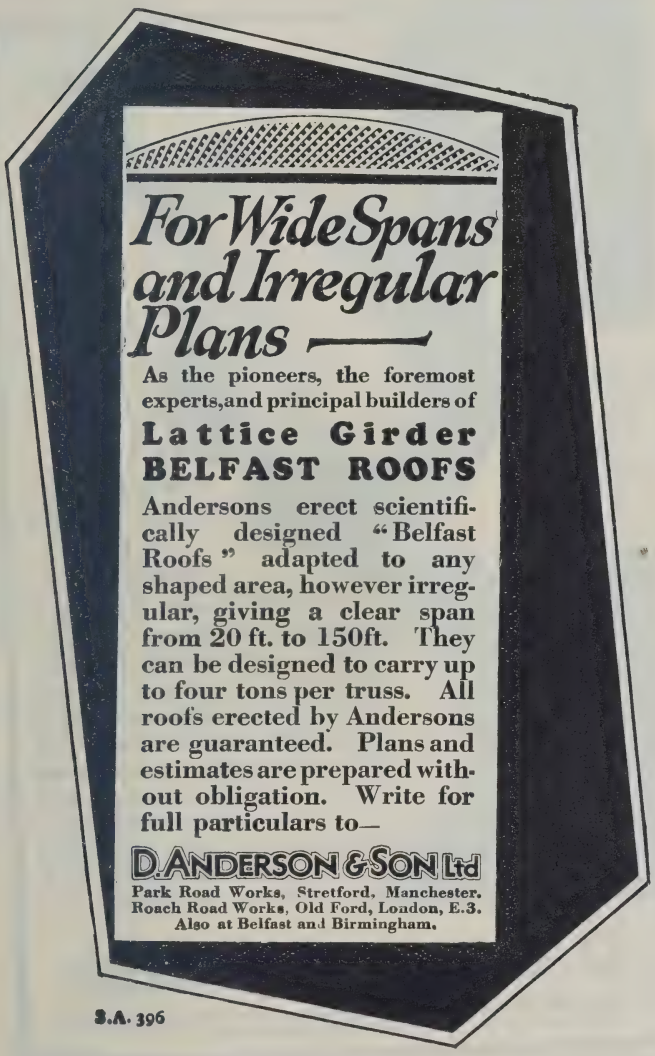
Weight of Components of Roofs.

	Per Square Foot.
Lead covering, including laps, but not boarding or rolls	5½ lbs. to 8½ lbs.
Zinc „ „ 14 to 16 zinc gauge	1½ „ „ 1¾ „
Corrugated iron, galvanised, 16 W.G.	3½ „
„ „ „ 18 W.G.	2½ „
„ „ „ 20 W.G.	2 „
Sheet iron, 16 W.G.	2½ „
„ „ 20 W.G.	1½ „
Slating laid with a 3-inch lap, including nails but not battens or iron laths :	
Doubles, 13 inches × 6 inches, at 18 cwt. per 1,200	8½ „
Ladies, 16 inches × 8 inches, at 31·5 cwt. per 1,200	8¼ „
Countesses, 20 inches × 10 inches, at 50 cwt. per 1,200	8 „
Duchesses, 24 inches × 12 inches, at 77 cwt. per 1,200	8½ „
Tiles, plain, 11 inches × 7 inches, laid with a 3-inch lap, and pointed with mortar, including laths and absorbed rain	18 „
„ pan, 13½ inches × 9½ inches, laid to 3-inch lap, and pointed with mortar, including laths and rain	12 „
„ Broomhall, laths excluded, rain included	8½ „
„ Italian (ridge and furrow), not including the boarding, but including the mortar and absorbed rain	14 „
Slate battens, 3½ inches × 1 inch :	
For doubles	2 „
For countesses	1¼ „
Slate boarding, ¾ inch thick	2½ „
„ „ 1 inch thick	3½ „
„ „ 1¼ inch thick	4¼ „
Wrought-iron laths, angle irons :	
For duchess slates	2 „
For countess slates	1⅝ „
Cast-iron plates, ⅜ inch thick	15 „
Thatch, including battens	6½ „
Felt, asphalted	3 „
Rok	½ lb. to ½ lb.
Ruberoid	½ lb. to ½ lb.
Lath-and-plaster ceiling	8 lbs.
Ceiling joists	3 „
Common rafters	3 „
Collar beams	2 „

Framing for wooden roofs, including purlins and ridge boards, but exclusive of tie beams :	
20 feet span, king post, rise one-fourth span	2 lbs. per foot super.
30 „ „ „ „	2 „ „ „
40 „ queen post „ „	2 „ „ „
50 „ „ „ „	3 „ „ „
60 „ „ „ „	4 „ „ „
Tie beams : 20 feet span, king post	11 lbs. per foot run of tie.
30 „ „	20 „ „ „
40 „ queen post	18 „ „ „
50 „ „	20 „ „ „
60 „ „	30 „ „ „

Angles of Roofs for Different Coverings.

Kind of Covering.	Inclination to the Horizon.	Height of Roof in Parts of Span.	Kind of Covering.	Inclination to the Horizon.	Height of Roof in Parts of Span.
	° ′			° ′	
Copper	3 50	$\frac{1}{30}$	Thin slabs of stone or flags	29 41	$\frac{2}{7}$
Lead	3 50	$\frac{1}{30}$	Pantiles	24 0	$\frac{2}{9}$
Zinc	4 0	$\frac{1}{29}$	Thatch of straw, &c.	45 0	$\frac{1}{2}$
Slates, large	22 0	$\frac{1}{5}$	Plain tiles	45 0	$\frac{1}{2}$
„ ordinary	26 33	$\frac{1}{4}$	Ruberoid	3 50	$\frac{1}{30}$
Asphalted felt	3 50	$\frac{1}{30}$			


For Wide Spans and Irregular Plans
As the pioneers, the foremost experts, and principal builders of
Lattice Girder BELFAST ROOFS
Andersons erect scientifically designed "Belfast Roofs" adapted to any shaped area, however irregular, giving a clear span from 20 ft. to 150ft. They can be designed to carry up to four tons per truss. All roofs erected by Andersons are guaranteed. Plans and estimates are prepared without obligation. Write for full particulars to—
D. ANDERSON & SON Ltd
Park Road Works, Stretford, Manchester.
Roach Road Works, Old Ford, Loadon, E.3.
Also at Belfast and Birmingham.
B.A. 396

THE ROOFING THAT DOES NOT CORRODE.

This flexible Bitumen Roofing is essentially of interest to Engineers. Undoubtedly the most efficient and economical covering for industrial buildings and other permanent structures. Easily and quickly laid on any type or shape of Roof. ROK is unaffected by moisture, salt and acid-laden atmospheres. Full particulars in detailed booklet from Mr. K. E. Stone.

D. ANDERSON & SON Ltd

Park Road Works, Stretford, Manchester. Roach Road Works, Old Ford, London, E.3. Also at Belfast and Birmingham.

S.A. 395

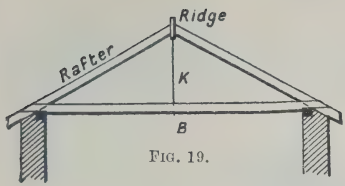


FIG. 19.

Single-span or Couple Roof.

May be used when walls can take the thrust without spreading; when they cannot, insert 4″ × 1″ binders as B (fig. 19), or ⅝′ iron ties, at about 10′ intervals, supported by a light iron king rod (K).

SCANTLINGS FOR COUPLE ROOFS.

Pitch up to 30°, with countess slates on 1″ boards.

Span.	Rafter.	Ridge Board.	*Ceiling Joist.	Span.	Rafter.	Ridge Board.	*Ceiling Joist.
Feet.	Inches.	Inches.	Inches.	Feet.	Inches.	Inches.	Inches.
8	3 × 2	7 × 1½	4 × 2	14	4½ × 2	7 × 1½	7 × 2
10	3½ × 2	7 × 1½	5 × 2	16	5 × 2	8 × 1½	8 × 2
12	4 × 2	7 × 1½	6 × 2	18	5½ × 2	8 × 1½	9 × 2

Collar-beam Roof.

Ceiling carried on under side of rafter and collar; collar best placed half-way up rafter; used for roofs up to 18′ span, under the same conditions as to wind, walls, and ties as given for couple roofs (fig. 20).

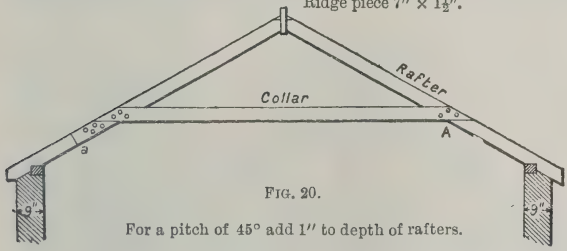


FIG. 20.

For a pitch of 45° add 1″ to depth of rafters.

The rafters are nailed to ridge-piece and bird's-mouthed over wall-plate, 1′ apart in the clear, or from centre to centre on very exposed sites.

SCANTLINGS FOR COLLAR-BEAM ROOFS.

Pitch up to 30°. If 45°, add 1″ to depth of rafters.

Span in Feet.	Rafters. For Countess Slates on ⅜″ Boards, 1′ clear apart, or centre to centre if very exposed Site.				Collars.	
	† Thrust taken by Walls or Ties, Collars ½ way up.		‡ Tied by Collars ¼ way up to prevent Walls spreading.		From ¼ to ½ way up.	At any Height.
	No Ceiling.	Ceiled to Collars.	No Ceiling.	Ceiled to Collars.	No Ceiling.	Ceiled to Collars.
	Inches.	Inches.	Inches.	Inches.	Inches.	2″ wide × ½″
8	1½ × 2½	1¾ × 3	2¼ × 3¼	2¼ × 3¾	1¾ × 2½	added to ½′
10	1¾ × 2½	1¾ × 3	2¼ × 4	2¼ × 4½	2 × 2½	for every foot
12	1¾ × 2¾	1¾ × 3¼	2¼ × 4¼	2¼ × 5	2 × 2¾	run of clear
14	1¾ × 3	1¾ × 3½	2½ × 5	2½ × 5½	2 × 3	length of
16	2 × 3¼	2 × 3¾	2½ × 5½	2½ × 6	2 × 3½	underside of
18	2 × 3¾	2 × 4¼	2½ × 6	2½ × 6½	2 × 4	collar.

* When used they act as ties, B (fig. 19), but without K. † Halve these collars on to rafters without cutting into latter. ‡ These rafters allow of the tension joint (fig. 21, page 318). If the collar is required halfway up, about one-fourth must be added to both breadth and depth of rafters, and ¾ in. to depth of collars. But with unstable walls, ties are far cheaper, and may be at long intervals if secured to wall-plates of sufficient width to take the thrust between the ties.

Fastenings of Collar-beam Roof.—Collars should never cut into rafters unless acting as ties; even then, the rafter being weakened at its weakest point, it has to be strengthened. If the walls or plates can take the thrust, the collars are struts, and under transverse stress if ceiled. Collar roofs to wooden huts constantly push out the walls, owing to the wall-plate or top sill not being stiff enough. Fig. 21 shows method of fastening collar C to rafter R, when collar acts as a tie,

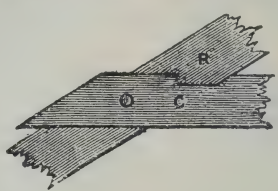


FIG. 21.

and, whether ceiled or not, it is better than a dove-tail; the collar and rafter are both notched about $\frac{1}{2}''$. When collar is a strut it may be halved on, as at A (fig. 20, p. 317); when intended also to be ceiled it should, in addition, rest on a bearer spiked to side of rafter, as at *a* (fig. 20, p. 317), the rafter being uncut in either case. The rafters abut on the ridge, and are nailed to it. They are bird's-mouthed on to wall-plate.

King-Post Roof.

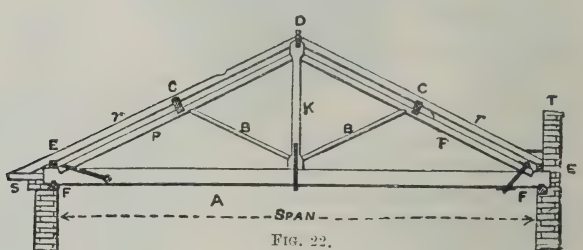


FIG. 22.

Used for spans from 20′ to 30′ (fig. 22).

A, tie-beam; B, braces or struts; C, purlins; D, ridge-board; E, pole-plate; F, wall-plate; K, king post; P, principal rafter; *r*, small or common rafter; S, eaves boarding; T, parapet.

SCANTLINGS FOR KING-POST ROOFS.

Span.	Tie Beam. A.	King Post. K.	Principal Rafters. P.	Braces. B.	Purlins. C.	Common Rafters. *r*.	Ridge. D.	Pole-plate E.
Feet.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
20	$9\frac{1}{2} \times 4$	4×3	4×4	$3\frac{1}{2} \times 2$	$8 \times 4\frac{3}{4}$	$3\frac{1}{2} \times 2$	6×2	4×4
22	$9\frac{1}{2} \times 5$	5×3	5×3	$3\frac{3}{4} \times 2\frac{1}{4}$	$8\frac{1}{2} \times 5$	$3\frac{3}{4} \times 2$	6×2	4×4
24	$10\frac{1}{2} \times 5$	$5 \times 3\frac{1}{2}$	$5 \times 3\frac{1}{2}$	$4 \times 2\frac{1}{2}$	$8\frac{1}{2} \times 5$	4×2	7×2	4×4
26	$11\frac{1}{2} \times 5$	5×4	$5 \times 4\frac{1}{4}$	$4\frac{1}{4} \times 2\frac{1}{2}$	$8\frac{3}{4} \times 5$	$4\frac{1}{4} \times 2$	8×2	4×4
28	$11\frac{1}{2} \times 6$	6×4	$6 \times 3\frac{1}{2}$	$4\frac{1}{2} \times 2\frac{3}{4}$	$8\frac{3}{4} \times 5\frac{1}{2}$	$4\frac{1}{2} \times 2$	8×2	4×4
30	$12\frac{1}{2} \times 6$	$6 \times 4\frac{1}{2}$	6×4	$4\frac{3}{4} \times 3$	$9 \times 5\frac{1}{2}$	$4\frac{3}{4} \times 2$	8×2	4×4

Trusses, 10′ apart; angle of roof, 26° 33′; covering, slate; wood, Baltic pine.

Wall-plates, F, $4'' \times 4''$ to $5'' \times 4''$; if the trusses are over openings the wall-plates should be increased.

Queen-Post Roof.

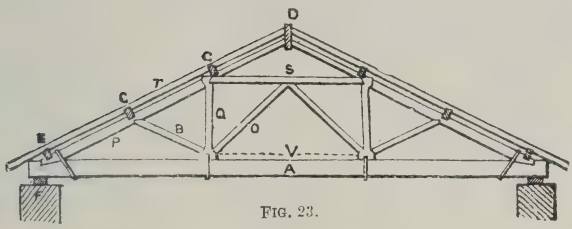


FIG. 23.

Used for spans over 30 and up to 46 feet (fig. 23). The braces, O, are generally omitted, and a straight straining-sill placed at V.

SCANTLINGS FOR QUEEN-POST ROOFS.

Span.	Tie Beam. A.	Queen Posts. Q.	Principal Rafters. P.	Straining Beam. S.	Braces. B.	Purlins. C.	Common Rafters r.	Ridge. D.	Pole-plate. E.
Feet.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
32	10 × 4½	4½ × 4	5 × 4½	6¾ × 4½	3¾ × 2½	8 × 4¾	3½ × 2	6 × 2	4 × 4
34	10 × 5	5 × 3½	5 × 5	6¾ × 5	4 × 2½	8½ × 5	3¾ × 2	6 × 2	4 × 4
36	10½ × 5	5 × 4	5 × 5¾	7 × 5	4¼ × 2½	8½ × 5	4 × 2	6 × 2	4 × 4
38	10 × 6	6 × 3¾	6 × 6	7¼ × 6	4¼ × 2½	8½ × 5	4 × 2	6 × 2	4 × 4
40	11 × 6	6 × 4	6 × 6	8 × 6	4⅝ × 2⅞	8½ × 5	4¼ × 2	7 × 2	4 × 4
42	11½ × 6	6 × 4½	6¼ × 6	8¼ × 6	4⅝ × 2¾	8¾ × 5¼	4½ × 2	7 × 2	4 × 4
44	12 × 6	6 × 5	6½ × 6	8½ × 6	4⅝ × 3	9 × 5	4¾ × 2	8 × 2	4 × 4
46	12½ × 6	6 × 5½	7 × 6	9 × 6	4¾ × 3	9 × 5½	5 × 2	8 × 2	4 × 4

Trusses, 10 apart ; pitch of roof 26° 33′, or ¼ ; countess slates ; timber, Baltic pine.
Wall-plates, F, 5″ × 4″ to 6″ × 4″ ; more if trusses are over openings.

Common Rafters.

12″ from centre to centre, or apart if not very exposed site.

Bearing in Feet.	Breadth in Inches.				
	1¾	2	2¼	2½	3
	Depth in Inches.	Depth in Inches.	Depth in Inches.	Depth in Inches.	Depth in Inches.
5	3	2⅞	2¾		
6	3½	3½	3½	3⅝	
8	4¾	4⅝	4⅝	4½	4
10	6	5¾	5½	5⅝	5
12	7	6¾	6½	6⅝	6
14	8¼	8	7¾	7½	7
16	9½	9¼	8¾	8½	8
18	10⅝	10¼	10	9½	9
20	11¾	11½	11	10½	10

Thin and deep rafters most economical ; should not be thinner than 1¾″ to 2″, or may split when boards are nailed to them. For stone or tile covering should be one-third stronger.

Joints of Wooden Roofs.

Joint at Foot of Principal Rafter (fig. 24).—Wood, Northern pine, *i.e.* red fir; rafter, $4'' \times 5''$; tie, $10'' \times 4''$; *ad*, $3''$; tenon, $\frac{1}{3}$ width of rafter.

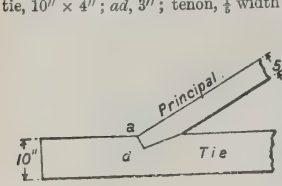


FIG. 24.

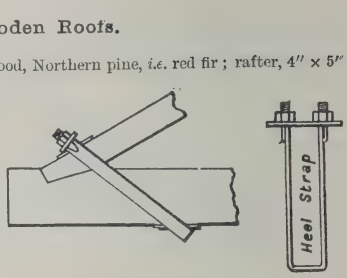


FIG. 25.

Iron Straps to Wood Roofs (fig. 25).—When longest unsupported part of tie beam

= 10 feet, strap should be . . . . . $1\frac{1}{2}'' \times \frac{3}{16}''$;
= 15 feet, " " . . . . . $1\frac{1}{2}'' \times \frac{1}{4}''$;
= 20 feet. " " . . . . . $2'' \times \frac{1}{4}''$.

The straps at the feet of the principal rafters may be of the same section as that for supporting the tie beam; a width less than $1\frac{1}{2}''$ is liable to cut into the fibres of the wood.

The strap in small roofs is generally fixed as shown on left-hand side of fig. 25.

Purlin.—Should be in pieces as long, and as nearly over strut B, as possible (fig. 26). The purlin, C, is notched over principal rafter, P, which is cogged to receive it; notches about $1'' \times 1''$. Common rafter, *r*, is notched on to purlin. The strut or brace, B, is tenoned into the principal rafter, P, as shown.

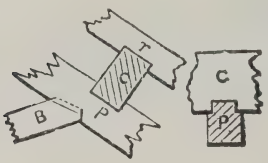


FIG. 26.

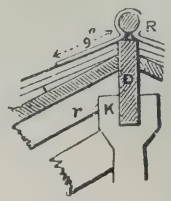


FIG. 27.

FIG. 28.

Ridge Board and Roll.—The ridge board, D (fig. 27), is let into king post, K, as shown. The common rafter, *r*, butts against and is nailed on to ridge board. R shows the ridge roll of wood with lead covering, ridge tiles, or Williams' patent ridge (fig. 28), in two pieces; the ends of the roll butt against each other, and are fixed with white lead, copper dowels, and $2''$ copper screws.

FIG. 29.

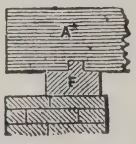


FIG. 30.

Wall-plate and Tie Beam.—Fig. 29 shows a wall-plate, F, cogged to receive tie beam, A, which should have a clear space round to prevent rotting. A stone template is sometimes substituted for the wall plate.

F, F, shows joint of wall-plates at angles of buildings, an arrangement which is better than a dovetail.

Rafter bird's-mouthed on to wall-plate or pole-plate, &c., spiked or skew-nailed down to it, is shown by fig. 30.

Pole-plate.—Generally rests on end of tie beam (fig. 22, page 318), or on foot of principal, being nailed or notched down. The position, however, depends upon the nature of the eaves.

King Post.—The king post, K (figs. 22 and 31), suspends the tie beam, with attached ceiling, if any, and also the principals, by means of the braces, B.

The principal rafter should abut square on the king post, as P. When head of post is not sufficiently long for this the joint may be made as P^1. The braces, B, should also abut perpendicularly on the king post. The ends of the principal rafters and braces are mortised into the king post.

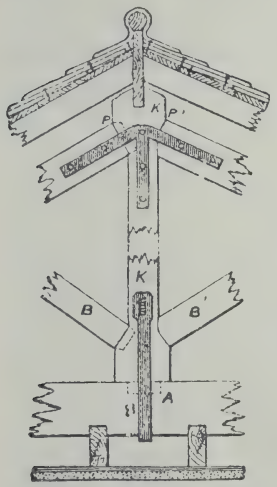


FIG. 31.

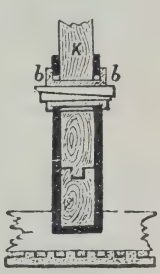


FIG. 32.

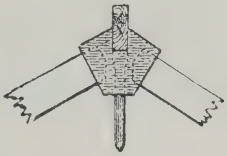


FIG. 33.

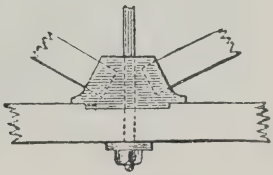


FIG. 34.

The king post is tenoned into the tie beam, A, with a short tenon, and fastened to it by an iron strap or *stirrup* iron. Gibs and cotters, as at *b*, *b* (fig. 32), enable the strap to be raised so as to tighten up the roof.

If a wrought-iron rod is substituted for a king post, its diameter in inches = square root of span of roof in feet × ·02.

Fig. 33 shows cast-iron head to receive rafters, and fig. 34 iron shoe to receive struts; a block of wood is often used for the feet of struts to butt against. When there are no gibs and cotters wedges may be driven in between underside of tie beam and foot of strap to force tie beam against foot of king post.

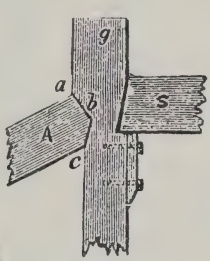


FIG. 35.

Queen Posts.—The abutments of the principal rafter, A, and struts of queen-post roof are the same as for the king-post. Joint of straining beam, S, with queen post, shown in fig. 35. If the head, *g*, of the post is narrow, the end of A should be cut away at an angle, *a b c*, so as to enable it to abut closely against the head; but if the latter is wide enough to admit of it, then the end of A may be cut square. If a wrought-iron rod is substituted for a queen post, diameter in inches = square root of length in feet of that portion of beam supported by queen bolt × ·29.

Pressure of Wind and Weight of Snow on Roofs.

The *Wind Pressure* is usually assumed to be equal to 40 lbs. per foot in cases where the inclination of the roof is 60°; for less angles than this the pressure is taken as follows :—

Inclination	5°	10°	15°	20°	25°	30°	35°	40°	45°	50°	55°	60°
Pressure in lbs. per sq. ft.	5·1	9·6	14·2	18·4	22·6	26·5	30·1	33·3	36·0	38·0	39·4	40·0

The *Weight of Snow* in Great Britain varies from 5 to 11 lbs. per cubic foot. It is not likely to accumulate on roofs to a greater depth than 6 inches; 5 lbs. per superficial foot of horizontal surface covered seems a quite sufficient allowance to make.

In Canada and the United States snow has been found in some instances to weigh as much as 30 lbs. per cubic foot some hours after it has fallen, and an allowance of at least 12 lbs. per superficial foot of horizontal surface covered should be allowed for it.

ROOFING MATERIALS.

Heat-Insulating Value of Roofing Materials.*

A National Physical Laboratory test of roofing by measuring the heat lost from a heated room leads to the remarkable conclusion that the rate of emission of heat by radiation from the covering surface has more effect on the inside temperature than the rate of conduction of heat through the material. This result is of importance in the roofing of large factories of a semi-permanent nature, where the temperature depends more upon the covering than in the case of buildings having a closed air space under the roof.

Old galvanised iron allows more heat to pass than glass, a remarkable result. Slates are good, but in proportion heavy. Deal boarding covered with asphalted felt, though thick, is not so heavy as Welsh slate, and is one of the best insulators. In this case its poor conductivity is evidently not negligible, and in all thick roofing materials it is probably of importance.

HEAT TRANSMITTED THROUGH ROOFING MATERIAL EXPOSED TO STRONG RADIATION.

Material.	Rise of Temperature in Testing Chamber Deg. C. per Minute.	British Thermal Units per 100 sq. ft. per Hour.	Thickness in Inches.	Weight in Pounds per Square Foot.
1. Bright galvanised iron sheet .	0·268	111	0·04	1·6
2. Galvanised iron, blackened below	0·40	108	0·04	1·6
3. Galvanised iron, blackened above	0·93	385	0·04	1·6
4. Galvanised iron, blackened above and below	1·40	581	0·04	1·6
5. Galvanised corrugated iron after one month's exposure to the weather	0·75	310	0·033	1·28
6. Do. one year	1·02	422	0·033	1·28
7. (6) painted black above . .	1·13	472	0·033	1·28
8. Roofing glass, serrated . . .	1·10	453	0·22	2·25
9. Welsh slate	0·81	337	0·17	2·9
10. Westmoreland slate . . .	0·60	248	0·25	4·8
11. ¼-in. T.G. deal covered with asphalted felt	0·30	124	1·0	2·6
12. Corrugated fibrocement after one month in use	0·78	325	0·2	1·8
13. Do. after one year in use . .	0·80	334	0·2	1·8
14. Do. painted dead black . .	0·82	341	0·2	1·8
15. Do. 'aluminium-finished' outside	0·50	207	0·2	1·8
16. Do. laid on top of thin asphalted felt	0·51	211	0·25	2·0

A suggestion was made in the National Physical Laboratory Report that air spaces in roofing materials might be advantageous. A satisfactory means of doing this is to lay corrugated asbestos cement plates on the thinnest asphalted felt or other light sheet material supported by the roof frames. The effect of this is shown by the table to be equal to that of an aluminium surface, and where heat insulation and lightness are of the first importance a strong and efficient roofing can be made in this way at a reasonable cost.

* W. M. Thornton, D.Sc., D.Eng., Armstrong College, Newcastle-on-Tyne; *Engineering*, October 19, 1917.

'Poilite.'

(*The Asbestos Cement Building Products, Ltd., Trafford Park, Manchester.*)

'Poilite' is a tough, weather-proof, fire-safe, and permanent material, the constituents of which are British Portland cement and asbestos, the latter being employed in its fiberised form, known as 'mineral flax,' serving the purpose of an aggregate, thereby increasing the initial strength of the cement tenfold.

'Poilite' is produced in specially designed rolling mills, in the form of layers or films of cement and asbestos uniformly distributed and interlaced so as to constitute a species of tough woven fabric. The layers adhere so firmly to each other that, after reaching the mature stage, a 'Poilite' asbestos cement sheet is found on fracture to be void of all stratification or lamination. The fibre not only gives the product a pre-eminent degree of incombustibility, resiliency, and durability in varying conditions of climate, but also gives great resistance to transverse mechanical and tensile stresses, which increases in ratio with its age.

Tensile strength is 5,973 lbs. per sq. in.; crushing is 7,000 lbs. per sq. in. It is impervious to all atmospheric conditions, age, or decay, is non-corrosive, and requires no preservative treatment at any time.

'Everite' Corrugated Sheeting.

The sheets are light for transport and fixing and reduce cost of sub-structure, require no painting or other treatment, improve with age, are not affected by atmospheric changes, and, therefore, ensure equable temperature of the interior, proof against fire and water.

'*Bigsix Everite*' is claimed to be the strongest asbestos-cement corrugated sheet. When placed on purlins at 4 ft. centres it has been found to support thirteen men without deflection. Its thickness of $\frac{5}{32}$ in. gives the advantage of improved insulation from external climatic conditions, and the storm-proof half corrugation side lap of 2 ins. only of material eliminates any necessity for providing against expansion and contraction of the roof structure.

The sheets are manufactured in successive 6-in. lengths, up to 10 ft. long; standard width, 41½ ins.; reputed thickness, $\frac{5}{32}$ in.; purlin spacing up to 4 ft. 6 in. centres; end lap, 6 ins.; side lap, 2 ins.; number of corrugations per sheet, 7½ ins.; pitch of corrugations, 5¾ ins.; depth of corrugations, 2 ins. Actual cover of an 8-ft. sheet as laid, 7 ft. 6 ins. by 3 ft. 3½ ins.—*i.e.* 24·69 sq. ft. Weight of 100 sq. ft. of finished roofing with fixing accessories is about 350 lbs.—*i.e.* about 3½ lbs. per sq. ft.

Special close nesting adjustable ridging is supplied for use with this material, as also apron pieces, barge boards, ridge ends, corner pieces, etc.

'*Standard Everite.*'—This is manufactured in successive 6-in. lengths up to 10 ft. long; Standard width, 30 ins.; reputed thickness, ¼ in.; number of corrugations, 10½ per sheet; pitch of corrugations, 2¾ ins.; horizontal lap, 6 ins. for roofs of ordinary pitch, say 30° or over; side lap, 4¼ ins. (1½ corrugations). Actual cover of 10 ft. sheet, 9 ft. 6 ins. by 2 ft. 2 ins., or say 20½ sq. ft.; 13½ sq. yds. of sheeting sufficient to cover 100 sq. ft. of roofing. Weight per sq. yd. of sheeting about 19 lbs. Weight per 100 sq. ft. of roofing about 256 lbs.

Sheets are so designed that 1½ corrugations side lap give the effective covering of 2 corrugations; this makes a 7 per cent. saving in the amount of material required.

Sheets up to 10 ft. by 2 ft. 6 ins.

'Poilite' Flat Sheets.

For outside work an excellent effect of old half-timber work and plaster can be obtained by the use of these sheets and creosoted boards for covering the joint. If preferred, the rough side of the sheet can be fixed to show. It is quite as impervious as the smooth side.

For inside work the joint can be covered with a wood moulding, thereby forming panelling to taste, which can be decorated with paper, paint, or distemper immediately after it has been primed with a suitable priming.

The sheets are available in the following sizes (from $\frac{5}{32}$ in. thick upwards): 8 ft. by 4 ft., 6 ft. by 4 ft., 4 ft. by 4 ft., 6 ft. by 3 ft., 4 ft. by 3 ft.

Sheets $\frac{3}{16}$ in. or ¼ in. thick are recommended for internal linings, but for external work ¼ in. or ⅜ in. sheeting should be used.

For lining ceilings, sheets 4 ft. by 4 ft. by $\frac{3}{16}$ in. are most suitable and easily handled.

'Poilite' Louvres.

'Poilite' louvres can be drilled ready for attaching to upright standards with bolts or screws and for an efficient fixing these standards, of either metal or wood, should be arranged at the junction of each length of louvre and should support same both inside and out. Alternatively, the louvres may be bolted to cleats in the sides of standards at each end of the louvre length, or a third method fixing sometimes adopted is to cut a notch with a saw in the corner of the louvre, which is then slipped into a slot in the timber uprights.

Supplied in 4-ft. lengths (standard section).

'Poilite' Wall Tiling.

'Poilite' wall tiling is available in all grey, all red, or checker pattern red and grey, and is supplied in 2-ft. squares of 16 tiles each.

The squares are easily applied to plaster, brick, or breeze walls by simply nailing direct, or to suitably spaced battens.

The nails are afterwards hidden by pointing the joints between the tiles with cement.

'Fibrent' Asbestos-Cement.

(*Asbestos Cement Building Products, Ltd., Trafford Park, Manchester.*)

'Fibrent,' the registered trade mark for asbestos-cement sheets and slates, sometimes termed Fibrocement, is a form of concrete or reinforced cement, the mineral fibres fulfilling this function of reinforcement, and the highest grade British Portland cement and specially selected asbestos fibre only being used for the purpose.

'Fibrent' Corrugated Asbestos-Cement Sheets

Combine the advantages of corrugated metallic roofing in point of economy, and in addition they are non-conductive, fire-resisting, practically everlasting in their original form, and require no treatment for preservation purposes, etc.

'Fibrent' corrugated sheets are manufactured in lengths of 10 ft., rising by increments of 6 ins., 2 ft. 6 ins. and 2 ft. 7½ ins. wide, 3-in. corrugations, ¼ in. thickness.

'Fibro' Supers Corrugated Sheets

'Fibrent' corrugated sheets are manufactured in lengths of 10 in. rising by increments of 6 ins., and 2 ft. 7½ ins. wide, comprising nine 3 in. corrugations, and ¼ in. thickness. The sheets allow for a 1½ corrugation vertical lap, giving covering capacity in width of each sheet of 2 ft. 3 ins.

'Fibro Supers' corrugated sheets give all the advantages of 'Fibrent' asbestos cement materials. They are the largest and strongest asbestos cement corrugated sheets manufactured, embodying new features by their method of construction, fulfilling all the requirements of a satisfactory roofing material, being also used with equal success for sides and ends of buildings.

'Fibro Supers' patent ridge and eave sheet supplied in conjunction with 'Fibro Supers' allow no space between the gutter and the sheeting, thus allowing free fall of water from roof into gutter, and give a well-secured fixture at the ridge, eliminating the chance of percolation in large span roofs through any blowing back of water.

'Fibro Supers' patent eave sheets besides the great advantage of their use at the ridge and eaves, can also be used over glazing, thus sealing the open corrugations.

Close fitting ridging, apron pieces, barge boards, corner sections, and cap finials are supplied in conjunction with 'Fibro Supers' to complete and give an adequate finish to all types of buildings.

'Fibrent' Flat Sheets

Have the same advantages claimed for the slates and corrugated sheets, but are not so heavily compressed. The sheets are used for both external and internal walls, partitions, and ceilings. The standard sizes of 'Fibrent' flat sheets are 8 ft. by 4 ins., 6 ft. by 4 ins., 6 ft. by 3 ft., and 4 ft. by 4 ft., in thicknesses varying from $\frac{5}{32}$ in. to ¼ in.; ¼ in. thickness sheets are generally used for external walls, $\frac{3}{16}$ in. thickness for internal, and $\frac{5}{32}$ in. thickness for ceilings. 'Fibrent' asbestos-cement flat sheets can be cut to requisite sizes by means of a good hand saw at the time of fixing, and, in consequence, waste is obviated. For panel effect, covered strips in required widths and thicknesses are obtainable, and when fixed the sheets can be distempered, enamelled, papered, or grained, after application of concentrated size, thus producing effects which are obtainable with other methods of forming partitions and ceilings. If fixed to timber studding, the supports are spaced at not more than 2-ft. centres.

'Fibrent' Panel Sheets

Have the same advantages and qualities as 'Fibrent' asbestos-cement flat sheets, and give a very artistic finish for any interior work for which they may be required. The extra advantage of 'Fibrent' panel sheets is that the use of a covered strip of the joints is obviated, the rib overlapping the panel by 1 in. These sheets are also embossed with the reproduction of a timber grain, and if waxed and polished giving an effective interior decoration.

'Fibrent' panel sheets can be supplied at any length up to 8 ft. by 4 ft., ¼ in. wide, each sheet containing five ribs 3 ins. wide. The sheets are manufactured in thicknesses of $\frac{3}{16}$ in. and ¼ in.

For 'Fibrent' Asbestos-Cement Slates, see p. 413.

'Turnall' Asbestos-Cement Building Sheets.

(*Asbestos Cement Building Products, Ltd., Trafford Park, Manchester.*)

These sheets, made in sizes 8 ft. × 4 ft., or 4 ft. × 4 ft. etc., are readily worked and quickly fixed with ordinary carpenter's tools.

USE ASBESTOS CEMENT BUILDING PRODUCTS
DURABLE & ECONOMICAL

Turners' Trafford Tiles (ASBESTOS-CEMENT)

"EVERITE"
Asbestos-Cement Rainwater goods. Soil and Vent Pipes for all types of buildings. Pressure Pipes for Water Mains.

"EVERITE"
Asbestos-Cement Flue Pipes and Fittings, specially manufactured for the Gas industry.

Decorated
"POILITE" *Sheets for Bathrooms, Cloakrooms, Lavatories, etc., provide permanent hygienic decoration at low cost.*

"DECOLITE"
Flooring—the perfect flooring for all Industrial Buildings.

POILITE RIGID RUBBER FLOOR TILES
for Houses, Shops, Offices and Ships.

"TURNALL"
Stipple Glaze Decorated Sheets—a permanent artistically decorated sheet at moderate cost.

"TURNALL"
Oak Grained Asbestos Sheet has the true oak grain impressed in the sheet and provides a permanent and economical panelling.

"POILITE"
Pantiles, russet colour, provide economical roofs of character and charm; depth of colour; light and shade.

"POILITE"
Asbestos-Cement Troughs for underground Cable carrying.

"EVERITE"
Ducting for ventilation.

FOR ROOFING AND WALLING

"TURNERS' TRAFFORD TILES" AND SHEETS.

"BIGSIX" CORRUGATED SHEETS.

"EVERITE" STANDARD CORRUGATED SHEETS.

"POILITE" FLAT BUILDING SHEETS.

"POILITE" STRAIGHT COVER AND DIAGONAL SLATES.

"SERVAL" SLATES.

"TURNALL" ASBESTOS WOOD.

"SERVAL" ASBESTOS FELT.

"TURNALL" HAND-MADE PLAIN TILES.

Sole Distributors:

ASBESTOS CEMENT BUILDING PRODUCTS LTD.

INCORPORATING THE SELLING ORGANISATIONS OF
BELL'S POILITE & EVERITE Co. Ltd. BRITISH FIBROCEMENT WORKS Ltd.
TURNER BROTHERS ASBESTOS CO. LTD. (For Asbestos-Cement Products)

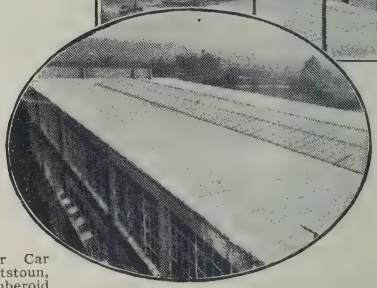
TRAFFORD PARK MANCHESTER — **ASBESTOS HOUSE, SOUTHWARK ST. LONDON, S.E.1**

BIRMINGHAM: 14 New Street — **NEWCASTLE-ON-TYNE:** Collingwood Buildings — **GLASGOW:** Castle Chambers, 65 Renfield Street

ACBP

RUBEROID
Roofing

M & N Warehouses, Southampton Docks, roofed with Ruberoid Built-up Roof.



Albion Motor Car Works, Scotstoun, roofed with Ruberoid on concrete.

DURING the 37 years in which it has been on the market, Ruberoid has been used on large commercial and industrial buildings in all parts of the world with uniform success. Ruberoid has the great advantage of being unaffected by vibration or stresses and is impervious to heat or cold, sea-air or the fumes of acids or alkalis. Equally efficient on flat, pitched or curved roofs of wood or concrete.

Write for our fully illustrated catalogue of Specifications (No. E.239), showing the various grades and types of Ruberoid Roof

TRADE-MARK RUBEROID REGISTERED

The RUBEROID Co. *Ltd*

28 Lincoln House, High Holborn, London, W.C. 1.

Branches:—Birmingham, Manchester, Newcastle, Leith and Dublin.

Ceilings.—For ceilings, sheets $\frac{5}{16}$ in. or $\frac{3}{8}$ in. thick are most suitable, and are nailed to ceiling joists with 1½ ins. galvanised or copper nails at not less than ½ in. from edge of sheets, and about 6 in. centres, care being taken that edges butt closely and nail heads are flush.

In forming ceilings, 4 ft. × 4 ft. sheets are recommended; these are quickly and easily handled by the erectors, and the ceilings may be subdivided into panels, the joints being covered with a moulded wood or fillet strip, the result giving a highly decorative effect.

Partitions.—These may be formed with sheets $\frac{3}{16}$ in. to $\frac{3}{8}$ in. thick, nailed as before described to timber studding, spaced at 1 ft. 4 ins. or 2 ft. centres. Between the standards, dwangs or struts should be placed to support the ends of the sheets. The joints may be filled before painting, and the whole surface rubbed down in the usual way, or as an alternative covered with moulded wood or fillet strips.

'Turner's Trafford Sheets.'

(Asbestos Cement Building Products, Ltd., Trafford Park, Manchester.)

'Turner's Trafford Sheets' embody the same principles in design as are found in 'Trafford Tiles' (p. 413), and form a durable and economical siding material that will last for many years without attention, and that will provide absolute protection against cold, damp, and heat.

'Turner's Trafford Sheets' for vertical work—standard size: 6 ft. × 3 ft. 8 ins. × ¼ in.

Nett weather surface (when fixed) 5 ft. 8 ins. × 3 ft. 4 ins.

Quantity required to cover 100 sq. ft., 5·30 sheets.

Approximate weight: 56 lbs.—40 sheets to the ton.

Special accessory parts have been designed for use in conjunction with the Trafford Tiles and Sheets; these include close-fitting and ventilating type ridging, expansion joint pieces, ventilating louvre blades, flashing pieces .etc., details of which will be found in the makers' catalogue.

'Serval' Asbestos Felt.

(Asbestos Cement Building Products, Ltd., Trafford Park, Manchester.)

This material is designed for use as a roofing felt, sarking felt, or underfelt for floors; the material has a pure asbestos fibre base impregnated with mineral bitumen.

The felt is impervious to damp, will not disintegrate, and will withstand years of exposure. It resists fumes, gas, and acid vapours, is tough and pliable, rot-proof, vermin proof, and odourless; it insulates from sound, heat, and cold, and is unaffected by extreme changes of temperature.

It is made in two grades—heavy weight, and standard weight. Heavy weight in rolls 12 yds. × 36 ins. wide. Standard weight in rolls 24 yds. by 36 ins. wide.

Ruberoid Roofing.

(Ruberoid Co. Ltd., Lincoln House, High Holborn, London, W.C. 1.)

Ruberoid, an exceptionally light material (made in England and used generally throughout the world for over 30 years), weighing only ⅛ to ½ lb. per sq. ft., is manufactured in rolls 36 ins. wide, 72 ft. long, containing sufficient material to cover 2 squares. On pitched roofs the Ruberoid is laid from ridge to eaves on wood sheathing boards, which should be close butted. The arris at ridges and hips should be slightly rounded. The strips of Ruberoid are joined by lapping at the seams at least 2 ins. (fig. 36), well coating with Ruberoid cement between the laps and nailing to the boards down centre of each seam, spacing the nails 2 ins. apart. The nail heads and seams are further protected by being painted with Ruberoid cement.

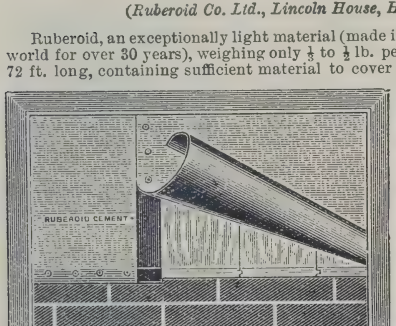


Fig. 36.

On flat roofs of wood the same method is employed. No drips or rolls are needed. Roofs should fall 2 ins. in 10 ft. to allow drainage. Ruberoid can be flashed to parapet walls in the same manner as lead, and box gutters, cesspools and outlets lined with it.

Flat concrete roofs are waterproofed with one or two layers of Ruberoid cemented to the surface of the concrete with Ruberoid compound; the joints in this case are lapped 3 inches. This method of waterproofing is also largely used for cracked concrete and asphalt roofs.

A Ruberoid roof being airtight, proper ventilation should be provided at the highest point.

As it is easily laid in position and is unaffected by moist heat it is largely used in tropical countries, particularly on the West Coast of Africa, in India, and South Africa.

Paint does not adhere well to it, and it should not be tarred ; if desired it can be coated with Ruberine roof paint, which is made in three colours, red, green, and black. Ruberoid, being acid and alkali proof, can be safely used on chemical works and other places where it is subjected to fumes, as well as on the sea-coast.

Price of Standard Ruberoid Roofing per roll, to cover 200 sq. ft. of roof, including accessories :—

½-ply Grey . 21*s.* 6*d.* per roll.	1-ply Red . 27*s.* 4*d.* per roll.	2-ply Grey . 30*s.* 9*d.* per roll
1-ply „ . 25*s.* 0*d.* „ „	2-ply Red . 34*s.* 4*d.* „ „	3-ply „ . 37*s.* 0*d.* „ „

The manufacturers, *The Ruberoid Co. Ltd.*, Lincoln House, High Holborn, London, W.C. 1, quote prices on application, fixed complete in any part of the United Kingdom.

'Ruberoid' roofing is supplied in rolls of 72 ft. by 3 ft. wide, in the following grades, per roll :—

½-ply, 45 lbs.; 1-ply, 60 lbs.; 2-ply, 85 lbs.; 3-ply, 105 lbs.; 1-ply red, 60 lbs.; 2-ply red, 85 lbs.

Approximate number of super ft. of roof which can be covered with one ton :—

½-ply, 9,400; 1-ply, 7,100; 2-ply, 5,300; 3-ply, 4,280; red, 5,000.

'Rok' Roofing.

(D. Anderson & Son, Ltd., Park Works, Stretford, Manchester: Roach Road, Bow, London, E., and Lagan Works, Belfast.)

'Rok' roofing forms a permanent and easily applied roof covering. Only the highest grade of wool felt is used in its manufacture. This is saturated with an elastic water-proofing compound which does not dry out or evaporate in any climate, as it contains no oils, or volatile matter. The coating on the surface is a permanent one, composed of a natural bitumen of very high melting point. There is nothing of an organic nature in either the saturating or coating compound. It is acid and alkali proof and not affected by white ants. It is suitable for pitched, flat, or circular ('Belfast') roofs, and for reinforced concrete roofs it will be found to make a thoroughly water-tight job.

Rain water from roofs covered with it can be used for domestic purposes.

Being an excellent non-conductor, it ensures an even temperature, and is therefore an ideal roofing for hot climates.

It is laid on close jointed boarding, which should not be less than ¾ in. thick, either plain edged or tongued and grooved. The purlins or rafters should be so spaced that there is no appreciable spring in the boarding.

It may be fixed either longitudinally from gable to gable, or vertically from eaves to eaves. It is, however, considered advisable to lay it crosswise to the boarding. The joints should be formed with at least 2 ins. overlap, being stuck together with 'Rok' cement and carefully nailed with tacks not more than 2 ins. apart placed in the centre of the seams. The heads of the tacks where exposed and the outside of the seams should be dressed with 'Rok' cement.

'Rok' roofing is supplied in rolls of 72 ft. by 3 ft. wide, in the following grades :—

½-ply average weight, 50 lbs. per roll, 21/6 per roll 2-ply average weight, 82 lbs. per roll, 30/- per roll
1-ply „ 62 „ 24/6 „ 3-ply „ 102 „ 36/- „

(The above weights do not include the nails and cements.)

The prices include the supplying of special galvanised broad-headed nails and liquid cement for fixing.

Sufficient tacks and cement to fix the roofing are supplied free by the manufacturers with each roll. Each roll, if carefully laid, should cover 200 sq. ft.

The manufacturers will undertake to carry out the fixing if desired.

'Rok' roofing may also be used on concrete, fixed with special 'Rok' mastic, and it is advisable to write the manufacturers for full directions as to the method of applying.

It may also be used for covering flats and gutters. In this case it is recommended to use two layers, the first layer being tacked to the boarding, and the top layer stuck to the under one with 'Rok' mastic. In this way no nail heads are exposed. Flashings to walls, round lights, etc., may be made in the material.

Mineral Surfaced Rok Roofing.

'Mineral Surfaced Rok' is a new variety of roofing specially produced for use on dwelling-houses and other buildings where a more attractive appearance is desired than that of the usual bitumen roofings. It is made of similar materials to 'Rok,' and is of the same high quality in every respect, but the surface is covered with crushed minerals, in either red or green natural unfading colours. These minerals are firmly embedded in the surface of this roofing, which is quite as attractive in appearance as are slates or tiles, with the advantage that there is no possibility of breaking or cracking, and that no wind or water can ever penetrate to the interior. Mineral-surfaced 'Rok' is, in fact, storm-proof.

Supplied in rolls 12 yds. by 1 yd., sufficient to cover 100 sq. ft. Weight 85 lbs. per 100 sq. ft., *i.e.* about ¾-lb. per sq. ft. Price, including nails and cement, red or green, 22*s.* 6*d.* per roll.

'Italit' Corrugated Asbestos-Cement Roofing.

(*Eternit Pietra Artificiale* (*London*), *Ltd.*, *Norfolk House*, *Laurence Pountney Hill*, *London*, *E.C.4.*)

'Italit' corrugated sheets are made in lengths of 4 ft. 0 in. to 8 ft. 0 in., rising by even 6-in. lengths. Greater lengths up to 10 ft. can be supplied in Superstrong and Standard only to special order. The sheets are 3 ft. 2 ins. wide and provide a cover of 3 ft. 0 in. to the weather as laid. The corrugations have a pitch of 6 ins. and a depth of 2 ins., providing 6½ corrugations per sheet. Side lap, 2 ins. wide, as illustrated in fig. 37.

The following show details:

	Superstrong.	Standard.	Lightweight.
Thickness of sheet . . .	¼ in. to $\frac{9}{32}$ in.	$\frac{7}{32}$ in. reputed	$\frac{3}{16}$ in. reputed
Minimum end lap . . .	6 ins.	4¾ ins.	4½ ins.
Weight allowance per square foot	3·25 lbs.	2·60 lbs.	2·20 lbs.

'Italit' corrugated sheets are available in (natural) grey and red colours.

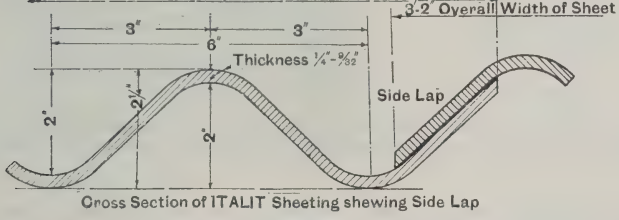


FIG. 37.

Purlin Spacings.

For roofs having a rise of 4 ins. or more, per ft., the following indicates the relationship between the sheets and spans of purlins, based upon a uniformly distributed load not exceeding 80 lbs. per sq. ft. of roof area:

Lightweight—$\frac{3}{16}$ in. thick, for spans up to 3 ft. 0 in.
Standard—$\frac{7}{32}$ in. reputed, for spans up to 4 ft. 0 in.
Superstrong—¼ in. to $\frac{9}{32}$ in., for spans up to 5 ft. 0 in.

These distances are of necessity conservative, and dependent upon the location and nature of the structure, the prevalence or otherwise of high winds, snow loads, and other contingencies. In practice, they are frequently exceeded by as much as 25 per cent.

Side Rail Spacings.

While the spacing of side rails varies with the nature and class of the structure, 'Italit' corrugated sheets for vertical work may be fixed to side rails or supports spaced at distances up to 25 per cent. in excess of the minimum purlin spacing indicated for roofs.

'Italit' Ridging and Flashings.

Ridges—Corrugated Ridges 3 ft. 2 ins. provide a cover of 3 ft. 0 in. as laid.
Adjustable Interlocking Ridges to closely fit any pitch of roof are supplied in pairs and provide a cover of 3 ft. 0 in. as laid.
Flashings—Apron Flashings 3 ft. 3 ins. long. Stop End Flashings, Side Wall Flashings, Corner Flashings, and Barge Boards are available.

FLOORS

Single-joisted Floor.

Fig. 38 shows a single set of bearers, *b*, called common or bridging joists, to which ceiling joists, *a*, are attached when desired. The ceiling is more liable to crack with a single than with a double floor, but this may be obviated by making every third or fourth joist deeper, and fastening the ceiling to them only. The ceiling joists are notched, and nailed to the joists *b*.

FIG. 38.

This is the strongest and cheapest floor, the maximum strength to be got from a wooden, untrussed floor being with deep tingle joists placed close together, and such a floor with a plaster ceiling below is fire resisting.

Double Floor.

Figs. 39, 40, show a double set of bearers, *i.e.*, binding joists or binders, *d*, and bridging joists, *b*. The binders are the chief supports of the floor, and are generally about 6′ apart. The bridging joists are notched over them, and any ceiling joists, *a*, under them. The bridging as

FIG. 39.

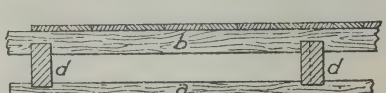


FIG. 40.

well as the ceiling joists can be framed between the binders, or supported by fillets spiked to their sides when it is required to reduce the depth of the floor. The binders rest on stone templates.

The ends of girders should be on stone templates (fig. 42) and should have a space round them to prevent decay.

Framed Floors

differ from double floors only in having the binders framed into a third set of bearers, called girders.

The girders, *a* (**fig. 41**), are the chief support; the binding joists, *d*, are tusk-tenoned into them (fig. 14, page 313); the latter support the bridging joists, *b*, and ceiling joists, *c*. The

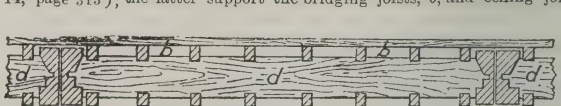


FIG. 41.

girders should not be more than 8′ to 10′ apart. When the breadth of a girder is considerable it is sometimes sawn down the middle, reversed, and bolted together with sawn sides outwards, slips being placed between to allow air to circulate freely. This allows centre of balk to be examined, and is less liable to rot; it slightly diminishes strength of girder. Such girders are sometimes strengthened by wrought-iron flitches bolted in between the two halves.

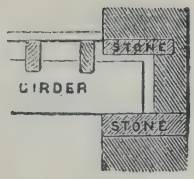


FIG. 42.

Camber of Joists.

Floors, when first framed, should always be kept about $\frac{3}{4}''$ higher in the middle than at the sides of a room; also the ceiling joists should be fixed about $\frac{3}{4}''$ in 20′ higher in the middle than at the sides of room. If laid level they are sure to sag when used.

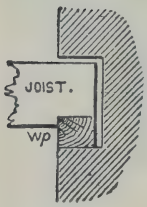


FIG. 43.

Wall-plates for Floors.

On ground-floor generally placed on offset. May be let into wall, care being taken that free space is left round end of joists or girders, as in fig. 43; but, as the wall-plate is perishable, the effective strength of the wall is reduced by the width of the plates. They should, if possible, rest on stone corbels or brick corbel courses. The joists are usually only nailed to the wall-plates, but sometimes notched (fig. 43) or cogged; the latter is quite unnecessary.

For 20′ bearing, wall-plate is $4\frac{1}{2}'' \times 3''$; for 30′ bearing, $6'' \times 4''$; for 40′, $7\frac{1}{2}'' \times 5''$.

(*Tredgold.*)

Wooden Girders for Floors,

10 feet from centre to centre.

b = breadth; d = depth, in inches; L = length, in feet.

$$d = \sqrt[3]{\frac{L^2}{b}} \times 4{\cdot}2 \text{ for fir; by } 4{\cdot}34 \text{ for oak.} \qquad b = \frac{L^2}{d^3} \times 74 \text{ for fir; by } 82 \text{ for oak.}$$

To obtain maximum stiffness make $b = \frac{2}{3}d$.

Girders of Wood with Wrought-iron Flitches.—The wood and iron of these girders bend differently; therefore, only half strength of wood is exerted if the iron is not unduly strained. If thickness of iron is less than $\frac{2}{17}$ that of wood, the beam will be weaker than without iron. If thickness of iron = $\frac{2}{17}$ wood, the beam will have the same strength as if without iron, and be twice as stiff. If thickness of iron exceeds $\frac{2}{17}$ wood, beam will be stronger than plain wooden beam.

Rolled-iron Joists.

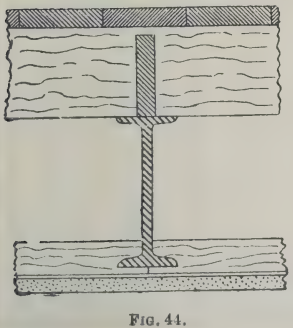


FIG. 44.

Rolled-iron joists are now extensively used as binders, the upper flange of which may support the floor joists, and the lower flange the ceiling joists, as in fig. 44.

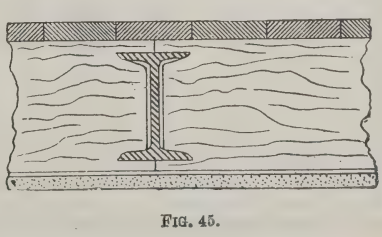


FIG. 45.

When the depth of the floor is required to be as shallow as possible, rolled-iron joists for binders are useful, as the floor may be constructed without increasing the depth required by the bridging joists, as shown in fig. 45.

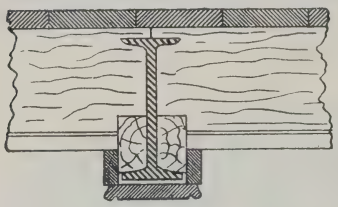


FIG. 46.

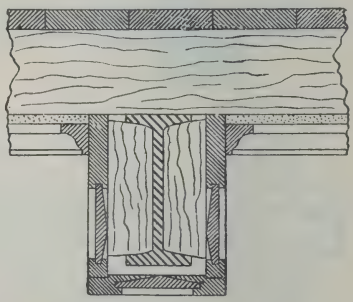


FIG. 47.

Figs. 46 and 47 show rolled-iron joists used as binders, and arranged so as to give the ceiling a panelled appearance.

GIRDERS OF BALTIC YELLOW FIR,

10 feet from centre to centre.

Bearing in Feet.	Depth, 10 Inches.	Depth, 11 Inches.	Depth, 12 Inches.	Depth, 13 Inches.	Depth, 14 Inches.	Depth, 15 Inches.	Depth, 16 Inches.	Depth, 17 Inches.	Depth, 18 Inches.	Depth, 19 Inches.	Depth, 20 Inches.	Depth, 21 Inches.
	Breadth in Inches.											
10	7½	5½	4	3½	2¾	2¼	1¾	..	1¼			
11	9	6¾	5	4¼	3¼	2¾	2¼	..	1¾			
12	10½	8	5¾	5	3¾	3¼	2¾	2¼	2			
13	12½	9½	6¾	5¾	4½	3¾	3	2½	2¼			
14	14½	10¾	7¾	6¾	5¼	4½	3½	3	2½			
15	16½	12½	9	7¾	6	5	4	3½	3			
16	18¾	14	10½	8¾	7	5¾	4¾	4	3¼			
17	..	16	11½	9¾	8	6½	5¼	4½	3¾			
18	..	17¾	13	11	8¾	7¼	6	5	4			
19	..	19¾	14½	12	9¾	8	6¾	5½	4¾			
20	..	..	16	13½	10¾	9	7¼	6	5½			
21	..	..	17½	15	12	9¾	8	6¾	5¾			
22	..	..	19½	16½	13	10¾	8¾	7½	6¼			
23	..	..	..	18	14¼	11⅛	9½	8	6¾			
24	..	..	..	19½	15½	12¾	10½	8¾	7½			
25	..	..	..	..	16¾	13¾	11¼	9½	8			
26	..	..	..	..	18¼	15	12¼	10¼	8½			
27	..	..	..	..	19¾	16	13	11	9¼			
28	..	..	..	..	..	17¼	14¼	11¾	10			
29	..	..	..	..	..	18¾	15¼	12¾	10¾			
30	..	..	..	..	..	19¾	16¼	13½	11¼	9¾	8½	7¼
31	..	..	..	..	..	..	17¼	14¼	12¼	10¼	9	7¾
32	..	..	..	..	..	..	18½	15½	13	11	9½	8¼
33	..	..	..	..	..	..	19¾	16½	14	11¾	10	8¾
34	..	..	..	..	..	..	..	17½	14¾	12¼	10¾	9½
35	..	..	..	..	..	..	..	18½	15¼	13¼	11½	10
36	..	..	..	..	..	..	..	19½	16½	14	12	10½

SINGLE OR BRIDGING JOISTS OF BALTIC YELLOW FIR,
12′ from centre to centre.

Bearing in Feet.	Breadth, 1½ Inch.	Breadth, 2 Inches.	Breadth, 2½ Inches.	Breadth, 3 Inches.	Breadth, 3⅜ Inches.	Breadth, 4 Inches.
	Depth in Inches.					
5	5¾	5½	4¾	4⅜	4½	4
6	6¼	5¾	5⅜	5	4¾	4½
7	7	6½	6	5½	5¼	5
8	7¾	7	6¼	6¼	5⅝	5⅜
9	8½	7½	6⅞	6½	6	5⅞
10	9	8	7½	7	6¾	6½
11	9½	8¾	8	7½	7	6¾
12	10	9¼	8½	8	7½	7¼
13	10¾	9¾	9	8½	8	7¾
14	11	10	9⅛	9	8½	8
15	11¼	10½	9¾	9¼	8¾	8½
16	12¼	11	10⅛	9¾	9¼	8⅞
17	12¾	11½	10¾	10¼	9¾	9¼
18	13¼	12	11¼	10½	10	9½
19	13¾	12½	11½	10¾	10¼	10
20	14¼	13	12	11¼	10¾	10¼
21	14¾	13½	12⅛	11¾	11	10¾
22	15	13¾	12¾	12	11¼	11
23	15½	14	13	12¼	11¾	11¼
24	16	14½	13½	12¾	12	11¾
25	16½	15	14	13	12½	12

For ordinary floors up to 16′ span, with the above scantlings, the joists may safely be spaced 12″ apart, instead of centre to centre. For spans over 16′ stiffness requires them 12″ from centre to centre, and over 20′ the scantlings are rather deficient in stiffness.

Ceiling Joists.

2″ thick is sufficient. They are nailed and notched to the joists.

Rule for Depth of Ceiling Joists.

L = length ; b = breadth ; d = depth.

If $b = 2''$, then $d = \frac{1}{2}''$ for each foot run of span $= \frac{L}{\sqrt[3]{b}} \times \cdot 64$ for fir ; $\times$ 0·67 for oak.

CEILING JOISTS OF BALTIC YELLOW FIR,
12″ from centre to centre.

Bearing in Feet.	Breadth, 1½ Inch.	Breadth, 1¾ Inch.	Breadth, 2 Inches.	Breadth, 2¼ Inches.	Breadth, 2½ Inches.	Breadth, 2¾ Inches.	Breadth, 3 Inches.
	Depth in Inches.						
4	2¼	2⅛	2				
5	2¾	2⅝	2½				
6	3⅜	3⅛	3	2⅞			
7	4	3¾	3½	3⅜	3¼		
8	4½	4⅛	4	3⅞	3¾	3⅝	3½
9	5	4⅝	4½	4⅜	4¼	4⅛	4
10	5½	5⅜	5	4⅞	4¾	4⅝	4½
11	6¼	5⅞	5½	5⅜	5¼	5⅛	5
12	6¾	6½	6	5⅞	5¾	5⅝	5½
13	7½	7	6½	6¼	6	5⅞	5¾
14	8	7½	7	6⅞	6½	6⅜	6¼
15	8½	7⅞	7½	7⅜	7¼	7	6¾

(*Tredgold.*)

Ceiling joists should never have very long bearings, particularly those for ceilings next the roof of a house, as long joists are subject to warp, which breaks the plastering.

The distance from centre to centre should never exceed 12″, otherwise the bearing for the laths becomes too long. If the distance from centre to centre exceeds 12″, double laths should be used.

Lintels.

Lintels over doors and windows are of wood, generally 1″ in depth for every foot of span in width, or = $\frac{1}{15}$ span; the weight of the wall above being taken off them by a relieving arch of brick or stone of span equal to their full length. If, as for very cheap and temporary work, no relieving arch is used, they should have a depth of 1½″ to 1½″ for every foot of span, or more if part of the weight of a floor or roof falls over them. The ends should bear 9″ on the wall, although sometimes only given 4½″.

Details of Flooring Boards.

Thickness of Flooring Boards.—1¼″ in ordinary houses.

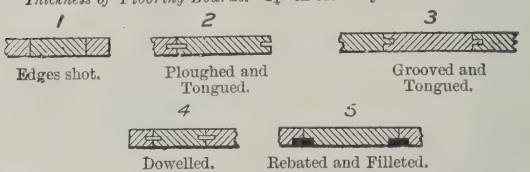


FIG. 48. Edge Joints.

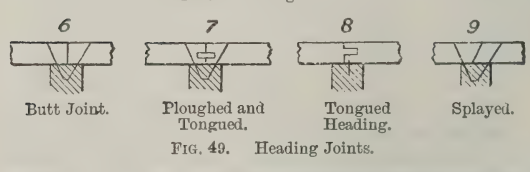


FIG. 49. Heading Joints.

Edge Joints (fig. 48.) —1 is used for inferior floors; 2 and 3 for ordinary floors of a high class; 4 for very superior floors; 5 where the traffic is heavy.

Tongues and Fillets are of wood or hoop iron; the latter, if for good work, should be galvanised. If of wood, should be ¼″ thick, cut across the grain; for strength as well as to prevent splitting, cross or feather tongues are used, the grain running at an angle across the tongue. Iron tongues for 1¼″ boards are 1⅛″ wide, 17 B.W.G.; for 1½″ boards, 1⅜″ wide, of 17 B.W.G.; for 2″ boards 1⅞″ wide, of 15 B.W.G.; for 2½″ and 3″ boards, 1½″ wide, of 15 B.W.G. Tongues are put one-third of depth from bottom of board to give more wear to the upper edges.

Heading Joints.—Fig. 49 shows various forms of heading joints, the inclined black lines being the securing nails.

Secret Joints.

Joints for important or polished floors are secured in such a manner as not to show their fixing, or present gaps or unsightly holes, as would occur in ordinarily fixed flooring, and as they are all fixed on one edge only, are free to shrink or swell without splitting.

Figs. 50, 51 show boards secured by dowel pins and screwed to joists through one edge only, a cheap and effective method, but one which does not prevent dust passing through.

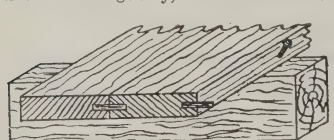


FIG. 50.

FIG. 51.

FIG. 52.

FIG. 53.

FIG. 54.

Figs. 52 and 53 show the method by rebate, groove and tongue, secured by screws on one edge only. These can be easily fixed or taken up when required, and prevent any dust passing through. The expense of these joints is the great objection to their use. The method shown in fig. 54 possesses the advantages of the two previous joints at a less cost.

Clasp nails or wrought or cut *flooring brads* are used, two to each board, over each joist. The nails should be from 2″ to 2¼″ longer than thickness of board; 3¾″ (52 lbs. to the 1,000) may be used for 1¼″ boarding.

Nail heads, for good work, should be punched down, and the holes stopped with lead putty.

Width of Floor Boards.—For important rooms, 4½″; in ordinary rooms, 7″; for common and

temporary purposes, 9″. The narrower the boards, the less they shrink. All the boards should be *listed*, or have the sapwood removed from edges. They should be very well seasoned; if not, they should be temporarily laid, and left a year before being finally laid and nailed down.

Ventilation of Floor Joists, &c.

The spaces between the timbers of floors and flat roofs should be well ventilated by air bricks in walls. Ground below a floor should be thoroughly drained, and covered in some cases with a layer of lime concrete, or with ashes and quicklime.

Fire-Resisting Floors.

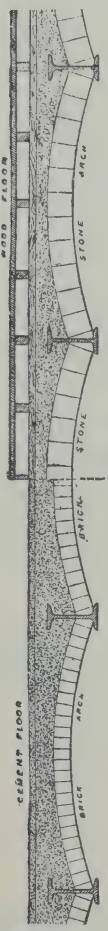


FIG. 55.

FIG. 56.

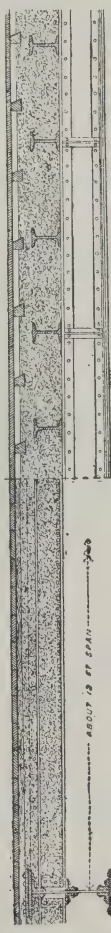


FIG. 57.

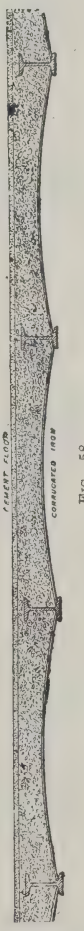


FIG. 58.

See also 'Fire Resistant Construction,' *Building Research Board Report No.* 8 (1927). Published by His Majesty's Stationery Office, Adastral House, Kingsway, London, W.C. 2; price 1*s.* 6*d.* net.

'Expamet' Expanded Metal.

'Expamet' Expanded Metal, as manufactured by *The Expanded Metal Company, Ltd.* (of London and West Hartlepool), is made from blank metal sheets and plates—steel, brass, copper, etc.—which are cut and expanded into meshes of various shapes and sizes. Each size of mesh is made in several weights, for, while a particular mesh remains constant in size, centre to centre, it can be produced from several thicknesses of sheets or plates, and strands of various widths can be made; the different styles and weights have all a distinguishing number assigned to them, and each may be specified and ordered under its List Reference No.

The material is machine-made, of meshwork formation, and has no joins, no welds, and no loose or separate strands or attachments; the junctions between the meshes remain uncut, and all the strands or members in a sheet are connected rigidly, and have continuous fibres—important features peculiar to Expanded Metal.

'Expamet' Diamond Mesh Expanded Steel Reinforcement for Concrete.

'Expamet' Expanded Steel has been in successful and ever-increasing use all over the world for some thirty years past. As reinforcement for concrete it has exceptional qualities, especially in large plain and curved areas. It is particularly suitable for reinforcing concrete in roads, foundations, walls, floors, roofs, lintels, arches, bridges, grain silos, coal pockets, tanks, reservoirs, dams, retaining walls, piers, abutments, pipes, sewers, conduits, etc. 'Expamet' Expanded Steel of the 3-in. mesh weights is used mostly for such work, and the 1½-in. and 6-in. meshes also may be used in some cases; the lighter weights of the ¾-in. and 1½-in. meshes are used frequently in concrete for encasing steelwork, and various weights of 6-in. mesh in concrete roadways, pavings, tramway tracks, and the like. It is supplied in rolls or flat sheets cut to size and ready for use, sheets up to 16 ft. × 8 ft. being sent in flat bundles and larger sheets in rolls of small diameter; it is simple, economical, and effective in practice, and has the great advantage for shipping purposes that it packs closely and is easily transported and quickly handled. Its unique feature is that it is a sheet of steel network, wherein all the strands or members are connected rigidly; thus they cannot be displaced, when in position, by the laying and tamping of concrete: it follows that expanded steel reinforcement is entirely to be relied upon in either skilled or unskilled hands, and the steel goes where it is planned to go without expensive setting out and labour. The respective meshes 'key' into each other, and interlock where sheets overlap; thus a layer of the reinforcement may be made absolutely continuous, no matter how large the area to be treated.

Effective mechanical and cross bond and end anchorage is obtained, for being a double-way reinforcement of peculiar formation, it cannot slip within the concrete.

As meshwork with a dependable grip is a most efficient form of reinforcement for resisting both tensile and thermal stresses, 'Expamet' Expanded Steel-Concrete slabbing is well adapted for heat and fire resistance, as well as for load carrying.

The distribution of stress by 'Expamet' Expanded Steel is complete, for there is steel to transmit in all directions the stresses due to a load; and, no matter where it may be imposed, the load does not affect merely the portion of slab directly under it. Also, as each mesh contains its own concrete, the mass is bound together very effectively; and the concrete enclosed within the mesh being brought into compression, the effective value of the reinforcement is increased.

'Expamet' Expanded Metal Lathings for Plasterwork.

'Expamet' Expanded Metal Lathings are made from mild steel sheets. They form an ideal key for plaster in ceilings, steelwork encasing, solid and hollow partitions, exterior walls, and other plasterwork. All lathing is coated with a mineral oil, and then dipped once in asphaltum paint. Unpainted lathing should not be used for plasterwork, and it is advisable to paint all clips, hangers, ceiling bars, etc., as well as the lathings.

The latest type is known under the trade name of 'BB' Expanded Metal Lathing, and it supersedes the older Diamond Mesh types, but for hairless plaster the 'BB' and ¼-in. Diamond Meshes are preferable, while for hair plaster the ⅜-in. mesh may be used. Both types are suitable for use for all purposes to which metal lathings can be applied. For special ceilings, heavy ornamental plasterwork, heavy facing such as tiling, mosaic, etc., and for exterior work, the heavier lathings should be employed, especially in horizontal work.

The 'BB' mesh is made from 26, 24, and 22 gauges, and is stocked in sheets 9 ft. × 2 ft.; the Diamond Mesh from 24, 22, and 20 gauges, and in sheets 6 ft. × 2 ft., 7 ft. × 2 ft., and 8 ft. × 2 ft. 3 ins.

'Exmet' Reinforcement for Brickwork.

'Exmet' Reinforcement is specially suitable for reinforcing brickwork, concrete-block work, partition slabs, asphalt, etc. It is manufactured from 24, 22, and 20 gauge mild steel, in 2½-in., 7-in., and 12-in. widths, and is supplied in standard coils containing 270 or 75 feet lineal. It is coated with a mineral oil, and then dipped in asphaltum paint, but can be obtained unpainted if ordered specially. Flat strips of any length up to 16 ft. can be supplied, either uncoated, painted, or galvanised.

' Ribmet ' for Concrete and Plaster Work.

' Ribmet ' (Regd.) is composed of the company's well-known ' BB ' Lathing, in 26, 24, and 22 gauge mild steel, with U-shaped ribs attached securely to the sheets, in the direction of their longway and parallel to the sides. Standard size sheets are 2 ft. to 9 ft. long inclusive, rising 6 ins. at a time ; sheets of other lengths between 2 ft. and 9 ft., and sheets over 9 ft. and up to 12 ft., can be supplied. The material is half an inch thick overall.

There are two types of ' Ribmet '—' X ' type and ' Z ' type ; in the first the lathing is grooved to receive the ribs ; in the second the ribs are attached to the flat sheet of lathing with the meshwork passing across the opening of the ribs—novel and unique features in material of its kind.

The standard ribs are made from 19 gauge mild steel, and are attached to the lathing, at frequent regular intervals, with strong half-round steel wire. The ribs can be spaced apart at practically any required dimensions, but 3 ins., 4 ins., and 5 ins. are the standard centres ; generally, the outer ribs are about 2 ins. from the outer sides of the sheets, and the intermediate ribs at the respective spacings, but ribs can be attached along the edges if required.

Concrete Flooring, Roofing, etc.—' X ' type ' Ribmet ' is to be preferred for suspended concrete flooring, roofing, and the like—particularly as a permanent centring ; the lath corrugation stiffens the rib, and the meshwork is small enough to retain the concrete while it is being poured.

For long spans and heavy superloads, 3-in. mesh ' Expamet ' reinforcement should be used along with the ' Ribmet ' ; laid on the ribs, it is distanced mechanically from the lathing ; thus the reinforcement is embedded within the concrete, and there is a lath face to receive a plaster finish.

In flooring and similar work, while temporary close-boarding is rendered unnecessary by the use of ' Ribmet,' temporary bearers are essential, as spacings of 18 ins. and upwards, according to the weight of the concrete to be imposed and whether the soffit is straight or curved.

For designs and estimates for ' Ribmet,' in any work for which it is suitable, application should be made to the company.

Plaster Ceilings, Partitions, etc.—' Z ' type ' Ribmet ' has the unique feature of the lathing passing across the openings of the ribs ; it is thus particularly desirable for suspended plasterwork in ceilings, walls, partitions, and the like ; it gives a continuous and unbroken lath face and ' key ' for the plaster, a uniform thickness of which is obtained throughout, and the ribs are filled with it. It follows that the possibility of the plaster cracking along the ribs is reduced to a minimum and, filled with plaster, there can be no condensation inside them.

Unless there is some very good reason to the contrary, the ribs should never be within the thickness of the plaster, but on the back face of the lathing, except in solid partitions and the like.

' Ribmet ' should be fixed securely, longway of the sheet, from bearer to bearer, to ensure a level and rigid ground to receive plaster. The strands in the various sheets should all slope in one direction ; in vertical work they should slope inwards and downwards from the plaster face. It should be secured, at each rib, to the bearers supporting it, and be wired together at 3-in. to 4-in. intervals between supports. The sheets should be overlapped not less than one mesh where they join ; overlaps should not be made at angles or curves—at such places the sheets should be bent, or angle pieces of plain lathing should be used to cover and connect butt-joints. Galvanised slice-cut staples or tenterhooks, not less than 1¾ ins. long, are recommended for timber bearers, and not less than 14 gauge galvanised soft iron wire for tying to steel bearers ; 18 gauge galvanised soft iron wire is sufficient for tying the sheets together at overlapped joins.

Practical tests and actual experience show that No. Z 244 ' Ribmet ' may be used for ordinary plasterwork, with entirely satisfactory results, up to 3 ft. spacings in horizontal work, and 4 ft. in vertical ; but the heavier weights are recommended for special, ornamental, and exterior work.

' Expamet ' Expanded Metal Make-up Goods.

Practically any of the meshes may be used for openwork partitions, shop divisions, fencing, etc. ; guards for machinery, windows, trees, etc. ; lockers for clothes, tools, etc. ; baskets for waste-paper, letters, etc. ; and similar articles.

' Expamet ' Expanded Metal Lathing and Plaster Partitions.

These are constructed solid or hollow, according to requirements.

The weight per super foot of surface area, measured one side only, is for 2-in. solid walling about 18 lbs. ; and for 6-in. hollow walling about 21 lbs., with a slight increase in weight if the outside skin is back-plastered.

EXPANDED METAL SUSPENDED CEILINGS.

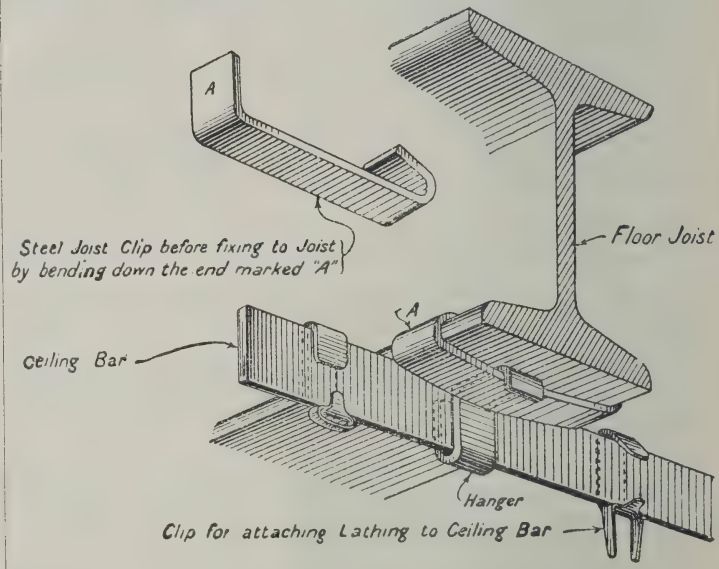


FIG. 59.

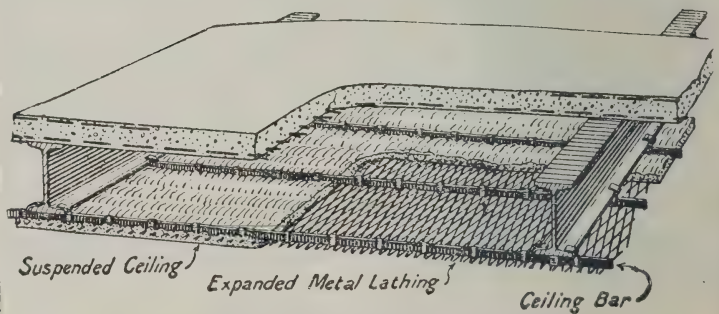


FIG. 60.

The lighter lathings may be used for both systems, but the heavier lathings are recommended for exterior work.

Solid Partitions, etc.—The metal grounds for solid walls, etc., are formed of 'BB' Expanded Metal Lathing securely fastened to ⅜-in. diameter steel rods, or other suitable sections, placed vertically 12 ins. to 15 ins. apart. Tension rods are fixed tautly at their ends with screws to timber, hangers or spikes to concrete, and clips to steelwork; in some cases it suffices if their ends are bent round structural members, as in steel-frame buildings.

When the metal groundwork is completed, it must be temporarily strutted and railed at half height on one side to keep it in a vertical plane. The first or 'render' coat should then be applied on the opposite side, ½ in. thick, and allowed to set hard previous to the second coat being applied. When these two coats are set the temporary supports may be removed, and the reverse side floated. The finishing coat is then applied in the ordinary way. Interior solid partitions are sometimes built only 1½ in. thick, but usually they are made 2 ins. thick; exterior solid walls are generally made 2½ ins. to 3 ins. thick.

Hollow Partitions, etc.—The metal grounds for hollow walls, etc., are formed of 'BB' Expanded Metal Lathing, securely fastened on both sides to steel studs spaced from 12 ins. to 18 ins. apart. These studs are formed of two angle bars, connected at intervals by distance straps, and by angle foot and head pieces, which are holed to allow of the studs being fixed to flooring, etc. If the studs have to be fixed to steelwork a clip is riveted to the foot and head pieces to form means of attachment. Overall thickness is regulated by the length of the cross straps and pieces; 6 ins. is a usual overall dimension, and ¾-in. of plaster is sufficient for the interior skin, and 1½-in. for the exterior. It is wise to back-plaster lathing in exterior work, and thus embed it.

Suspended Ceilings.

These ceilings (figs. 59 and 60) can be suspended at any desired level and to any desired shape. They may be fixed flush with the soffits of floor joists, roof principals, etc., or they can be suspended below them at suitable distance to accommodate pipes, electric wires, etc. The ceilings may be carried to the side walls, in bays or coves of any desired form, for obtaining artistic or decorative effects.

A flat ceiling with hangers varying in length, or a curved ceiling with hangers made to follow the contour, may be suspended below arched flooring, etc.; ceilings suspended from roof principals may be dealt with in the same manner.

For spans of up to 8 feet, flat steel bars, of suitable sections, are suspended on edge from the floor joists, etc., at intervals according to the weight of 'Expamet' or 'BB' lathing and plaster to be used, by means of mild steel hangers made for the purpose. The section of bar to be used will depend upon the character of the work to be done in each case. For spans of over 8 feet it is more economical to use tee or angle bars supports, at say 6-ft. centres, and from them to suspend the ceiling bars.

Where bars join they should overlap, at least 2 ins., in a hanger. In wall-end bays the bars should be pinned not less than 2 ins. into the masonry. In corridors and other places, where the spans are not too long, intermediate support is not necessary—the bars may be pinned at their ends into the brickwork, etc. In long spans, it is wise, at the half-span, to tie a bearer bar on top of, and at right angles to, those on edge, to maintain them in a vertical plane.

The hangers should be secured by screws to timber joists, attached to R.S.J.'s by means of mild-steel clips, bent over tie members of roof trusses, and pinned into reinforced concrete beams. The bars should be fastened in the hangers with wire, and also at lapped joints, to prevent the metal grounds from rising.

The lathing should be fixed to the ceiling bars with lathing clips or with galvanised soft iron wire, and when this is completed the ceiling grounds are ready to receive plaster.

The weight of these suspended ceilings, including metal grounds and plaster, ranges from 9 to 12 lbs. per foot super; and the air space provided affords a perfect fire and sound-resisting construction.

Steelwork Encasing.

For plasterwork, the 'Expamet' Lathings, and sometimes the lighter weights in ⅜-in. 'Expamet' expanded steel, such as No. 38, are used; while for concrete encasing, formed with temporary centring, the lighter weights in 1½-in. 'Expamet,' such as No. 6, are recommended.

Structural Steelwork Encasings.—In many cases lathing is simply wrapped round the steelwork, secured in position when necessary, and wired together at lapped joints, and then covered with plaster. A better system, however, is to use metal cradling, such as ¼-in. or ⅜-in. round

DIAMOND MESH EXPANDED STEEL-CONCRETE SLABS, FOR FLOORING, ROOFING, ETC. SAFE SUPERIMPOSED LOADS PER FOOT SUPER.

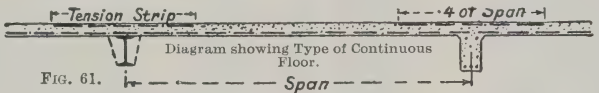


Diagram showing Type of Continuous Floor.

FIG. 61.

Continuous Slabs. $BM = \frac{WL}{12}$

SPAN Ft. Ins.	56 lbs.		70 lbs.		84 lbs.		100 lbs.		112 lbs.		150 lbs.		224 lbs.		280 lbs.		336 lbs.		392 lbs.		448 lbs.	
	Concrete Thickness Inches.	Expanded Steel List Ref. No.	Concrete Thickness Inches.	Expanded Steel List Ref. No.	Concrete Thickness Inches.	Expanded Steel List Ref. No.	Concrete Thickness Inches.	Expanded Steel List Ref. No.	Concrete Thickness Inches.	Expanded Steel List Ref. No.	Concrete Thickness Inches.	Expanded Steel List Ref. No.	Concrete Thickness Inches.	Expanded Steel List Ref. No.	Concrete Thickness Inches.	Expanded Steel List Ref. No.	Concrete Thickness Inches.	Expanded Steel List Ref. No.	Concrete Thickness Inches.	Expanded Steel List Ref. No.	Concrete Thickness Inches.	Expanded Steel List Ref. No.
4 0	3	15	3	15	3	15	3	15	3	15	3	15	3	15	3	9	3	9	3½	9	3½	9
4 6	3	15	3	15	3	15	3	15	3	15	3	15	3	9	3	9	3½	9	3½	8	3½	8
5 0	3	15	3	15	3	15	3	15	3	15	3	15	3	9	3½	9	3½	8	4	8	4	8
5 6	3	15	3	15	3	15	3	15	3	15	3	9	3½	9	3½	8	4	8	4	11	4½	11
6 0	3	15	3	15	3	15	3	9	3	9	3	9	3½	8	4	8	4	11	4½	11	5	11
6 6	3	15	3	15	3	9	3	9	3	9	3½	9	4	8	4	11	4½	61	5	61	5½	61
7 0	3	15	3	9	3	9	3	9	3½	9	3½	8	4	11	4½	61	5	61	5½	61	5½	10
7 6	3	9	3	9	3	9	3½	9	3½	8	3½	8	4½	11	5	61	5½	61	5½	10	6	10
8 0	3	9	3½	9	3½	9	3½	8	3½	8	4	8	4½	61	5½	61	5½	10	6	10	6½	62
8 6	3½	9	3½	9	3½	8	3½	8	4	8	4½	11	5	61	5½	10	6	10	6½	62	6½	62
9 0	3½	9	3½	8	3½	8	4	8	4	11	4½	61	6	61	6	10	6½	62	6½	62	7	62
9 6	3½	8	3½	8	4	8	4	11	4½	11	5	61		10	6½	10	6½	62	7	62	7½	31
10 0	3½	8	4	8	4	11	4½	11	4½	61	5½	61	6	10	6½	62	7	62	7½	31	8	30
10 6	4	8	4	11	4½	11	4½	61	5	61	6	61	6½	62	6½	62	7½	62	8	31	8	30
11 0	4	8	4½	11	4½	61	5	61	5½	61	6	10	6½	62	7½	62	8	31	8	30	9	30
11 6	4	11	4½	61	5	61	5½	61	6	61	6	10	6½	62	7½	31	8	30	9	30	10	30
12 0	4½	11	4½	61	5	61	6	61	6	10	6½	10	7½	62	8	30	8½	30	9½	30	10	68
12 6	4½	61	5	61	6	61	6	10	6	10	6½	62	7½	31	8	30	9½	30	10	68	10	68
13 0	5	61	5½	61	6	10	6	10	6½	10	6½	62	8	30	9	30	10	30	10	68	10	68
13 6	5½	61	5½	10	6	10	6½	10	6½	62	7	62	8	30	9½	30	10	68	10	68	10½	68
14 0	5½	10	6	10	6½	10	6½	62	6½	62	7½	31	8½	30	10	30	10	68	10	68	11	68
14 6	6	10	6	10	6½	62	6½	62	6½	2	8	31	9½	30	10	68	10	68	11	68	12	68
15 0	6	10	6½	62	6½	62	7	62	7½	62	8	30	10	30	10	68	10½	68	12	68		
15 6	6½	10	6½	62	6½	62	7½	62	7½	31	8	30	10	68	10	68	11	68				

Note.—Generally it is not economical to use thicknesses shown below the zigzag line in the table. *(The Expanded Metal Co., Ltd.)*

DIAMOND MESH EXPANDED STEEL-CONCRETE SLABS, FOR FLOORING, ROOFING, ETC. SAFE SUPERIMPOSED LOADS PER FOOT SUPER.

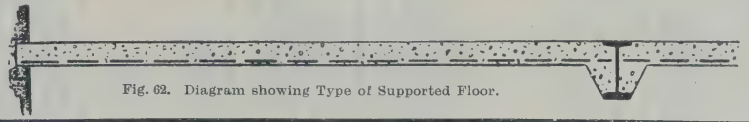


Fig. 62. Diagram showing Type of Supported Floor.

Supported Slabs. $BM = \frac{WL}{8}$

SPAN Ft. Ins.	56 lbs.		70 lbs.		84 lbs.		100 lbs.		112 lbs.		150 lbs.		224 lbs.		280 lbs.		336 lbs.		392 lbs.		448 lbs.	
	Concrete Thickness Inches.	Expanded Steel List Ref. No.	Concrete Thickness Inches.	Expanded Steel List Ref. No.	Concrete Thickness Inches.	Expanded Steel List Ref. No.	Concrete Thickness Inches.	Expanded Steel List Ref. No.	Concrete Thickness Inches.	Expanded Steel List Ref. No.	Concrete Thickness Inches.	Expanded Steel List Ref. No.	Concrete Thickness Inches.	Expanded Steel List Ref. No.	Concrete Thickness Inches.	Expanded Steel List Ref. No.	Concrete Thickness Inches.	Expanded Steel List Ref. No.	Concrete Thickness Inches.	Expanded Steel List Ref. No.	Concrete Thickness Inches.	Expanded Steel List Ref. No.
4 0	3	15	3	15	3	15	3	15	3	15	3	15	3	9	3	9	3½	8	3½	8	4	8
4 6	3	15	3	15	3	15	3	15	3	15	3	9	3½	9	3½	8	4	8	4	11	4½	11
5 0	3	15	3	15	3	15	3	9	3	9	3	9	3½	8	4	8	4½	11	4½	61	5	61
5 6	3	15	3	9	3	9	3	9	3	9	3½	9	4	8	4½	11	4½	61	5	61	6	61
6 0	3	9	3	9	3	9	3½	9	3½	9	3½	8	4½	11	5	11	5½	61	5½	10	6	10
6 6	3	9	3	9	3½	9	3½	8	3½	8	4	8	4⅝	61	5½	61	5½	10	6	10	6½	62
7 0	3½	9	3½	9	3½	8	3½	8	4	8	4½	11	5	61	5½	10	6	10	6½	62	6½	62
7 6	3½	9	3½	8	3½	8	4	8	4	11	4½	61	6	61	6	10	6½	62	6½	62	7½	62
8 0	3½	8	3½	8	4	8	4½	11	4½	11	5	61	6	10	6½	62	6½	62	7½	62	8	31
8 6	4	8	4	8	4½	11	4½	61	4½	61	6	61	6½	10	6½	62	7½	62	8	31	8	30
9 0	4	8	4½	11	4½	61	5	61	5	61	6	10	6½	62	7	62	8	31	8	30	9	30
9 6	4½	11	4½	61	5	61	5½	61	6	61	6½	10	7	62	8	62	8	30	9	30	10	30
10 0	4½	61	5	61	5½	61	5½	10	6	10	6½	62	7½	62	8	30	9	30	10	30	10	68
10 6	5	61	5½	61	5½	10	6	10	6½	10	6½	62	8	31	8½	30	10	30	10	68	10	68
11 0	5½	61	5½	10	6	10	6½	10	6½	62	7	62	8	30	9½	30	10	68	10	68	10½	68
11 6	5½	10	6	10	6½	10	6½	62	6½	62	7½	62	8½	30	10	30	10	68	10½	68	11½	68
12 0	6	10	6	10	6½	62	6½	62	7	62	8	31	9½	30	10	68	10	68	11	68	12	68
12 6	6½	10	6½	10	6½	62	7	62	7½	31	8	30	10	68	10	68	11	68	12½	68		
13 0	6½	62	6½	62	7	62	7½	62	8	31	8½	30	10	68	10	68	12	68				
13 6	6½	62	7	62	7½	62	8	62	8	30	9½	30	10	68	11	68						
14 0	6½	62	7½	62	8	31	8	30	8½	30	10	30	10	68	12	68						
14 6	7½	62	8	62	8	30	8½	30	9	30	10	68	11	68								
15 0	8	62	8	30	8½	30	9½	30	10	30	10	68	12	68								
15 6	8	30	8	30	9	30	10	68	10	68	10	68										

Note.—Generally it is not economical to use thicknesses shown below the zigzag line in the table.

(*The Expanded Metal Co., Ltd.*)

rods, bent to the shape of the steelwork, and secured in position, to which to tie the lathing, and to allow of the plaster having a better 'key.' In high-class work, the encasing should be in one body and not less than 2 ins. thick; that thickness can be obtained in the following ways: for plaster finish, by fixing the lathing 2 ins. from the face of the steelwork, filling in the space with concrete, and covering the lathing with plaster; for concrete finish, by using temporary centring and filling in the concrete encasing, with 1½-in. 'Expamet' expanded steel at the centre of its thickness.

Girder Encasing.—For boxing-in exposed girders, etc., not more than 12 ins. on side or soffit, 'Expamet' expanded metal lathing can be wrapped round the beam and secured by staples, tenter hooks, or spikes to the underside of the floor above; or, preferably, it can be wired to ⅜-in. round rod cradling fixed at 12-in. to 15-in. centres along the beam. In the first system the long way of the mesh in the lathing should be vertical on the sides, and in the second it should be horizontal. For larger section beams, for high-class and for heavy work, the second system should always be used.

Wall Casing or Deafening is constructed by means of holdfasts, set into the wall at about 4 ft. centres vertically, and spaced horizontally according to the weight of the lathing to be used. These holdfasts are holed or hooked at their outer end to receive flat vertical bars, which are either bolted or wired in position, the lathing and plaster being applied as described elsewhere.

Air-ducts, False Beams, Cornices, Pilasters, etc., can be constructed by means of light steel framework fixed to receive the lathing, and if the cradles cannot be placed sufficiently near each other themselves to take the lathing, they should be aligned with ⅜-in. diameter rods wired to them, horizontally or vertically, as the case may be, to form backing for the lathing as well as stiffeners for the cradles.

Concrete Pipes, Conduits, Sewers, etc., built in situ.

'Expamet' Expanded Steel sheets can be cambered either long way or short way of the mesh to suit any requirements. The diagonals of the meshes should run one with, and the other at right angles to, the periphery of the pipes. The reinforcement should not only be laid, but it should also be fixed in place, so that there may be no possibility of its losing its correct position; it is advisable to clip or wire the sheets together at overlapped joins, and for pressure pipes to allow an overlap of not less than two meshes.

With particular regard to pressure pipes, the concrete forming the main shell should not be depended upon for impermeability, for experience has shown that the tamping seams are to be feared, no matter how carefully fresh material may be added to that which has already been laid.

For pressures of up to 50 feet, a ¾-in. facing of cement and sand, 1 to 2, laid in one body on the centring immediately before the concrete or on the concrete after the removal of the forms, has proved effective. But for greater pressures, say up to 100 ft., the facing should be laminated or put on in coats, the first coat not less than ½ in. of cement and sand 1 to 2; the second of about ¼-in., 1 to 1, troweled to a hard polished surface, and a final wash of thick grout.

Generally speaking, the thickness of the concrete shell should in no place be less than one-twelfth of the greatest internal diameter of the pipe, with an allowance of 1 in. for cover to the reinforcement, whilst the steel should be designed to resist the whole of the tensile stress.

Concrete Mixtures for 'Expamet' Expanded Steel Reinforcement.

The following mixtures of concrete, all parts by measure, are recommended for use with 'Expamet' expanded steel reinforcement:—*For floors, walls, etc., up to 3½ ins. thick:* 3 parts of aggregate, 1½ parts of sand, to 1 part of Portland cement; *thicker than 3½ ins.:* 4 parts of aggregate, 2 parts of sand, to 1 part of Portland cement. *For general concrete work such as foundations and heavy work:* 5 parts of aggregate, 2½ parts of sand, to 1 part of Portland cement. *For concrete to retain liquid pressures:* 3 parts of aggregate, 2 parts of sand, to 1 part of Portland cement. The aggregate should be clean sharp broken flint gravel, or hard and durable broken stone. For steelwork encasing, flooring and similar work, if such aggregates are not readily obtainable hard broken brick or hard furnace clinker, or a mixture of the two, free from combustible materials, old mortar, ashes and dust, may be used. Cinders and coke breeze are not recommended for reinforced concrete work. The aggregate should be of various sizes between ¾ and ¼ in. for the 4½, 5, and 6–1 mixtures, and from 1½ in. to ⅛ in. for the 7½–1 mixture, all fine stuff below ⅛ in. may be used in place of sand. The sand should be clean sharp gritty sand consisting of mixed or graded sized grains from ⅛ in. down, with the coarse grains predominating. Portland cement must be used, and it should be in accordance with the requirements of the current British Standard Specification. The concrete should be mixed three times dry and three times after water has been added, and must be deposited before the cement has reached its initial set, and must not afterwards be disturbed.

Fire-Resisting Materials.

The best materials to resist fire are brickwork, burnt clay in any form, plaster, and concrete.

Girders and Arches form a heavy floor; the girders may be protected by blocks of fireclay, or plaster, or concrete. Fire-brick arches 4′ to 12′ span, spandrels filled in and backed up with concrete, or arches may be of concrete only.

Figs. 55 to 58 (page 333) show various forms of fire-resisting floors. The concrete in some cases is made of calcined gypsum, which does not lose its cohesive power even when raised to a white heat and drenched.

The American Concrete Institute recommends that for fire-resistive construction, limestone, trap rock, blast-furnace slag, and burned clay be given a preference over highly siliceous gravels; that in cases where gravel aggregate is to be used, with no additional protective material over the concrete, round columns be given a preference over rectangular ones; and that where gravel aggregate is used all columns, but especially rectangular columns and round columns with spiral reinforcement, be given the additional protection of approximately 1 in. of Portland cement plaster either on metal lath or reinforced by light expanded metal.

Bracket.

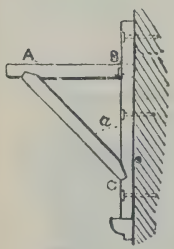


FIG. 63.

If a weight, W, is placed at point A, fig. 63, tension tending to pull bracket out of socket at B

$$= W \tan a = W \frac{AB}{OB}.$$

Compression on $AC = \frac{W}{\cos a} = W \frac{AC}{BC}$. If W is distributed over AB, these strains will be $\frac{1}{2}$ W tan a and $\frac{1}{2} \frac{W}{\cos a}$; and there will be a maximum bending moment on $AB = \frac{W \times AB}{8}$, which is lessened by the tensile stress $\frac{1}{4}$ W tan a tending to prevent bending and to draw it out at B.

Vertical strain on BC = W

Partitions.

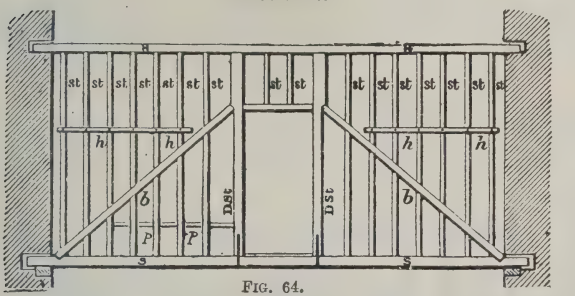


FIG. 64.

Partitions (fig. 64) should be framed so as to be supported only by the walls. *b* braces. H head, S sill, *st* studs. D*st* door-studs.

BLOCKS FOR PARTITION WORK.

A form of block suitable for partition work has been developed in the United States. The material is composed of five parts by volume of ground quicklime, ten parts of slaked lime, and one of wood fibre. The block can be removed from a mould in ten minutes, and in twenty minutes can be handled. After seven days it has a compression strength of 100 lbs. per sq. in. It can be sawn, and nails can be driven into it.

Dimensions of principal timbers, *b*, H, D*st* :—

4″ × 3″	for a bearing not exceeding	20	feet.
5″ × 3½″	„	„	„	30 „
6′ × 4″	„	„	„	40 „

The filling-in pieces, ' studs, 'or ' quarterings,' need not be thicker than 2″, or just sufficient to nail the laths to. They are tennoned to the top and bottom plates, but butted and nailed on to the braces. If they exceed 3′ or 4′ in length they should be strengthened by short struts or horizontal pieces, as at *p*, *p*, called ' nogging pieces,' or,what is better, by a continued rail, *h h*, notched across the uprights and nailed to each. In a brick nogged partition the nogging pieces are essential. They are much thinner than in a lath-and-plaster partition, being about ¾″ thick and 2″ wide, placed at every fourth course of the brickwork in height.

The studs or quarterings for a lath-and-plaster partition should be spaced at from 12″ to 18″ from centre to centre ; in a brick nogged partition they should be 18″, 2′ 3″, or 3′ apart. If a partition is not well framed, or is made with unseasoned timber, the plastering will crack. Weight of a square (100 feet super) of partitioning may be taken at 1,480 to 2,000 lbs.

Slag wool, packed in the full depth of the joists between ceiling and floors, is an effective deadener of sound, and this method has the advantage of leaving no hollow spaces in which vermin can congregate, or in which fire can work its way unseen. Sprips of hair felt should be laid along the joists to prevent continuity of woodwork, otherwise sound will be transmitted through the woodwork itself. The approximate quantity of slag wool required to fill a square yard of flooring, say 9 inches deep, will be 80 lbs. This method is perhaps a little more costly than some architects care to adopt, and lesser thicknesses of slag wool produce very satisfactory results.

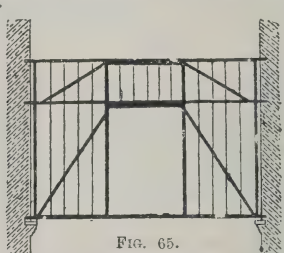


FIG. 65.

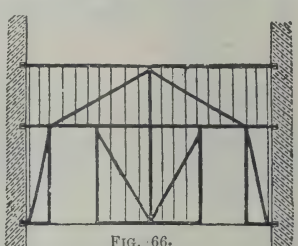


FIG. 66.

Figs. 65 and 66 show arrangements for partitions having doorways ; in both cases the main portion of the partition is suspended from the truss forming the upper part of it.

For rendering partitions, as far as possible, sound-proof, 'Willesden Paper' (*The Willesden Paper and Canvas Works, Ltd.*) may be used with advantage ; it is also damp-proof.

Stairs.

Stairs (fig. 67) are described according to their form in plan, viz. :—

1. *Straight—a.*

2. *Dog-legged—b* ; when the flights rise alternately in opposite directions with no interval between the flights.

3. *Open newel—c* ; when the flights are ranged round an open well-hole, with a newel post at each angle.

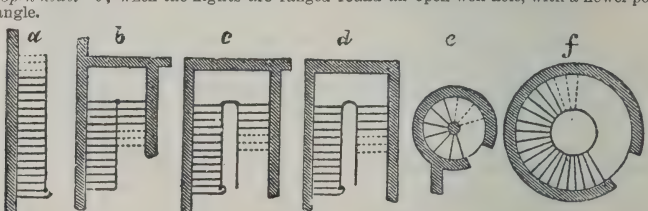


FIG. 67.

4. *Geometrical—d* ; similar to the last but with no newel posts ; the handrail running uninterruptedly from bottom to top. The angles of the well-hole are generally rounded.

5. *Circular newel—e.*

6. *Circular geometrical—f* ; when without a central newel.

Width—Stairs should be 3′ 6″ wide to allow two persons to pass easily.

Tread and Rise.—Tread should be from 9″ to 12″, rise from 6″ to 7½″. Tread increases as rise decreases. The following rules are in use for proportioning rise and tread of steps :—

1st. Tread in inches + twice rise in inches = 24. 2nd. Tread × rise = 66.

Flights.—It is dangerous to have a stair between two floors in one continuous flight. Considering convenience and effect, it is desirable to limit height of one flight to about 7′, and to avoid having two flights in one direction. *Winders* should be avoided if possible.

Handrail.—Upper surface should be from 2′ 7″ to 2′ 9″ above the surface of the tread at the nosings, and on landings from 3′ to 3′ 3″. Or, taking the height at landings as 3′ subtract ½ the rise for height of rail from nosings of steps.

Materials.—In lofty dwellings it is desirable that stairs should be of stone or concrete, to be fireproof, at any rate in the lower storeys.

Stone Stairs.—Steps are supported by a wall at one or both sides. When at one side only, they are called hanging steps, and should be built at least 9″ into the wall. The bottom step is bedded on the floor, and the lower front edge of each other step rests on the upper back edge of the step below. The best form for the joint is shown in fig. 68, where the face *ab* is at right angles to the general inclination of the stair, and about 1″ wide. Spandrel steps should have the sides which are built into the wall left square.

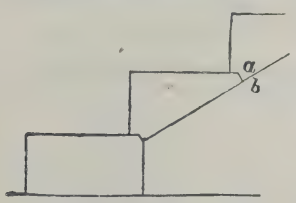
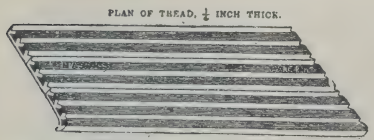


FIG. 68.

Repairing Stone Steps when Worn.—One plan is to fill up the hollows with Portland cement concrete made with fine sharp washed gravel about the size of a pea, 1 part gravel, 1 part cement. The worn part should be chiselled down, so as to give at least ¾″ thickness of concrete in every part. The edge of concrete steps should always be rounded. Another method is to insert and cement slabs of artificial stone instead of concrete; the stone known as the 'Silex' brand is very suitable for the purpose.



Lines thus ▒▒▒ *represent Lead—other parts Chilled Steel, Brass, or other Hardened Metals.*

FIG. 69.

MASON'S TREADS.

(*Patent.*)

These treads, fig. 69, which have remarkable unwearable and non-slipping qualities, are adapted to either wood or stone stairs. The arrangement is such that a hard metal withstands the wear, and a soft one gives the foothold.

Wooden Stairs.—Fig. 70. —Treads should be of hard wood, 1⅛″ thick, for steps 4′ long, increasing ⅛″ for every additional 6″ of length.

The treads and risers are supported by a wall string and an outer string, the former 1½″ thick, the latter 2″. Treads and risers are housed into the wall string, and secured by wedges

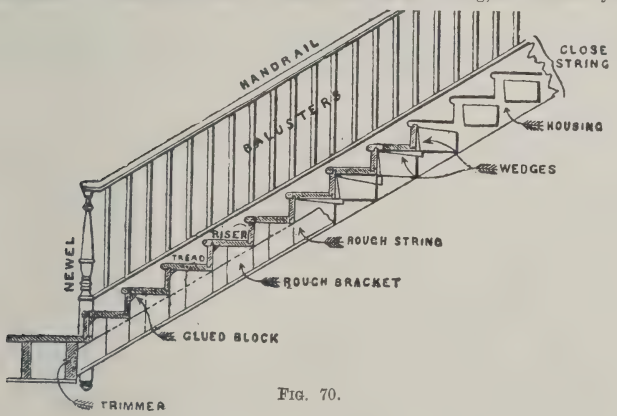


FIG. 70.

The outer ends may be either housed into a close string, or may rest on and be mitred to the notches of an open string.

When stairs are more than 3 ft. wide, treads should be supported by rough strings or carriages fixed underneath and framed into the trimming joists of the landings. Scantlings of rough strings are calculated as bridging joists. Rough brackets are nailed to the side of the rough string to support the treads.

Joints between treads and risers are grooved and tongued, feathered or rebated. The last is the best joint.

Heat Transmission through Walls, Concretes and Plasters.

See Building Research Board, Special Report, No. 7, 1923 (published by His Majesty's Stationery Office, Adastral House, Kingsway, London, W.C. 2; price 1*s.* 6*d.* net).

This report gives the results of experiments carried out at the National Physical Laboratory by Ezer Griffiths, D.Sc.

A reprint (Technical Paper No. 6), price 9*d.*, postage extra, has recently been printed.

A Special Report *re* the general subject is in course of preparation.

SECTION IX

BUILDING MATERIALS AND CONSTRUCTION

(With Contributions by A. L. Howard, F. A. W. Philips, M.Inst.C.E., J. T. Rea, A. B. Searle, Robert S. Vesey, G. Stanley Walpole, D.Sc., F.I.C. and other authorities.)

PART I

TIMBER — WOODEN PILLARS — ROOFS — BRICKWORK — WALLS — LONDON BUILDING ACTS — FOUNDATIONS — MASONRY (pp. 347-382)

PART II

MORTARS—CONCRETES—CEMENTS (pp. 383-408)

PART III

PLASTERING—LATHING—MOULDINGS—SLATING—TILING — GLAZING — PAINTING — PAPER-HANGING — PLUMBING — BOLT FIXING (pp. 409-422)

SECTION IX

BUILDING MATERIALS AND CONSTRUCTION.

PART I

TIMBER—SEASONING—CREOSOTING—WOODEN PILLARS—ROOFS—BRICKWORK—WALLS—LONDON BUILDING ACTS—FOUNDATIONS—MASONRY.

TIMBER.

(Contributed by Alexander L. Howard.)

In all specifications for building materials and construction, it is customary to use the terms 'Timbering,' 'Joiners' work,' and 'Cabinet work.' The first-named applies only to the wood used in constructional work. These terms are again subdivided into the following :—

Baulks.—Small-sized squares up to 9 ins.

Timber.—Includes everything, whether hewn or sawn, above the size of baulks.

Planks.—Includes everything above 1½ ins. in thickness, and more than 9 ins. in width.

Deals.—Confined entirely to widths of 9 ins. of thicknesses above 1½ ins. Thicknesses less than this are termed deal boards.

Battens.—Includes any thickness above 1½ ins., and over 4 and under 9 ins. in width.

Boards.—Includes everything, whatever the width, of 1½ ins. in thickness and under.

Scantlings.—An inclusive term which covers the whole of the requirements with the exception of prepared floorings, skirting or joiners' work, but which is generally used more particularly to describe small-sized pieces which are not named specifically above, such as 3 ins. × 4 ins., 2½ ins. × 3 ins., etc.

Lumber.—The term used in Canada and the United States for sawn wood of all descriptions.

It is customary in some specifications to require that the boards shall be of deal widths or of batten widths. In such case the above limitations are observed.

For scaffolding, etc., round timber is described by the following terms :—

Scaffold poles.—These range in size from 2½ to 6 ins. in diameter in the middle.

Spars.—Above 6 ins. in diameter in the middle.

Rickers.—Sizes less than the above.

In specifying thicknesses and widths it is customary, except in the case of Government contracts, to accept these after planing; thus, where 1-in. thickness is specified, the completed work is ⅞ in. in thickness, while a width of 12 ins. should actually finish a full 11¾ ins.

Timber Measures.

BAULKS AND TIMBER.—Sold at per load of 50 ft. cube. Deals, planks, battens, and scantlings at per Leningrad standard. (One standard = 165 ft. cube.)

PREPARED FLOOR BOARDS.—These are usually sold at per square. This is a standard measurement which equals 100 ft. superficial. Till recently it was the trade custom to supply a specific number of feet, according to the width of the board, which should approximate to the square. These were as follows :—

4-in. boards . . .	300 ft.	6-in. boards . . .	200 ft.
4½ „ . . .	270 „	6½ „ . . .	190 „
5 „ . . .	240 „	7 „ . . .	180 „
5½ „ . . .	220 „		

This practice, which was always unsatisfactory owing to the inaccuracy of the figures, has now been discontinued.

LUMBER (see above) AND AUSTRALIAN WOOD.—Sold at per 1,000 ft. of 1-in. superficial measure for all thicknesses of 1 in. and over. Under 1 in. at per 1,000 ft. superficial of their thicknesses. Teak at per load of 50 cub. ft.

Descriptions of Soft Woods.

PINE, SCOTS (*Pinus sylvestris*, Linn.).—Commonly called 'red' or 'yellow,' 'red' or 'yellow fir,' 'yellow deal,' 'deal.' Used for joists, floors, roofs, sash-frames, sashes, doors, centering, scaffolding, piles, etc., that is, for all timbering work in buildings excepting where baulks or logs hewn or square sawn are necessary. When reasonably free from sap and in bright condition is very durable. Archangel, Onega, Leningrad, and a few of the Swedish ports supply the best quality, but good qualities also come from other districts.

SPRUCE (*Picea excelsa*, Link.).—Commonly called 'white,' 'white fir,' 'white deal,' or 'white battens.' Used in Scotland and the north-west districts generally, for timbering work and for floorings, skirtings, and joiners' work of the cheaper class. In the midlands and south used for the latter purposes, but there usually considered unsuitable for timbering work. When used for boards, skirtings, joiners' work, etc., and properly ventilated, is fairly durable, though not in the same degree as Scots pine. Archangel, Onega, Leningrad, and some Swedish ports supply, as with Scots pine, the best quality, though good timber also comes from other districts.

SPRUCE, CANADIAN AND AMERICAN (*Picea alba*, Link.).—Although used to a certain extent in Scotland and the north-west districts, and occasionally elsewhere, it is not suitable for building construction as it is not durable, nor does it possess a sufficient degree of strength for such work.

LARCH (*Larix europœa*, DC.).—Not commonly used for building construction, though its strength and durability would recommend it equally with Scots pine. Although the supplies of British-grown larch are not ordinarily made use of for such work, yet its good qualities recommend it for more general consideration.

PINE, PITCH (*Pinus palustris*, Mill.).—Commonly used for large timbering sizes, joiners' work and flooring. Not usually employed for ordinary timbering work. Stronger than the descriptions named above, and equally durable, but only when completely ventilated. Should not be used for constructional work when thorough ventilation is impossible. Imported in all sizes and descriptions sawn and hewn square of large sizes and long lengths. Planks, deals, battens and boards are imported in three qualities described as 'prime,' 'unsorted or rio,' and 'merchantable.' In the import of square sawn or hewn timber the further standard of quality is indicated by the terms 'long leaf' or 'short leaf.' It is probable that the best quality only, i.e. 'long leaf,' is the produce of *Pinus palustris*, while that described as 'short-leaf' is the produce of many sorts, including the Loblolly pine (*P. taeda*, Linn.).

PINE, RED CANADIAN (*Pinus rubra*, Mich.).—Used commonly in Scotland and the north-west districts, and only occasionally in the south and the midlands. Generally shipped in lengths of from 10-16 ft., though occasionally in shorter lengths; on this account generally unsuitable for timbering work: also in deal and plank sizes, and small quantities of other scantlings. Similar to Scots pine in appearance, though slightly redder in colour. Excellent in constructional and joiners' work, though for the latter, perhaps, rather hard in texture. Very durable in all situations, but requires good ventilation.

PINE, AMERICAN WHITE (*Pinus strobus*, Linn.).—Known in England as 'yellow pine,' in Scotland as 'white pine,' or 'yellow pine,' but in America always as 'white pine.' Formerly used for joiners' work in England and largely so in Scotland and the north-west districts, but of late years has been too expensive for such uses.

PINE, OREGON (*Pseudotsuga Douglasii*, Carr.).—Known also as 'British Columbian pine,' and 'Douglas fir.' Obtainable in very long lengths and large sizes, very free from knots and generally only carrying a moderate amount of sapwood. Not very favourably received hitherto as it is not considered durable. Mostly used for roofing timbers for public buildings and for floorings; also a restricted amount for joiners' work; for this purpose not much in favour as the hard grain rises or the soft grain sinks, so presenting a ridgy surface under paint or varnish.

Descriptions of Hard Woods.

ACACIA (*Robinia Pseudacacia*, Linn.).—Weight, 51 lbs.* Europe. Golden-yellow. Hard, very durable, especially in contact with the ground, and in alternate wet and dry. When well seasoned is stronger, stiffer, and tougher than oak. Used for sills, wall-plates, posts, palings, trenails for ships. Would be well adapted for decorative cabinet work.

ACAPU (*Andira aubletii*).—Weight, 70 lbs. Brazil. Dark brown, very hard, durable and strong; excellent for flooring and frame-work.

ALDER (*Alnus glutinosa*, Gaert.).—Weight, 30 lbs. Great Britain. Brown; very durable under water, works well and easily. Valuable for piles, planking, sluices, pumps, etc. Used also in turnery and for other light purposes; large quantities also in the form of plywood.

ALMOND WOOD, INDIAN (*Chickrassia tabularis*, Adr. Juss.).—Weight, 49 lbs. India. Yellowish to reddish-brown, with beautiful satiny lustre and fine silver grain. Seasons and works well. Used for furniture and carving.

ANJAN (*Hardwickia binata*, Roxb.).—Weight, 82 lbs. India. Dark-red, streaked with black or purple; cross-grained, extremely durable, very hard; liable to split, does not warp. Excellent for sleepers but difficult to work; used for naves for cart-wheels, and for agricultural implements.

---

* Weights given are average dry weight per cubic foot.

(*Anogeissus latifolia*, Wall.).—Weight, 62 lbs. India. Purple-brown, shining, very hard, strong, and tough but liable to split in seasoning. Used in shipbuilding, for axles of carts and also for furniture.

ASH, ENGLISH, CANADIAN AND AMERICAN (*Fraxinus excelsior*, Linn., *F. americana*, Linn. etc.).—Weight, 41–46 lbs. The British Isles and North America. White to brownish-white, sometimes with darker streaks. Easily worked, very flexible; strong, tough and elastic, therefore well adapted for work exposed to sudden shocks and strains. Not durable unless thoroughly ventilated. Largely used in air-craft construction.

ASH, JAPANESE (*F. mandschurica*, Rupr.).—Weight, 34 lbs. Japan. Not so strong or tough as the English species, although strong enough for many purposes, including carriage building. Takes a high finish; useful for interior fittings and cabinet work. Liable to shrink more in seasoning than English ash, but stands well when seasoned; has an average weight of from 12 to 15 per cent. less.

BASSWOOD (*Tilia americana*, Linn.).—Weight, 30 lbs. North America. Pure white to whitish-brown, soft, easily worked, durable after seasoning though liable to shrink in the process. Used for turnery, carpentry and furniture, and is especially useful in the pianoforte trade. Should not be confounded (although it generally is) with canary whitewood (*q.v.*).

BEECH (*Fagus sylvatica*, Linn.).—Weight, 42 lbs. Europe. Light, yellowish-brown, showing when cut on the quarter the silvery flecks of the medullary rays. Hard, close, fine and even-grained. Very durable when submerged, therefore suitable for piling; used also for machinery, chair-making, tool-handles, turnery, and a great variety of other purposes.

BIRCH (*Betula lenta*, Linn.).—Weight, 48 lbs. Europe and North America. Firm, tough; works up easily to a satiny lustre. Used for wheelwrights' work and for furniture; large quantities also for plywood.

(*Bischofia javanica*, Blume).—Weight, 45 lbs. India. Red, tough, moderately hard. Though liable to warp and crack above ground, yet when in wet ground or completely submerged is almost imperishable. Suitable for piling and all constructional work, and for building.

BLACKBUTT (*Eucalyptus pilularis*, Sm.).—Weight, 57 lbs. Australia. Light-brown, heavy and somewhat difficult to work owing to interlocked grain; very strong and durable. Used in large quantities for street paving and for sleepers; suitable for all constructional work.

BLACKWOOD, AFRICAN (*Dalbergia Melanoxylon*, Guill. and Perr.).—Weight, 74 lbs. Tropical Africa. Sometimes known as Senegal ebony. Dark-brown to jet-black with slightly fragrant scent. Used in turnery and for musical instruments.

BLACKWOOD, AUSTRALIAN (*Acacia Melanoxylon*, R.Br.).—Weight, 45 lbs. Australia. Dark reddish-brown with a lustrous sheen, often beautifully figured. Easily worked, liable to warp unless thoroughly seasoned, but when seasoned stands exceedingly well under the most trying conditions. Useful for all kinds of construction, carpentry and cabinet work. The figured wood is used in the form of veneer for panelling, etc.

BLACKWOOD, BOMBAY. See Rosewood, Indian.

(*Boehmeria rugulosa*, Wedd.).—Weight, 41 lbs. India. Red, moderately hard, smooth and even-grained; seasons well and is easily cut. A splendid wood for carving; suitable for turnery.

BOXWOOD (*Buxus sempervirens*, Linn.).—Weight, 30 to 60 lbs. Europe, Asia Minor, etc. Exceedingly hard, very dense and close in the grain. Cuts well without splintering; susceptible of a fine polish; liable to split unless carefully and thoroughly seasoned. Used largely for blocks for wood-engraving, also in turnery, for instruments for sports and games, etc., and for a great variety of other purposes.

BOXWOOD, KNYSNA, OR EAST LONDON (*Gonioma Kamassi*, E. Mey.).—Weight, 58 lbs. South Africa. Uses as above.

(*Calophyllum spectabile*, Willd.).—Weight, 38 lbs. India and Ceylon. Shining, light-red, cross-grained; squares up to 25 ft., siding 18 ins. Used for masts, spars and planking.

CAMPHORWOOD (*Cinnamomum camphora*, Nees.).—Japan. A medium-grained, fragrant, cinnamon-coloured wood. Used for the linings of drawers and cabinets.

CAMPHORWOOD, BORNEO, or KAPOR (*Dryobalanops aromatica*, Gaert.).—Weight, 47 lbs. Malay Peninsula. Straight-grained, hard, tough, and durable. Used for heavy constructional work, planks, beams, piles, etc., and for joiners' work and shipbuilding.

CEDAR.—A name given erroneously to many species of *Juniperus* and *Cedrela*, etc. True cedars are the Himalayan (Deodar), Atlantic and Lebanon cedars.

CEDAR, CIGAR BOX (*Cedrela odorata*, Linn.).—Weight, 30 lbs. Tropical America. Also known as West Indian cedar. A soft, red, fragrant wood; used as its name implies.

CEDAR, MOULMEIN (*Cedrela Toona*, Roxb.).—Weight, 30 lbs. India. Warm red, durable and easy to work. Used for cabinet-making, furniture, cigar boxes and carving.

CEDAR, PENCIL (*Juniperus virginiana*, Linn.).—Alternatively known as red cedar and Virginian cedar. Weight, 30 lbs. United States. Light, soft and fine-grained. Very durable, especially in contact with the soil. Used chiefly for pencil-making; also employed, though to a limited extent, for cabinet work and panelling.

CHAPLASH (*Artocarpus Chaplasha*, Roxb.).—Weight, 34 lbs. India. Yellow to brown, even-grained; seasons well and is durable. Suitable for building and joiners' work; also for furniture.

CHESTNUT (*Castanea vesca*, Gaert.).—Weight, 30–40 lbs. Europe. Commonly called 'Sweet Chestnut,' or 'Spanish Chestnut.' Light to dark brown; resembles oak, from which it is only distinguished by the absence of the pronounced medullary ray so evident in that timber. A very valuable wood, insufficiently appreciated. Easier to work than oak, although more tough

not so strong or stiff. Very durable in water when wholly submerged, or in outdoor work where ventilated, and in joiners', cabinet and panelling work. Does not shrink or swell much; gives way suddenly under cross-strains. Used in the past for roof timbers and panelling; often mixed with oak (a very intelligent manner of use, as, unlike oak, insects, fungus, wood-boring beetles, etc., dislike it, so that its use mixed with oak has helped to preserve the latter from such attacks). Also largely employed for fencing, posts and rails, hop-poles and furniture. Large-sized trees are much spoilt by spiral growth and cup and heart shakes, suggesting careless forestry during growth.

CHESTNUT, HORSE (*Aesculus hippocastanum*, Linn.).—Weight, 30 lbs. Europe. Yellowish-white, soft, and not very durable. Used for rollers in cotton mills.

COCOBOLO.—Weight, 70 lbs. South America. A handsome, rich dark-red wood streaked with black or orange; hard, strong and durable. Used chiefly in the form of veneer for decorative work, such as brush backs, handles, and turnery.

COCOBOLO, INDIAN (*Dalbergia cultrata*, Grah.).—Weight, 70 lbs. Burma. Dark red to purple, streaked with black; very hard. Uses as above.

COCUS-WOOD (*Brya ebenus*, DC.).—Weight, 70 lbs. The West Indies. Dark bronze to brown or purple; very heavy, fine and even-grained, durable. Used for musical instruments and general turnery.

COROMANDEL or CALAMANDER-WOOD (*Diospyros quaesita*, Thw.).—Weight, 57 lbs. Ceylon. Greyish-brown streaked with black. A most valuable ornamental wood, in demand for cabinet work.

CUTCH or KHAIR (*Acacia Catechu*, Willd.).—Weight, 65 lbs. India. A hard wood, varying from light to dark red; seasons well, takes a fine polish and is durable. Employed in house-building, wheelwrights' work and for agricultural implements.

DILO (*Calophyllum inophyllum*, Linn.).—Weight, 42 lbs. India and Burma. Sometimes known as the Alexandrian laurel. Reddish-brown, close-grained. Used in India for such work as shipbuilding, machinery and masts, but would be excellent for cabinet-making.

EBONY (*Diospyros ebenum*, Koenig, etc.).—Weight, 60–80 lbs. India, Ceylon, Africa. Ebonies vary in their degree of blackness, few are jet black; all, however, are heavy, hard and close-grained, and generally durable. Often used in the form of veneer for decorative cabinet work; also employed for inlay and turnery.

ELM, ENGLISH AND AMERICAN (*Ulmus campestris*, Linn., *U. americana*, Linn., etc.).—Weight, 40 lbs. Europe, N. America. Dark-brown, tough, twisted in grain, and therefore difficult to work. Durable both when completely submerged, as in piles, and when perfectly dry, but soon rots when exposed to alternate wet and dry. Warps and twists in seasoning and shrinks very much, but is not liable to split; takes nails well; not so strong, stiff, or tough, as oak. Used largely for piles, water-pipes, pumps, etc., also for coffins, agricultural tools, naves of wheels, sheaves of blocks, keels of ships, and also for chair-making.

ELM, ROCK (*Ulmus racemosa*, Thomas).—Weight, 45 lbs. Canada and the United States. Light yellowish-brown, becoming paler on exposure. Tough, elastic, very strong; durable when submerged or when thoroughly ventilated. Used for boat and ship building, ladders for ships, rubbing pieces, hoops for casks, and many other important works. Supplies of good quality growing scarce.

ELM, WYCH OR SCOTCH (*Ulmus montana*, Smith).—Lighter-brown in colour, softer and more straight-grained and therefore more easily split than English elm. Used largely in chair-making, boat-building, and for agricultural implements.

ENG or IN (*Dipterocarpus tuberculatus*, Roxb.).—Weight, 54 lbs. Burma. This dark reddish-brown wood is one of the valuable timbers of Burma. Very durable and in great demand for house and boat-building. Has been used in this country very successfully for parquet flooring; excellent for all constructional work.

(*Gmelina arborea*, Roxb.).—Weight, 36 lbs. India, Burma. Golden-yellow to greyish-white, with a glossy sheen; soft, even-grained, durable; does not warp or crack, easily worked. Used for furniture, door-panels, carriage-building, boats, light planking, etc., and in all works where shrinkage is to be avoided. Roxburgh, writing considerably over a century ago, spoke of the great value of this wood. Will probably be most appreciated for cabinet work.

GREENHEART, DEMERARA AND SURINAM (*Nectandra rodioei*, Schomb.).—Weight, 70 lbs. British and Dutch Guiana and tropical South America. Dark greenish-brown, clean and straight in the grain, hard and heavy; tough, strong and elastic. Heartwood extremely durable. Resists teredo and therefore largely used for piles, wharves and jetties; also in shipbuilding, machinery and planking; one of the best woods for fishing rods. In strength exceeds most other woods, whether tried by transverse or tensile strain. When tested for crushing force in the direction of the fibres it bears the addition of weight after weight till the crushing force is obtained. It then, with a loud report, gives way suddenly and nothing is left but a loose mass of shapeless fibres.

GREENHEART, BURMA (*Heterophragma adenophyllum*, Seem.).—Weight, 48 lbs. India, Burma, Andaman Islands. Bronze to greenish-yellow, greatly resembling the Surinam. Does not warp or split. Suitable for cabinet work.

GUM, BLUE (*Eucalyptus Globulus*, Labill.).—Weight, 50 lbs. Australia. Light-brown, hard, heavy, tough, elastic; somewhat interlocked grain, durable. Used for shipbuilding, telegraph poles, felloes, shafts, spokes and piles. Employed for the piles in Dover Harbour.

GUM, RED (*E. calophylla*, R.Br.).—Weight, 60 lbs. Australia. Yellowish-red, fairly dense

and hard, splits readily; the prevalence of gum veins impairs its suitability for constructional work. Not durable underground. Used for axe and other handles, spokes, naves, fencing rails, and agricultural implements.

GURJUN (*Dipterocarpus turbinatus*, Gaert.).—Weight, 50 lbs. Andaman Islands and Burma. Reddish-brown, capable of a smooth finish, durable; possessed of an agreeable aromatic scent which is fairly persistent. Has a close, hard, non-fibrous grain which makes the wood very suitable for a number of purposes where smoothness of surface and hard wear are required. Long lengths and large sizes can be obtained free from all defects, and the supply is expected to be on a very extensive scale. The wood seasons slowly, but although unreliable if unseasoned, when properly dried stands well. Where used for joinery work, cabinet work, ships' fittings, and floorings should always be artificially seasoned. Should be quarter sawn where employed in works of importance. Suitable for all the above named purposes and particularly for floorings, solid and parquet, especially where hard wear is required.

HALDU, INDIAN (*Adina cordifolia*, Hook. f.).—Weight, 42 lbs. India, Burma. Lustrous golden-yellow, even-grained, capable of a fine finish from the tool; durable, seasons well. A beautiful wood for cabinet-making and turnery, and is especially suitable for carving.

HICKORY (*Hicoria alba*, Britt., etc.).—Weight, 50 lbs. North America. Brownish-white to reddish-brown, hard, strong, exceedingly tough, springy and elastic; smooth and straight-grained. Liable to shrink and warp, not durable if exposed. Used very largely for spokes and rims of carriage wheels; also for gunstocks, chair-making, and carriage-building; the best wood yet discovered for golf shafts.

HINOKI (*Cupressus obtusa*, Koch.).—Japan. Yellow to rose-red, sweet-scented, fine-grained, takes a high polish. Used in Japan for decorative work in buildings, etc.

HORNBEAM (*Carpinus betulus*, Linn.).—Weight, 50 lbs. Europe. Yellowish-white; hard, heavy, close-grained; loses much weight in drying, tough and difficult to split. Durable if kept dry, but decays quickly if exposed to wet. Used for cogs, handles, mallets and pianoforte keys.

IROKO (*Chlorophora excelsa*, Benth. and Hook. f.).—Often miscalled African teak. Weight, 54 lbs. Tropical Africa. Yellow to brown, darkening on exposure; strong, straight-grained, very durable. Used as a substitute for teak for gun carriages, joinery and cabinet work, building and constructional work.

IRONBARK (*Eucalyptus leucoxylon*, F. v. M.).—New South Wales. Deep red, hard, heavy, strong, difficult to work. Suitable for piers, wharves, etc., and constructional work generally.

JARRAH (*E. marginata*, Sm.).—Weight, 60 lbs. Western Australia. Reddish in colour, hard, dense, straight-grained, with somewhat interlocked fibres; very durable both in and above ground, and almost non-inflammable. An excellent timber, obtainable in very long lengths and wide widths. Used for all heavy constructional and engineering works, piles for docks, wharves, jetties, and fencing. Excellent for flooring, house-building and for street paving blocks; suitable also for cabinet work.

JARUL (*Lagerstroemia Flos-reginae*, Retz.).—Weight, 43 lbs. India, Burma. Light red and lustrous. Used for shipbuilding, constructional work, and for carts; suitable for cabinet-making.

KAKI (*Diospyros Kaki*, Linn. f.).—Japan. Dark grey streaked with black. This is the Japanese ebony. Used in turnery and ornamental work.

KARRI (*Eucalyptus diversicolor*, F. v. M.).—Weight, 63 lbs. Western Australia. Greatly resembles jarrah in appearance; red, dense, tough, elastic; produces timber of exceptional size. Suitable for most of the uses to which jarrah is put, though it is not so durable for underground work or in damp situations. Somewhat prone to dry rot; almost non-inflammable. The heart-wood core should be avoided. When 'Powellised,' is now reported as exceedingly durable and termite-proof for sleepers, etc.

KAURI PINE or COWDIE PINE (*Agathis australis*, Salisb.).—Weight, 38 lbs. New Zealand. Yellowish-white or straw colour, strong, fine, and even-grained; planes well and takes a good polish with a silky lustre on the surface; shrinks very little and is remarkably free from defects. Very valuable for deck planking and for fitments in ships; also used for joiners' work and ornamental purposes.

KEYAKI (*Zelkowa acuminata*, Planch.).—Japan. Hard and strong; lustrous and often highly-figured, when it is used for fine cabinet work. Also employed in building and naval architecture; excellent for carving.

KINGWOOD.—Brazil. A handsome dark brown streaked with crimson to violet and purple, fine grained. Used in fancy cabinet work and turnery, generally in the form of veneer.

LABURNUM (*Cytisus laburnum*, Linn.).—Europe. Dark brown to bronze, hard and close grained; takes a good polish. Used in turnery and for pulleys and blocks.

LANCEWOOD (*Guatteria virgata*, Dun.).—Weight, 50 lbs. Central and South America. Light yellow, hard and close grained, elastic. Used for carts, carriages, and shafts, and also for fishing rods.

LANCEWOOD, BURMA (*Homalium tomentosum*, Benth.).—Weight, 59 lbs. India and Burma. Greyish-brown, fine, close and even grained. Takes a very smooth surface, is apt to split in seasoning, durable. Suitable for furniture.

LANCEWOOD, INDIAN GREY (*Vitex glabrata*, Br.).—Weight, 42 lbs. India, Ceylon. Light greyish-brown with a lustrous sheen and a ripple-like figure. Takes a good polish and would be excellent for furniture.

LAUAN (*Dipterocarpus thurifer*, Blanco).—The Philippine Islands. Dark brown, strong, and heavy. Suitable for shipbuilding and furniture.

LAUREL-WOOD. See Walnut, Indian.

LEZA-WOOD (*Lagerstroemia tomentosa*, Presl.).—Weight, 50 lbs. Burma. Grey to greyish-brown, soft, works easily and takes a good polish. Would be suitable for furniture.

LIGNUM-VITAE (*Guaiacum officinale*, Linn.).—Weight, 70 lbs. Central and tropical South America. Dark greenish-brown to black, hard, heavy, strong and close-grained; almost imperishable; liable to cup-shake. Used for pestles, mortars, croquet, and other mallets, and for a great variety of purposes. Has a unique value for propeller shafts for steamers on account of its lubricating qualities.

LIGNUM-VITAE, INDIAN (*Berrya ammonilla*, Roxb.).—Weight, 62 lbs. India. Brown to purplish-red, flecked as in English beech; firm, hard texture, and satiny lustre; has a sticky feeling to the touch, planes well. Suitable for decorative cabinet work and turnery.

LIME (*Tilia cordata*, Miller.).—Weight, 30 lbs. Europe. Pale yellowish-white, soft and light; easily cut and carved; not very durable when exposed to the weather. Excellent for carving, for pianofortes and models of buildings.

LUMBAYAO (*Tarrietia javanica*, Bl.).—Weight, 44 lbs. The Philippines. Dark to light red, light and easy to work. Used for furniture.

MAGNOLIA (*Magnolia acuminata*, Linn.).—Weight, 30 lbs. United States. White, soft, not strong, but very durable. Used for interior fittings and occasionally in cabinet work. Somewhat similar to canary whitewood.

MAHOGANY.—Species uncertain. Supplies received from St. Domingo (Spanish), Cuba, Honduras, Mexico, Panama, Nicaragua, Colombia and the districts of Central America generally; the West Coast of Africa. One of the most valuable of all decorative woods, the merits of which are so universally acknowledged that further description is unnecessary.

MAHWA (*Bassia latifolia*, Roxb.).—Weight, 62 lbs. India, Burma. Reddish brown; used for house-building, furniture, naves of wheels, etc.

MALLET-WOOD, INDIAN (*Schleichera trijuga*, Willd.).—Weight, 68 lbs. India and Burma. Light reddish-brown, seasons well, takes a good polish; strong and durable but somewhat difficult to work. Suitable for handles, agricultural implements, and for furniture.

MAPLE (*Acer saccharinum*, Wang.).—Weight, 40 lbs. Canada and the United States. Creamy to yellowish-white; has a fine, smooth surface and a dense, even grain; tough and strong. Excellent for flooring, especially for hard wear, and for veneering and interior finish; also for furniture. Birds-eye maple is the figured variety, and is appreciated for cabinet work.

MARBLE-WOOD, ANDAMAN (*Diospyros Kurzii*, Hiern.).—Weight, 61 lbs. Andaman Islands. A unique and handsome wood, grey streaked with black, smooth and cold to the touch; liable to shrink and warp unless thoroughly seasoned. Valuable for cabinet work and for walking-sticks, etc.

(*Melia composita*, Willd.).—Weight, 28 lbs. India. Pale brown, resembling cedar in this, as also in grain and texture; durable and light in weight. Suitable for sugar boxes, and for boat and yacht building.

(*Mesua ferrea*, Linn.).—Weight, 72 lbs. India, Burma. Dark red, extremely hard, strong and heavy; good sizes procurable. Suitable for sleepers, wood-paving blocks, etc.

MINGRIS (*Koompassia Beccariana*, Taub.).—Borneo. Light red, very hard and heavy; not durable.

MIRABOW or MIRABOO (*Intsia Bakeri*, Prain.).—Weight, 54 lbs. Borneo, Malay States. Reddish-brown, darkening with age, hard, fine, even-grained; often beautifully figured. Tough, durable, and susceptible of a fine polish. Suitable for decorative furniture.

MORA (*Mora excelsa*, Baill. and Benth.).—Weight, 60 lbs. British and Dutch Guiana. Chestnut-brown, hard, heavy, tough. Usually straight in the grain, but sometimes wavy, which gives a beautiful figure. Takes a good polish, and therefore suitable for furniture. Extremely durable; useful for sleepers, and for beams, keelsons, etc., in shipbuilding; and also in a general way in large scantlings for civil and naval architecture.

MULBERRY, YELLOW (*Morus laevigata*, Wall.).—Weight, 45 lbs. India, Burma. Very beautiful golden to chrome yellow, shining and lustrous; has a pretty silver grain; hard and close grained. Suitable for furniture, turnery, and decorative work generally.

MYALL or VIOLET-WOOD (*Acacia pendula*, A. Cunn.).—Weight, 76 lbs. Australia. Rich dark brown, hard, heavy, close grained, often figured. Has a faint scent of violets. Suitable for veneer, turnery, etc.

NIRI (*Xylocarpus borneensis*, Becc.).—Borneo and the Philippines. Dark red, hard and heavy; fine grained and durable, shrinks little in seasoning. Useful for furniture.

OAK (*Quercus pedunculata*, Ehrh., and *Q. sessiliflora*, Sm., etc.).—Europe, etc.

British.—Weight 53 lbs. Light-brown in colour, hard, tough, very strong. Does not easily splinter; durable under water, above ground and in alternate wet and dry. Apt to warp and split in seasoning, but when once thoroughly seasoned few woods stand better. Must not be used in immediate contact with iron. Uses so extensive that it would be impossible to enumerate. Should be more extensively used, however, for panelling and decorative work generally, in place of other supplies which are not so good.

Austrian.—So-called Austrian oak was largely used before the war for decorative panelling, floorings, etc., although in point of fact it was more often specified than supplied. A clean, mild oak, generally free from much defect.

Dantzig.—Similar to the above, though slightly stronger and harder; possesses elasticity in a marked degree.

Riga, Libau, and *Odessa* oak are similar to Austrian. The last-named yields slightly larger dimensions.

French.—Resembles supplies of British, but is generally milder in character.

Italian.—Heavier and less elastic than the British; harder and more difficult to work. No supplies have been forthcoming of late years.

American (*Quercus alba, etc.*).—Weight, 42 lbs. Supplies of American oak include the produce of over twenty-two different botanical species, consequently supplies vary largely in quality, texture, durability and colour. American oak is not employed in the best descriptions of work.

Japanese (*Quercus grossesserrata,* Bl., *Q. crispula, etc.*).—Weight, 40 lbs. Similar in appearance to the European, generally closer in texture; slightly milder than the Austrian, from which it is almost impossible to distinguish it except by examination with the lens. Possesses a very uniform colour. Suitable for the best descriptions of work.

Wainscot.—A term used to describe the produce of any oak when hewn, sawn, or reft on the quarter.

PADAUK, ANDAMAN (*Pterocarpus dalbergioides,* Roxb.).—Weight, 48 lbs. Andaman Islands. A handsome crimson-coloured wood, sometimes streaked with black or brown, often finely figured. Seasons and works well, takes a good polish; shrinks very little, does not crack or split; not subject to dry rot. Excellent for all interior decoration and fittings and for furniture. One of the best woods for gun-carriages and ordnance work; has given good results for flooring. Used in America for such work as the trimming of Pullman cars, the decoration of public and private buildings and for furniture.

PADAUK, BURMA (*P. macrocarpus,* Kurz.).—Weight, 60 lbs. Burma. Resembles the Andaman variety, but is darker in colour, harder and heavier. Used for the same purposes, though possibly a slightly inferior wood.

PARTRIDGE WOOD (*Andira inermis,* Kunth.).—Brazil. Reddish-brown to black, figured and mottled. Used for cabinet work, turnery, umbrella and walking-sticks, etc.

PEAR TREE (*Pyrus communis,* Linn.).—Weight, 50 lbs. Europe. Pale brown, tough and difficult to split; close grained. Cuts with great facility in any direction without splitting, and therefore used largely for carving and wood-engraving; also for musical instruments and for turnery. Needs careful and thorough seasoning.

PLANE (*Platanus occidentalis,* Linn.).—Weight, 30 lbs. North America. Sometimes known as button-wood. Light brownish-yellow, shows a fine figure when cut on the quarter. Used for blocks, screws, etc.

POON (*Calophyllum tomentosum,* Wight.).—Weight, 40 lbs. India. Pale, rather reddish mahogany colour with darker red veins. Close, firm, hard grain; takes a smooth surface, stands well. Suitable for decorative cabinet work.

POPLAR (*Populus alba,* Linn., and *P. nigra,* Linn.).—Weight, 30 lbs. Europe. Light reddish-brown, soft, light; neither strong nor durable. Used for such work as turnery, clog-making, and toys; latterly in much demand for plywood.

PRIMA VERA or WHITE MAHOGANY.—North America. Light creamy yellow, often figured Used for interior decoration.

PURIRI (*Vitex littoralis,* A. Cunn.).—Weight, 60 lbs. New Zealand. Dark brown, hard heavy, close grained; very strong and durable. Suitable for heavy constructional work such as piles, posts and sleepers.

PURPLEHEART (*Peltogyne paniculata,* Benth.).—Weight, 50 lbs. Tropical South America. Rich purple, hard, heavy, and lustrous; works well. Used for flooring and all ornamental work such as marquetry and turnery.

PYINMA (*Lagerstroemia hypoleuca,* Kurz.).—Weight, 40 lbs. Andaman Islands. Red, hard, seasons well, durable; easy to work. Suitable for house-building, and for spokes and felloes of wheels.

PYINKADO (*Xylia dolabriformis,* Benth.).—Weight, 60 lbs. India. Rich, warm reddish-brown; close, firm, hard, rather cross-grained texture; cold and sticky to the touch. Very heavy, and becomes harder with age. Extremely durable, probably on account of the resinous substances contained in it. Largely used for railway sleepers. Suitable also for paving blocks, telegraph posts, building, piles, and beams of bridges and similar constructional work.

RASPBERRY-JAM WOOD (*Acacia acuminata,* Benth.).—Weight, 60 lbs. Australia. Dark reddish-brown to purple, streaked with black; has a scent like crushed raspberries; somewhat difficult to work. Used in ornamental cabinet work and turnery.

ROSEWOOD, BAHIA AND RIO (*Dalbergia nigra,* Fr. Allem.).—Weight, 60 lbs. Brazil. The first-named is the superior wood. Dark reddish or purplish-brown, streaked with black; faint scent like rosewater. Dense and heavy, takes a fine polish; liable to heart-shake. Used both solid and in veneer for ornamental cabinet work. Less used now than formerly.

ROSEWOOD, INDIAN (*Dalbergia latifolia,* Roxb.).—Weight, 50 lbs. India. Alternatively known as East Indian or Bombay rosewood and Bombay blackwood. Dark purple streaked with black, hard and close grained; takes a good polish. Available in long lengths and very large sizes of perfect quality. Useful for cabinet work; would be excellent for panelling.

SABICU (*Lysiloma Sabicu,* Benth.).—Weight, 50 lbs. The West Indies. Dark brown, sometimes streaked with black, often finely figured; hard, heavy, close grained; susceptible of a good polish. Used for shipbuilding, interior fittings, cabinet work, furniture, etc.

SAL (*Shorea robusta*, Wall.).—Weight, 60 lbs. Burma. Similar in colour to Indian sal, but more even grained; hard and durable. Used for building and also for small work such as tool handles.

SALMON WOOD, INDIAN (*Eriolaena Candollei*, Wall.).—Weight, 51 lbs. Burma. Bright salmon-red, handsomely marked; hard, shining, close grained, takes a fine polish, seasons well. Suitable for decorative cabinet work, bandings, and inlay.

SANDERS, RED (*Pterocarpus santalinus*, Linn. f.).—Weight, 75 lbs. India. Dark claret-red, sometimes nearly black, streaked with dark violet. Valuable for decorative cabinet work and inlay.

SATINEE (*Ferolia guianensis*, Aubl.).—Weight, 55 lbs. South America. Light reddish-brown, hard, tough, elastic; takes a good polish. Suitable for furniture, also for fishing rods.

SATIN WALNUT (*Liquidambar styraciflua*, Linn.).—Weight, 40 lbs. North America. Should be called gum, by which name it is known in America; has been called hazel pine, sweet gum, and Californian red gum in England. Has no qualities which justify its being called either walnut or pine. Pale reddish-brown, soft, close grained, has a smooth, satiny sheen. Warps and twists badly in drying, and even when seasoned is susceptible to change in the atmosphere. Used for cheap furniture; quite unfitted for street-paving, for which purpose it was tried in London some years ago.

SATINWOOD, WEST INDIAN (*Zanthoxylum*, sp.).—Porto Rico, St. Domingo and the West Indies generally. Chrome yellow to golden, maturing with age to a rose yellow; has a beautiful satiny lustre, often finely figured. Used for the finest cabinet work, brush backs, etc.

SATINWOOD, EAST INDIAN (*Chloroxylon Swietenia*, DC.).—Weight, 60 lbs. Ceylon and occasionally from India. Uses similar to the above.

SISSOO (*Dalbergia Sissoo*, Roxb.).—Weight, 50 lbs. India. Rich warm brown, sometimes streaked with golden or darker brown. Firm, hard, even texture; takes a very smooth surface and a good polish; strong and durable, seasons well. Capable of being carved to any depth in the most intricate shapes without splintering. Suitable for furniture and carving, also for gun-carriage wheels.

STRINGYBARK (*Eucalyptus obliqua*, L'Hérit.).—Weight, 60 lbs. Australia and Tasmania. Brown, hard, heavy, strong, tough and close grained. Splits easily; requires careful seasoning; liable to shakes and gum veins. Useful for a large variety of purposes, including fencing, flooring, piling, paving, mine timbering, and shipbuilding.

SYCAMORE (*Acer Pseudo-platanus*, Linn.).—Weight, 40 lbs. Europe. Yellowish-white, often finely-figured, even grained, lustrous, easily worked; apt to crack and warp during seasoning. Used for rollers for calico printing, also for washing machines, and, in the form of veneer, for panelling in carriages and railway coaches. When stained grey forms the 'greywood' of commerce.

TALLOW-WOOD (*Eucalyptus microcorys*, F. v. M.).—Weight, 60 lbs. Australia. Yellowish-brown, sometimes wavy in grain, strong and durable; liable to shakes. Used for wheelwrights' work, piling, and flooring.

TALLOW-WOOD, INDIAN (*Terminalia Chebula*, Retz.).—Weight, 48 lbs. India. Purplish-grey to brown, with a fine ripple marking. Fairly durable, takes a good polish; cross grained and hard, somewhat difficult to work. Suitable for furniture, house-building, and golf heads.

TAPANG (*Koompassia excelsa*, Taub.).—Borneo. Dark reddish-brown, dense and heavy; durable and strong, but very brittle; obtainable in large sizes. Suitable for heavy constructional work.

TEAK, BORNEO.—This term is an entire misnomer, having been wrongfully applied to a Borneo wood which is certainly not teak; *Tectona grandis* has never been found in Borneo.

TEAK, JAVA (*Tectona grandis*, Linn. f.).—Weight, 43 lbs. Java. Shorter in length and inferior in quality to Indian teak; otherwise similar.

TEAK, MOULMEIN, RANGOON, and BANGKOK (*Tectona grandis*, Linn. f.).—Weight, 45 lbs. India, Burma and Siam. Yellow to brown, darkening with age; moderately hard and strong. Contains an essential oil which acts as a preservative; extremely durable; often unsound at the centre. The chief export timber of India and Burma. Used for an immense variety of purposes; for the backing of armour plates in battleships, also house and boatbuilding and all constructional work; flooring, sleepers, carpentry, carriage-building, etc.

THINGAN (*Hopea odorata*, Roxb.).—Weight, 40 lbs. India. Bright canary yellow, resembling satinwood; mild, straight grain similar to mahogany and of the same texture; very durable. large supplies available in good sizes. Excellent for cabinet work, etc.

THITKA (*Pentace burmanica*, Kurz.).—Weight, 44 lbs. India. Light red, resembling cedar, with a pretty silver grain. Strong and durable; takes a very smooth surface and a fine polish. Suitable for artistic decorative work.

TOTARA (*Podocarpus Totara*, A. Cunn.).—Weight, 30 lbs. New Zealand. Red, fine, and even-grained, hard, heavy, tough, and very durable. Does not warp or twist; resists teredo. Used for piles, sleepers, telegraph poles and general building purposes; suitable also for interior fittings, etc.

TUART (*Eucalyptus gomphocephala*, DC.).—Weight, 70 lbs. Australia. Creamy-yellow, hard, very heavy, tough, durable; has an interlocked grain. One of the strongest timbers known and in demand where special strength and hardness are necessary. Used for railway wagons, engine-bearers, ships' beams, planking, shafts, naves and felloes, and wheelwrights' work generally.

TULIP-WOOD (*Physocalymma scaberrimum*, Pohl.).—Brazil. Striped pink to rose colour. Used chiefly for bandings, inlay, and marquetry.

TULIP-WOOD, INDIAN (*Dalbergia Oliveri*, Gamble).—Weight, 66 lbs. India. Striped pinkish-red; very hard and cross-grained. Suitable for similar purposes to the above.

TUPELO (*Nyssa sylvatica*, Marsh.).—Weight, 40 lbs. North America. Also called tupelo gum. White to light yellow, paling somewhat on exposure to air; smooth and close-grained; liable to warp. Used in a general way as a substitute for whitewood, also for electric casings and for furniture.

VENGAI (*Pterocarpus marsupium*, Roxb.).—Weight, 55 lbs. India. Golden yellowish-brown with lighter and darker streaks; smooth and lustrous; firm and hard with contrary grain; has a good figure when cut on the quarter. Used for cabinet work and interior fittings.

WALNUT (*Juglans regia*, Linn.).—Weight, 40 lbs. Europe and Asia. Greyish-brown, often finely marked and figured; hard, firm, close-grained; takes a good polish. Liable to the attacks of worm (beetle) when used for furniture. The finest wood for gun-stocks, as it withstands shock and does not warp. Used for pianofortes and furniture generally.

WALNUT, BLACK (*Juglans nigra*, Linn.).—Weight, 40 lbs. North America. Chocolate-brown to grey, darkening with age. Easily worked and more durable than European walnut; also less liable to insect attack.

WALNUT, INDIAN (*Albizzia Lebbek*, Benth., and *A. procera*, Benth., *A. odoratissima*, Benth.).—Weight, 46 lbs. India. Rich warm brown, streaked with yellow and black; often has wavy marks and broken roe and mottle. Firm, hard texture; takes a good surface from the tool and has a bright lustre; seasons and stands well; available in large sizes. Suitable for furniture, panelling, veneer, turnery, and all decorative work.

WALNUT, INDIAN BLACK (*Terminalia tomentosa*, W. and A.).—Weight, 55 lbs. India. Dark brown, variegated with darker streaks; needs careful seasoning; large supplies available. Suitable for decorative, cabinet, and instrument work.

WANDOO (*Eucalyptus redunca*, Schan.).—Weight, 70 lbs. Australia. Brownish-red, very hard, strong, dense and durable. Equal to jarrah for railway sleepers; also used for piles, bridge-building, wheelwrights' work, boat-building, fencing, and general building work.

WHITEWOOD, CANARY (*Liriodendron tulipifera*, Linn.).—Weight, 30 lbs. North America. Called in America, poplar, yellow poplar, tulip tree, etc.; known in England as whitewood, canary whitewood, and canary wood; in Scotland, wrongly called basswood. Pale yellow, soft, light, tough; easily worked, takes a good polish and stains well; liable to shrink and warp. Usefu lfor indoor building, joinery, wainscoting, chairmaking, carriage-building, etc.

WILLOW (*Salix alba*, Linn., and *S. coerulea*, Linn.).—Weight, 30 lbs. Europe. White, light, soft, smooth-grained; very durable in water. Used for brake blocks on railway wagons, as it does not readily fire with friction. Largely employed for the making of artificial limbs and for cricket bats.

YACAL (*Shorea*, sp.).—The Philippines, Borneo. Brown, darkening on exposure; good and durable. Used for railroad ties and for constructional work generally.

YANG (*Dipterocarpus*, sp.).—Siam. Similar in all respects to the Eng of Burma (*q.v.*).

YELLOW HEART (*Fagraea fragrans*, Roxb.).—Weight, 60 lbs. Burma. Light yellowish-brown, hard, close-grained, very durable. Used in Burma for house-building, piles and joinery, but in this country, where supplies are now arriving, it will probably be most appreciated as a first-class cabinet-wood.

YELLOW JACK-WOOD (*Artocarpus integrifolia*, Linn. f.).—Weight, 40 lbs. India, Burma. Rich gamboge yellow, darkening somewhat on exposure; has a hard and soft contrary grain. Used for decorative furniture work, brush backs, etc.

YEW (*Taxus baccata*, Linn.).—Weight, 50 lbs. Europe, India, etc. Reddish-brown to amber, often finely figured; very hard, close-grained and heavy. Said to be more durable than other European wood. Tough and elastic. Used for chair-making, walking sticks, furniture; and in the form of veneer for decorative work generally; also for axles, cogs for mill-wheels, etc.

ZEBRA WOOD, RED (*Melanorrhœa usitata*, Wall.).—Weight, 56 lbs. India. Red, with darker red or golden streaks; very hard. Useful for tool handles, and general decorative work. Has been recommended for building, sleepers, gun-stocks, etc.

Durability of Timber.

WOODS WHICH RESIST THE SEA-WORM (*Teredo navalis*).—No complete list of such timbers is possible, for reports vary as to this quality being possessed by any particular timber, while the increase of scientific research and the resultant preventive measures might cause such a list to be considerably enlarged. It is therefore recommended to institute enquiries concerning the timber at the time of purchase.

This same principle should be applied when considering the durability of timber in other particular circumstances.

Strength of Australian Timber.

The following table, compiled by Professor Warren, of the Sydney University, sets forth the respective values of all the Australian best hardwoods in comparison with the Western Australian Jarrah, Karri, and Red Gum :—

Timber.	Weight per Cubic Foot.	Average Strength in Lbs. per Square Inch.			
		Transverse Strength. Modulus of Rupture. 54 ins. × 6 ins. × 4 ins.	Tensile Strength.	Compressive Strength 12 ins. × 3 ins. × 3 ins.	Shearing Strength 6 ins. × 6 ins. × 3 ins.
N S.W. Timbers :	Lbs.				
Ironbark	75·18	18,204	17,900	10,500	2,200
Tallow-wood	77·06	15,300	13,400	8,500	1,660
Blackbutt	66·69	14,800	21,000	8,600	1,700
Blue gum	73·64	14,700	15,900	7,800	1,800
Grey gum	67·32	17,600	22,000	9,500	2,100
Spotted gum	62·19	15,300	16,400	8,500	2,200
Turpentine	69·34	14,400	17,800	8,500	1,600
Grey or white box	73·62	15,950	21,564	8,021	2,110
Murray red gum	62·19	11,267	13,618	9,397	2,039
Stringybark	71·33	13,083	15,976	6,098	2,089
W.A. Timbers :					
Jarrah	62·0	12,040	16,047	6,839	1,878
Karri	—	11,208	13,920	7,132	1,637
Red Gum	62·5	10,739	22,725	5,476	1,556

Content Sizes and Weights of Timber.

40 cubic feet of unhewn timber
50 „ squared „
600 super. feet of 1-inch planks or deals
400 „ 1½ „ „
300 „ 2 „ „

240 super. feet of 2½-inch planks or deals
200 „ 3 „ „
170 „ 3½ „ „
150 „ 4 „ „
} = 1 load.

120 deals = 1 hundred. Battens are not over 7 inches wide. Deals are not over 9 inches wide. Planks are mostly 11 inches wide.

One square (100 *feet superficial*) *of flooring requires :—*

12-feet deal boards :—	Number.
Rough	12½
Edges shot	12½
Wrought and laid folding	13
Wrought and laid straight joint	13½
Wrought, ploughed, and tongued	14

12-feet battens :—	Number
Rough	16
Edges shot	16½
Wrought and laid folding	17
Wrought and laid straight joint.	18

Number of cubic feet in one 1 of timber : Ash, 45 ; elm, 60 ; mahogany, 34 ; beech, 51 ; fir, 65 ; oak, 39.

Standards by which Deals are Sold.

Planks, deals, and battens are usually sold by the number required to make up the same cubic content as 6 score or a long hundred of some standard size; thus, in London the Leningrad standard is used, which with others is given below :—

Standard.	No. of Pieces.	Size.	Feet Cube.	Feet Superficial.
Leningrad	120	12 ft. × 11 ins. × 1½ ins.	=165	=1,320
London and Dublin	120	12 ft. × 9 ins. × 3 ins.	=270	=1,080
Christiana hundred	120	11 ft. × 9 ins. × 1½ ins.	=103⅛	= 990
„ „	60	15 ft. × 11 ins. × 1½ ins.	=103⅛	= 825
Quebec short hundred	100	12 ft. × 11 ins. × 2½ ins.	=229⅙	=1,100
„ long „	120	10 ft. × 11 ins. × 3 ins.	=275	=1,000
Drammen	120	9 ft. × 6½ ins. × 2½ ins.	=151⅝	= 585

Prepared floorboards are sold by the customary square, a given number of feet run, as below, varying with the width, so as to approximate to the square of 100 feet superficial :—140 feet run of 9-inch boards, 160 feet of 8-inch, 170 feet of 7½-inch, 180 feet of 7-inch, 185 feet of 6¾-inch, 200 feet of 6-inch, 210 feet of 5⅝-inch, 220 feet of 5½-inch, 230 feet of 5¼-inch, 240 feet of 5-inch.

To cut the Stoutest Rectangular Beam from a Log.

Divide the diameter into 3 equal parts, *a*, *a*, *a*, and draw lines at right angles as represented. $h = 1 \cdot 414\, b$. b = diameter ÷ 3.

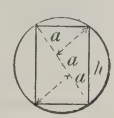


FIG. 1

To cut the Stiffest Rectangular Beam from a Log.

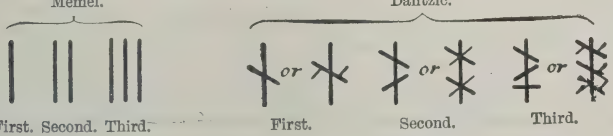
Divide the diameter into 4 equal parts, and draw lines at right angles as represented. $h = 1 \cdot 732\, b$. b = diameter ÷ 2.

Selection of Timber.

In selecting timber the most convenient sizes are 12 inches square. Choose the highest in colour, where the strong red grain appears to rise to the surface; avoid spongy hearts, porous grain, and dead knots.

SCRIBE MARKS.

The following are the 'scribe marks' indicating the quality of Memel and Dantzic timber :—



Memel timber is generally about 13½ or 14 inches square; Dantzic timber about 18 or 19 inches. When the *slab deals*, or outsides, have been cut away, the log that remains of this timber is generally 14 to 15 inches square.

Seasoning Timber.

When freshly hewn timber is allowed to dry it shrinks considerably, the effect of the shrinkage depending on the shape of the timber. Thus, a sector, A (fig. 4), becomes narrower, a beam of square section, B, becomes rhomboidal, and a pole of circular section, C, becomes elliptical. If a tree (fig. 2) is sawn lengthwise into nine pieces, the outer layers, being youngest and

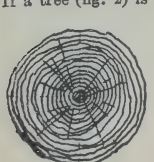


FIG. 2.

FIG. 3.

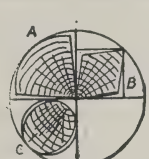


FIG. 4.

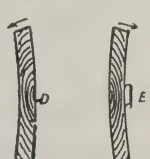


FIG. 5.

more full of sap, will shrink the most, so that the planks, instead of being straight, will be curved (fig. 3), the centre plank contracting chiefly at its ends, and the side planks as at D (fig. 5). Even when a plank has been thoroughly dried in the open air, if a thick shaving is taken off it the newly exposed surface will shrink, drawing the wood inwards as in E. In consequence of its shrinkage, freshly felled timber will warp badly in use, unless it has been *seasoned*—i.e. treated in such a manner that all the shrinkage takes place before the wood is used. For this purpose the timber is sawn into planks and 'seasoned' by (*a*) exposing it to the weather, (*b*) immersing it in water for some weeks and then allowing it to dry, (*c*) drying it by artificial heat in suitable chambers known as 'kilns.' The first method is the most effective, but is crude and very slow, as the drying is frequently suspended for long periods by adverse weather.

Prolonged immersion in water is unnecessarily severe for most purposes, and adds greatly to the time subsequently required for drying. Moreover, if the timber is immersed too long, it subsequently becomes brashy and useless.

Seasoning by artificial heat is, for most purposes, quite as good as natural seasoning; if the kilns are well designed and properly managed they reproduce the most favourable atmospheric conditions, eliminate the adverse ones, and will do in a week or so what would otherwise require several months. For woodwork which is to be exposed to the weather, natural seasoning is preferable, though it may be assisted by a preliminary artificial seasoning.

It is not possible by any rapid tests to compare the liability of woods which have been seasoned by different methods, but the following results, obtained by a Government Department, are interesting:

Nature of Timber.	Natural Seasoning.		Artificial Seasoning.	
	Breaking Load. lbs./sq. in.	Elastic Limit. lbs./sq. in.	Breaking Load. lbs./sq. in.	Elastic Limit. lbs./sq. in.
American Walnut	15,300	7,030	15,000	7,000
Mahogany	11,700	6,230	12,700	6,880
English Ash	10,900	5,500	14,000	7,000
Spruce	8,500	4,500	9,000	4,000

TIME REQUIRED FOR SEASONING TIMBER.

Oak.—From 4 to 8 square inches, about 6 months; under 12 square inches, 10 months; under 16 square inches, 14 months; under 20 square inches, 18 months; under 24 square inches, 22 months; above 24 square inches, 26 months.

Fir.—Half the foregoing time in each case.

DECREASE IN DIMENSIONS OF TIMBER BY SEASONING.

Wood.	Shrinkage	Wood.	Shrinkage.
	Inches.		Inches.
Canada cedar	14 to 13¼	Southern pitch pine	18⅜ to 18½
Elm	11 „ 10⅜	Spruce	8½ „ 8⅜
English oak	12 „ 11⅝	American white pine	12 „ 11⅞
Northern pitch pine	10×10 „ 9¾×9¾	Northern yellow pine	18 „ 17⅝

COMPARATIVE WEIGHT OF GREEN AND SEASONED TIMBER.

Wood.	Weight of Cubic Foot in Lbs.		Wood.	Weight of Cubic Foot in Lbs.	
	Green.	Seasoned.		Green.	Seasoned.
Ash	43⅜–63½	32½–59	Oak	58½–63	43–62½*
Beech	56¼–63½	41–52	Riga fir	23⅜–63¾	—

(*Nördlinger.*)

PERCENTAGE OF LOSS OF WEIGHT IN SEASONING.

Red pine	12 to 25	Larch	16 to 25
American yellow pine	18 to 27	Oak, British	16 to 30
Elm	About 40		
Mahogany	16 to 25		

See also 'FOREST PRODUCTS RESEARCH' (Department of Scientific and Industrial Research); Project 1—Mechanical and Physical Properties of Timbers.†

Progress Report 1—Tests of Some Home-Grown Timbers in their Green Condition.‡

* The similarity in the maximum figures for green weight of all four species, and the near approximation to the weight of a cubic foot of water (62½ lbs.) should be noted.

† Price 2*s.* net. Postage extra.

‡ Price 9*d.* net. Postage extra; London, H.M. Stationery Office, Adastral House, Kingsway, W.C. 2.

Artificial Seasoning of Timber.

(Contributed by G. Stanley Walpole, D.Sc., F.I.C.)

In all systems of artificial seasoning the timber is stacked in a kiln or chamber in such a manner that practically the whole surface of the timber is exposed to currents of air warmed to the temperature and supplied with such an amount of steam, determined by experience, which ensure that evaporation of moisture takes place at a 'safe' rate, so that the timber is not deteriorated in any way.

Timber dryers can be divided into two classes: (*a*) *Progressive* or *Tunnel Dryers*, in which the timber is moved through zones of air each differing in temperature and humidity from the neighbouring one; (*b*) *Compartment Dryers*, in which the timber remains stationary, the air conditions being varied.

Progressive dryers may be divided into two sections: first those employing the use of fan air circulation (Sturtevant type), and secondly those in which the air is circulated automatically without any power cost (Erith type). In compartment dryers, fans are almost invariably used, as thermal circulation would not be satisfactory for that type.

Where the duty of the seasoning plant is definitely known in advance, and where such duty consists of the drying of quantities of timber of more or less uniform grade and thickness, the progressive type dryer is by far the most economical. Where it is impossible to predict the exact duty of the plant, as where many different kinds and thicknesses of timber have to be dealt with, the compartment system may be preferred, as when properly installed it is very 'elastic' in nature. If there is a possibility that the nature of the trade will alter, the compartment system may be preferable to the progressive type, by reason of the fact that, should different classes of timber have to be dealt with the fans and heater coils originally installed may require little, if any, modification.

Progressive or Tunnel Dryers.

In this type of dryer the timber is piled on trucks which run on suitable rails right through a tunnel constructed of brickwork or concrete or, in some cases, of timber, with doors at both ends. The tunnel is first completely filled with trucks stacked with timber; afterwards the introduction of a truck of green timber displaces a similar truck of seasoned material at the opposite end. The movement of the trucks is thus said to be 'progressive.' In the 'fan-less' progressive automatic dryers (Erith type) there are a number of steam pipe coils arranged below the rails. Where the rail level of the kiln must be the same as the general ground level, the space for the coils must be excavated, but sometimes the rail level of the kiln may be a few feet above ground level in order to give the necessary space for the pipes under the tracks.

The steam pipe coils are arranged in such a manner that circulation of air is obtained from end to end of the kiln, and the majority of the heating coils are grouped together near the 'dry' end of the plant. The air in passing over the timber absorbs moisture, and the plant is so arranged that the trucks of green timber meet first air having a low temperature and a high humidity. As each truck passes through the dryer it meets air of gradually increasing temperature and decreasing humidity.

Another type of progressive dryer (Sturtevant) depends for air circulation upon centrifugal fans, the air circulation and temperature being under *separate* control. The arrangement of trucks and rails in this type is generally similar to those for the 'fan-less' system, but in this case the fans and heater are fixed entirely outside the dryer in a small room or shed adjoining. As previously indicated, both systems of tunnel or progressive dryers are suitable for large quantities of uniform timber such as floorboards, match boarding, etc. The application of the progressive systems is, however, somewhat limited, and for small installations, and in cases where the ground space is insufficient for progressive working, compartment dryers have to be adopted.

Compartment Dryers.

In this system a number of compartments or kilns, generally of brick or concrete, are filled with timber either by hand piling or by truck piling. It should be noted, however, that if truck piling is adopted it is only for convenience in handling the material, and is not essential to the working of the dryer as such. The number of compartments may vary from one to (say) six, all being supplied by one central heating plant. For a larger number of compartments it is generally desirable to divide the same into groups, each dealt with by a separate heating plant.

Air is introduced through a duct in each compartment along the side walls near the floor, the air passing over and under the piled timber (which is arranged with its length in the direction of the length of the chamber), and then out through a corresponding duct on the opposite side of the kiln. The air circulation is made positive by means of centrifugal fans arranged in a small chamber either at one side or on top of the dryer, and means are provided to enable each compartment to be under separate and independent control both as to temperature and humidity. Thus it is quite possible to deal with different classes of timber in adjacent compartments simultaneously. Further, the stage of drying reached in each of the different compartments may be quite different, and the benefits of this must be at once apparent to those interested.

A very important adjunct to the compartment system is the *Guide Chart System of Control* (Sturtevant patent), the object of which is to reproduce in each kiln, during the whole period of seasoning, conditions previously determined by investigation and experience. The working of the system is briefly as follows: In each compartment, or kiln, is a self-recording hygrometer comprising wet and dry bulb thermometers actuating levers terminating in pen points which rest against a paper chart on a revolving drum rotated by clockwork. Upon this chart are marked guide lines indicating both the dry bulb and wet bulb temperatures which should be given from day to day in that particular compartment. The dampers, etc., in connection with each compartment are operated by hand at intervals, so that the conditions in the compartment correspond to those given by the guide lines on the chart. Thus not only do the guide lines give the conditions to be aimed at, but the pen pointers make a permanent record showing how nearly the actual conditions have approached the ideal condition. In this way the Guide Chart System acts also as a check on the attention of the kiln operator.

A large number of guide charts have already been prepared by the Sturtevant Engineering Co. Ltd., to suit various classes and thicknesses of timber, both hard and soft, and once the best conditions for a particular grade of timber have been established, similar conditions for subsequent batches can be reproduced as required.

See also 'The Principles of Kiln-Seasoning of Timber,' Department of Scientific and Industrial Research, Forest Products Research, Special Report No. 2, H.M. Stationery Office, Adastral House, Kingsway, W.C. 2, price 9d. net.

Storage and Preservation of Timber.

Seasoned timber should be stored under shelter and kept dry. The bottom timbers should be 2 ft. above the ground, and each overlying timber should have 1 in. space between it and the timber below. A free, gentle circulation of air should play around it, so as to keep it well ventilated. Some timbers, such as oak, beech, elm, and others, will last for a very long time if kept constantly under water; oak piles have been known, under these conditions, to last hundreds of years. Timber exposed to alternate moisture and dryness lasts a very short time.

Slaked lime hastens decay of timber, which should therefore be protected from contact with mortar.

Timber is destroyed by either wet- or dry-rot. Both wet- and dry-rot are due to fungoid growths, and in both cases gases, principally carbon dioxide and hydrogen, are given off. Dry-rot is usually caused by want of ventilation; damp and moderate warmth encourage it. The sap-wood is usually attacked first. It often occurs where ends of beams are built into walls, where wet and unseasoned timber is painted, where a damp floor is covered with oil-cloth or other impervious covering, and in warm cellars. The seeds of the dry-rot fungi are carried about readily, and once it breaks out in one part of a building the timber in other parts may become infected.

Painting or tarring will preserve timber, provided the latter is thoroughly dry and well seasoned; otherwise it only confines the sap and moisture and causes rot.

Timber is also preserved by creosoting prior to use. Tredgold doubted the utility of painting timber which was not exposed to the weather.

See also 'Forest Products Research' (Department of Scientific and Industrial Research); Bulletin No. 1—Dry Rot in Wood.* This Bulletin gives valuable details in regard to 'Precautions to be taken in the use of Timber in New Buildings to Prevent Outbreaks of Dry-Rot.'

Storing Logs.

Logs stored on skidways or left in the woods during the summer months may be damaged in a number of ways, principally through sap-staining, insect attack, decay, and checking.

Storing under water will prevent 'blue stain,' checking, insect attack, and decay, except that logs in sea-water, where marine borers are active, would be subject to attack. Wood of any species completely submerged in water will resist decay indefinitely. Alternate wetting and drying favour decay. Storing on skids in such a way that the air can circulate freely around each log will prevent the accumulation of moisture and thus retard decay. Such storing, however, is liable to increase checking, and, unless the bark is removed, will have little effect in preventing insect attack. The skids should be situated where there is good air circulation, and they should be raised off the ground. Weeds and brush should be cut down. Peeling the bark completely from the logs will do much to eliminate insect attack and retard decay by removing the protection required by many insects and by allowing the logs to dry more rapidly. It will favour checking, however. Painting the ends of the logs will very materially retard the loss of moisture and thus retard end checking. If the logs are peeled and properly piled on skids, painting should not increase the danger from decay or sap stain. A yellow ochre or barn paint will do fairly well for the purpose. Painting the peeled surfaces with coal-tar creosote will be useful in preventing sap decay, and if applied soon enough may be effective in retarding sap stain. Any grade of creosote in common use for wood preservation is suitable and expensive oils are unnecessary. (*'Bulletin of the Hydro-Electric Commission of Ontario.'*)

* Price 1s. 6d. net; London, H.M. Stationery Office, Adastral House, Kingsway, W.C. 2.

Preservative Processes.

Charring timber.—The ends of posts sunk into the ground are often charred in order to prevent rot, but recent experiments appear to show that the charring is ineffective and useless.

Vulcanising.—The wood is heated in a closed cylinder from 8 to 12 hours under a pressure of 150 to 200 lbs. per square inch and a temperature of from 300° to 500° F.

Drying.—Compressed, superheated and dried air is circulated in order to remove excessive moisture: it is claimed that the sap undergoes a chemical change and solidifies in the pores as an antiseptic matter in cooling.

Sugared timber.—A paper was read before the British Association at their meeting in 1903 on the preservation, seasoning, and strengthening of timber by means of boiling it in a saccharine solution and afterwards evaporating the moisture in a drying chamber. This process filled the interstices of the wood with a substance akin to the wood itself, rendering it more solid, firm and compact. The dried sap of the sugar-cane acted in timber much as mortar did in a stone wall, binding the otherwise loose materials strongly together.

Different methods of preserving timber by impregnation with solutions of metallic salts, such as bichloride of mercury (*Kyanising*), sulphate of copper (*Boucherising*) and chloride of zinc (*Burnetising*), have met with little success.

In the 'Ferell' process, the principal constituents of the injected solution are sulphate and chloride of aluminium.

Creosoting.

(Contributed by Robert S. Vesey, Member of the Society of Chemical Industry.)

Modern Methods of Creosoting.

Cresoting is recognised as the most efficient method of preserving timber, and all Railway Companies in this country have their sleepers and crossing timbers treated in this way. The process is also generally used for the preservation of telegraph poles, piles, posts, and the like. Softwood sleepers untreated seldom last more than three to four years, whereas their average life when creosoted is about fifteen years, and even then, in most cases, the sleepers are only removed from the track owing to their having become chair-worn, and thereafter the bulk have many years life in them as siding sleepers or fence posts.

The 'Full Cell' Process of creosoting is generally preferred to the 'Empty Cell' or 'Rüping' Process, which latter principally aims at penetration and the withdrawing of two-thirds of the creosote at the end of the operation. Although only about one-third of the creosote used for the 'Full Cell' Process is left in the timber in the case of the Rüping Process, the cost of this treatment is nearly two-thirds the cost of the 'Full Cell,' and in this country it has seldom been made use of.

Construction of Creosoting Plants.

Creosoting is carried out in cylinders or retorts 75 to 105 ft. in length by 7 ft. in diameter, the average plant consisting of three, which are erected upon underground covered storage tanks of $\frac{3}{8}$-in. steel, and usually 7 ft. deep, two of these tanks being used in conjunction with each cylinder. The cylinders are of $\frac{7}{8}$-in. steel plate tested to stand at least 400 lbs. pressure per sq. in., and are each fitted at the bottom with two suction pipes for lifting and dropping the creosote to and from the cylinders; at the top of each there is a vacuum valve, pressure valve, overflow pipe, and safety valve. They are fitted with a railway track of about 2 ft. 6 ins. gauge, to take the steel bogies loaded with the timber which is to be creosoted, and 'holding-down' bars are fitted throughout the entire length to prevent the bogies floating while the process is being carried out; each load is either chained or fixed to the bogie by means of arms. Modern plants are fitted with doors at each end of the cylinders to allow the same to be charged or withdrawn from either end, the doors being fixed to the cylinder and swinging on a projecting arm fixed to the top; they can be easily opened and closed by two men. These doors are sealed with about twenty screws 12 ins. long by $2\frac{1}{2}$ ins. diameter, and are tightened by means of a large hand spanner.

The storage tanks cover the whole area occupied by the cylinders, and generally project about 2 ft. to 3 ft. at the sides, and about the same distance at the ends. This additional length at the ends is left open for a space of about 3 sq. ft. and is necessary for allowing the drainage from the cylinders, at the end of the process, to enter the tanks. A portable bridge of rails connects the creosoting cylinders with the bogie roads in the works, and this is of course removed when the cylinder doors are being opened or closed. The cylinders are usually covered with asbestos to enable the heat of the creosote to be maintained, although in impregnating the timber there is always a reduction in the temperature of the oil; in some cases steam coils are fitted in the cylinders. The storage tanks should always be fitted with coils to enable the creosote to be kept at a temperature of 140° to 150° F., although in America very much higher temperatures are used. These tanks are used for the filling of the cylinders, and to obtain additional creosote or impregnation a smaller underground tank is installed, which is usually built in concrete and is placed in the boiler-house; it is known as the 'Pressing' tank, and on it are fitted the vacuum and pressure pumps.

Working Process.

About 50 cu. ft. of the timber to be creosoted is selected as an average to test the amount of creosote absorbed, and this is weighed before and after creosoting. Immediately the doors have been screwed down, the vacuum pump is started and the air extracted from the cylinder, about 25 ins. of mercury being obtained. This vacuum being maintained, the suction cocks are opened and the cylinder is nearly filled from the two storage tanks immediately underneath; but to complete the filling, the suction cocks are closed, and the pressure pump used for about two minutes to pump in sufficient creosote, the additional oil being taken from the pressing tank. The operator then records from the gauge the number of gallons of oil in the pressing tank, and continues working the pressure pump until the required number of gallons to impregnate the timber have been injected, the amount varying according to the specification. In the case of well-seasoned Baltic Red Fir or Scots Fir, 8 to 10 lbs. per cu. ft. may be injected with 60 to 80 lbs. pressure per sq. in. As soon as sufficient oil has been injected the pressure pump is stopped, the suction valves opened, and the creosote dropped back into the storage tanks. The cylinder doors are then opened and superfluous oil allowed to drain from the timber, and although circumstances will not always permit, no charge should be withdrawn for at least two hours.

Time taken for Carrying out Process.

However well stowed the bogies might be with the timber to be creosoted, the cylinders are only from 50 to 55 per cent. filled, so that a large quantity of creosote is required in connection with each charge. The following is about the average time taken to carry out the creosoting of seasoned Baltic Red Fir with a modern plant:—

Obtaining 25 in. vacuum and lifting oil from storage tanks to cylinder, and completely filling by means of pressure pump—8 to 10 minutes.

Injecting 10 lbs. of cresote per cu. ft. with pressure pump at 60 to 80 lbs. pressure per sq. in. (not maintained, but which is sufficient to inject this quantity of oil)—6 to 10 minutes.

Dropping creosote to storage tanks at end of process—15 minutes.

During the years 1919 and 1920 when supplies of Baltic Red Fir were very short, considerable quantities of Oregon Pine and Pitch Pine sleepers were imported, and very much higher pressures were required for the efficient creosoting of these, and a much longer time was taken to carry out the process. Generally speaking, however, Baltic Red Fir is much preferred to Oregon and Pitch Pine, now that large supplies of the first named are available. Seasoned Oregon Pine will seldom take more than 2 to 4 lbs. of creosote per cu. ft. at 200 lbs. pressure maintained for 4 hours with the creosote at 150° F., although if sappy pieces are selected it is possible to inject up to 10 lbs. per cu. ft., but these cases are very exceptional in this country. With the various classes of Pitch Pine met with in England ('Longleaf,' 'Shortleaf,' and 'Loblolly') the absorption varies very considerably, and seasoned hard grown 'Longleaf' Pitch Pine will seldom absorb more than an average of 5 lbs. per cu. ft. with a pressure of 200 lbs. per sq. in. maintained for 4 hours. In the case of 'Shortleaf' the absorption would average 8 to 12 lbs. per cu. ft. with the same pressure and time, but 'Loblolly,' which is very sappy and unsuitable for use where strength is required, will absorb up to 20 lbs. of creosote per cu. ft. Seasoned Portuguese Pine will also absorb 10 to 14 lbs. of creosote with low pressure, but this timber is seldom met with in this country. Seasoned Larchwood (Northern Europe and home-grown) will on the average only absorb 5 to 7 lbs. per cu. ft. with high pressure maintained, as in the case of Oregon Pine. Although other species of timber are creosoted in this country, at least 95 per cent. are of Baltic Red Fir, Scots Fir, Oregon, and Pitch Pine, the majority being Baltic Red Fir, which is nearly always preferred for sleepers.

Specification of Creosote Oil.

There has been a good deal of controversy over the question of the best grade of creosote for use in the preservation of timber, some authorities favouring the naphthalene bases, and others placing more importance upon the tar acids. The following is the *average* specification stipulated by those who have creosoting to carry out in this country:—

1. The creosote used, to be obtained from the distillation of coal-tar, free from all impurities and to be fluid at 75° F.

2. Specific gravity to be between 1·040 and 1·080 (water at 60° F. being 1·000).

3. To yield not more than 75 per cent. upon being distilled to 600° F.

4. Tar acids, not less than 8 per cent.

5. Water, 3 per cent. maximum.

6. In some cases the percentage of naphthalene is not mentioned; in others a minimum of 10 per cent., and a maximum of 30 per cent., and sometimes the specification states that this is to be kept as low as possible. There is no doubt that in the case of piles which are in contact with water they should be creosoted with oil containing a percentage of naphthalene, as this prevents the exudation of the more fluid parts of the creosote, and also assists in preventing the entrance of the water.

British Standard Specification for Creosote.*

Suitable for the treatment of Railway Sleepers and Telegraph, Telephone, and Hangar Poles.
(No. 144—*August*, 1921.) (*Abstract.*)

1. The material shall consist essentially of a distillate of coal tar, and shall be free from any admixture of petroleum or similar oils.

2. The specific gravity shall not be less than 1·015, and not more than 1·07 at 38° C. (100° F.) when compared with water at the same temperature.

3. The material shall become completely liquid on being slowly warmed to 38° C. (100° F.) with stirring, and on cooling down shall remain completely liquid after standing for two hours at 32° C. (90° F.). If the creosote is required for brush application it shall become completely fluid on being slowly warmed to 38° C. (100° F.) with stirring, and shall, after cooling down, remain completely fluid on standing for two hours at 15° C. (59° F.).

4. The amount of water in the creosote shall not exceed 3 per cent.

5. When 100 c.c. measured at 38° C. (100° F.) of the dry creosote are distilled from a 250 c.c. distillation flask at such a rate that the distillation is complete in about twenty minutes, there shall distil at 760 m.m. pressure,

up to 205° C. (401° F.) not more than 7 c.c.
„ 230° C. (446° F.) „ 40 c.c.
„ 315° C. (599° F.) „ 78 c.c.

the volumes of all fractions being measured at 38° C. (100° F.).

The residue above 315° C. (599° F.) shall be soft and not sticky, and its weight shall be not less than 22 gms.

6. The amount of tar acids shall be not less than 5 per cent. and not more than 16 per cent. by volume.

7. The amount of matter insoluble in benzol (benzene) shall not exceed 0·4 per cent. by weight.

B.S.S. Alternative for Scotch Creosote.

Scotch creosote shall conform to the above specification with the following exceptions:—

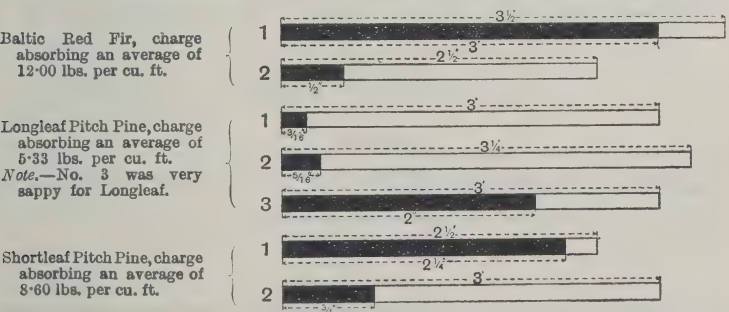
2. The specific gravity shall be not less than 1 at 38° C. (100° F.). In the case of the blast furnace oil the specific gravity may be lower, but shall not be less than 0·940 at 38° C. (100° F.)

5. The distillate at 315° C. (599° F.) shall be not more than 85 c.c. and the residue not less than 15 grms.

6. There shall be no upper limit to the amount of tar acids.

General Notes on Creosoting.

It is erroneous to think that in creosoting under pressure the oil penetrates throughout the timber, as this is not the case, and only occurs when timber is of small dimensions. It may be taken as a general rule in the case of Baltic Red Fir, Pitch Pine, and Oregon, that the timber showing sapwood will readily absorb the creosote, and the reverse is the case with that showing little sapwood. The heart penetration is from $\frac{1}{32}$ in. to $\frac{1}{4}$ in., according to the growth and species. In the case of Baltic Red Fir the impregnation varies considerably, according to the nature of the timber, and in all specifications for creosoting, care should be taken that the words 'average per cu. ft.' are inserted, as to obtain identical impregnation in each piece is impossible. Presuming that an average of 10 lbs. of oil per cu. ft. has been injected into the charge of timber, it does not necessarily mean that the timber which has absorbed 12 lbs. per cu. ft. will last longer than pieces with 8 lbs. per cu. ft., as the latter has a greater percentage of the heart of the tree, which prevents the admission of a large amount of creosote. The oil has done its work as far as possible, and with the smaller impregnation has considerably extended the life of the timber. The following illustrations (which are extremes) show test cores or borings taken from timber of 10 ins. by 5 ins. section showing varying penetrations, the species being creosoted separately:—



* By permission of the British Engineering Standards Association.

The cores are two-thirds the actual size, and the impregnation shown in black.

A Cresoting Plant of the foregoing description would be capable of annually treating over two million cu. ft. of the softwood species referred to, and one million cu. ft. of the hardwood, provided the works were of about 15 acres extent with 5 or 6 travelling cranes for the transportation of the timber.

See also p. 539, 'Creosote Treatment of Railway Sleepers.'

To ascertain Weight of Timber-work, Timber Flooring.

Multiply breadth in feet by length in feet by the thickness in inches, and by one of the following factors, according to the material: For elm, use 2·92 lbs.; for yellow pine, 3 42 lbs.; for white pine, 2·97 lbs.; for dry oak, 4·95 lbs.

To find the Weight of Timber Beams, Posts, and Joists.

Multiply length in feet by the breadth in inches and the depth in inches, and the product by one of the following factors: For elm, ·243 lbs.; yellow pine, ·285 lbs.; white pine, ·247 lbs.; dry oak, ·404 lbs.

WEIGHT PER THOUSAND (M), FEET BOARD MEASURE.

Wood.	Dry.	Partially Seasoned.	Green.
	Lbs.	Lbs.	Lbs.
Pine and hemlock	2,500	2,750	3,000
Norway and yellow pine . .	3,000	4,000	5,000
Oak and walnut	4,000	5,000	—
Ash and maple	3,500	4,000	—

AVERAGE WEIGHT PER CUBIC FOOT OF WOODS.

Wood.	Weight.	Wood.	Weight.	Wood.	Weight.	Wood.	Weight.	Wood.	Weight.
	Lbs.		Lbs.		Lbs.		Lbs.		Lbs.
Ash	45	Birch	44·5	Larch .	35	Yellow pine	42	White pine	35
Beech	43	Elm	35	Mahogany	50	Oak . .	58	Yew.	50

Wooden Pillars.

APPROXIMATE RULE FOR THE DIMENSIONS OF WOODEN PILLARS.

W = weight in lbs., to be supported by the pillars. L = length of the pillar, in feet. D = diameter of pillar, in inches.

$$D = x \sqrt{L} \sqrt{W}.$$

x = ·80 for oak, ·87 for red deal.

Shingles.

1 bundle of 16-inch shingles will cover 30 sq. ft., and 1 bundle of 18-inch shingles, 33 sq. ft. when laid 5½ inches exposed to the weather.

6 pounds of fourpenny nails will lay 1,000 split-pine shingles.

Clapboards.

1 bundle laid 3½ inches exposed to the weather will cover 26 square feet. To be laid with fivepenny nails.

To find Solidity of Round Timber.

WHEN ALL DIMENSIONS ARE IN FEET.
Length × (¼ mean girth)² = cubic feet.

WHEN LENGTH IN FEET, GIRTH IN INCHES.
Length × (¼mean girth)² ÷ 144 = cubic feet.

WHEN ALL DIMENSIONS ARE IN INCHES.
Length × (¼ mean girth)² ÷ 1728 = cubic feet.

To find Solidity of Square Timber.

WHEN ALL DIMENSIONS ARE IN FEET.
Length × breadth × depth = cubic feet.

WHEN ONE DIMENSION IS IN INCHES.
Length × breadth × depth ÷ 12 = cubic feet.

WHEN TWO DIMENSIONS ARE IN INCHES.
Length × breadth × depth ÷ 144 = cubic feet.

For BOARD MEASURE depth always equals 1 inch.

To FIND SURFACE IN SQUARE FEET, proceed as per rules for solidity of square timber.

Roofs.

LOAD ON ROOFS.

Description of Covering.	Per Square of 100 Superficial Feet.	Minimum Slope.	Description of Covering.	Per Square of 100 Superficial Feet.
	Lbs.	Degrees.		Lbs.
Lead covering weighs .	784	4	Timber framing for	
Zinc	168	4	slated or tiled	
Corrugated iron	336	4	roofs	560 to 672
Slates	840 to 1,008	25½ to 30	Additional load for	
Tiles	896 to 1,680	26½ to 30	pressure of wind .	4,032
Boarding ¾ inch thick .	290	25	Ditto for Hurricanes,	
Boarding 1½ inch thick .	580	25	say	8,960

The angle of Gothic roofs should not exceed 60°.

CORRUGATED-IRON ROOFING.

B. Wire Gauge.	Size of Sheets.	Weight per Square.	Square Feet per Ton.	B. Wire Gauge.	Size of Sheets.	Weight per Square.	Square Feet per Ton.
No.	Feet.	Lbs.		No.	Feet.	Lbs.	
16	6 × 2 to 8 × 3	350	800	22	6 × 2 to 7 × 2½	175	1,550
18	6 × 2 to 8 × 3	258	1,000	24	6 × 2 to 7 × 2½	136	1,880
20	6 × 2 to 8 × 3	204	1,250	26	6 × 2 to 7 × 2½	118	2,170

One-tenth of the weight to be added for lappage. Sheets should overlap about 6 inches, and be double riveted at joints. 3 lbs. of rivets required per square of roofing. Purlins should be 6 feet apart. Curved roofs may be made up to 20 feet span without framing; tie-rods 12 feet apart.

ANGLES OF ROOFS.

Proportion of Rise to Span.	Angle.	Slope.	Proportion of Rise to Span.	Angle.	Slope.
	° ′			° ′	
⅛	18 25	3 to 1	⅝	53 0	¾ to 1
¼	26 35	2 to 1	¾	56 20	⅔ to 1
⅜	33 42	1½ to 1	1	63 30	½ to 1
½	45 0	1 to 1			

(See also page 316).

DIMENSIONS OF PARTS OF WOODEN ROOFS.

Span.	Section.						
	Principal.	Tie Beam.	King Posts.	Queen Posts.	Small Queens.	Straining Beam.	Struts.
Feet.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
20	4 × 4	9 × 4	4 × 4	..	..	..	3 × 3
25	5 × 4	10 × 5	5 × 5	..	..	..	5 × 3
30	6 × 4	11 × 6	6 × 6	..	..	..	6 × 3
35	5 × 4	11 × 4	..	4 × 4	..	7 × 4	4 × 2
45	6 × 5	13 × 6	..	6 × 6	..	7 × 6	5 × 3
50	8 × 6	13 × 8	..	8 × 8	8 × 4	9 × 6	5 × 3
55	8 × 7	14 × 9	..	9 × 8	9 × 4	10 × 6	5½ × 3
60	8 × 8	15 × 10	..	10 × 8	10 × 4	11 × 6	6 × 3

BRICKWORK.

Bricks.

Description.	Size.	Weight.	Weight Per 1,000.
London stocks . . .	8¾ × 4¼ × 2¾ ins.	6·8 lbs.	60¾ cwt.
Red kiln	8¾ × 4¼ × 2¾ „	7·0 „	63 „
Welsh fire	9 × 4½ × 2¾ „	8·0 „	72 „
Paving	9 × 4½ × 1¾ „	5·0 „	45 „
Dutch clinkers . . .	6¼ × 3 × 1½ „	1·5 „	14 „

8¾ × 4¼ × 2¾ bricks, when laid in cement, occupy a space of 9 × 4½ × 3 ins. approximately.

The thickness of walls is regulated by the size of the bricks, and walls are 4½ ins. or half-brick thick, 9 ins. or 1 brick thick, 14 ins. or 1½ bricks thick. The standard thickness of reduced brickwork is 14 ins., or 1½ bricks.

1 rod of brickwork × 272 superficial ft., 1½ bricks thick; = 11½ cubic yards = 306 cub. ft.; = 235 ft. of bricks and 71 of mortar; = 4,300 stock bricks; = 4,500 place bricks; = 5,370 stock bricks laid dry; = 4,900 stock bricks laid dry in wells and circular cesspools; = 15 tons average weight.

1 load of mortar = 1 cubic yard. 1 load of sand = 1 cubic yard. 1 bag of cement = 3 bushels. 1 sack of cement = 5 bushels.

1 rod of brickwork requires about { 1 cubic yard of stone lime, 3½ cubic yards of sand; or 1½ yards of chalk lime, 3 yards of drift; or 36 bushels of cement, 36 bushels of sharp sand.

1 cubic yard of brickwork requires about { 6½ cubic feet of sand. 2½ „ „ lime.

1 builder's cart load of bricks = 500 bricks. 1,000 stock bricks, closely stacked, = 55 ft. cube. 1,000 old bricks, cleaned and loosely stacked, = 70 ft. cube. 1 foot of reduced brickwork = 16 bricks. 1 foot of superficial facing = 7 bricks. 1 foot superficial of gauged arches = 10 bricks. 1 yard superficial in bricknogging, laid flat, = 45 bricks and ¾ cubic foot of mortar. 1 yard superficial in bricknogging, laid on edge, = 30 bricks and ½ cubic foot of mortar.

1 yard of paving = 36 stock bricks laid flat, = 52 bricks on edge, = 82 paving bricks on edge, = 125 Dutch clinkers laid flat, = 140 Dutch clinkers on edge, = 136 Dutch clinkers laid herring-bone.

Bricks absorb one-fifth of their weight of water, but the proportion varies greatly. Engineering bricks should be quite impervious to water.

A bricklayer's hod will hold 20 bricks, ⅔ cubic foot of mortar, or nearly ½ bushel.

1 rod of brickwork requires 130 gallons of water to slack the lime and mix the mortar.

In a day of 10 hours a bricklayer can lay about 1,500 to 1,600 bricks where the joints are left rough, about 1,000 when both faces are worked fair, or 500 when carefully pointed and faced with picked bricks.

For ordinary purposes, assume roughly that a bricklayer and labourer can, if they choose, turn out 2 cubic yards per diem, containing 800 bricks.

Bonds for Brickwork.

(Revised by E. A. W. Phillips, M.Inst.C.E., M.C.I.)

ENGLISH THIN-WALL BOND (fig. 6) is the simplest for all ordinary work where the walls do not exceed three brick-lengths in thickness. The heading and stretching courses always alternate in this class of bond. No stretchers occur except those seen on faces of walls. No bricks in the same course should break joint with each other. All vertical joints in any course should be covered (by one-quarter brick laps at least) by the courses above and below, and this whether offsets occur or not, or other differences in the thicknesses of the walls.

FLEMISH BOND (fig. 7) shows headers and stretchers alternately in each course on the faces of walls. Is quite as strong as *English Thin-wall Bond*, since the quarter laps are in the

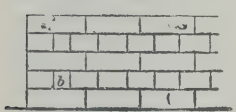


FIG. 6. English Bond.

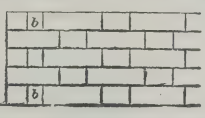


FIG. 7. Flemish Bond.

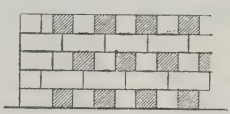


FIG. 8. English Thick-wall Bond.

same manner equally well maintained. *Flemish Bond* gives a better appearance for 9-inch walls, as a smoother face can be shown on both sides. In *double Flemish Bond*, both back and front of walls show Flemish Bond; in *single Flemish Bond*, one face shows Flemish, the other English bond, as above.

ENGLISH THICK-WALL BOND (fig. 8). In this bond, every brick in a stretcher course should be a stretcher, and every brick in a header course should be a header. The bricks not only break joint, by half their lengths, with the bricks in the same description of course (i.e., stretcher or header) next above or below them, but also break joint, by half their lengths, in the same course. This is the strongest form of bond known, and should invariably be used in heavy walling, of $3\frac{1}{2}$ bricks thickness and over. The quoin difficulty is best overcome by having a few specially moulded bricks, one brick-length in length, and $\frac{3}{4}$ of a brick-length in width, and of the usual thickness. The half-bricks on faces in fig. 8 are shaded.

GARDEN-WALL BOND is the *English Thick-wall Bond* adapted to thin one-brick walling, several courses of stretchers occurring to one course of headers. The same thing may be done with *chimneys*, where the walls are 9 ins. thick.

DIAGONAL BOND and HERRING BONE BOND are only useful for paving. *English Thick-wall Bond* is stronger than either.

HEADER BOND consists entirely of headers, and is used for sharp curved or quick-sweep walls.

STRETCHER BOND, entirely of stretchers, is used in $\frac{1}{2}$-brick walls of all kinds.

Other bonds, Roman bond, and the Sussex bonds, are described in 'Indian Engineering.' See also papers in the Minutes of Proceedings of Civil Engineers, vol. clxxi.

CLOSERS, *b* (figs. 6, 7), are necessary in every alternate course to close the work on to any vertical line, or to start the bond from it; they run through the thickness of the wall.

QUEEN CLOSERS, as *b*, are $\frac{1}{2}$-headers formed by cutting a brick or a half-brick, called a *bat*, longitudinally in half; $\frac{3}{4}$-stretchers, or $\frac{3}{4}$-*bats*, can be used as closers, but not so economically.

KING CLOSERS are bricks cut to a splay, so as to obtain as strong a bond as possible in closing on recessed openings, as for doors and windows.

HOOP-IRON BOND, for walls, is generally formed of hoop iron, from 1 in. to $1\frac{1}{2}$ in. wide and 18 B.W.G. ($\frac{1}{20}$ in.) to 16 B.W.G. ($\frac{1}{16}$ in.) thick; it should be clean bright and bedded in cement, for if allowed to rust it swells and injures the wall; it holds better if the edges are jagged. Useful in erecting works on bad foundations, but must not be relied on to give permanent strength, unless laid as carefully as reinforcement.

WALLS.

General Rules for Building Walls.

To prevent unequal settlement the whole building should be run up together, no part rising more than about 3 ft. above the rest. Work, unless done in cement, should take its bearings gradually; at least a fortnight should be allowed for completing each storey of a high building. New work should never be bonded to old, unless built in cement. The new work should be built into a chase cut in the old work. Bricks and stones should be clean, to allow the mortar to adhere to them, and well wetted, to prevent them absorbing moisture from the mortar. Building should not be carried on in frost, but rain is good. If obliged to be carried on in frost, cement should be used, and it should be covered up with straw or boards every night, for cement which freezes before it has set is worthless.

Terra-cotta air bricks should be built into the wall, below the level of the ground-floor, for ventilation. Holes should be left in the sleeper walls.

Hollow Walls

keep houses dry, and tend to maintain a more equable temperature within them. They may be of two types: capacity walls and walls built of hollow blocks. A *cavity wall* consists of two thicknesses of brickwork separated by an air-space of 2½ ins., sometimes 4½ ins. Exterior wall, 4½ ins. thick; interior wall, of such thickness as will give the whole the necessary stability (9 ins. for two-storeyed building, 4½ ins. for ordinary one-storeyed). The two thicknesses should be tied together by bonding-ties of brick or slate, or wrought-iron ties 1 in. by $\frac{3}{16}$ in. (fig. 9), at horizontal distances of 3 ft., and alternately at vertical intervals of three courses; this requires five ties to the superficial yard. The half-brick thickness is sometimes put on the inside, in which case the weight of roof should fall on the outer thickness of wall. The air-space should be carried round all angles and up to the sides of door- and window-frames, commencing at damp-proof course, which will form its bottom, and ending in a covering course of brickwork which closes the top, so that the air-space has no communication with open air.

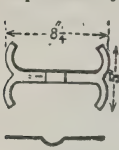


FIG. 9.

Walls of *hollow blocks* (fig. 10) are less liable to accommodate vermin and are less risky as regards fire, as the hollow portions can be divided into a series of independent cells. The blocks may be of concrete or burned clay, the latter being usually preferred, not only on account of their porosity, but also on account of their possessing the advantageous properties of bricks, whilst being much cheaper to erect, as they commonly only measure 12 or 18 ins. by 12 ins. by 9 ins., and so are equivalent to 10 or 16 common bricks. Hollow blocks are usually rectangular tubes, which must be laid in such a manner as to provide a seal where required. Various patents have been granted for making blocks with a sealed hollow cell, but though ingenious they are not much used. For further information respecting hollow blocks see page 531.

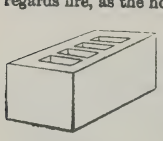


FIG. 10.

Hollow blocks have ample strength for all ordinary requirements or brickwork, especially if laid in a 1 : 6 cement mortar. Reinforcement may be used if desired, but otherwise the hollows or cores are left 'empty.' The blocks are usually arranged with the hollow or tubular portion vertical, the joints being broken as with masonry and common brickwork. Care should be taken to arrange the detailed dimensions of the buildings so that the blocks need not be cut.

Although the hollow portion of each block may be completely seated, if desired, it is usually better to allow a small opening through which a sufficient current of air may pass, so as to secure efficient removal of moisture and to allow the air, when heated, to expand without being under pressure.

The Damp Course

is placed in a wall, 6 ins. or 9 ins. above the ground line, in order to prevent moisture from the foundations rising in the walls by capillary attraction. The following are the most satisfactory forms of damp course:—

(i) Two courses of impervious bricks (preferably 'Staffordshire blues') laid flat in a mortar composed of 1 part of Portland cement, 5 parts of sand and a waterproofing agent such as 'Pudlo' (page 496). Ordinary lime mortar is useless, as it is too porous.

(ii) One or two courses of slate or thin impervious stone laid in the cement mortar mentioned in (i).

(iii) A layer of waterproofed cement mortar (see (i)) ½ in. to 1 in. thick.

(iv) A mixture of gravel or fine grit and asphalt or tar (preferably heated together) and laid in a thickness of ⅝ in. to 1 in.

(v) An elastic waterproof material such as bitumen (*George M. Callender & Co. Ltd.*), in roll form, has distinct advantages in cases where settlement in the foundations would break any rigid material, such as slates or mastic asphalt, the material being sufficiently pliable to conform to such settlement. Bitumen damp course is easily handled, even by unskilled workmen, as the material simply requires to be unrolled on to the foundations, a small lap joint being made at the end of each length. It is not necessary to seal the joint, as the weight of the next course of bricks is quite sufficient to keep the damp course in position.

(vi) Hollow tubes of stoneware or other impervious material, set vertically and laid in waterproofed cement mortar as mentioned in (i).

The essential feature of a damp course is that it must be wholly impervious to water, as otherwise it is useless. This important fact is often overlooked, as in the case where good, impervious bricks are laid in a porous lime mortar or in a porous cement-mortar.

BRICK NOGGINGS

are thin walls or partitions, usually 4½ ins. thick, constructed with vertical posts framed into top and bottom plates, filled in with brickwork in mortar, either flat or on edge. They require—

Per yard super 45 stock bricks laid flat, and ¾ cubic. ft. mortar.
„ 30 „ on edge, and ½ „ „

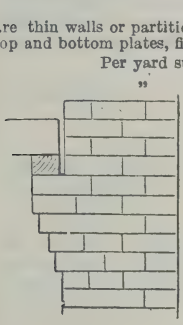


FIG. 11.

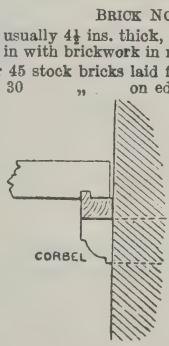


FIG. 12.

CORBELLING

(Figs. 11. and 12) keeps joists clear of masonry, and prevents walls from being weakened by the insertion of timber; in brickwork the projections of the courses should not be more than a quarter brick. When great strength is required, they may project only one-eighth brick. Stone corbels should be of York or equally hard stone, about 4 feet from centre to centre, according to depth of wall plate.

COPING.

If of brick, should be the hardest and least porous, set in cement. In stone, should be of as long stones as possible. Fig. 13 shows brick-on-edge coping. The course A may be two courses of

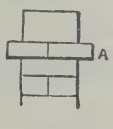


FIG. 13.

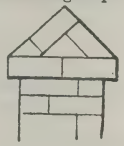


FIG. 14.

FIG. 15.

tiles laid in cement, breaking-joint, called *tile-creasing*. Fig. 14 shows another form; but moulded brick copings are preferable, being made of various patterns to suit walls of different thicknesses. Fig. 15 is a stone saddle-back coping.

Thickness of Walls.

In walls not more than 30 feet high top storey may be 8½ inches thick.

Thickness of higher walls at top and 16 feet below = 13 inches.

From 16 feet below top to base the wall not to be less than the space contained in the two straight lines drawn from each side of the walls, at the base, to each side of the wall 16 feet below the top.

The thickness to be in no case less than one-fourteenth of the height of the storey; but any additional thickness may be confined to piers whose collective width equals one-fourth length of wall. Footings to be double the width of the base, diminishing in regular offsets half the width of their height.

Cross-walls to be two-thirds of the thickness of the external or party walls, but never less than 8½ inches.

THICKNESS OF WALLS OF BUILDINGS (GWILT'S RULE).

H = height of wall; L = length between party walls; D = diagonal of face of wall = $\sqrt{H^2+L^2}$; T = thickness of wall; all in feet.

Then

$$T = \frac{HL}{nD},$$

where $n = 22$ for dwelling-houses of brick, 20 for warehouses of brick, 18 for public buildings of brick.

When the height of wall is less than half its length, then n may be increased to 27, 23, and 20 respectively.

The dimensions in the following Tables may be safely adopted.

THICKNESS OF BRICK EXTERNAL WALLS FOR DIFFERENT SORTS OF ONE-STOREYED BUILDINGS.

Description.	Main Wall.	Piers 2 ft. 3 in. Wide, 10 Feet from Centre to Centre.
Single-storeyed buildings up to 10 feet high and 30 feet wide, with cross-walls every 14 feet	9 inches	
Ditto, with principals, but without cross-walls, any length	9 „	14 inches
Ditto, over 10 feet and not exceeding 20 feet high, with principals, but not cross-walls, any length, width up to 30 feet	14 „	18 „

NOTE.—Where it is not considered advisable to have piers, the greater thickness, *i.e.*, the thickness of the piers, should be adopted.

THICKNESS OF BRICK EXTERNAL WALLS FOR DIFFERENT SORTS OF TWO-STOREYED BUILDINGS.

Description.	Lower Storey.	Upper Storey.
Two-storeyed buildings with cross-walls every 14 feet up to 30-feet span	14 inches	9 inches
Ditto, 30-feet span, with principals, but not cross-walls .	18 „	14 „

Stonework should be, as a rule, one-third thicker than brickwork.

For inhabited buildings the walls are better built hollow.

When concrete (1 Portland cement, 6 ballast) is used, the wall is usually about one-third thicker than brickwork.

Rules for the Walls of Buildings not Public and not of the Warehouse Class.

(*London Building Act*, 57 & 58 *Vict.* 1894.)

According to this Act the external and party walls of dwelling-houses must be made throughout the different storeys of the thickness shown in the following Table, arranged according to the heights and lengths of the walls, and arranged for walls up to 120 feet in height, and supposed to be built of bricks not less than 8½ inches long or of stone or other blocks of hard and incombustible substance, the beds or courses being horizontal, the heights of the storeys being subject to the conditions given after the Table.

TABLE I.

1. Height up to 25 Feet.	Length up to 30 Feet. Not more than two storeys.	Length unlimited. More than two storeys.	
	From base to top of wall, 8½ ins.	Wall below topmost storey, 13 ins. Topmost storey, 8½ ins. Remainder 8½ ins.	
2. Height up to 40 feet.	Length up to 35 Feet.	Length unlimited.	
	Wall below two topmost storeys, 13 ins. Remainder, 8½ ins.	One storey, 17½ ins. Rest of wall below topmost storey, 13 ins. Remainder, 8½ ins.	
3. Height up to 50 Feet.	Length up to 30 Feet.	Length up to 45 Feet.	Length unlimited.
	One storey, 17½ ins. Rest of wall below topmost storey, 13 ins. Remainder, 8½ ins.	Two storeys, 17½ ins. Remainder, 13 ins.	One storey, 21½ ins. One storey, 17½ ins. Remainder, 13 ins.

TABLE I. (*continued*).

4. Height up to 60 Feet.	Length up to 45 Feet. Two storeys, 17½ ins. Remainder, 13 ins.	Length unlimited. One storey, 21½ ins. Two storeys, 17½ ins. Remainder, 13 ins.
5. Height up to 70 Feet.	Length up to 45 Feet. One storey, 21½ ins. Two storeys, 17½ ins. Remainder, 13 ins.	Length unlimited. Each storey below uppermost two storeys to be increased 4½ ins. beyond thicknesses for length up to 45 feet.
6. Height up to 80 Feet.	Length up to 45 Feet. One storey, 21½ ins. Three storeys, 17½ ins. Remainder, 13 ins.	Length unlimited. Each storey below uppermost two storeys to be increased 4½ ins. beyond thicknesses for length up to 45 feet.
7. Height up to 90 Feet.	Length up to 45 Feet. One storey, 26 ins. One storey, 21½ ins. Three storeys, 17½ ins. Remainder, 13 ins.	Length unlimited. Each storey below uppermost two storeys to be increased 4½ ins. beyond thicknesses for length up to 45 feet.
8. Height up to 100 Feet.	Length up to 45 Feet. One storey, 26 ins. Two storeys, 21½ ins. Three storeys, 17½ ins. Remainder, 13 ins.	Length unlimited. Each storey below uppermost two storeys to be increased 4½ ins. beyond thicknesses for length up to 45 feet.
9. Height up to 120 Feet.	Length up to 45 Feet. One storey, 30 ins. Two storeys, 26 ins. Two storeys, 21½ ins. Three storeys, 17½ ins. Remainder, 13 ins.	Length unlimited. Each storey below uppermost two storeys to be increased 4½ ins. beyond thicknesses for length up to 45 feet.

10. If any storey exceeds in height sixteen times the thickness prescribed for the walls of such storey in the above Table, the thickness of each external and party wall throughout such storey shall be increased to one-sixteenth part of the height of the storey, and the thickness of each external and party wall below that storey shall be increased to a like extent; but any such additional thickness may be confined to piers, properly distributed, of which the collective widths amount to one-fourth part of the length of the wall.

11. No storey enclosed with walls less than 13 inches in thickness shall be more than 10 feet in height between the floor and the ceiling thereof, or between the floor and the tie of the roof.

12. All buildings, excepting public buildings and such buildings as are in the Act defined to be buildings of the warehouse class, shall, as respects the thickness of their walls, be subject to the foregoing rules.

Rules for the Walls of Buildings of the Warehouse Class.

(London Building Act.)

The external and party walls of buildings of the warehouse class must at the base be made of not less thickness than that shown in the following Table, arranged for walls up to 120 feet in height, and supposed to be built of bricks not less than 8½ inches, or of stone or other blocks of hard and incombustible substance, the beds or courses being horizontal.

TABLE II.

1. Height up to 25 Feet.	Length unlimited. Base, 13 inches.		
2. Height up to 30 Feet.	Length up to 45 Feet. Base, 13 ins.	Length unlimited. Base, 17½ ins.	
3. Height up to 40 Feet.	Length up to 35 Feet. Base, 13 ins.	Length up to 45 Feet. Base, 17½ ins.	Length unlimited. Base, 21½ ins.
4. Height up to 50 Feet.	Length up to 30 Feet. Base, 17½ ins.	Length up to 45 Feet. Base, 21½ ins.	Length unlimited. Base, 26 ins.
5. Height up to 60 Feet.	Length up to 45 Feet. Base, 21 ins.	Length unlimited. Base, 26 ins.	
6. Height up to 70 Feet.	Length up to 45 Feet. Base, 21½ ins.	Length unlimited. Length to be increased in thickness from the base up to within 16 ft. from the top of the wall by 4½ ins.	
7. Height up to 80 Feet.	Length up to 45 Feet. Base, 21½ ins.	Length unlimited. Length to be increased in thickness from the base up to within 16 ft. from the top of the wall by 4½ ins.	
8. Height up to 90 Feet.	Length up to 45 Feet. Base, 26 ins.	Length unlimited. Length to be increased in thickness from the base up to within 16 ft. from the top of the wall by 4½ ins.	
9. Height up to 100 Feet.	Length up to 45 Feet. Base, 26 ins.	Length unlimited. Length to be increased in thickness from the base up to within 16 ft. from the top of the wall by 4½ ins.	
10. Height up to 120 Feet.	Length up to 45 Feet. Base, 31 ins.	Length unlimited. Length to be increased in thickness from the base up to within 16 ft. from the top of the wall by 4½ ins.	

11. The thickness of the wall at the top, and for 16 feet below the top, shall be 13½ inches; and the intermediate parts of the wall between the base and such 16 feet below the top shall be built solid throughout the space between straight lines drawn on each side of the wall, and joining the thickness at the base to the thickness at 16 feet below the top; nevertheless, in walls not exceeding 30 feet in height the walls of the topmost storey may be 9 inches thick, provided the height of that storey does not exceed 10 feet.

12. If in any storey of a building of the warehouse class the thickness of the wall, as determined by the foregoing Table, is less than one-fourteenth part of the height of such storey, the thickness of the wall shall be increased to one-fourteenth part of the height of the storey, and the thickness of each external and party wall below that storey shall be increased to a like extent; but any such additional thickness may be confined to piers, properly distributed, of which the collective widths amount to one-fourth part of the length of the wall.

13. The thickness of any wall of a building of the warehouse class, if built of materials other than such bricks as aforesaid, shall be deemed to be sufficient if made of the thickness required by the above Tables, or of such other thickness as may be approved by the County Council.

MISCELLANEOUS.

(*London Building Act.*)

The thickness of a cross-wall shall be two-thirds of the thickness hereinbefore required for an external or party wall of the same dimensions, and belonging to the same class of buildings, but never less than 8½ inches; and no wall subdividing any building shall be deemed to be a cross-wall unless it is carried up to the floor of the topmost storey, and unless in each storey the aggregate extent of the vertical faces or elevations of all the recesses and that of all the openings therein taken together does not exceed one-half of the whole extent of the vertical face or elevation of the wall.

Rules for Structural Steelwork for Buildings.

London County Council Act (1909).

Part IV., section 22, of the London County Council (General Powers) Act, 1909, which came into operation August 16, 1909, makes certain provisions, summarised below, with respect to buildings of iron and steel skeleton construction, and with the foundations, walls, floors, staircases, etc. of such buildings.

(1) All rolled steel shall comply with the British standard specification for structural steel for bridges and general building construction, and every pillar or girder shall be of iron or steel or any other metal which may be standardised by the Engineering Standards Committee.*

(2) The skeleton framework of a building, together with the party wall or walls upon which such framework bears, shall be capable of safely and independently sustaining the whole dead load and superimposed load bearing thereon.

(3) Pillars in external walls to be completely enclosed and protected from the action of fire by a casing of brickwork, etc., or incombustible material, at least 4 inches thick, properly bonded.

(4) Girders in external walls to be similarly encased with brickwork or other incombustible material at least 4 inches thick, properly secured to the adjoining work. The casing on the underside of such girders and to the edges of the flanges, plates, and angles may be not less than 2 inches thick.

(5) All other pillars and girders to be protected from the action of fire by being encased to the satisfaction of the district surveyor to a thickness of not less than 2 inches. The casing on the upper surface of girders and on the lower surface of subsidiary joists may be not less than 1 inch. Wood firrings must not be used in connection with any casing. This subsection does not apply in the case of buildings of only one storey and not more than 25 feet in height.

(6) Girders to be secured against buckling if the length of the girder exceeds thirty times the width of the compression flange. The web of every girder to be secured against buckling when the depth of the web exceeds sixty times the thickness.

(7) The span of a girder must not exceed twenty-four times the depth of the girder unless the calculated deflection is less than one four-hundredth part of the span.

(8) Two or more girders arranged alongside and closely adjacent to one another, so as to act together, must be fixed together by means of iron separators and bolts, or in any efficient manner approved by the district surveyor. Separators or other members acting as separators must not be placed at a distance apart exceeding five times the depth of the girders, and must be placed immediately over all supports and immediately under all concentrated loads.

(9) Girders for supporting external walls must be placed at the floor level of a storey or not more than 5 feet above or below floor level.

(10) Rivets must be used where reasonably practicable. Where bolts are used they must extend through the full thickness of the nuts, and the nuts must be so secured as to avoid risk of their becoming loose. The distance from the edge of a rivet or bolt hole to the edge of the member may not be less than the diameter of the rivet or bolt. Rivets must be so placed that their centres shall not be closer together than three times the diameter of the rivets. The pitch of rivets measured in a continuous straight line in girders, pillars, and roofwork must not exceed sixteen times the thickness of the thinnest member through which they pass.

(11)—(A) An external wall may not be less than 8½ inches thick for the topmost 20 feet of its height, and 13 inches for the remainder of its height below such topmost 20 feet. A less thickness may be allowed in any case in which under the London Building Act, 1894, such less thickness is prescribed.

(B) All party walls shall be of the thicknesses prescribed by the principal Acts.

(C) All brickwork and work in which terra-cotta, concrete, or other similar materials are used must be executed in Portland cement (British standard specification) mortar bedded close up to the metal framework without intervening cavity, and all joints must be made full and solid. In party walls (exclusive of any parts thereof immediately surrounding metal framework), or other internal brickwork not constructed to carry loads or stresses, lime mortar may be used.

(12)—(A) No plate or bar in any steel or wrought-iron pillar may in any part be less than ¼ inch thick, and the bases of all such pillars must be at right angles to the axis.

(B) Joints in such pillars must be close-butted with riveted cover-plates, and joints between such pillars must be properly fixed and made, and, unless unavoidable, no joint may be made between such pillars except as near as practicable to the level of a girder secured to such pillars.

* Now British Engineering Standards Association.

(c) The foot of every such pillar must have a proper base-plate riveted thereto with sufficient gusset pieces to distribute properly the load on the foundations, and the gusset pieces shall have sufficient rivets to transmit the whole of the load on to the base-plates.

(d) Where any pillars are built up hollow the cavities must be filled with concrete or be covered at both ends by metal plates riveted thereto.

(13)—(a) The width of cast-iron pillars may not be less than 5 inches, and the metal may not be in any part less than ¾ inch thick or one-twelfth of the least width of such pillar (whichever is the greater).

(b) The cap and base of every pillar must be in one piece with the pillar, or be connected thereto with a turned and bored joint.

(c) The ends of all pillars must be at right angles to the axis.

(d) Joints between such pillars shall be at or as near as practicable to the level of a girder secured to such pillars, and shall be fixed with not fewer than four bolts of not less diameter than the least thickness of metal in the pillar. If more than four bolts are used the diameter of the bolts may be reduced proportionately, but no bolt may be less than ¾ inch in diameter.

(e) The base of every such pillar shall have sufficient area to distribute properly the load on the foundations.

(14) The base of every pillar shall be bedded so that the load may be transmitted uniformly to the foundations.

(15) The stress in any metal interposed between the ends of a superimposed pillar and a pillar beneath must not exceed the stress on the superimposed pillar, and the least width across such interposed metal may not be less than the least width of the superimposed pillar.

(16) All floors and staircases (together with their enclosing walls) shall be constructed throughout of fire-resisting materials and be carried upon fire-resisting supports.

(17) All structural metalwork comprised in the skeleton framework of a building must be cleaned of all scale, etc., and be coated with one coat of boiled oil, tar, or paint before erection, and after erection shall receive at least one additional coat. Where embedded or encased in brickwork or other incombustible materials one coat of Portland cement wash applied after erection may be used.

(18)—(a) The dead load of a building consists of the actual weight of walls, floors, roofs, partitions, and all other permanent construction comprised in such building.

(b) The superimposed load in respect of a building consists of all other loads.

(c) For calculating the loads on constructions carrying loads in buildings the superimposed load on each floor and on the roof shall be estimated as equivalent to the following dead load :—

For a floor intended to be used for the purposes of human habitation or domestic purposes, 70 lbs. per square foot of floor area ;

For a floor intended to be used for the purpose of an office, or for any similar purpose, 100 lbs. per square foot.

For a floor intended to be used for the purpose of a workshop or retail shop, 112 lbs. per square foot ;

For every floor in a building of the warehouse class not intended to be used for any of the purposes aforesaid, not less than 224 lbs. per square foot. In every building of the warehouse class a notice shall be exhibited in a conspicuous place on each storey stating the maximum superimposed load per square foot which may be carried on such floor;

For a roof the plane of which inclines upwards at a greater angle than 20° with the horizontal, the superimposed load (which includes wind pressure) shall be estimated at 28 lbs. per square foot of sloping surface ;

For all other roofs the superimposed load shall be estimated at 56 lbs. per square foot measured on a horizontal plane.

If the superimposed load on any floor or roof is to exceed that specified, such greater load shall be provided for pursuant to subsection (2).

In the case of any floor intended to be used for a purpose for which a superimposed load is not specified, the superimposed load shall be provided for pursuant to subsection (2).

(19) For calculating the total load to be carried on foundations, pillars (including brick pillars), piers, and walls in buildings of more than two storeys in height, the superimposed loads for the roof and topmost storey shall be calculated in full in accordance with the last preceding subsection ; but for the lower storeys a reduction may be allowed, as follows :—

For the storey next below the topmost storey a reduction of 5 per cent. of the full superimposed load ;

For the next succeeding lower storey a reduction of 10 per cent. of the full superimposed load, and for each succeeding lower storey a further reduction of 5 per cent. of the full superimposed load. Provided always that the total reduction in respect of any storey shall not exceed 50 per cent. of the full superimposed load for such storey ;

No such reduction as aforesaid shall be allowed in the case of a building of the warehouse class.

(20) All buildings shall be so designed as to resist safely a wind pressure in any horizontal direction of not less than 30 lbs. per square foot of the upper two-thirds of the surface of such buildings exposed to wind pressure.

(21)—(a) The working stresses on pillars of cast iron or mild steel due to the loads thereon (other than stresses induced by wind pressure) shall not exceed those specified in the two following tables, according to the several ratios therein specified, or a proportionate load for other ratios :—

CAST-IRON PILLARS.

Ratio of Length to least Radius of Gyration.	Working Stresses in Tons per Square Inch of Net Section.		
	Hinged ends.	One end hinged and one end fixed.	Both ends fixed.
20	3·5	4·0	4·5
30	3·0	3·5	4·0
40	2·5	3·0	3·5
50	2·0	2·5	3·0
60	1·5	2·0	2·5
70	1·0	1·5	2·0
80	0·5	1·0	1·5

MILD STEEL PILLARS.

Ratio of Length to least Radius of Gyration.	Working Stresses in Tons per Square Inch of Section.		
	Hinged ends.	One end hinged and one end fixed.	Both ends fixed.
20	4·0	5·0	6·0
40	3·5	4·5	5·5
60	3·0	4·0	5·0
80	2·5	3·5	4·5
100	2·0	3·0	4·0
120	1·0	2·5	3·5
140	0·0	2·0	3·0
160		1·0	2·5
180		0·0	1·5
200			0·5
210			0·0

(B) The working stresses on wrought-iron pillars due to the loads (other than wind-pressure stresses) may not exceed two-thirds of the stresses specified for mild steel pillars.

(C) Any pillar eccentrically loaded must have the combined stresses resulting from such eccentricity at any part added to all other stresses at that part, and the sum of the stresses may in no case exceed the working stresses specified.

Provided that working stresses exceeding those specified in paragraphs (A), (B), and (C) by not more than 25 per cent. may be allowed where such excess is due to wind pressure.

(D) The eccentric load of a pillar may be considered to be distributed uniformly over its cross-section area at the next lower level at which it is fixed and secured in the direction of eccentricity.

(22) The working stresses of iron and steel (except in the case of pillars as hereinbefore provided) may not exceed the following :—

Metal.	Working Stresses in Tons per Square Inch.			
	Tension.	Compression.	Shearing.	Bearing.
Cast iron	1·5	8	1·5	10
Wrought iron	5	5	4	7
Mild steel	7·5	7·5	5·5	11

(23) Any rivet used in double shear the working shear may not exceed one and three-quarter times the shear allowed on a rivet used in single shear.

(24) The pressures of foundations on the natural ground may not exceed the following :—

Material.	Tons per Square Foot.
Natural bed of soft clay or wet or loose sand	1
Natural bed of ordinary clay or confined sand	2
Natural bed of compact gravel, London blue clay, or chalk	4

(25) The pressure on concrete foundations may not exceed 12 tons per square foot.

(26) No disengaged brick pillar without lateral supports may have a height of more than six times its least width ; but a pillar with lateral supports may have a height between such supports not more than twelve times its least width, but in no case less than 13½ inches.

(27) The pressure on any brickwork may not exceed the following :—

Material.	Tons per Square Foot.
Blue brick in cement mortar	12
Hard brick (including London stock) in cement mortar	8
Ordinary brick in cement mortar	5

(28) The Council may prescribe the materials and the proportions to be used in concrete provided under the provisions of this section.

(29) It shall be lawful to make any addition, alteration, or do other work upon a building in accordance with this section, provided that the loads and stresses in a building so altered are transmitted from the roof to the foundations by a skeleton framework of metal, or partly by a skeleton framework and a party wall, and the foregoing provisions shall apply as if the part so altered were a separate building.

Foundations.

Brick walls are made wider at the base, in order to spread the weight of the building over a larger area and produce an equality of settlement.

Figs. 16 to 22 show sections of footings (the wide courses at the base of a wall) of walls in English bond from one to three bricks in thickness. Trenches are excavated for beds of concrete usually from 2 feet to 7 feet in depth, and 12 inches wider than base of footings (6 inches on each

FIG. 16. Elevation.

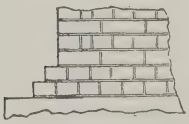
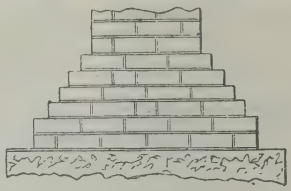


FIG. 17. Section.

FIG. 18. Frontage for 2½-Brick Wall.



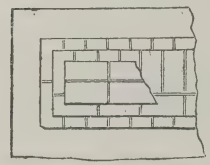


FIG. 19. Plan.

FIG. 20.

FIG. 21.

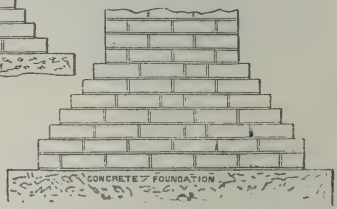


FIG. 22.

side). If the site on which a building is to be built be gravel, sand, or natural virgin soil, no concrete bedding must be used ; but otherwise the whole site should be covered with a layer of concrete not less than 6 inches thick.

The foundations of the walls of a building should be on a bed of good concrete not less than 9 inches thick, and projecting at least 4 inches on each side of the lowest course of footings of walls. If the site be upon a natural bed of gravel, concrete is not required.

MASONRY.

Lifting Stones

There are various ways of laying hold of stones which are too heavy to be moved by hand. The following are those most frequently used :—

1. By a pair of nippers or iron claws or hooks, as in fig. 23, the ends of which catch in holes

made, one in each side of the block, in the same vertical plane as, and at a convenient distance above, the centre of gravity of the block. 2. By a *Lewis*, or iron dovetail, as in fig. 24, made in three pieces, the two outer ones being first dropped into a corresponding dovetailed hole cut in the stone directly over its centre of gravity, then the centre-piece slipped in between them, the suspending-tackle put in its place, and finally a bolt passed through the whole and keyed up. 3. By two curved iron plugs, with a chain passing through eyes in their heads, placed in a dovetailed hole directly over the centre of gravity of the stone, as shown in fig. 25. 4. By an iron conical plug, with an eye at the top for a chain or hook to pass through, placed in a rather

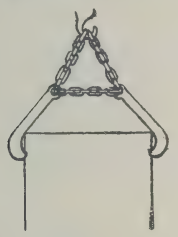


FIG. 23.

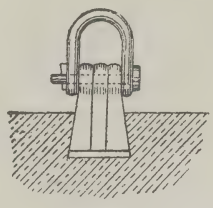


FIG. 24.

larger conical hole directly over the centre of gravity of the stone, with an iron wedge curved so as to fit the plug and the circumference of the hole, tightly driven in by its side, as shown in fig. 26. This plug was used for setting the heavy granite coping on the Thames Embankment

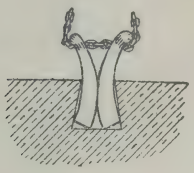


FIG. 25.

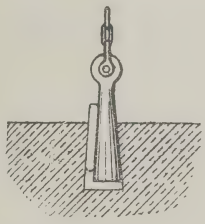


FIG. 26.

wall. 5. By a single iron plug, very slightly tapered, and driven into a cylindrical hole directly over the centre of gravity of the stone. By this simple means hard stones, such as granite, may be lifted with ease, a few side taps being sufficient to loosen the plug when required to be withdrawn.

Setting Stones.

All stones, except under peculiar circumstances, should be laid on their natural or quarry beds, or with their natural beds as far as possible perpendicular to the pressures they have to bear.

All dry and porous stones should be well wetted before being laid, so as not to absorb the moisture required for the proper setting of the mortar. Iron should never be placed in contact with stonework where, by rusting, it might disfigure it with stains, or split the stone, either by its increase in bulk during the process of oxidation, or by its expanding and contracting under the influence of heat and cold.

Descriptions of Masonry.

There are three descriptions of masonry in ordinary use, viz. :—

Rubble. Block-in-Course. Ashlar.

If the stone at disposal is thinly bedded, rough, or intractable, it should be used as *Rubble work*; if obtainable in blocks, and more or less easily wrought, it should be used as *Block-in-Course* or *Ashlar*, according to circumstances.

The various systems are illustrated by figs. 27 to 36.

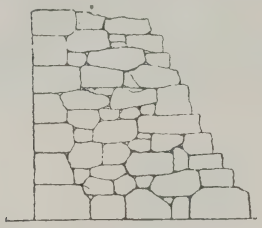


FIG. 27. Common, Rough, Uncoursed, or Random Rubble.

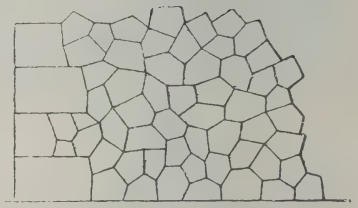


FIG. 28. Random Rubble, with hammer-dressed joints and no spalls on face.

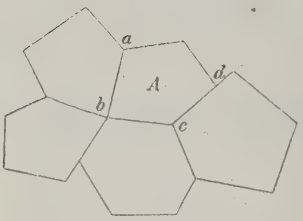


FIG. 29. Hammer-dressed Joints for Random Rubble.

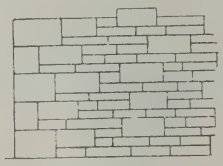


FIG. 30. Irregular-coursed, Random-coursed, Snecked, or Squared Rubble.

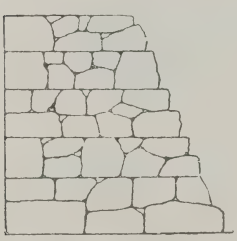


FIG. 31. Built up to Courses.

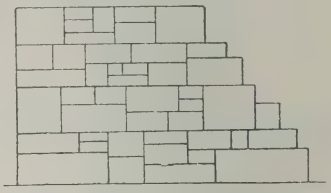


FIG. 32. Built up to Courses, with beds horizontal and joints vertical.

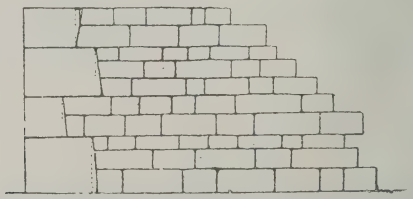


FIG. 33. Coursed Rubble.

Rubble Masonry.

The best stones for rubble masonry are those that scabble freely, and such as lie in 4 or 5 in. beds. Basalts and stones of a crystalline structure are troublesome to use, as they fly under the hammer; but granite and sandstones work in well.

Rubble may be either *uncoursed*, *irregular*, or *random-coursed*, *worked up to courses*, or *coursed*, chiefly depending upon the character of the stone at disposal; some stones, such as Kentish rag, from their intractable nature, and the absence of any distinct lines of bedding, are specially adapted for *uncoursed* rubble, whilst others readily work into courses, and therefore should be used as *coursed* rubble.

In common, rough, uncoursed, or random rubble (fig. 27) the stones are laid at random, without being brought up to any level courses, the only tools required being a trowel, waller's hammer, and plumb or battering rules, as the case may require.

One bond stone should be used to at least every yard super of the face, and these should run about two-thirds to three-fourths through the wall, alternately from the opposite sides; this, in the case of dwelling-houses, diminishes the chances of damp being transmitted to the interior and is in all cases much preferable to a smaller number of *through* stones. In boundary and similar walls *through* stones may, of course, be used. In thick walls, such as piers, abutments, and retaining walls, the bond stones should tail into the work as far as possible; and the whole should be bonded together in the best way the sizes of the blocks will admit of.

It is necessary to see the bond stones actually built into the wall, for builders are apt to set large stones on edge, running only 3 or 4 ins. into the wall, instead of true bond stones—a trick which cannot well be detected after they have once been covered up. In random work (figs. 28 and 29) the stones cleave into irregular shapes, and are fitted closely together. Thus, in order to get A (fig. 29) into place the angles *abc*, *dcb*, are taken with the mason's bevel, and the joints are hammered and punched to fit truly. This is the most expensive kind of rubble work, and requires much more skill to build properly than any other class of rubble walling.

If the rubble stone is distinctly bedded, the beds varying much in depth, and often running very thin, the stones should be put together as in fig. 30, in what is called *irregular* or *random-coursed* rubble. The joints in this case are not necessarily vertical; but in a superior class of the same work all the joints are specified to be vertical and the beds horizontal, by which a more regular appearance is given to the work. The stones are all shaped and roughly squared as required, and hammer or axe faced, in the gross, ready for the waller to set in the wall. This class of masonry is very bold and effective for engineering purposes, when a good proportion of large stones are used, roughly hammer-dressed on face.

Both of the kinds of rubble masonry just described are very commonly worked up at intervals, with the line and level, to level courses, either regular or irregular in depth, the latter being preferable on account of its appearance and more suitable to the irregular sizes of the stones used. From 18 to 14 ins. are the ordinary depths of the courses, according to the nature of the stone. Figs. 31 and 32 show both common and irregular coursed rubble worked up to courses; the latter being also called *coursed header work* when the headers, occurring at intervals, run the full depth of the course. In fig. 33 a superior class of work is shown, the joints being all vertical, by which means a more regular appearance is obtained.

Rankine's 'Civil Engineering,' speaking of coursed rubble (such as fig. 33), says: '*One-fourth part* at least of the face in each course should consist of bond stones or headers, each header to be of the entire depth of the course, of a depth ranging from one and a half times to double that depth, and of a length extending into the building to from three to five times that depth, as in ashlar. These headers should be roughly squared with the hammer, and their beds hammer-dressed to approximate planes; and care should be taken not to place the headers of successive courses above each other, for that arrangement would cause a deficiency of bond in the intermediate parts of the course. Between the headers each course is to be built of smaller stones, of which there may be one, two, or more in the depth of the course. These are sometimes roughly squared, so as to have vertical side-joints; sometimes the stones are taken as they come, so that the side-joints are irregular; but no side-joint should form an angle with a bed-joint sharper than 60°. Care should be taken, not only that each stone shall rest on its natural bed, but that the sides parallel to that natural bed shall be the largest, so that the stone may be flat, and not be set on edge or on end. However small and irregular the stones may be, care should be taken to make the course break joint. Hollows between the larger stones should be carefully filled with smaller stones, completely embedded in mortar.'

The same authority gives the resistance of good coursed rubble masonry to crushing at about four-tenths of that of single blocks of the stone it is built with.

Coursed, or regular coursed rubble, as shown in fig. 33, is applicable where the beds, though thin, are pretty regular, so that a sufficient number of stones of a uniform depth can be got to allow of their being laid in regular courses of one stone only in depth. Larger stones, often of cut stone, equal in depth to two or three courses of the rubble work, are generally used at the quoins.

Whenever cut-stone is used in connection with rubble work, as at the quoins of a building, the ends adjoining or tailing into the rubble should never be cut square, but be left comparatively rough, so as to harmonise with the masonry in the body of the wall, and not appear as if cut off from it by straight lines.

Dry-walling is the simplest class of rubble work, and consists of stone roughly hammered, and bedded by pinning with spalls, without any mortar. It requires considerable skill to build properly, as its stability depends entirely on the firm bedding and bonding of the different stones. It is chiefly used for fencing land, for railway and canal embankments and cuttings, or at the backs of retaining-walls, to diminish the pressure of the earth on them, and to allow of water finding its way down to the drains below. Such walls—which are generally about 5 ft. high, increasing in thickness towards the bottom, and averaging about 18 ins. thick—are usually built to lines strained between trestles (fig. 34), in order to avoid plumbing the face.

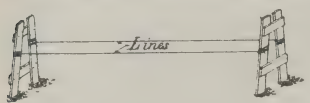


FIG. 34.

Dry-rubble walling is generally built in courses about 12 ins. high, and should have a waterproof top, or *coping*, to keep the water from getting into the body of the work and bursting it in frosty weather. The coping may be made of stones laid on edge in mortar, bituminous concrete, or, for want of anything better, clay puddle, or even sods.

When the stones come out in thin slabs the walls may be specified to be built of 'good sound, flat bedded stones of a fair average size (none less than 6 ins. broad across the narrowest part), set dry, and coped with stones, not less than 9 ins. deep, set on edge; each coping stone to extend across the full breadth of the wall, and to be properly set in good hydraulic mortar.' A high joint with a projecting ledge is very frequently used to mark the joints of rubble work; this is a bad system, and is sure to peel off under the influence of wet and frost.

A cubic yard of rubble masonry will, as a rule, require $\frac{1}{5}$ cubic yard of mortar and $1\frac{1}{5}$ cubic yard of stone.

FIG. 35.—Block-in-Course.

FIG. 36.—Ashlar.

BLOCK-IN-COURSE MASONRY.

Block-in-Course (fig. 35) is a term applied by engineers to a class of masonry much used in piers, abutments, wing walls, &c., when there is a suitable stone in the district and good solid work is required. The stones are all squared and brought to good fair joints, the faces being more or less worked, according to taste, but generally merely hammer-dressed; and the courses consist of single stones only in depth. It chiefly differs from ashlar in the smaller size of the stones and depth of the courses, the stones at command being—ordinarily rougher and more thinly bedded—chiefly *shoddies*, or stones under 12 ins. deep, whereas *ashlar* blocks range, as a rule, from 12 ins. to 14 ins. in depth.

The best stone for the purpose is a hard, self-bedded stone which will work easily into courses from 3 ins. to 10 ins. or 12 ins. in depth, the length of each stone being four or five times its depth. No attention is paid to uniformity in the depth of the courses.

ASHLAR MASONRY.

The term *ashlar* is applied to masonry built up of a thick-bedded stone which admits of being scabbled, or sawn into square blocks of given dimensions. It is also used to denote roughly squared blocks of stone over 12 ins. deep, as delivered from the quarry or stonecutter, or as prepared for setting. Ashlar work (fig. 36) is built in courses of uniform, or nearly uniform, depth—generally from 10 ins. to 14 ins.—ranging throughout with the quoins and dressings. It goes by different names according to the face put upon the stone, from quarry-pitched or rock ashlar up to wrought ashlar.

No stone should be laid in mortar without being first fitted into place and any irregularities in shape corrected. Large ashlars are first accurately fitted into place, and then set on a bed of mortar carefully spread out to receive them; after which the vertical joints have to be filled by stopping them up on the outsides with cement, and pouring in cement or mortar grout, which should be worked about with a piece of hoop iron, so as to ensure its completely filling the joints. The beds and joints are worked to plane surfaces, so as to allow of close-fitting joints. These surfaces should not be too smooth, otherwise the mortar will not adhere so well to the stones, but in good work should be true enough to allow of joints not over $\frac{1}{8}$ in. thick; they are generally

either left as they come from the saw (if sufficiently true), or taken out of winding by running chisel draughts round the margins, and dressing them down with the point, as in half-plain work, taking off any parts left projecting beyond the level of the chisel draughts with a chisel.

Proportions for Stones, Bond, &c.

In both block-in-course and ashlar masonry it is necessary to proportion the length, breadth, and thickness of the stones according to their hardness, in order to guard against their breaking across. Rankine says that in the weaker sandstones and granular limestones the length of a stone should not exceed three times its depth, nor the breadth one and a-half to double its depth. In harder stones—such as require 2½ or more tons to the inch to crush them—the length may by four or five times the depth, and the breadth three times the depth.

The best bond in masonry is that which shows on the face of the work alternate headers and stretchers in each course, as in Flemish bond in brickwork, each header coming over the centre of a stretcher in the course below. In such work one-third of the face consists of headers, if the length of the stretchers is twice the breadth of the headers; but as stones are rarely cut to exactly the same dimensions, it may be laid down that not less than one-fourth of the face of the wall should consist of headers, and that the stones should break joint from one to one and a-half times the depth of a course.

The thickness of the joints will vary from ½ to ⅛ inch, according to the smoothness of, or amount of work bestowed upon, the beds, as it must be sufficient to transmit the pressures from stone to stone without permitting of actual contact at any point of their surfaces.

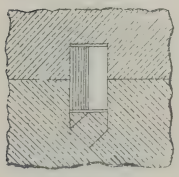


FIG. 37.

Dowelling.

For securing stones in places where the adhesion of the mortar or cement is not sufficient *dowelling* is the best method. Fig. 37 shows a method of doing this. It consists in making exactly opposite each other, in the two stones two holes, either circular or square in section, and fitting into them a dowel or pin. These may preferably be of good hard slate, run with either brimstone or cement.

Measurement and Valuation of Masonry.

In districts where stone abounds, and forms the principal material for building purposes, ordinary walls are built of rubble stone, and measured by the cubic yard, or some other local standard, such as, in Ireland, by the running perch (21 feet), of a given height and thickness, or by the square perch of 21 feet super, at a standard thickness of 18 inches. In the West of England the square perch is employed, of 18 feet super, at a standard thickness of 2 feet, to which walls of any thickness are reduced; or the price per perch may vary with the thickness of the wall. In other parts masonry walling is measured by the rood of 36 square yards, or 24 cubic yards, the standard thickness being 2 feet; or by the rod of 272 feet super, as in brickwork, at a standard thickness of 18 inches or 2 feet.

In engineering, as distinct from architectural constructions, masonry is rarely valued otherwise than by the cubic yard for the roughest descriptions of work, and by the cubic foot for superior work.

The price in all cases varies with the quality and description of work, as 'coursed' or 'uncoursed,' in 'walling' or 'foundations.'

The superficial content of the surface-work is paid for separately by the square foot, yard, or rood of 36 square yards, including jointing or pointing, and such squaring to beds and joints as may be required, quoins, &c., of selected stones being often paid for at so much extra per foot run.

Superior work, such as squared masonry, whether in wallings or dressings, is usually valued by the foot cube as block stone, merely scabbled or sawn to the required dimensions and set. The work upon it is taken separately by the foot super.

When different descriptions of masonry occur in a wall, such as rubble faced with ashlar, instead of cubing out the rubble and ashlar work separately—in which case the latter is valued by the cubic foot prepared and set, including all beds and joints, the facework being taken by the foot superficial as tooled, pointed, &c.—the whole may be taken as rubble masonry, and so much extra allowed per foot superficial for the superior facing, including all extra labour, dressing, and pointing.

When walls are built of rubble masonry, it is invariably cheaper to buy bricks, and build them into such parts as jambs and arches, the labour upon which would be much less than in those shown on the face of the wall.

The block stone for the cut stonework in a building is ordinarily sold at the quarries, or delivered, by the cubic foot, or in large rough blocks by the ton, or from 13 to 17 cubic feet, according to the description of the stone, 1 inch being allowed each way for irregularities and waste.

QUANTITY OF STONE EQUIVALENT TO 1 TON IN WEIGHT.

Description.	Feet Cube.	Description.	Feet Cube.
Vein marble	13	Craigleith	14¾
Statuary marble and granite	13½	Portland	15 to 16
Purbeck	14	Derby	15
Yorkshire	12 to 15	Bath	16 to 17

Description.	Feet Super.	Description.	Feet Super.
3½-inch York paving (tooled)	58½	3-inch granite	54
3 „ „ „ „	49½	6 „ „	27
2½ „ Purbeck paving	68	7 „ curb	23
3 „ „ „	56		

SECTION IX

PART II

MORTARS—CONCRETES—CEMENTS—PLASTERS—GLUES

Revised and amplified by Alfred B. Searle
(Consulting Advisor on Cementitious Materials).

MORTARS.

Mortars are pasty substances which gradually harden on exposure; they are used for uniting stones or bricks in the construction of masonry or brickwork.

The chief kinds of mortar are:

Lime mortar, composed of lime and sand.

Trass mortar, composed of lime, sand, and trass or brick-dust, or of lime and trass without sand.

Cement mortar, composed of Portland cement (or other silicious cement) and sand.

Hydraulic mortar, composed of either hydraulic lime or cement, and sand, and characterised by its power of hardening under water.

Clay mortar, consisting of a mixture of fireclay and crushed firebricks, and used chiefly in the construction of furnaces, kilns, and other structures which have to be heated to high temperatures.

Lime Mortar.

The chief ingredients of ordinary lime mortar are lime and sand, but other materials, such as crushed bricks, ground stone or trass, and waste products such as ground slag, ashes, or coke, may, under suitable conditions, replace some of the sand. The tenacity of the mortar may be increased by the addition of hair or other fibrous material.

The best mortar for use in air * consists of a mixture of lime, sand, and brick-dust, but the latter is usually omitted on account of its cost.

Lime mortars made of fat or pure lime should not be used for roads, bridges, viaducts, waterworks, or any structure exposed to water, and some authorities maintain that even for churches, houses, and other buildings, it should not be used without the addition of trass or its equivalent, so as to prevent the flaking, crusts, and canker which are produced when too fat a lime is used.

Hydraulic lime mortar is composed of hydraulic lime and sand, the proportions varying from 3 to 6 measures for each measure of the lime (see *Lime Concrete*, p. 388). The chief use of this kind of mortar is for structures exposed to water, but unless the hydraulic lime is well selected, the structures may develop crusts and canker. Portland cement mortar is usually much superior to mortar made with hydraulic lime.

Trass Mortar.

Trass mortar is but little used in Great Britain, but has been used extensively abroad, and particularly in the East. The term 'trass mortar' is not confined to mortars containing true trass, but includes those containing similar materials, such as brick- or tile-dust, pozzolana, santorin earth, and other metamorphosed alumino-silicates, which combine with free lime to form an indefinite compound having cementitious properties. The use of trass or similar material overcomes the disadvantages which accompany the use of fat-lime. The best proportions for trass mortars are shown in the table on p. 386. An excess of trass does no harm.

Cement Mortar.

Cement mortar consists of a mixture of sand and Portland cement, the usual proportions being 2 to 6 measures of sand to 1 of cement, according to the strength required. The best proportions for various purposes are shown in the table on p. 386. For structures exposed to the air there should be more than twice as much sand as cement, but for completely waterproof structures the proportion of cement should not be less than half the sand used.

Lime is sometimes added to cement mortar to increase its plasticity and to make it easier to use. The addition of not more than 20 per cent. of hydrated lime slightly increases the strength of the concrete, but if the latter is exposed to water (inch rain), the free lime may be washed out, leaving a porous material which soon flakes and disintegrates when exposed to frost. These drawbacks can be avoided by adding trass instead of lime—a cement mortar composed of two measures of cement, one of trass and six measures of sand, being of wide application and easy to work.

All cement mortars are hydraulic—i.e. they harden readily under water.

* Mortars for use under water are termed *hydraulic* mortars.

Raw Materials for Mortars.

The chief materials for mortars are *lime* and *sand*, but *Portland Cement* (p. 401) is sometimes used to replace part or all of the lime.

The *lime* should be made by burning a calcareous limestone low in magnesia; magnesian limestone, or dolomite, is much less suitable. The burning should be effected in such a manner that the lime contains at least 85 per cent. of calcium oxide in the form of quicklime. Limes which have been imperfectly burned, and so contain 15 per cent. or more of unaltered calcium carbonate, are not suitable for mortar to be used on important work. A very pure lime (containing 95 per cent. of free calcium oxide) is not necessary for building and is usually too costly. The idea that a pure lime is, in itself, objectionable, is incorrect, provided a suitable sand is used. If a very pure lime is used with a very coarse sand, the absence of fine siliceous matter may result in a weak mortar which chiefly hardens externally.

Lean (i.e. less pure) lime * and hydraulic lime, on the contrary, harden in a manner similar to cement. The addition of trass or brick-dust to slaked lime has a similar effect on the manner in which the mortar hardens.

Hydraulic limes differ from the 'fat' or quick limes in consisting of an indefinite compound of lime and clay, which may be regarded as a crude Portland cement (p. 401), containing a large excess of free lime. The hydraulic limes owe their power of setting and hardening under water to this lime-clay compound, and consequently form a stronger brickwork or masonry than is produced by common mortar. Some hydraulic limes consist of a mixture of quicklime, true hydraulic lime, and inert or unburned material; for important work they should be tested and the proportions of these ingredients ascertained, as in hydraulic lime the proportion of cementitious matter present is more important than the percentage of lime.

Lias lime is hydraulic lime obtained from clayey limestones occurring in the Lias formations.

Quicklime should be slaked or hydrated before use; hydraulic lime should be bought in the form of a powder. To save the trouble and time involved in slaking and the risk of 'blowing' and other defects due to imperfectly slaked lime, it is becoming increasingly the custom to use *hydrated lime*. This has been slaked under skilled supervision, so as to produce a fine powder which is free from all coarse material and quicklime; it is sold in bags of 1 cwt. and is ready for immediate use. Slaked or Hydrated lime should pass completely through a 60-mesh sieve, and should not leave more than 8 per cent. residue on a 100-mesh sieve. Hydraulic lime should not leave more than 8 per cent. of residue on a 100-mesh sieve.†

A hydraulic lime, when mixed with 30–50 per cent. of its weight of water and stored under water at 60° F., should harden sufficiently not to be indented with the finger-nail in about three days.†

The *sand* used for mortars should consist of sharp (angular) grains of various sizes. Rounded grains do not interlock sufficiently to produce a strong mortar.†

Crystalline quartz sand is to be preferred to mica or felspar sand; the usual description 'good, clean, sharp sand' is much too wide. The grains of sand which are angular give the best results of tensile strength, those that are round give the highest compression strength. It is advisable to submit the sand to a microscopic examination to ascertain its suitability, as a considerable variation in the strength of mortar may occur, owing to the form and variety of the sand particles. From trials made with mortar composed of 1 of cement and 3 of sand a difference of strength varying from 171 lbs. to 313 lbs. per sq. in. was found, owing entirely to the difference in the sand; normal sand mixed in above proportions gave 256 lbs. per sq. in. in 28 days.

It is very important that the sand used for mortars and concrete should be free from clay and decomposed vegetable matter (*humus*). A simple test, devised by D. Abrams and O. Harder, consists in shaking the sand vigorously with a dilute solution of caustic soda, allowing the sand to settle and noting the colour of the overlying liquid. If it is dark-coloured, the sand is probably unsuitable for use; a very pale yellow colour may be neglected.

The grading of the sand, i.e. the proportions of grains of different sizes, deserves far more attention than has yet been paid to it. By the use of a suitably graded sand, it is easy to reduce the proportion of the more expensive lime or cement by about 33 per cent. of that required for a less suitable sand. For most purposes the sand should contain about 30 per cent. of air-spaces or voids between the grains. The proportion of voids may be estimated by filling a measure with sand, and then pouring in water until the water is level with the top of the sand. The ratio of the volume of water, added to the volume of the sand, is that of the volume of the voids to that of the sand.

The *water* used in the preparation of mortar should be free from decaying substances; hence, peaty water should not be used. Apart from this, its source and character are of small importance. Brackish water, and even sea water, does little harm as regards the strength of the mortar, but may cause scum or efflorescence on the structure.† If both are available, soft water is better than hard. For cement-sand mortar the average will be about 22 per cent. of the cement and sand by volume, or about 20 per cent. by weight, for trass-cements and mortar 8 to 10 per cent. by volume.

* Limes which produce less than twice their volume of slaked lime in the form of an apparently dry powder are termed 'lean' or 'poor' limes.

† See Tentative Specifications (p. 385), which give different figures.

*(Building Research Board Report No. 9—1927).**

	Quicklime.			Hydrated Lime.		Hydraulic Lime.		
	Class A.	High Magnesium. Class A.	Class B.	Class A.	Class B.	Class C1.	Class C2.	Class C3.
Composition	Not less than 95% $CaO + MgO$.† Not more than 10% CO_2.	Not less than 95%. $CaO + MgO$.† More than 20% MgO.† Not more than 10% CO_2.	Not less than 85% $CaO + MgO$.† Not more than 10% CO_2.	Not less than 95% $CaO + MgO$.† Not more than 5% CO_2.	Not less than 85% $CaO + MgO$.† Not more than 5% CO_2.	Not less than 60% $CaO + MgO$.† Not more than 10% CO_2 and 15% $SiO_2 + Al_2O_3$.†	Not less than 60% $CaO + MgO$.† Not more than 10% CO_2. Not more than 25% or less than 15% $SiO_2 + Al_2O_3$.†	Not less than 60% $CaO + MgO$.† Not more than 10% CO_2. Not more than 35%, or less than 25% $SiO_2 + Al_2O_3$.†
Fineness and residue.	Not more than 5% when slaked and washed through I.M.M. Standard No. 20 sieve.	(As Class A)	(As Class A)	All through No. 20 sieve. Not more than 10% on No. 200 sieve. Not more than 1% on No. 50 sieve.	All through No. 20 sieve. Not more than 15% on No. 200 sieve.	*Either* not more than 5% when washed through No. 20 sieve. *Or* ground to pass through No. 20 sieve, and not more than 15% residue on No. 200 sieve.	(As Class C1)	(As Class C1)
Soundness	—	—	—	See below ‡	—	—	—	—
Tensile strength	—	—	—	—	—	—	See below §	See below §

‡ Hydrated Lime (Class A)—

Soundness: Accelerated steaming test on 4 in. × ¼ in. pat, made with lime 1 part, standard sand 5 parts, allowed to set in air 24 hours, covered with lime paste and kept 24 hours, then immersed 5 hours in steam to show no signs of cracking, pitting, or disintegration.

§ Hydraulic Lime (Classes C2 and C3)—

Tensile Strength: On lime slaked, stored 24 hours, mixed with three times its weight of standard sand and sufficient water, and briquettes stored in moist air 72 hours and in water for 28 days. Briquettes same shape and size as for cement.

Average of six briquettes not less than 75 lbs. per sq. in. after 28 days.

* Published by His Majesty's Stationery Office, Adastral House, Kingsway, London, W.C. 2; price 1*s.* 9*d.* net.

† On ignited sample.

Proportions for Mortars.

Mortars are usually made largely by guess-work, but to secure the best results—which are often the cheapest, as a large saving can be effected by preventing a waste of lime—it is necessary to determine the best proportion of sand for the particular lime used.

For many purposes, the most suitable proportions are from 2 to 5 measures of sand to 1 measure each of lime and brick-dust, or trass, but the latter is usually omitted, and large quantities of mortar consist of 1 measure of lime to 3 measures of sharp sand. When ashes * or coke dust are used, one measure of these materials usually replaces an equal amount of sand. For the cheapest mortar, 1 part of lime and 4 parts of sand is used, but such a mortar is weaker than is desirable for general work.

According to J. A. van der Kloes, who has investigated the matter very thoroughly, the best proportions for mortars for various purposes are as shown in the following table :—

Number.	Lean Lime.	Fat Lime.	Lime-Putty.		Trass.	Portland Cement.	Trass or Pozzolana Cement.	Sand.
			Solid.	Fluid.				
A.—Waterproof Mortars for Work which is continually under Water.								
I.	1	—	—	—	1½	—	—	1½
II.	—	1	—	—	1½	—	—	2
III. (*a*)	—	—	1	—	3	—	—	4
III. (*b*)	—	—	—	1	2¼	—	—	3
IV.	—	—	—	—	—	1	—	2
V.	—	—	—	—	1	1	—	2½
VI.	—	—	—	—	—	—	1	1–1½
B.—Mortars for Quays, Sluice Walls, Damp-proof Courses, etc.								
VII.	1	—	—	—	1½	—	—	2–2½
VIII.	—	1	—	—	1½	—	—	2½–3
IX. (*a*)	—	—	1	—	3	—	—	5–6
IX. (*b*)	—	—	—	1	2¼	—	—	3¾–4½
X.	—	—	—	—	—	1	—	3
XI.	—	—	—	—	1	1	—	4
XII.	—	—	—	—	—	—	1	2
C.—Mortars for Foundations and Buildings.								
XIII.	1	—	—	—	1½	—	—	3–4
XIV.	—	1	—	—	1½	—	—	4–5
XV. (*a*)	—	—	1	—	3	—	—	8–10
XV. (*b*)	—	—	—	1	2¼	—	—	6–7½
XVI.	—	—	—	—	—	1	—	3
XVII.	—	—	—	—	1	1	—	4–5
XVIII.	—	—	—	—	—	—	1	2½–3

For 1 cu. yd. of cement mortar, the following quantities of sand and Portland cement will be required :—

Proportions.		Sand.	Cement.
Cement.	Sand.	Cub. ft.	Lbs.
1	0	—	2,800
1	1	18·8	1,695
1	2	24·9	1,120
1	3	28·7	860
1	4	30·3	682
1	5	31·4	565
1	6	32·2	483

* The use of ashes in mortar is a frequent cause of scum or efflorescence in brickwork.

0·87 cu. ft. of neat cement mortar will result from mixing 0·36 cu. ft. of water with 1 cu. ft. of loose cement.

1 cu. ft. of loose Portland cement will make about—

4·1	cu. ft. of	concrete mixed	1 : 2 : 4
5·0	,,	,,	1 : 2½ : 5
5·8	,,	,,	1 : 3 : 6
7·5	,,	,,	1 : 4 : 8

1 cu. ft. of loose Portland cement neat as cement mortar will cover about 10·4 sq. ft., 1 in. thick.
1 cu. ft. of loose Portland cement to 1 sand will cover about 17 sq. ft. 1 in. thick.
1 cu. ft. of loose Portland cement to 2 sand will cover about 25 sq. ft. 1 in. thick.
1 cu. ft. of loose Portland cement to 3 sand will cover about 34 sq. ft. 1 in. thick.
1 cu. ft. of loose Portland cement to 2 sand will lay about 146 bricks with ⅜-in. joint, and 247 bricks with ½-in. joint.
1 cu. ft. of loose Portland cement to 3 sand will lay about 212 bricks with ⅜-in joint, and 317 bricks with ½-in. joint.
1 yd. super. of pointing brickwork in neat cement requires about 8 lbs. of cement.

The proportion of water required varies with the nature of the lime and sand used. (See the par. on *water*, on p. 384.)

Mixing Mortars.

If a mortar is to answer its purpose satisfactorily, it is obvious that the various constituents must be mixed as uniformly and intimately as possible, for wherever sand or other powdery or gritty substances accumulate in the mortar, no hardening or adhesion will take place, and the mortar will be almost useless at such points.

Small quantities of mortar may be mixed by repeatedly turning over the materials with a shovel, and afterwards with a trowel, so as to mix them very thoroughly. As the labour is hard, and workmen are not always sufficiently conscientious, machine-made mortar is often more reliable than that mixed by hand. The most suitable machine for mixing mortar consists of an edge-runner or pan-mill of moderately light construction, in which the materials are treated for 20 to 30 minutes. Some builders introduce large pieces of material into such a mill, and grind them therein. This is not desirable. No pieces larger than ¼ in. diameter should be introduced into a mortar mill, and if any materials more than $\frac{1}{16}$ in. diameter are present, the mixing must be prolonged until they have all been reduced to $\frac{1}{32}$ in. or less in diameter.

Choice of Mortar

Mortar made with ordinary lime should not be exposed to water.

Mortar made with hydraulic lime should not be exposed to running water, unless the mortar also contains trass.

Cement-mortar is best where great strength is required a short time after completing the structure, but where a longer period of hardening is permissible, lime-trass mortar is cheaper and equally durable.

For the interior of kilns, furnaces, and flues, clay mortar should be used.

Coloured Mortars.

TINTS GIVEN TO PORTLAND CEMENT MORTAR CONTAINING TWO PARTS OF RIVER SAND TO ONE PART OF CEMENT.

Pigment.	Weight of Dry Pigment to 100 Pounds of Cement.			
	½ Lb.	1 Lb.	2 Lbs.	4 Lbs.
Lamp black	Light slate	Light gray	Blue gray	Dark blue slate
Prussian blue	Light green slate	Light blue slate	Blue slate	Bright blue slate
Ultramarine blue.	—	Light blue slate	Blue slate	Bright blue slate
Yellow ochre	Light green	—	—	Light buff
Burnt umber	Light pinkish slate	Pinkish slate	Dull lavender pink	Chocolate
Venetian red	Slate, pink tinge	Bright pinkish slate	Light dull pink	Dull pink
Chattanooga iron ore	Light pinkish slate	Dull pink	Light terra cotta	Light brick red
Red iron ore	Pinkish slate	Dull pink	Terra cotta	Light brick red

(*L. G. Sabin.*)

Specifications of Mortars.

There are no official specifications for mortar in the British Isles, but the Department of Scientific and Industrial Research has issued in *Building Research Board Report No.* 9—1927 (p. 385), the following tentative specifications:

Quicklime for mortar—quicklime, Class A or B.

Hydrated lime for mortar—hydrated lime, Class A or B. Hydrated hydraulic lime is *not* suitable.

Hydraulic lime for mortar—hydraulic lime, C2 or C3.

Mortar for structural work (*i.e.* exclusive of plaster, stucco, and parging) to consist of a plastic mixture of lime, putty, or hydrated lime, sand, and water, with or without pozzolanic material. The *lime* for work above ground-level should be Class C1, or preferably C2 or C3 (see table), and for below ground or water level, C2, C3, or a lime-cement mixture. Lime and hydrated lime of Class A should not be used without a pozzolanic material in such proportion that the lime, when mixed with three parts by weight of sand and the proposed proportion of pozzolanic material, shall have a minimum tensile strength of 50 lbs. per sq. in. The pozzolanic material may replace an equal weight of sand up to one-third of the weight of sand stated above. Mixtures containing less sand must have higher tensile strengths (see the *Report*). The *sand* shall not pass more than 10 per cent. of its weight through a Standard I.M.M. No. 80 sieve, and the proportions of particles of different sizes shall be well distributed. In lieu of sand, hard rock, slag, clean bricks, clean clinker, coal-cinder of approved origin or hard-burned clay, may be used if crushed so that all the material passes through a Standard I.M.M. No. 5 sieve. The *water* must be free from unusual proportions of dissolved salts and gases and must not be brackish.

In structures below ground or water level and subject to load, not more than one volume of sand shall be used to one volume of hydraulic lime. In structures above ground-level, subject to load, two volumes of sand may be used to one volume of hydraulic lime. In structures above ground not subject to load and in work of less importance, three volumes of sand to one volume of hydraulic lime may be used. Pozzolanic material may replace part of the lime by agreement with the architect.

CONCRETES.

Concretes are very coarse mortars, containing stones, gravel or ballast, sand, and a binding agent such as lime or cement. They are known by various names, according to the binding agent employed.

Lime concrete usually consists of a mixture of 1 part lime, 4 parts gravel, and 2 parts sand.

The best way of making lime concrete is by mixing together unslaked or hot, fresh-ground lime with sand and ballast, in the required proportions, on a stone, brick, or wooden floor, turning it over twice to dry, and then, as it is shovelled to a third heap, adding from the rose of a watering-can sufficient water only to slake the lime and make the ingredients cling together in a pasty mass, turning it over well as the water is added. Generally speaking, it will take about ¾ gallon to each cubic foot of ballast, but much will depend on the nature of the concrete and the dryness of the ballast employed.

Although frequently employed, fat or quick lime should not be used for lime concrete exposed to the weather, only hydraulic lime being suitable.

Beton is a French term, originally applied to lime concrete, but now used for any kind of concrete.

Cement concrete (more commonly known as *concrete* or *mass concrete*) is a mixture of stones, gravel or ballast, sand, and cement. Coke, clinker, and other materials are sometimes used to replace part of the stones, etc.

Breeze concrete consists of a mixture of 3 measures of small pieces of coke (preferably gas-works coke breeze), 1 of sand, and 1 of Portland cement. It is cheap, and nails can readily be driven into it. Its great disadvantage is that, in the event of a conflagration, it may take fire and increase the resulting damage instead of diminishing it.

Concrete made of ashes or clinker is sometimes, but erroneously, termed 'breeze concrete.'

Slag concrete is a mixture of crushed slag and Portland cement. It has been reported as having a greater compressive strength, less weight per cubic yard, and greater resistance to impact than concrete made with natural stone, but various failures which have occurred when slag concrete has been used have made many engineers unwilling to employ it. According to Dr. J. E. Stead, these failures were chiefly due to (1) using too little cement, and (2) the use of fine slag contaminated with ashes and other deleterious substances. In Dr. Stead's opinion, pieces of slag of less than ½-in. diameter should not be employed in concrete. Fine slag, when moistened, is liable to evolve sulphuretted hydrogen, which has a very objectionable odour.

Reinforced concrete or *ferro-concrete* consists of a concrete structure, the strength of which is increased by the immersion of steel rods or wires in the concrete (see Section XI, p. 439).

Raw Materials for Concrete.

All concretes consist essentially of a coarse aggregate, a fine aggregate, and a binding material or cement.

The *coarse aggregate* should consist of suitable stone, gravel or ballast, coke breeze,* clinker, or other hard material of ample resistance to crushing, low porosity (absorbing less than 6 per

* On the danger of coke breeze, cinders, and other coal residues, see pp. 393 and 450.

cent. of its weight of water), and not liable to rapid disintegration by frost. The aggregate should be clean, and if necessary it should be washed before use. Except for the largest and roughest work, all the pieces of coarse aggregate should pass through a 2-in. ring.

Gravel is probably used to a greater extent for concrete than any other one material. As it is a natural substance of very variable composition, even in the same deposit, care should be taken to ascertain its suitability before use. Some gravels contain too large a proportion of soft sandstone to be satisfactory and should not be used.

The best gravel is that with the lowest proportion of voids, *i.e.* that which contains stones of many different sizes in suitable proportion. The preliminary washing and screening of the gravel is usually done by the supplier, but the user should test the proportions of pieces of various sizes, the hardness of the softer pieces, and the percentage of voids, so as to ensure a suitable gravel being used.

The *fine aggregate* should consist of sharp, quartzose sand, similar to that used for mortar (p. 384). If trass is added, the concrete will be more resistant to water. Wet sand requires more cement than dry sand. The sand should be free from clay, and if necessary should be washed before use.

The *cement* may consist of hydraulic lime, Portland cement, or one of the cruder cements, such as *natural cement*, *Roman cement*, *Grappier cement*, *slag cement* (*iron Portland cement*), or various mixtures of Portland cement with sand, blast furnace slag (*Montan cement*), iron ore, etc.

Of these, Portland cement is the strongest and most reliable. It should conform to the Standard Specification (p. 403).

The *water* used for concrete should be the same as that for mortar (p. 384).

Proportions for Concrete.

To obtain the best results, the proportion of coarse aggregate (stones, gravel, etc.) should be such that the interstices or voids between its particles are wholly filled with sand, the interstices or voids in the sand being then filled with cement; additional cement, sufficient to cover all the particles of aggregate with an adhesive coating, should also be present.

As the determination of the voids is often troublesome, many engineers adopt definite proportions of coarse and fine aggregate and cement. For reinforced concrete work where very good concrete is required, the proportions are usually 1 of Portland cement to 2 of sand to 4 of coarse material. For a medium quality concrete, such as is used for foundations, walls, arches, ordinary floors and stairs, the proportions should be 1 : 2½ : 5. For heavy walls, retaining walls, piers, and abutments which are subjected to considerable strain, the concrete can, however, be poorer, 1 : 3 : 6. Wherever concrete is employed in large masses, and is subjected to simple compression of small intensity, or is used for a backing for brickwork or masonry, or for covering the site of a building to prevent the damp rising from the soil, a still poorer concrete can be used of 1 : 4 : 8. In such work and in foundations large stones (commonly called 'plums') can often be placed by hand in the mass some little distance apart so as to allow them to be surrounded by the concrete; these economise considerably, without greatly reducing the strength of the mass.

For concrete to be used under water, a mixture corresponding to a cement-trass mortar is preferable (see the table on p. 386).

The following table shows the relative weights required to produce the proportions mentioned in the first column; from it the cost of materials per cu. yd. of concrete for various mixtures can be ascertained.

MATERIALS FOR 1 CU. YD. OF CONCRETE.

Based on loose cement weighing 90 lbs. per cu. ft., with an average specific gravity of 3·12 and a cu. ft. of loose, moist, coarse sand weighing 89 lbs. when dried.

Proportions.	Kind of Coarse Material.	Lbs. Portland Cement in 1 cu. yd.	Sand cu. yds. in 1 cu. yd.	Coarse Material, cu. yds. in 1 cu. yd.
1 : 1½ : 3	Shingle (40 per cent. voids)	666	·41	·82
Do.	Broken stone (45 per cent. voids)	697	·43	·86
1 : 1¾ : 3½	Shingle	610	·42	·84
Do.	Broken stone	640	·44	·88
1 : 2 : 4	Shingle	520	·43	·86
Do.	Broken stone	548	·45	·90
1 : 2½ : 5	Shingle	430	·44	·88
Do.	Broken stone	450	·46	·92
1 : 3 : 6	Shingle	364	45	·90
Do.	Broken stone	383	·47	·94
1 : 4 : 8	Shingle	280	·46	·92
Do.	Broken stone	294	·48	·97

This table (p. 389) does not apply to concrete made with a single aggregate, such as Thames ballast, to which no sand is added, but where proper scientifically proportioned concrete is employed, as should always be the case, the values given in the table are accurate and correspond with experience.

In measuring out the materials, the fact must not be overlooked that about 34 cu. ft. of stone, sand, and cement will be required for each cu. yd. of concrete, as the sand and cement occupy the interstices between the pieces of coarse aggregate.

Gravel passing same screens as stone always has less voids than the stone.

Fine sand concrete has smaller weight per cubic foot than coarse sand concrete.

Fine sand plus large aggregate (without cement) gives *smaller* volume than coarse. Fine sand plus large aggregate (with cement) gives *larger* volume than coarse.

Fine sand concrete is easier worked than coarse sand concrete for equal amounts of sand.

The finer the aggregate the more deleterious material and air it carries with it into the concrete.

The finer the sand the less should be used.

Mixing Concrete

Concrete may be mixed by hand, as described on p. 387, but large quantities are preferably mixed by machinery. Various types of mixing machines have proved satisfactory, but the ones most largely used consist of a cylinder, fitted with baffle plates internally and mounted on trunnions so as to be revolved rapidly.

COMBINED TABLE OF WORKING CAPACITIES AND LEADING DIMENSIONS OF CONCRETE MIXERS.

No. of Mixer.	00	0	1	2	3	4	5
Nominal batch capacity (cu. yds.)	0·08	¼	½	¾	1	1½	2
Hopper batch (cu. ft.)	3	6·75	13·5	20	30	56	80
Approximate output per batch of broken stone (cu. ft.)	2	5	11	16	24	38	54
Approximate output per batch of ballast or gravel (cu. ft.)	2·25	5·5	12	18	27	42	60
Approximate output per 10-hour day of broken stone (cu. yds.)	22	55	118	180	270	415	592
Approximate output per 10-hour day of ballast or gravel (cu. yds.)	25	60	133	200	300	470	660
Capacity output which has been obtained per 10-hour day (cu. yds.)	50	120	266	400	600	940	1,320
R.P.M. (driving shaft)	174	132	118	122	94	99	90
R.P.M. (drum)	24	18	16	15	14½	14	12
Mean speed of elevator (ft. per min.)	50	50	45	47	50	50	50
Size of pulleys (ins.)	15×2½	18×4½	21×6½	24×7½	32×9½	42×10½	60×12½
Diameter of driving shaft (ins.)	1½	1¾	2⅛	2⅛	2½	3	3½
Capacity of water tank (galls.)	4	10	20	30	35	35	70
Diameter of drum	2′ 9″	3′ 6″	4′ 6″	5′ 0″	5′ 6″	6′ 6″	7′ 6″
Width of drum	2′ 0″	2′ 6″	3′ 0″	3′ 6″	4′ 6″	5′ 0″	5′ 9″
Approximate B.H.P. absorbed (varies according to material)	2	5	7	12	15	20	40

NOTE.—In the production of concrete it is impossible to guarantee definitely the exact amount of mixed material which will be produced from a given quantity of aggregate, owing to the varied nature and size of the stone used, but the above figures, based on actual results, may be taken as approximately correct.

The mixing should be as rapid as possible, consistent with thoroughness, as the concrete should be placed in position before it has begun to set.

Remixed concrete should never be used.

For descriptions of mechanical mixers and further information on mixing concrete, see 'Cement, Concrete, and Bricks,' by A. B. Searle (Constable).

Placing Concrete.

In ordinary foundations, the concrete should be deposited in horizontal layers, not more than a foot thick, and care should be taken to cover any joints in one layer by the succeeding one, as the joint between two days' work is always a weak part; moreover, the last-laid layer should be well wetted, to ensure a proper connection with the next.

The chief mistake made in placing mass concrete is in failing to secure sufficient adhesion to the previously placed material. In reinforced concrete there is often a danger of incomplete filling, particularly around the more complex portions of the reinforcement; this may be minimised by the use of a wetter mixture or by efficient damping or vibration. If the latter is effected mechanically, care must be taken not to continue it so long that the setting of the concrete is disturbed.

Fine concrete or cement mortar may be sprayed on to the surface to be covered by means of a *Cement Gun* operated by compressed air (see p. 394). This method is particularly effective for 'plastering' with cement mortar, or in the construction of walls about 2 ins. thick on a basis of expanded metal or wire netting.

As the right placing of concrete is of great importance, it may be desirable to consult one or more text-books on the subject.

Various articles formerly made by carving stone are now made of concrete at a much lower cost (see 'Artificial Stone,' p. 407).

Volume of Concrete.

The volume of a mass of concrete is much less than that of the separate ingredients, owing to the voids or interstices between the particles of coarse aggregate being occupied, in the concrete, by the sand and cement. Roughly,

Mixture.	Volume of coarse aggregate.	Volume of concrete.
1 : 2 : 4	1 cu. yd.	1·07 cu. yd.
1 : 2½ : 5	1 cu. yd.	1·02 cu. yd.
1 : 2 : 8	1 cu. yd.	1 cu. yd.

1 cu. ft. of loose Portland cement will make about :—

4 cu. ft. of 1 : 2 : 4 concrete.	6 cu. ft. of 1 : 3 : 6 concrete.
5 cu. ft. of 1 : 2½ : 5 „	7½ cu. ft. of 1 : 4 : 8 „

Weight of Concrete.

The weight of 1 cu. ft. of concrete varies with the proportion and nature of the aggregate used.

The average weight of 1 : 2 : 4 concrete, using sand as fine aggregate and one of the following coarse aggregates, is: coke breeze, 100 lbs. per cu. ft.; clinker, 110 lbs. per cu. ft.; sandstone, 120 lbs. per cu. ft.; bricks, 125 lbs. per cu. ft.; limestone, 135 lbs. per cu. ft.; shingle, 145 lbs. per cu. ft.; granite, 145–150 lbs. per cu. ft.

Strength of Concrete.

The proportion of water used in mixing concrete is much more important than is commonly realised, as it has a great influence on the strength of the concrete. According to D. A. Abrams,

$$\text{Compressive strength of concrete in lbs. per sq. in.} = \frac{14{,}000}{7x},$$

where x is the ratio (by volume) of water : cement. If this is the case, a comparatively small excess of water must have a very harmful effect. The American practice of using a very wet mixture is not to be recommended where great strength is required.

Too little water, as well as a large excess of water, are adverse to the production of strong concrete.

Concrete containing too much cement may be weaker than that containing less cement.

Badly mixed concrete is weak in some parts and strong in others.

The addition of 10 per cent. calcium chloride to the mixing water increases the strength of mass concrete, but cannot be used for reinforced concrete as it corrodes the reinforcement.

The compressive strength of most concretes, as commercially made, can be increased 25 to 100 per cent. or more by employing rigid inspection, which will ensure proper methods of fabrication of the materials.

For equal working consistency and equal cement, gravel concrete is as strong as stone concrete.

30–40 per cent. higher strengths are obtained with 3·16 in. cubes than with 6 in. cubes.

Small cubes are more uncertain and inconsistent in the strength values than larger cubes.

The requirements of the London County Council and the Specification drawn up by a Joint Committee of the Royal Institute of British Architects, District Surveyors' Association, Institute of Builders, Institute of Municipal and County Engineers, War Office, Admiralty, London County Council, and the Concrete Institute, call for a crushing strength of 1,800 lbs. per sq. in. at 28 days and 2,400 lbs. per sq. in. at 90 days, for a 1 : 2 : 4 mixture.

The figures 1,400 at 28 days and 2,000 at 90 days more nearly represent the results obtained under ordinary working conditions, with an aggregate like Thames ballast.

The strength of concrete continues to increase long after it has been made, and is by no means at its maximum after 90 days.

According to the U.S.A. Bureau of Standards, if S_{28} be the strength of concrete at 28 days, and S_7 the strength at 7 days, then :

$$S_{28} = S_7 + 30\sqrt{S_7}.$$

Weather conditions have a marked effect on concrete work. Heat hastens the hardening process; cold delays it. The effect of cold becomes noticeable when temperatures fall below 50° F., and becomes more marked with lower temperatures. As a rule, concrete does not show any serious effects from having once been frozen if, after it thaws out, it is not again frozen until early hardening is complete. It is better, however, to protect the concrete from freezing for from forty-eight hours to four or five days, depending upon the degree of the cold, rather than to expose it to the possibility of freezing. Warmth and moisture are necessary to the proper hardening of concrete.

It is especially important to keep concrete thoroughly saturated with water for at least a week, as if it is allowed to dry too soon, it never develops its full strength.

Effect of Addition of Pigments.

Trials have recently (1927) been made by the British Portland Cement Association to test the effect on the strength of concrete produced by the addition of pigments. The tests were made, at the age of seven days, on briquettes 2¼ in. cube. Red oxide of iron produced no effect; green chromium oxide, ultramarine blue, yellow ochre, and manganese dioxide slightly reduced the strength. Carbon black reduced the strength very noticeably.

Effect of Sea-water on Concrete.

The effect of sea-water on concrete depends on the permeability of the latter, dense, waterproof concrete being largely resistant to sea-water, whilst porous concrete corrodes readily. For exposure to sea-water, the concrete should contain a considerable proportion of trass, or equivalent, a suitable mixture consisting of 1 measure of trass, 1 of Portland cement, and 2½ to 3 measures of aggregate. The aggregate should consist of coarse and moderately fine particles, in such proportions as to produce a concrete of maximum impermeability.

Condensation on Concrete.

Interior plastering will prevent condensation on any sort of concrete. A good method is to cover the interior of the concrete walls with ordinary plasterer's lime and sand mortar (about 3 of sand to 1 of lime), lightly keyed in the usual manner to receive the final setting coat. This coat must be left until strong pressure from the thumb makes no impression. It may be necessary to leave it for several days, according to the weather. The finishing coat of lime, putty, sand and plaster of Paris may then be applied with safety. Immediately before this latter coat sets, it should be finished with a soft hair brush, which produces a granular surface, a further method of absorbing moisture.

The best and most easily obtainable finishing coat is the following :

3 parts of lime putty or chalk lime; 6 parts of washed sand; 1 part of plaster of Paris. It gives a good finish, sets well, and works quickly.

Instead of making dense concrete for walls, it is much cheaper and more hygienic to use a very porous concrete, and to render it with a cement waterproofed exterior coating, for the fiercest driving rain cannot penetrate a ¾-in. pudloed cement rendering. Porous concrete, like all porous materials, has also the quality of retaining the heat engendered in the room, thus giving a warmer dwelling. Therefore porous concrete has the dual advantage of assisting the absorption of condensation and of conserving heat.

Preventing Scaling and Cracking of Concrete.

The best method of preventing *scaling* is to keep the concrete thoroughly wet for a week or more after it has been placed, the surface being protected with wet cloths, sawdust, or other porous material. The surface may also be covered with about 2 in. of calcium chloride.

Cracking may be prevented in the same manner, except in those cases where insufficient allowance has been made for stresses due to load or to expansion.

Permeability of Concrete.

The Swiss Hydraulic Association Report (issued in 1922) states that only highly plastic loams of very fine grain can be recommended for making the concrete or gravel used in lining canal bottoms and rendering walls impermeable, and is in any case only likely to form a strong protective coating on surfaces which are always under water; layers which are exposed to the action of the sun rays, rain, and frost are apt to develop cracks. Rammed loam, even when mixed with sand or gravel, will crack in drying, and cannot be much improved by subsequent wetting. But loose gravel, suitably treated with loam water in such a way that the loam particles settle first in the lower gravel layers, will harden to a stony mass, not liable to cracking. The amount of water percolating per day through ordinary concrete under a pressure of 5 atmospheres was found to diminish gradually.

Colouring Concrete.

A method of producing colour in concrete has been introduced, in which coloured ground glass is mixed with the cement, which gives an unlimited variety of patterns and colour effects. It has been suggested that this concrete be used for brightening roads, pavements, and house fronts, and, since the colour will not fade, it will eliminate the drab appearance of buildings and highways so prevalent in big towns.

Rust Prevention in Reinforced Concrete.

Various methods have been proposed for preventing the reinforcement from rusting. Under ordinary circumstances, no special precautions are necessary, and if the corrosion is due to electric currents no application of chemicals can be satisfactory.

The use of galvanised reinforcement has been stated to prevent rusting, and also to secure greater adhesion between the metal and the concrete.

Steel cannot rust when embedded in a strongly alkaline substance. In the setting of Portland cement concrete a relatively large quantity of caustic lime (lime hydrate) is produced, and the concrete becomes saturated with it. This is strongly alkaline, and by decreasing the number of hydrogen ions present makes any corrosion of the steel impossible.

Coal Residues in Concrete.

A Joint Committee appointed by certain institutes and associations to investigate the subject of the use of coal residues in concrete, has issued a report * giving a summary of the objections and dangers attendant on the use of such, especially in contact with steel.

The objections to this material arise mainly under two headings: (1) The corrosive effect on the steel; (2) expansion.

The report gives in Part 1 the existing laws in Great Britain. Part 2 gives the results of an exhaustive examination as to corrosion, and Part 3 as to expansion. Part 4 contains personal commentaries arising on the two foregoing, together with practical experience of various authorities and individuals on the subject.

Reg. No. 151 of the London County Council bye-laws, which apply only to concrete in reinforced concrete construction, gives a list of 'Prohibited Materials' (see page 450):—

There is no regulation whatever as to material used in other concrete in contact with steel.

Treatment of Dusty Concrete Floors.

Various hardening solutions are sold for preventing concrete floors from being dusty, and to give a better wearing surface. The composition of most of these solutions is a trade secret.

Two excellent hardening solutions are the following:—

(1) Mix 1 part silicate of soda to 4 or 5 parts water. Brush or sprinkle the floor well with this solution, and allow to dry. Then, after 4 hours and within 24 hours, wash clean with clean water. Repeat the treatment three times.

(2) The following solution has been recommended by *Concrete*:—

To 10 gallons of cold water add 1 fluid ounce of sulphuric acid. Heat the liquid to boiling point and stir in 25 lbs. of sulphate of alumina. Let the solution cool and strain. Sweep and

* Report of Joint Committee of R.I.B.A., Inst. C.E., and other Institutions in conjunction with the Association of Floor Constructors, 53 Victoria St., London, S.W. 1. Price 1*s.* 6*d.*, postage 1½*d.*

thoroughly wash the concrete surface. After drying, lay on the hardener in four applications of varying strength, ranging from 30 per cent. solution and 70 per cent. water, up to 100 per cent. solution. The above quantity suffices for the complete treatment of an area of from 50 square yards to 100 square yards, depending upon the porosity of the surface treated.

(See also p. 771.)

To Waterproof Concrete.

Concrete may be made waterproof by taking care to use such proportions of the various ingredients as will produce a dense and impermeable material.

As this is not always practicable, it is often preferable to add a waterproofing agent prior to mixing the various ingredients. The chief of these additions are:

(1) *Oil* or a solution of paraffin wax in benzoline or other cheap volatile solvent.

(2) *Insoluble soap* made by impregnating the concrete with a solution of alum, and afterwards with a solution of soap. Suitable proportions are:—

One part of cement, 2 parts of sand, ¾ lb. of dry powdered alum to each cubic foot of sand. These are mixed and dried, and to them is added water in which has been dissolved ¾ lb. soap to each gallon. This, it is said, is nearly as strong as ordinary cement, and is quite impervious to water, and does not effloresce. For a wash, a mixture of 1 lb. of lye and 2 lbs. of alum in 2 gallons of water is often used.

(3) *Water-glass* (silicate of soda) applied in the form of a wash or spray, several applications being necessary at suitable intervals.

(4) *Special mixtures* sold under various trade names, such as Pudlo, Waterex, Prufitol, Prufit, etc.

Hadfields (Merton) Ltd., Western Road, Merton, supply a preparation called PRUFITOL for applying by brush or spraying machine to porous brick, stone, cement, or stucco work, which will keep out driving rains.

The integral system of incorporating a waterproofing agent with the cement and aggregate may be performed by mixing BAREAU cement waterproofing powder with the dry cement, or PRUFIT cement waterproofing paste with the gauging water.

'CEMENT GUN' MACHINE FOR APPLICATION OF HYDRAULIC MATERIALS.

In this machine, sand and cement, previously mixed dry, are placed in a hopper, from which the mixture is automatically fed through an air-locking device and hose to a nozzle, where water is automatically added, and by the aid of compressed air the 'Gunite' (as the mixture is called) is sprayed in a perfectly hydrated state on the surface to be coated.

The propelment of the material by the pneumatic pressure against the surface to be covered, ensures a very dense, hard, and durable mortar. It also means the elimination of heavy and expensive forms, for the material builds up against the backing to a considerable thickness without losing its high quality.

Tests showed that the modulus of elasticity of 'Gunite' was at the end of ninety days, on one part cement to three parts sand mixture, 4,705,000; on one part cement to two-and-a-half parts sand mixture, 4,760,000.

On prisms made of mixtures consisting of one part cement and three parts sand, at the end of ninety days an average ultimate compressive strength of 4,000 lbs. per sq. in. was obtained; on prisms consisting of mixtures containing one part cement to two-and-a-half sand, 4,600 lbs. per sq. in.

It was found that the adhesion between new and old surfaces was absolutely permanent.

'Gunite' has a very high resistance to salt water action.

The cement gun itself is supplied by the *Cement Gun (Great Britain), Ltd., 53 Victoria St., Westminster, S.W.* 1, whose firm also undertake constructional and repair work of a kind for which the cement gun is peculiarly suitable.

The cement gun is now used regularly in twenty-five collieries in this country, and many effective repairs to reservoirs, various concrete structures, and even to damaged factory chimneys, have been made by its use, under conditions where no other method could have been so satisfactory.

CONCRETE BLOCKS, TILES, AND OTHER ARTICLES.

In addition to masses of concrete moulded *in situ*, large quantities of blocks, tiles, garden vases fencing posts, troughs, and other articles made of concrete or of cement-sand mortar (p. 383) are in regular use, and afford clear proof of the manifold uses of this material.*

For *Concrete floors*, see p. 773.

* For further details see *Concrete and Constructional Engineering, Everyday Uses of Portland Cement*, and various text-books on concrete.

Asbestos Building Sheets.

The use of sheets composed of Portland cement, asbestos, and a neutral filler, has greatly increased during recent years. These sheets require costly plant for their production, so that it is usually necessary to buy them ready-made. They are usually sold under various trade names.

Plain sheets are used for walls and roofs ; corrugated sheets are largely used in place of those made of corrugated iron. Small sheets are used instead of roofing tiles.

See also pp. 323–324.

CEMENTS.

Cements are materials used for uniting other materials or articles so that they adhere, permanently. There are many kinds of cements, but the more important ones may be grouped, as follows :

(1) Mastic Cements.

Mastic cements, in which the chief ingredient is a gum (such as resin, shellac, wax, or rubber), boiled oil, or bituminous material to which is added a ' filler,' such as sand, rock dust, or other inert powder. If boiled oil is used red lead must usually be added as well as the filler. Useful mastic cements include :

A *strong general cement*, composed of 4 ozs. clear gum arabic, 3 ozs. fine starch, 2 ozs. white sugar. Dissolve gum in water ; then dissolve starch and sugar in gum solution ; boil mixture in a water-bath until starch becomes clear. Add a few drops of oil of cloves.

A cement for *fixing paper to metal*, composed of gum tragacanth, 30 grammes ; gum acacia, 120 grammes ; water, 500 cubic centimetres. Dissolve, filter, and add 2½ grammes of thymol in 120 cubic centimetres of glycerine. Add water to make up bulk of 1 litre.

A cement for *attaching rubber to metal*, made by soaking pulverised shellac in 10 times its weight of strong ammonia. The result will be a transparent mass, which becomes liquid in 3 to 4 weeks without the use of water. This fluid makes the rubber soft, but after the evaporation of the ammonia it becomes hard, and impermeable to gases and fluids.

A cement for *joining leather driving belts*, made by mixing 10 parts of bisulphide of carbon and 1 of oil of turpentine, and dissolving in it a sufficient quantity of gutta-percha to form a paste. The pieces of leather to be joined are cleansed from oil and grease by laying a rag upon their surfaces and placing a hot iron upon it. Both pieces are then spread with the cement, and subjected to pressure until the cement has become dry.

A cement for *joining wood to tin*, made by melting 1 part of yellow wax, and stirring in 2 parts gutta-percha chips until quite dissolved, then dissolving therein 2 parts shellac and ½th part boiled linseed oil. Apply the melted cement to the dried surface of wood and tin, and press the two surfaces together for 24 hours.

A cement for *quickly closing leaks in barrels* is composed of tallow, 25 parts ; wax, 20 parts ; lard, 40 parts ; sifted wood ashes, 25 parts. Mix together by heating, and apply to the defective place by means of a heated knife-blade.

Cutlers' cements for fastening the blades of knives into the handles :

1. Rosin, 4 parts ; beeswax, 1 part ; brick dust, 1 part.
2. Rosin, 4 parts ; pitch, 4 parts ; tallow, 2 parts ; brick dust, 2 parts.

An *oil-resisting cement* which will withstand the action of paraffin or petroleum jelly can be made by boiling rosin with one-third its weight in caustic soda and one and two-thirds its weight of water. After cooling, the water separates and can be thrown away, the residue being then made into a plastic state with plaster of Paris.

A cement for *glass* is made (*a*) by dissolving 15 parts caoutchouc in 60 parts chloroform, adding about 35 parts mastic, and allowing to stand without heat for eight days. The cement is elastic and transparent ; additional caoutchouc increases its elasticity ; (*b*) of equal parts of wax and vaseline.

A cement for *marble* consists of four parts finely powdered gypsum and 1 part finely powdered gum arabic. Mix intimately, then with a cold solution of borax make into a mortar-like paste. Use the paste as a cement. Leave the cemented parts for several days undisturbed, so that the cement shall thoroughly set.

Strong cements for *stone*, having little body and applied in layers, consist of :

(*a*) Resin, 8 parts ; wax, 1 part ; turpentine, 1 part. For nitric and hydrochloric acid vapours.

(*b*) Resin, 1 part ; sulphur, 1 part ; fireclay, 2 parts. Sulphur gives great hardness and permanency in resin cements.

Good waterproofing cements are :

(*c*) Resin, 1 part ; wax, 1 part ; powdered stone, 2 parts.

(*d*) Shellac, 5 parts ; wax, 1 part ; turpentine, 1 part ; chalk, 8 to 10 parts.

A strong cement for *metals* (not copper), *porcelain, and glass*, is made of powdered shellac, 1 part ; ammonia water, 10 parts. Allow to stand until solution is effected.

A cement for *leather* may consist of : Asphalt, 1 part ; resin, 1 part ; gutta-percha, 4 parts ; carbon disulphide, 20 parts.

A cement resistant to *acid vapours* may consist of : Rubber, 1 part ; linseed oil, 2 parts ; fireclay, 3 parts.

A *rubber cement* may be made by cutting crude rubber in small pieces and then adding carbon disulphide or benzol, and allowing the rubber to dissolve. Corks and wood are made impervious to water by soaking them in the above solution.

Typical *oil cements*, *i.e.* cements rich in oil, are :

(*a*) A mixture of china clay and linseed oil, made into a stiff paste ; this is useful for sealing to prevent the escape of steam or other vapours.

(*b*) A mixture of lime and linseed oil—commonly known as *putty*.

(*c*) A mixture of red lead or white lead and linseed oil, made into a paste.

All the above become very strong when set, and may, for some purposes, be diluted with powdered glass, clay, graphite, or other inert filler.

(*d*) A mixture of oxide of iron and linseed oil is sometimes used to fill spaces between metals, or holes in metal.

(2) Fish-glues or Liquid Glues.

Fish-glues or liquid glues are made by treating fish bones with acetic acid. *Seccotine* is of this nature.

(3) Casein Cements.

Casein cements consist of casein (milk curd) and an alkali ; they are resistant to moderate heat, water, and oil vapours, but not to acid fumes.

(*a*) Finely powdered casein, 12 parts ; fresh slaked lime, 50 parts ; fine sand, 50 parts ; and enough water to make a thick mass.

(*b*) Casein in fine powder, 1 part ; rubbed with silicate of soda, 3 parts (useful for ground unions standing a moderate heat).

(*c*) White of egg made into a paste with slaked lime. This should be used immediately after being made up.

Unless properly compounded, casein cements deteriorate if kept. The ' First Report of the Adhesive Research Committee, 1922 ' (published by His Majesty's Stationery Office, Adastral House, Kingsway, London, W.C. 2 ; price 4*s.* net), gives the following as satisfactory :

Casein, 78 per cent. ; dry sodium carbonate, 4·5 per cent. ; sodium fluoride, 4·0 per cent. ; freshly slaked lime, 12·5 per cent. ; sodium arsenate, 1·0 per cent. All ingredients must be ground to pass through a 90-mesh sieve. For use, mix the cement with water ; if mixed with a weak solution of gelatine glue the strength will be about 10 per cent. greater. Average breaking stress, about 1,000 lbs. per sq. in.

(4) Bituminous Cements.

Bituminous cements are those in which the chief ingredient is asphalt, tar, pitch, or bitumen (see also *Mastic cements*). Asphalt in benzol is useful as an adhesive for uniting glass for photographic and microscopical uses, also for coating wood, concrete, brick work, steel, etc., where the melted asphalt would be too thick to apply readily. Benzol is the cheapest and most satisfactory solvent. For waterproofing, melted asphalt with a small amount of paraffin added, and in particular cases boiled oil, is used :

(*a*) Refined lake asphalt.
(*b*) Asphalt, 4 parts ; paraffin, 1 part.
(*c*) Asphalt, 10 parts ; paraffin, 2 parts ; boiled oil, 1 part.

} Any of these may be thinned with hot benzol or toluol.

(*d*) Pitch, 8 parts ; resin, 6 parts ; wax, 1 part ; plaster, ¼ to ½ part.
(*e*) Pitch, 8 parts ; resin, 7 parts ; sulphur, 2 parts ; stone-powder, 1 part.

} Useful for uniting slate slabs and stoneware.

The following are plastic cements for *caulking* :

(*f*) Pitch, 3 parts ; shellac, 2 parts ; pure crude rubber, 1 part.
(*g*) Pitch, 1 part ; shellac, 1 part ; rubber substitute, 1 part.

} Mix by melting.

(5) Water-glass or Silicate Cements.

Water-glass cements consist chiefly of an inert material mixed with water-glass (syrupy silicate of soda). Typical mixtures are :

(*a*) A paste made of silicate of soda and asbestos.
(*b*) A paste made of silicate of soda and fine sand.
(*c*) A paste made of silicate of soda and powdered glass.
(*d*) Equal parts of water-glass and zinc oxide, or iron oxide or litharge.

Sometimes lime is added to make the cement set more rapidly, a typical mixture being silicate of soda 50 parts, asbestos 15 parts, slaked lime 10 parts.

(6) Calcareous Cements.

Calcareous cements, such as hydraulic lime, Roman cement, the various natural cements, and Portland cement. *Hydraulic limes* are described on p. 384. *Roman cement* is a brown, quick-setting cement, made by calcining and grinding the septaria or calcareous claystones found in London clay. It is one of the natural cements.

Natural cements (so-called) are made by heating natural mixtures of clay and limestone to redness, so as to effect a combination between the lime and clay similar to that in Portland cement. Such natural cements are comparable to a mixture of Portland cement with either an excess of lime or clay, as the case may be. Their composition is irregular and uncertain, so that they are inferior to Portland cement for all work of importance. They can be greatly improved by adjusting their composition by the addition of limestone or clay, as may be required, and heating until the mixture is clinkered. Such adjustment greatly increases the cost of production, but produces a Portland cement.

Portland cement is described on pp. 401–405.

Ciment Fondu or *Electric Cement* is composed of lime and alumina, heated to incipient fusion in an electric furnace and afterwards ground to powder ; it has a composition as follows : silica, 10 per cent. ; alumina, 42 per cent. ; lime, 42 per cent. ; iron, 6 per cent. ; as compared with Portland cement, which has an average analysis of silica, 23 per cent. ; alumina, 7 per cent. ; lime, 64 per cent. ; iron, 4 per cent. ; and 2 per cent. other matter ; in each case iron figures only as a non-deleterious impurity. The cement attains a high strength very much more quickly than Portland cement, and is said to be unaffected by sea-water. Its use is increasing rapidly, notwithstanding the results of investigations by German chemists, which seem to show that Portland cement can be made having similar properties, but far more cheaply.

(7) Gypsum Cements.

Gypsum cements include plaster of Paris, Keene's cement, Parian cement (see *Plasters*, p. 406).

(8) Oxychloride Cements.

Oxychloride cements, of which the best known is Sorel cement, are composed of dense magnesia and syrupy magnesia chloride, or zinc oxide and zinc phosphate. They depend very largely on the use of suitably calcined oxides and on the chloride or phosphate being of suitable density, so that it is wise to follow the manufacturers' instructions. Typical mixtures are :

(*a*) Magnesium oxide, 2 parts ; magnesium chloride, 1 part ; water to make a paste.
(*b*) Zinc oxide, 2 parts ; zinc chloride, 1 part ; water to make a paste.

Magnesium oxychloride cements may be diluted with fillers such as china clay, or may be coloured with mineral pigments such as ultramarine, ochre, etc. They are largely used for forming *floors*.

(9) Flour and Starch Pastes.

Pastes made of flour or starch are largely used for fastening papers to each other or to walls, and ceilings. A good paste for this kind is made :

(*a*) By dissolving ¾ oz. of alum in a quart of water, adding flour to make it the consistence of cream, then bring it to a boil, stirring it all the while, then add a teaspoonful of powdered resin, and also a few drops of oil of cloves. This paste will not turn sour by keeping.
(*b*) Flaxseed makes a very tough cement, but does not withstand water.
(*c*) Flour and molasses made into a stiff paste is sometimes used.
(*d*) A permanent cement for ordinary temperatures, impervious, but attacked by condensing steam and nitric vapours, consists of a mixture of stiff flour paste and concentrated zinc chloride solution.
(*e*) Dextrine and fine sand, made into a paste with water is sometimes used in foundries.
(*f*) A paste made by mixing dextrine with water is useful for fastening papers. The ' gum ' on labels and stamps is of this nature.
(*g*) *Rice glue* is made by mixing rice flour into a paste with cold water and then heating it, with constant stirring, until it boils.

(10) Celluloid Cements.

For celluloid goods a cement may be made :

(*a*) By dissolving one part of camphor in four parts of alcohol, and adding an equal amount of shellac to the camphor solution. The cement must be applied hot. The celluloid goods must have the fractured surfaces kept pressed together until the cement has cooled.

(*b*) Dissolve celluloid in amylacetate to make a thick jelly. This cement may be used cold, It can also be used to cement china, silk or cotton fabrics, leather, etc.; it is absolutely waterproof. A piece of paper smeared with the jelly makes an excellent sticking-plaster.

Celluloid cements for various purposes are on the market; they are sometimes known as *dopes*, and are chiefly of the (*b*) class.

(11) Refractory and Fire-resisting Cements.

Refractory cements are those which are specially resistant to heat and are used in repairing furnaces, etc. (see p. 430). Other refractory cements include:

(*a*) A mixture of fireclay and plumbago made into a paste with water; it may also be used to coat glass to protect it from flame.

(*b*) Equal parts of flour and lime and half a part of fireclay are made into a paste with water; white of egg is sometimes added. This is a fire-resistant cement and not highly refractory. It is sometimes used for patching retorts.

For patching *crucibles and melting pots*:

(*c*) Sift brickdust and mix with equal quantity of red lead: rub together with boiled linseed-oil, which has been mixed with sand to a stiffness of cement. In covering dishes apply the paste first, then apply sand and heat.

(*d*) Put freshly slaked lime with concentrated solution of borax, apply with a stiff brush and allow to dry. Upon heating, a fused glaze is obtained.

(*e*) Litharge, 6 parts; fresh burnt lime, 4 parts; china clay, 2 parts; and mix with cold linseed-oil. This is only fire-resisting.

(*f*) Alumina, 1 part; sand, 4 parts; slaked lime, 1 part; borax, ½ part; water to make a stiff paste.

(*g*) A fire-resisting cement for *iron pipes*: Iron filings, 10 parts; brick clay, 20 parts; china clay or fireclay, 15 parts; common salt, 8 parts. The salt is dissolved in water and mixed with the other ingredients.

(12) Glues

(1)

Commercial glue is made by boiling animal flesh, bones, etc., in water, removing fat and other impurities and concentrating by evaporating the greater part of the water. It is usually sold in thin cakes or flakes, or in the powdered form, and should be kept in a dry place so that it will not mould. In whatever form it is purchased, the glue should, prior to use, be soaked in cold water for several hours before it is heated. The water used for this purpose ought to be distilled; the drip from steam radiators may be utilised, if desired. The use of distilled water guards against chemical action due to impurities in the water, which may contain sulphur, lime, or iron, all of which are injurious to glue. If glue is soaked in hot water it quickly decomposes and its strength is impaired. The amount of water required should be stated by the makers.

Glue cake, ¼ to ½ in. thick, requires about twelve hours to dissolve; it should be broken into small pieces before being put to soak. Flake glue requires from six to eight hours, according to the thickness of the flake. Pulverised glue may be dissolved in from three to three and a half hours. The soaking process should not be hurried.

After being heated, the glue may be diluted with hot water to the required consistency. Glue should never be heated sufficiently to make it boil. It is also bad practice to prepare large quantities of glue at a time, because it loses its strength rapidly after being under heat for about four hours.

A satisfactory glue-heater should have ample accommodations for all the glue that will be prepared at one time, say, six or more glue-kettles. The kettles in which the glue is heated should be made of aluminium, and their size and number should provide for handling glue of various consistencies in sufficient quantities to last not more than four hours. Aluminium is the most suitable metal for glue-pots, because the scum on the glue does not adhere to the inside of the pots, so that they are easily cleaned.

A thermometer should be part of the equipment of a glue-heater, for the temperature at the bottom of a pot 6 ins. deep should not be over 150° F., which will bring the temperature at the top to about 140° F. Higher temperatures than this destroy the adhesive power of glue. The pots should be thoroughly cleaned at least once each day The glue that has become cooled should never be reheated, as reheated glue loses about 50 per cent. of its strength. It is good practice to have a new lot of glue ready for use before the glue already prepared is consumed.

(2)

Glue loses 5 per cent. of its strength for every hour of heating. Laboratory tests have shown a loss of 67 per cent. of strength after twelve hours under heat. Tests of a high-grade glue, made by the United States Forestry Bureau, showed a loss of one-half the glue strength in seven hours at a temperature of 176° F. Glue is strongest at a temperature of 135° F.

Ground glue should be soaked in cold water at least three hours; flake glue, eight hours; and cake glue, twelve hours. After the glue is soaked it does no harm if it stands twenty-four or even forty-eight hours, provided the temperature of the room is not over 70° F. The glue, when placed in the heater, should be melted, not cooked or boiled.

Where glue is used in large quantities the glue apparatus, and, in fact, the whole glue room,

should be kept as clean as a dairy. Glue brushes, when not in use, should be kept in a weak solution of carbolic acid in order to destroy the bacteria which may accumulate in them; the bacteria in glue which is left overnight will contaminate fresh glue the next morning and decrease its strength probably 50 per cent. Bacterial action is one of the reasons why glue loses its strength under prolonged heat.

The glue-heater should be of a type which will furnish a constant supply of fresh glue in its strongest condition. Glue should be mixed and used in small batches so that it may be consumed in four hours from the time it is melted. Otherwise it deteriorates rapidly. The individual glue-pot should be of a size, therefore, to contain only a four-hour supply of glue.

A little powdered *chalk* added to common glue is said to increase its strength.

Core Glues.

The following are largely used:

(*a*) Dextrine, 1 part; sand, 10 parts; water to form a thick paste.

(*b*) Anthracite coal powdered with molasses added to thick paste.

(*c*) Resin, partly saponified by soda lye, 1 part; flour, 2 parts; sand 4 parts; with water as a diluent.

(*d*) Powdered glue, 1 part; flour, 4 parts; sand, 6 parts; water.

Waterproof Glues.

(*a*) A very cheap waterproof glue is made by melting common glue with the smallest quantity of water possible. Add to this, by degrees, linseed-oil made 'drying' by boiling it with litharge. While the oil is added the ingredients must be well stirred so as to mix them thoroughly. Another plan: Take a very thick solution of glue 10 parts, linseed-oil varnish 5 parts, and litharge 1 part. Boil these together for ten minutes and use while hot. The additional ingredients have no effect whatever on the adhesive properties of the glue, which is rendered entirely waterproof.

(*b*) Gum shellac 3 parts and india-rubber 1 part by weight, these constituents being dissolved in separate vessels in ether, free from alcohol, subject to a gentle heat. When thoroughly dissolved, the two solutions are mixed and kept for some time in a vessel tightly sealed. This glue resists the action of water, both hot and cold, as well as most acids and alkalis. If the glue is thinned by the admixture of ether, and applied as a varnish to leather along the seams where this has been sewn together, it renders the joint or seam water-tight, and almost impossible to separate.

(*c*) A formula for waterproof glue giving a degree of water resistance comparable with the best casein glues is as follows: A grade of animal glue suitable for woodworking is soaked in such a proportion of cold water as is required by its grade and is then 'melted' in the glue-pot. Paraformaldehyde of 50-mesh fineness, and oxalic acid are then added in the proportions respectively of 10 per cent. and 5 per cent. on the weight of dry glue taken. As soon as the oxalic acid has dissolved, the glue is ready to use and will remain in a suitable condition for application for seven to nine hours at a temperature not exceeding 45° C.

Fire-resisting Glue.

Mix 4 parts linseed-oil and 1 part quicklime, then boil slowly until a paste. Spread the mixture on a plate and dry in the open air. It will become very hard, but can be dissolved over a fire, or in a glue-pot like common glue. This composition really is not glue at all, but a sort of waterproof cement that is extremely tenacious. (See also *Refractory Cements*.)

Liquid Glues.

(*a*) 1 part glue, 1 part water, 1 part vinegar. Dissolve by gentle heat, and then add 1 part alcohol.

(*b*) White glue, 16 ozs.; dry white lead, 4 ozs.; rain-water, 2 pints. Dissolve by gentle heat and then add alchohol, 4 ozs.

(*c*) Clear gelatine, 100 parts; glue, 100 parts; alcohol, 25 parts; alum, 6 parts. Mix whole with 200 parts of 20 per cent. acetic acid, and heat in water-bath for 6 hours.

Marine Glues.

(*a*) India-rubber, 4 parts; coal-tar naphtha, 34 parts; powdered shellac, 64 parts.

(*b*) Glue, 12 parts; dissolve in water; add yellow resin, 3 parts. When dissolved, add turpentine, 4 parts. This glue is inferior in quality to (*a*).

Glue for Parchment.

Parchment shavings, 1 lb.; water, 6 quarts. Boil until dissolved then strain, and evaporate to consistency of ordinary glue.

Glue for Leather.

A little tannin, added in sufficient quantity to hot glue until it has the appearance of white of egg, is said to improve it when used for cementing two pieces of leather together.

Glue for Fastening Leather to Metal.

A recipe for fastening leather to metal is: Digest one part of coarsely crushed gall-nuts with 8 parts of distilled water about six hours, and filter through linen. Then pour 1 part of

cold water over 1 part of glue, leave it to stand for twenty-four hours, and heat the whole, whereby a concentrated glue solution is obtained. Now coat the leather with the warm gall-nut extract, bring the glue solution on the roughened and warm metal, lay the leather on it, press it firmly, and allow it to dry in the air. The leather will adhere so firmly to the metal that it cannot be separated without tearing it.

To cement leather to iron pulleys, roughen the pulley with an old file; glue the leather to the pulley with glue, 1 part; glycerine, 1 part.

GLUE FOR FASTENING CLOTH TO WOOD.

Equal parts of glue and isinglass made into a liquid like common glue, and then thickened with tannin, has been recommended for fastening *cloth to wood*.

(13) Metallic Cements.

Rust Joint Cements.

(*a*) *Quick setting.*—1 lb. sal-ammoniac in powder; 2 lbs. flowers of sulphur; 80 lbs. iron filings or borings. Mix to a paste with water.

(*b*) *Slow-setting.*—2 lbs. sal-ammoniac in powder; 1 lb. flowers of sulphur; 200 lbs. iron filings or borings. Mix to a paste with water. This cement is better than the quick-setting, if the joint is not required for immediate use.

Cements for Steam Boilers, Steam Pipes, etc.

(*a*) *Soft.*—Red or white lead in oil, 4 parts; iron borings, 2 to 3 parts.

(*b*) *Hard.*—Iron borings and salt water, and a small quantity of sal-ammoniac with fresh water.

(*c*) 5 lbs. Paris white, 5 lbs. yellow ochre, 10 lbs. litharge, 5 lbs. red lead, 4 lbs. black oxide of manganese. The whole is to be well mixed, and a little asbestos and boiled oil added. This soon becomes nearly as hard as the iron itself.

This cement is also excellent for making fluid-pressure joints.

(*d*) Take white lead ground in oil, and add dry red lead sufficient to make a stiff putty. Pound this on a block of iron or stone until it becomes soft, continuing to add red lead until the mass will no longer soften or stick to the fingers. Interpose this putty between the flanges of the steam-pipe joints. This will make the best joint known.

For Leaky Iron Pipes.

In cases where caulking, plugging, or other mechanical means of stopping leaks in iron pipes cannot be adopted, a cement made by mixing finely pulverised black manganese oxide and raw linseed-oil into a thick paste will usually be found effective. It should be thoroughly forced into the leak, and the pipe should be kept warm enough to absorb or remove the oil from the manganese. In about twenty-four hours the cement should be as hard as the iron, and should adhere thoroughly, so that leakage does not again occur.

For Iron Stove-pipes and for Filling Cracks in Stoves.

Pulverised brick clay, 1 part; wood ashes, 1 part. Add a little salt, and mix to consistence of syrup.

To Fasten Iron to Iron.

(*a*) Sulphur, 6 parts; white lead, 6 parts; borax, 1 part. When composition is to be applied, wet with strong sulphuric acid and smear on surfaces, which must be at once pressed together. Allow one week to dry.

(*b*) Equal parts of litharge, red lead, and sufficient glycerine to form a paste. This cement is said to be waterproof, fireproof, and to resist the action of alkalies.

For Holes in Castings.

Flowers of sulphur, 1 part; sal-ammoniac, 2 parts; iron filings or borings, 80 parts. Mix to a thick paste with water.

To Fasten Iron to Stone.

(*a*) Iron filings, 20 parts; plaster of Paris, 50 parts; sal-ammoniac, 3 parts. Mix to a thin paste with weak vinegar.

(*b*) Sulphur, 1 part; pitch, 1 part.

To Fasten Iron to Marble.

A cement for making repairs on switchboards when iron or other metal has to be fastened to marble may be made from 30 parts plaster of Paris, 10 parts iron filings, and ½-part sal-ammoniac. These, states the *American Machinist*, are mixed with acetic acid to form a thin paste. This cement must always be used immediately after being mixed.

(14) Miscellaneous Cements.

Cement for Mending Rubber Goods.

Bisulphide of carbon, 26 parts; gutta-percha, 2 parts; caoutchouc, 4 parts; fish glue, 1 part.

Composition to make Corks Impervious.

A composition for soaking corks, wood packing, etc., to render impervious to oil vapours, consists of: gelatine, 2 parts; glycerine, 1 part; water, 6 parts.

To Fasten Metal to Wood.

In order to fasten metal to wood without the use of nails or screws, the metal parts may be roughened by the use of dilute sulphuric acid. As soon as the metal has been wiped dry, it is glued on with the best joiner's glue to which a little glycerine has been added. This process is not only adapted for small plates or slabs, but also larger ones, but care must be taken that they touch the roughened wood surface evenly throughout.

To Fasten Metal Letters upon Glass, Marble, Wood, etc.

(*a*) Copal varnish, 15 parts; linseed-oil varnish, 5 parts; oil of turpentine, 5 parts; glue, 5 parts. The glue is dissolved by placing the mixture in a water-bath. When the solution is complete 10 parts of slaked lime are added to it.

(*b*) A good cement for brass and glass can be made by mixing together 2 parts litharge, 1 part white lead, 3 parts linseed-oil, and 1 part gum copal. Use at once.

Portland Cement.

(*Partly contributed by A. B. Searle, Author of 'Cement, Concrete, and Bricks,' etc.*).

Portland cement is usually made from a mixture of chalk, or limestone, and clay ground together to an extreme degree of fineness, and subsequently dried, calcined, and ground to the fineness of flour. Pats or briquettes made from it should, when broken, exhibit a bluish-grey colour in the fracture.

Portland cement * may be obtained quick or slow setting. If there are no special reasons which make quick setting desirable, it is advisable to employ a cement which requires at least two hours to pass from the pasty state into the hard, solid one, as quick-setting cements require exceptional care and skill in gauging: otherwise they begin to set and to spoil before they have been placed in their proper position in the structure.

Most manufacturers add a little gypsum when grinding the cement clinker, and thereby adjust the rate of setting of the cement to suit the user's requirements. If required, the time taken for setting may be increased by exposing the cement to moist air before use, but this is not altogether satisfactory, and it is better to purchase a cement which sets at the desired rate.

Temperature has a very great effect upon the setting time of cement. A sample tested in a temperature of 61° Fahr. and gauged with water at the same temperature took six hours to set hard. The same sample gauged in a temperature of 90° Fahr. and gauged with water of equal temperature set hard in one hour fifty minutes. This is a point which is often lost sight of.

Testing Portland Cement.

The tests usually applied relate to its

- Chemical composition.
- Specific gravity.
- Fineness.
- Crushing or compression strength.
- Tensile strength. (Neat cement.)
- Tensile strength. (Cement and sand.)
- Time of setting.
- Time of hardening.
- Soundness.

The tests suggested by the British Standard Specification (p. 403) are usually regarded as sufficient, but the others are used for special purposes. The following notes indicate the purpose of the various tests:

Chemical Composition.—This test is for the detection of adulterants, or to determine whether certain constituents are present in amounts exceeding that believed to be safe.

Specific Gravity.—This test is designed to detect under-burning and adulteration, and should only be used in connection with the chemical analysis; low specific gravity is often caused by 'ageing' of the cement, so that to reject a cement because of a low specific gravity may be to reject it because it has been well seasoned.

Fineness.—The fineness of the cement is a measure of its cementing value; a fine cement will be much stronger when mixed into a mortar, or it can be mixed with a larger proportion of sand than a coarse one, and yet attain the same strength.

Crushing or Compression Strength.—The most generally comprehensive test of a Portland cement or of a mixture of it with sand would be a determination of the crushing strength of a cube of 4 in. side. Unfortunately, this requires a very costly machine, so that it is usual to substitute a tensile test which can be made much more cheaply.

Tensile Strength is determined in order to gain some idea of the cohesion between the particles. This property is not required in cement and concrete, but as it is roughly proportional to the crushing strength and the tensile strength is easier to determine, it is usually substituted for a crushing test.

A neat briquette rupture is practically a plane section.

* The composition of Portland cements is highly complex. Some further indications of it Cement, are given in 'Concrete, and Bricks,' by A. B. Searle (Constable & Co., Ltd., London).

Time of Setting.—This is determined in order to ascertain whether a cement sets at a rate suitable for a given piece of work. If a cement sets before it has been properly placed in position the work will be defective; hence where there is any doubt, a slow-setting cement should be used. There is no necessary relationship between the time of setting and that of hardening or attaining the maximum strength; a slow-setting cement may harden more rapidly than a quick-setting one, and *vice versâ*.

Time of Hardening.—This is sometimes determined to ascertain what is the minimum time which must elapse before a cement or concrete can be used, *e.g.* if a concrete road is being made, how long will it be before traffic can safely pass over it? The aluminous cements, such as *Ciment Fondu*, set slowly but harden very rapidly.

Soundness.—The object of this test is to hasten the rate at which a cement or concrete attains its maximum strength and to subject it to conditions more severe than are likely to occur in regular use. By thus exaggerating the conditions some idea is gained of the lasting properties of the material much more rapidly than would otherwise be the case. The soundness test is, in some respects, the most important of all, for if a sample passes all other tests satisfactorily yet fails in the soundness test, it is worthless as a constructional material. To pass the soundness test a cement or concrete must retain a *constant volume* under the conditions of the test; failure is revealed by the test-piece cracking, swelling, or disintegrating in any other way.

The actual testing of cement and concrete requires considerable skill. The methods described in the British Standard Specification for Portland Cement (see p. 403) are satisfactory, but require considerable practice before accurate results can be obtained.

Cement Testing Machines.

Tension tests of cement briquettes are generally made on special testing machines of various types, single and compound levers being used. The application of the load at a steady and definite rate is in some cases accomplished by running water or fine shot through a controllable opening into a vessel hanging from the end of the lever; in other cases a travelling poise is caused to move along a graduated lever at a steady speed.

A single-lever testing machine made by Mr. Adie, with a regulated travelling poise D, is shown in fig. 1. The briquette is placed in the clips B, C, and the hand-wheel R serves to tighten the screw attached to the lower clip C, so as to raise the lever F into a position of balance between the stops on the standard E. The poise D may be moved along the lever F, by means of the handle at the end of the beam. For loading at a uniform rate the traverse of the poise is accomplished by the pull of a suspended weight W, regulated by a dashpot cylinder below the pulley 5; the rate of traverse is regulated by adjusting a cock in the piston of the dashpot.

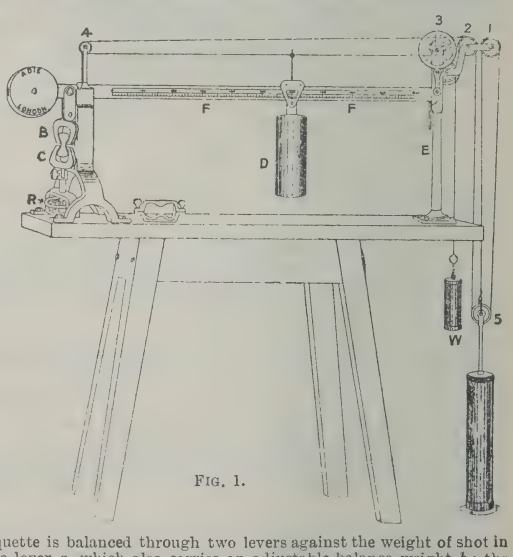


Fig. 1.

Fig. 2, p. 403, shows the form of another of Mr. Adie's cement-testing machines, designed by Dr. Michaelis. This machine, which is of the compound-lever type and is loaded by shot, may be taken as typical of German practice. The briquette *i* is held between clips *d* and *e*, the lower one of which is attached to a straining screw turned by the hand-wheel *f*. The pull on the briquette is balanced through two levers against the weight of shot in the vessel *c* at the end of the lever *a*, which also carries an adjustable balance weight *b*; the combined leverage of the two levers is 50 to 1. The shot runs into the vessel *c* from a reservoir *g*

through a channel, which is regulated at *o* for any desired rate of loading by a lever *n*. The breaking of the briquette and consequent fall of the lever automatically shuts off the supply of shot, and 50 times the weight of shot in *c* gives the breaking load, this quantity being recorded by a spring balance at *l*.

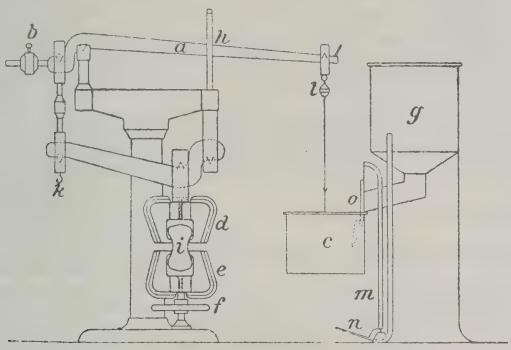


FIG. 2.

BRITISH STANDARD SPECIFICATION FOR PORTLAND CEMENT.*

(No. 12—1925.) (*Abstract.*)

1. The cement shall be manufactured by intimately mixing together calcareous and argillaceous or other silica and alumina bearing materials, burning them at a clinkering temperature and grinding the resulting clinker, so as to produce a cement capable of complying with the specification.

No addition of any material shall be made after burning other than calcium sulphate, or water, or both.

7. The sample or samples shall be tested for :—(*a*) fineness; (*b*) chemical composition; (*c*) tensile strength (neat cement); (*d*) tensile strength (cement and sand); (*e*) setting time; (*f*) soundness; and before any sample is submitted to tests (*c*), (*d*), and (*f*), it shall be spread out for a depth of 3 ins. for 24 hours in a temperature of from 58 to 64 degrees Fahrenheit.

8. The cement shall comply with the following conditions of fineness—100 grammes (or say, 4 ozs.) of cement shall be continuously sifted for a period of 15 minutes on each of the undermentioned sieves and in the order of succession given below, with the following results :—

(1) The residue on a sieve 180 × 180 = 32,400 meshes per sq. in., shall not exceed 10 per cent.
(2) The residue on a sieve 76 × 76 = 5,776 meshes per sq. in., shall not exceed 1 per cent.

The sieves shall be prepared from wire-cloth, and the diameter of the wire for the 32,400 mesh shall be ·00171 in., and for the 5,776 mesh, ·0044 in. The wire-cloth shall be woven (not twilled), the cloth being carefully mounted on the frames without distortion.

9. The cement shall comply with the following conditions as to its chemical composition. The proportion of lime, after deduction of the proportion necessary to combine with the sulphuric anhydride present, to silica and alumina, when calculated (in chemical equivalents) by the formula $\frac{CaO}{SiO_2+Al_2O_3}$, shall be not greater than 2·90 nor less than 2·0. The percentage of insoluble residue shall not exceed 1·5 per cent.; that of magnesia shall not exceed 4 per cent.; and the total sulphur content calculated as sulphuric anhydride (SO_3) shall not exceed 2·75 per cent. The total loss on ignition shall not exceed 3 per cent.

10. The breaking strength of neat cement shall be ascertained from briquettes of the specified shape. The briquettes shall be prepared in the following manner :—

The cement shall be mixed with such a proportion of water that the mixture shall be plastic when filled into the moulds used for forming the briquettes. The cement, gauged as stated, shall be filled into moulds of the form required to produce briquettes of the specified shape, each mould resting upon a non-porous plate. In filling the moulds the operator's hands and the blade of the ordinary gauging trowel shall alone be used. The trowel shall weigh about 7½ ozs.

* By permission of the British Engineering Standards Association.

No ramming or hammering in any form will be permitted, nor shall any other instrument or apparatus other than the trowel before described be employed for this operation. The moulds, after being filled, may be shaken to the extent necessary for expelling the air. Clean appliances shall be used for gauging, and the temperature of the water, and that of the test room at the time the above operations are being performed, shall be from 58 to 64 degrees Fahrenheit. The briquettes shall be kept in a damp atmosphere for 24 hours after gauging, when they shall be removed from the moulds and immediately submerged in clean fresh water and left there until taken out for breaking. The water in which they are submerged shall be maintained at a temperature of between 58 and 64 degrees Fahrenheit (in tropical climates any higher temperature up to 95 degrees Fahrenheit may be substituted). After they have been so taken out and until they are broken the briquettes shall not be allowed to become dry.

Six briquettes shall be tested for breaking strength at 7 days after gauging. The breaking strength shall be the average tensile breaking strength of the six briquettes. The briquettes to be tested shall be held in strong metal jaws of the specified shape, and the load shall be steadily and uniformly applied, starting from zero, and increased at the rate of 100 lbs. per sq. in. of section in 12 seconds.

The breaking strength of the briquettes at 7 days after gauging shall be not less than 600 lbs. per sq. in. of section.

11. The breaking strength of cement and sand shall be ascertained from briquettes of the specified shape. The briquettes shall be prepared in the following manner :—

A mixture of cement and sand in the proportion of one part by weight of cement to three parts by weight of the standard sand hereafter specified, shall be gauged with water, the percentage of water to be used being determined by the formula :—

$$\tfrac{1}{4} P + 2{\cdot}50,$$

where P is the percentage of water required for the preparation of the neat cement briquettes. The mixture thus gauged shall be evenly distributed in moulds of the form required to produce briquettes of the specified shape, each mould resting upon a non-porous plate. After filling a mould a small heap of the mixture shall be placed upon that in the mould and patted down with a standard spatula until the mixture is level with the top of the mould. This last operation shall be repeated on the other side and the mixture patted down until water appears on the surface ; the flat only of the standard spatula is to be used and no other instrument or apparatus is to be employed for this operation. The mould after being filled may be shaken to the extent necessary for expelling the air. No ramming or hammering in any form will be permitted during the preparation of the briquettes, which shall then be finished off in the moulds by smoothing the surface with the blade of a trowel. Clean appliances shall be employed for gauging, and the temperature of the water and that of the test room at the time the above operations are performed shall be from 58 to 64 degrees Fahrenheit (in tropical climates any higher temperature up to 95 degrees Fahrenheit may be substituted). The briquettes shall be kept in a damp atmosphere for 24 hours after gauging, when they shall be removed from the moulds and immediately submerged in clean fresh water, and left there until taken out for breaking. The water in which they are submerged shall be renewed every seven days, and maintained at a temperature of between 58 and 64 degrees Fahrenheit (in tropical climates any higher temperature up to 95 degrees Fahrenheit may be substituted). After they have been so taken out and until they are broken the briquettes shall not be allowed to become dry.

The briquettes shall be tested for breaking strength at 7 and 28 days respectively after gauging, six briquettes for each period. The breaking strength shall be the average tensile breaking strength of the six briquettes for each period. The briquettes to be tested shall be held in strong metal jaws of the specified shape, and the load steadily and uniformly applied, starting from zero, and increased at the rate of 100 lbs. per sq. in. of section in 12 seconds.

The breaking strength of the briquettes at 7 days after gauging shall be not less than 325 lbs. per sq. in. of section.

The breaking strength of the briquettes at 28 days after gauging shall show an increase on the breaking strength at 7 days and shall be not less than the number of lbs. per sq. in. of section arrived at from the following formula :—

$$\text{Breaking strength at 7 days} + \frac{10{,}000 \text{ lbs.}}{\text{Breaking strength at 7 days}}.$$

The standard sand shall be obtained from Leighton Buzzard, be thoroughly washed and dried, and shall pass through a sieve of 20 × 20 meshes per sq. in., and be retained on a sieve of 30 × 30 meshes per sq. in. The sieves shall be prepared from wire-cloth, the wires being ·0164 in. and ·0108 in. in diameter respectively. The wire-cloth shall be woven (not twilled), the cloth being carefully mounted on the frames without distortion.

12. Unless a specially quick setting cement is specified or required, it shall have an initial setting time of not less than 30 minutes and a final setting time of not more than 10 hours.

If a special quick setting cememt is specified or required, it shall have an initial setting time of not less than 2 minutes, and a final setting time of not more than 30 minutes.

The initial and final setting times of cement gauged as above shall be determined by means of the Vicat needle apparatus, in the manner specified.

13. The cement shall be tested for soundness by the Le Chatelier method, in the manner specified.

British Standard Specification for Portland Blast-Furnace Cement.*

(*No.* 146—*Revised* 1926.) (*Abstract.*)

1. This cement shall consist of a mixture of Portland cement clinker and granulated blast-furnace slag in such proportions as the manufacturer may prefer, but in no case shall the proportion of slag exceed 65 per cent.

The Portland cement clinker and the granulated blast-furnace slag shall be thoroughly and intimately mixed, and shall produce a cement capable of complying with the specification.

8. The cement shall comply with the following conditions of fineness :—100 grammes (or, say, 4 ozs.) of cement shall be continuously sifted for a period of 15 minutes on a 180 × 180 sieve and for a period of 5 minutes on a 76 × 76 sieve and in the order of succession given below, with the following results :—

(1) The residue on a sieve 180 × 180 = 32,400 meshes per square inch (5,022 meshes per cm.^{2}) shall not exceed 10 per cent.

(2) The residue on a sieve 76 × 76 = 5,776 meshes per square inch (895 meshes per cm.^{2}) shall not exceed 1 per cent.

[The dimensions of the wires used for the sieves are fully detailed in the specification.]

9. The cement shall comply with the following conditions as to its chemical composition. The percentage of insoluble residue shall not exceed 1·5 per cent. ; that of magnesia shall not exceed 5 per cent. Sulphur present as, and calculated as, sulphuric anhydride shall not exceed 2 per cent., and sulphur present as sulphide shall not exceed 1·2 per cent. sulphur, these being equivalent to a maximum total of 5·00 per cent. of sulphuric anhydride. The total loss on ignition shall not exceed 3 per cent.

10. The breaking strength of the testing briquettes of neat cement at 7 days after gauging shall be not less than 600 lb. per square inch of section (42·18 kg. per cm.^{2}).

The breaking strength of briquettes made of one part of cement to three parts of sand, with the proportion of water, shall

(*a*) be not less than 325 lbs. per sq. in. after 7 days ;

(*b*) show an increase over (*a*) of not less than the number of lbs. per sq. in. arrived at from the formula :—

$$\text{Breaking strength at 7 days} + \frac{10{,}000}{\text{Breaking strength at 7 days.}}$$

The methods of making the tests are the same as for Portland cement (see p. 403).

Rapid Hardening Cements.

Concrete made with aluminous cement (*Ciment Fondu*) supplied by the Lafarge Aluminous Cement Co., Ltd., 296–302 High Holborn, W.C. 1, is as strong after 24 hours as ordinary Portland cement concrete which is 90 days old.

'Ferrocrete,' the rapid hardening Portland cement (*The Cement Marketing Company, Limited, Portland House, Tothill Street, Westminster, S.W.* 1), gives in 3 to 4 days the same strength as concrete made with ordinary cement does in 28 days, while there is also a continuous growth of strength.

American Specification for Waterproofing Material for Portland Cement Concrete.

A master specification for integral waterproofing material for use with Portland cement mortar or concrete, which has recently (1928) been issued by the Bureau of Standards of the United States Government, contains the following provisions (*Abstract*) :—

(*c*) The material when mixed with Portland cement mortar or concrete shall render such mortar or concrete more completely impervious to water and shall not impair their durability or reduce the compressive strength more than 10 per cent.

(*d*) The waterproofing properties shall be such that when combined in the manner, and in the proportions of waterproofing material to cement, specified by the manufacturers for use with Portland cement mortar and concrete, the treated test-pieces shall show an absorption of not more than 3 per cent. and shall reduce the permeability—when tested under a pressure of 100 lbs. —at least 95 per cent. of test-pieces made without the admixture of waterproofing materials, but in all other respects of identical materials, composition, and age.

(*f*) Mortar for tests of waterproofing materials shall be made from Portland cement and sand passing a No. 20 sieve, and held on a No. 50 sieve in the proportion of one volume of cement to four volumes of sand, with only sufficient water thoroughly worked to form a mixture which has a

* By permission of the British Engineering Standards Association.

flow of 90 as measured by a cement laboratory flow table. All materials shall be mixed at, and the test-pieces be stored in, a temperature of approximately 70° F.

(*g*) The waterproofing materials shall be incorporated in accordance with the methods, and in the proportions to the amount of cement, as specified by the manufacturers in their standard directions for the use of their materials.

(*k*) The tests shall be made when test-pieces have aged twenty-eight and ninety days. In each test each of the treated and untreated pieces shall be tested under identical conditions.

(*l*) Pieces for the absorption tests shall be dried at 212° and 230° F. until no further loss in weight is shown. They shall then be allowed to cool in the air to room temperature. The pieces shall be completely immersed in water for twenty-four hours, then be removed, the surfaces dried with a cloth, and weighed immediately. This process shall be repeated daily until the pieces show less than one-tenth of 1 per cent. daily increase in weight. The increased weight of the treated and the untreated pieces shall then be compared.

(*m*) The impermeability tests shall be made by subjecting one side of the test-pieces to a water pressure of 100 lbs. per sq. in., using water free from any visible discoloration or matter in suspension. The test shall be continued twenty-four hours unless a measurable degree of permeability is shown in less time.

(*n*) The degree of permeability shall be calculated in cubic centimetres of water passing through the piece per sq. cm. of surface exposed to water pressure. The impermeability of the treated and the untreated piece shall be then compared.

PLASTERS.

Plasters are pasty materials applied to walls, ceilings, etc., to give a hard, smooth finish. They resemble lime mortars in many respects but serve a different purpose. Common plaster is made of lime and sand, with or without ½ lb. of hair to each 1 cu. ft. of plaster. It is made harder and has smoother finish if 20 per cent. of plaster of Paris is added.

According to a tentative specification issued by the Department of Scientific and Industrial Research (*Building Research, Special Report, No.* 9, 1927) the lime to be used for plaster, fine and coarse stuff, should be *Quicklime, Class A*, or *High Magnesian Quicklime, Class A*, and, for coarse stuff only, *Quicklime, Class B* (see table in *Mortars*, p. 385).

For *floors*, a hard-burned plaster of Paris is used, or a mixture of ordinary plaster with a slight excess of lime is laid, and then treated with zinc sulphate (white), iron sulphate (red), boiled oil (mahogany), or copal varnish.

Stucco for external use consists of 1 part of hydraulic lime to 3 parts of sand, but for more durable work a cement mortar (p. 383) is often used.

Rough cast is sometimes made of the same materials as stucco, or of cement mortar (p. 383).

Plaster of Paris is made by calcining gypsum and grinding the product. The resulting powder, when mixed with water, forms a smooth paste which rapidly sets to a white, solid mass.

Hard Plasters.

Keene's, Martin's, and Parian cements are hard-setting forms of plaster of Paris. Keene's cement is made by soaking calcined gypsum in a solution of alum or borax and cream of tartar. For Martin's cement, a solution of potassium carbonate is used, and Parian cement is an intimate mixture of gypsum and borax which has been calcined and then ground to powder.

Another method of hardening plaster consists in intimately mixing 4 per cent. of powdered marsh mallow root and knead to a dough with 40 per cent. of water. The mass will harden in about one hour, and can be cut, filed, turned, or drilled. Eight per cent. of marsh mallow renders it still tougher.

Plaster of Paris articles may be hardened by means of a solution of borate of ammonia (ammonium borate), made by dissolving boracic acid in hot water, and then adding ammonia to it until the solution turns red litmus paper blue (or so that the solution smells faintly of ammonia). The best results are obtained when very little free ammonia is present. The borate of ammonia may be applied to the plaster of Paris article after it has been cast or before mixing. The most satisfactory method is to treat the cast. The solution is applied to the dry plaster object and the absorption allowed to take place. The surface gradually hardens, and finally the interior. The plaster so hardened may be employed for a large number of uses.

The following process may be used for hardening plaster, so that it may be used for flooring:—About 6 parts of good-quality plaster are intimately mixed with 1 part of freshly slaked white lime, finely sifted. This mixture is then laid down as quickly as possible, care being taken that the trowel is not used on it for too long a time. The floor should then be allowed to become very dry, and afterwards be thoroughly saturated with sulphate of iron or zinc—the iron giving the strongest surface, the resistance to breaking being twenty times the strength of ordinary plaster. With sulphate of zinc the floor remains white, when iron is used it becomes the colour of rusted iron; but if linseed-oil, boiled with litharge, be applied to the surface, it becomes a beautiful mahogany colour; this is especially the case if a coat of copal varnish be added.

Artificial Stone.*

ARTIFICIAL BUILDING STONE, No. 1.—Mix 100 parts of hydraulic lime which has fallen to a powder with water to form a paste. To this add 250 parts of gravel and 50 of coal ashes or lixiviated wood ashes. The mass is then thoroughly mixed, and a sufficient quantity of water added to make the volume of the mass equal to 500 parts. It is then poured into moulds made of pine-boards, where it is allowed to remain until set.

No. 2.—125 parts of hydraulic lime which has fallen to a powder are mixed with a sufficient quantity of water to form a paste. To this are added 250 parts of ground oyster shells and 150 parts of ground peat ashes, and a sufficient quantity of water to make the bulk of the mass equal to 500 parts. It is then poured into moulds as in No. 1, and dried.

ARTIFICIAL STONE FROM QUARTZ SAND AND PLUMBIC OXIDE.—Ground quartz sand is mixed with 2 to 10 per cent. of finely-ground plumbic oxide. The harder the stones are to be, the more plumbic oxide must be used. The mixture is moistened with water-glass, again thoroughly mixed, and then pressed firmly into moulds. The resulting stone is dried and then burned.

VICTORIA STONE.—The refuse of granite quarries is broken up into pieces of suitable size, and 4 parts of the fragments thus obtained are mixed with 1 part of Portland cement, and sufficient water added to bring the whole to the consistency of dough. The mass is run into moulds, in which it is allowed to remain for several days, or until it has set solid. The blocks are then immersed in a solution of silicate of soda.

RANSOME'S ARTIFICIAL STONE.—Clean and dry sand and other suitable siliceous and earthy ingredients are thoroughly incorporated in a mixing mill with silicate of soda. The resulting pasty mass is then pressed into moulds of any required pattern or size, and when set sufficiently immersed in a solution of chloride of calcium. In the case of large pieces the saturation with chloride of calcium is facilitated by the use of an air-pump. The resulting reaction is the formation, by double decomposition of the ingredients of an insoluble calcium silicate out of sodium chloride. The first-named forms a solid and indurate binding material for the stone, and the sodium chloride is removed by a subsequent thorough washing with water. This last operation is important, since, if the sodium chloride is not completely removed from the stone, it will make its appearance subsequently in the form of a white efflorescence on the surface.

TO IMITATE VARIEGATED MARBLE.—Mix hydraulic lime and ground marble and incorporate with the mixture a solution of alum and suitable colouring substances. Differently coloured masses are then mixed together and cut into slabs.

ARTIFICIAL MARBLE.—The following mixtures have been recommended for making artificial marble. Grind and thoroughly mix comminuted stone, 280 parts; limestone or chalk, 140 parts; burned calamine, 5 parts; calcined felspar, 3 parts; fluorspar, 2 parts; calcium phosphate, 2 parts; water-glass, 40 parts.

On the addition of the water-glass the ingredients are quickly mixed, and thereupon pressed into moulds. The finished pieces are dried at a temperature gradually rising to 125° F.

Asphalt Flooring.

8 lbs. of asphalt composition (1 ton asphalt, 20 lbs. mineral tar) will cover 1 superficial foot $\frac{3}{4}$ in. thick. For goods warehouses thickness should be 1 in. For damp-course wall, $\frac{3}{4}$ in.

To Remedy Dampness in Walls.

1. If house shows damp on all sides just where walls rise from ground, cut out a course of bricks foot by foot, insert an asphalt damp-course, and re-brick up.

2. If wall becomes saturated by a stopped-up drain-pipe, remove plastering and re-point wall outside after wall has become dry. If it is inconvenient to renew plaster, the latter (although wet) may be varnished over with a solution of shellac in naphtha (4 ozs. shellac, 1 quart naphtha); this almost immediately hardens, and paper may be pasted over it.

3. A good coating of hot coal-tar laid on the exterior of a brick wall will prevent the entrance of damp, but should only be applied where its appearance is not objectionable.

4. Another plan for walls above-ground is to mix 1 part of Portland cement with 2 parts of fine sand, and add water enough to bring the ingredients to the consistency of thick cream. Cover the brickwork with two coats of the mixture, and when it is quite dry finish with a coat of paint.

5. Re-plaster the inside of the wall with sand and cement impregnated with one of the special waterproofing agents, such as 'Pudlo,' 'Bareau,' 'Waterex,' or 'Silicon Ester.'

* The manufacture of artificial stones, *lime-sand* bricks and blocks, is dealt with at length in 'Bricks and Artificial Stones of Non-Plastic Materials' by A. B. Searle (J.& A.Churchill, London).

6. Many special preparations for making damp walls waterproof are obtainable; the best of them consist essentially of: (*a*) a solution of water-glass (sodium, silicate) which is applied to the wall and is followed by a solution of calcium chloride, which forms an insoluble silicate; (*b*) a solution of various fluosilicates which with the mortar or with a subsequent application of calcium chloride form an insoluble calcium fluosilicate in the pores of the wall; or (*c*) various substances, dissolved in acetone, alcohol, or other volatile solvent, which remain in the pores when the solvent has evaporated.

Each of these methods is suitable for certain walls, but some walls appear to resist every kind of treatment yet applied to them.

Effects of Moisture Changes on Building Materials.

A Bulletin (No. 3) on this subject has been issued by the Building Research branch of the Department of Scientific and Industrial Research. Price 1*s.* net; postage extra. H.M. Stationery Office; London, Adastral House, Kingsway, W.C. 2.

ADDENDUM.

Proportions for Concrete. (See p. 389.)

It is being increasingly realised that the best concrete is that in which the coarse and fine aggregates are most completely immersed in a paste made of cement and water, no excessive amount of such paste being present. A concrete made on this basis is often much cheaper than those compounded on a 'voids' basis. For the newer method no definite proportions can be prescribed in advance; the best ones must be found by trial, as different batches of coarse and fine aggregate differ greatly.

To secure uniformity of consistency a *slump test* is generally used. In this, a conical tube 12 ins. high, with a lower diameter of 8 ins. and an upper one of 4 ins., is placed upright and filled with the concrete to be tested. The filling is effected in three stages, the layer formed in each stage being lightly tamped with a metal rod 21 ins. long and $\frac{5}{8}$ in. in diameter. Immediately afterwards the tube is lifted off and the slump or settlement is measured.

According to Duff Abrams, a concrete with a slump of 1 in. or less will contain a little more water than is necessary for the maximum strength but will be too stiff for most constructional work. If 10 per cent. more water is present, the slump will be increased to 3–4 ins., 25 per cent. more water will increase the slump to 6–7 ins.

Instead of using the amount of subsidence or slump as a direct measure, some engineers prefer the term *consistency*. The following table shows the relation between the two :—

Slump in inches.	Consistency.
0–$\frac{1}{2}$	0·90
$\frac{1}{2}$–1	1·00
1–2	1·05
2–3	1·08
3–4	1·10
5–6	1·15
6–7	1·25
8–10	1.50.

SECTION IX

PART III

LATHING—PLASTERING—MOULDINGS—SLATING—TILING—GLAZING—PAINTING—PAPER-HANGING—PLUMBING—BOLT FIXING.

Lathing and Plastering.

LATHING

is the first operation for ceilings and partitions; $\frac{3}{8}$-inch intervals are left between the laths, to key the plaster. Stronger for ceilings than for partitions.

One bundle of laths contains (nominally) 500 feet. Single fir laths are about $\frac{3}{16}$ in. thick, lath-and-a-half laths are full $\frac{1}{4}$ in. thick; double laths are about $\frac{3}{8}$ thick. They are about $1\frac{1}{4}$ in. broad. 1 bundle of 3-foot laths and 670 nails, or 1 bundle of 4-foot laths and 625 nails, will cover about 5 superficial yards.

Lath Nails are either wrought, cut, or cast; the latter, being cheapest, are generally used, 1 inch, $\frac{7}{8}$ inch, or $\frac{3}{4}$ inch long, according as double, one-and-a-half, or single laths are used. $\frac{7}{8}$-inch cut nails run about 380 to the pound.

PLASTERING.

Coarse stuff is mortar composed of 1 part lime (carefully-slaked fat or pure lime is generally used) and 1 or 2 of sharp fresh-water sand, with 1 lb. hair to 3 cubic ft. of mortar.

Hydrated lime is usually superior to lime which has been slaked by hand and can be used at once, whereas ordinary slaked lime must be kept in the form of a putty until slaking is complete.

Fine stuff is made of pure lime slaked with a little water, after which water is added to bring it to the consistency of cream; it is then left to settle, the superfluous water poured off, and the water evaporated until it is of proper thickness. In ceilings a small portion of white hair is mixed with it.

Bricklayers' and plasterers' putty is fine stuff brought to the proper consistency.

Gauged stuff is $\frac{1}{4}$ fine stuff mixed with $\frac{1}{4}$ plaster of Paris.

Common stucco is 1 lime to 3 or 4 clean-washed sand.

Bastard stucco is $\frac{1}{3}$ fine stuff and $\frac{2}{3}$ clean sand, with a little hair.

Lath-and-Plaster, or *One-Coat Work*.—A coat of coarse stuff, $\frac{1}{4}$ inch to $\frac{3}{8}$ inch thick, laid on the laths and smoothed off with the trowel.

Lath, Plaster, and Set, or *Two-coat Work*.—The first, or 'pricking-up' coat of coarse stuff is scored with a birch broom or undercut with a lath, and a thin finishing coat of fine stuff or gauged stuff added.

Lath, Plaster, Float, and Set, or *Three-coat Work*.—When the pricking-up coat is dry, the surface is scored, and divided into 8-foot squares by *screeds* of coarse stuff, 8 inches wide, carefully levelled. These are then filled in with coarse stuff, floated carefully level. The finishing coat is then put on as before.

Materials.	1 in. Thick.	$\frac{3}{4}$ in. Thick.	$\frac{1}{2}$ in. Thick.
1 bushel of cement, or 1·28 cubic feet, will cover .	$1\frac{1}{2}$ sq. yds.	$1\frac{1}{4}$ sq. yds.	$2\frac{1}{4}$ sq. yds.
1 cement and 1 sand " .	$2\frac{1}{4}$ "	3 "	$4\frac{1}{2}$ "
1 cement and 2 sand " .	$3\frac{1}{3}$ "	$4\frac{1}{2}$ "	$6\frac{3}{4}$ "

1 cubic yard of lime, 2 yards of sand, and 3 bushels of hair will cover 75 yards superficial, render and set, on brick, or 70 yards on lath.

Mouldings.

Mouldings are made of (*a*) plaster of Paris with or without fibrous material, etc., added to strengthen them and to save plaster; (*b*) burned clay or *terra cotta* (Section X.); or (*c*) wood. Mouldings of plaster may be bought ready for use, and are affixed with plaster of Paris or by means of nails. Wooden mouldings are supplied in various lengths and are sawn to the size of the job.

Nos. 1 to 32 (fig. 1) are forms of mouldings in general use:

Nos. 2, 17, 26.—For inner face of window-frames, and the outer and inner faces of door-frames.

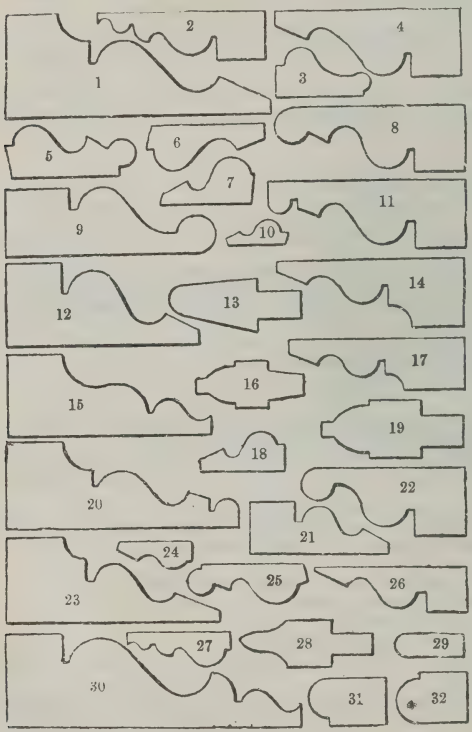


FIG. 1.—Mouldings.

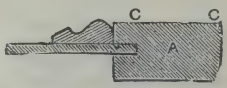


FIG. 2.—Moulding.

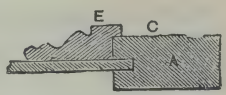


FIG. 3.—Bolection Moulding.

Nos. 18 and 24.—For edging the framing of panels of doors formed of styles and rails.

Nos. 13, 16, 19, 28.—Rebated mouldings for the reception of glass; may be used as sash-bars or for the bars of window-frames.

Nos. 31 and 32.—Beadings for capping matchboarding carried part way up a wall from the flooring.

No. 29.—Beading or slip for placing on either side of a window-frame, to confine the sashes in their proper place.

If the moulding does not project beyond the face, C, C, of the frame, A, as in fig. 2, it is simply described as a *moulding*.

If the moulding, fig. 3, projects beyond the face of the frame, A, making a narrow projecting framing, E, between the frame of the door and the curved part of the moulding, it is described as a *bolection* moulding.

Slating.

The following terms are used as in slaters' work :—

Head : the upper or edge top of slate. *Back* : the upper or exposed surface of slate when laid. *Red* : the lower or under surface of slate. *Tail* : the lower or bottom edge of slate. *Margin* : the part of each course of slates exposed to view (*a–b*, fig. 4). *Gauge*: the distance battens are apart from centre to centre ; required to determine the position of battens to receive the nails by which the slates are secured. *Bond.*—When a joint of two adjacent slates is immediately in the centre of the slate, the tail end of which rests upon them, the slates are said to bond. *Lap* : the distance the tail of one slate overlaps the head of second course below, when slates are nailed near the centre, or the distance the tail of slate overlaps the nail-hole of second course below when slates are nailed near the head. The lap, in practice, ranges from 2½ inches to 4 inches, and averages as 3 inches. *Holeing*: the piercing of slates to receive nails. *Sorting.*—When roof is to be covered by slates of different lengths, they are regulated to proper dimensions, so that the largest slates may be nailed near the eaves, and the smallest at the ridge. *Eaves* : the lower part of slating hanging over a wall.

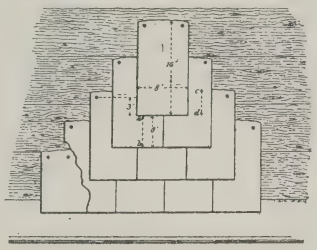


FIG. 4.

Sizes of Slates.

The sizes of slates in common use are as follows :—' Duchess ' slates, 24 inches by 12 inches, for moderate flat roofs of ½ pitch ; ' Countess ' slates, 20 inches by 10 inches, for roofs of ½ pitch ; and ' Ladies,' 16 inches by 8 inches, for roofs of ½ pitch, the rule being, the steeper the pitch, the smaller the slate, and the more exposed the building, the greater the pitch of the roof.

Slates are laid on *boards* (fig. 4) or *battens* (fig. 5), the latter being the cheapest. Boards, ¾ inch thick for Countess slates, and rafters not more than 18 inches in the clear, 1 inch or thicker, otherwise. Battens, 2 inches by ¾ inch for Countess, with rafters 12 inches interval : 3 inches by 1 inch for Duchess. Slates are often *holed* 1 inch or 1½ inch from the head, as in fig. 4, but are better centre-holed, i.e. just clear of head of slate below, as in fig. 5.

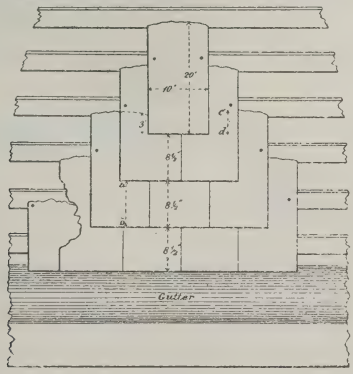


FIG. 5.

If a *good slate* is stood half its depth in water for several hours the moisture should not rise to the top.

Slating Nails.

Composition nails are recommended for all good work ; they are stiffer than copper or zinc, present a brassy appearance, and are cheaper than copper nails. If iron nails are used, they should be galvanised, or boiled in linseed oil. Two to each slate, 1⅜ inch long for small slates, 1½ inch long for Countess, and 1¾ for Duchess slates.

Zinc and Copper Clips.—When a slate gets broken that is nailed to boarding and has to be repaired, narrow clips, about ½ inch wide, are pushed between the slates to catch the head of

slate below. The new slate is then inserted, and the end of clip turned as a hook over the tail of the slate. When slates are nailed to battens, and the interior is accessible, they may be repaired by means of a piece of copper wire which is pointed at one end and driven into the batten or boarding at the head of under slate, passed between the side-joints of slates immediately above it, and lapped over the tail of the next slate but one above it. A man under the battens will secure a piece of copper wire to the battens, and then push the wire between the slates for the man above to clip over the tail of the new slate.

Size and Weight of Slates.

A thousand slates means 1,200. A square is 100 super feet.

Name.	Size.	Gauge for 3-in. Lap Nailed in Centre.	No. of Squares covered by 1,000 of 1,200.	Weight per 1,000 of 1,200.	Weight* per Square.	No. to Cover One Square.	Nails required per Square.	
							Iron.	Copper.
	Ins.	Ins.		Cwts.	Cwts.		No.	Lbs.
Smalls . . .	12 × 8	4½	2·8	17½	6¼	430	860	5
Doubles . . .	13 × 6	5	2·5	18	6	480	960	6
Ladies . . .	14 × 12	5½	5	31	6¼	240	480	3¼
„ . . .	16 × 8	6½	4·5	25	5·5	300	600	3½
Viscountesses . .	18 × 10	7½	6	36	6	200	400	2¾
Countesses . .	20 × 10	8½	7	40	5·7	171	342	4
Marchionesses . .	22 × 12	9½	9·4	55	6	130	260	3¼
Duchesses . . .	24 × 12	10½	10	63	6	125	250	3
Imperials . . .	33 × 24	13½	25	..	..	48	96	3
Rags	36 × 24	16½	30	..	..	40	80	3½
Queens . . .	36 × 24	16¼	30	..	..	40	80	2

* The weights are for first quality Penrhyn or Bangor: second quality weigh about one-third more. As the sizes of slates of the same name sometimes differ slightly, the size required should be given as well as the name, when of importance. This Table allows about $\frac{1}{10}$th for waste.

'Turnall' Asbestos-Cement Roofing Slates.

(*Asbestos Cement Building Products, Ltd., Trafford Park, Manchester.*)

These slates are made rectangular, honeycomb (fig. 6), and diamond (fig. 7) in pattern. They may be obtained in five rich colours, red, blue, grey, russet brown, and brindled.

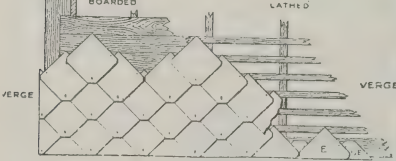


Fig. 6.

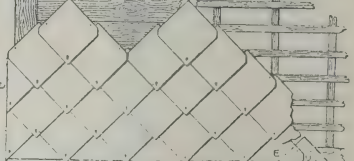


Fig. 7.

The original weather-resisting qualities of 'Turnall' Asbestos Cement Slates are guaranteed for twenty years.

'Turnall' slates can be cut to fit hips, valleys, verges, round chimneys, and roof lights, etc.

Rectangular sizes, 24 ins. × 12 ins. and 16 ins. × 8 ins. Diamond and honeycomb sizes, 15¾ ins. × 15¾ ins. and 11¾ ins. × 11¾ ins.

The lap to be given is determined by the pitch of the roof, position of building, and prevailing weather conditions.

'Poilite' Straight Cover Slates.

(*Asbestos Cement Building Products, Ltd., Trafford Park, Manchester.*)

The slates can be supplied either with or without the chamfered corners, and, when laid, their appearance is equal to the best North Wales or Westmorland slates.

Slates 24 ins. by 12 ins. are made in grey, blue, red, or russet-brown colours. Slates 15¾ ins. by 15¾ ins. or 15¾ ins. by 7¾ ins. can be supplied.

'Fibrent' Asbestos-Cement Slates.

(*Asbestos Cement Building Products, Ltd., Trafford Park, Manchester.*)

These slates are manufactured in straight, diagonal, or honeycomb patterns, 15¾ ins. by 7⅞ ins., 12 ins. by 12 ins., 15¾ ins. by 15¾ ins., 24 ins. by 12 ins., and 24 ins. by 24 ins., cut and holed for various laps as required.

By exposure to climatic extremes the slates become absolutely impervious. They are non-conductive of heat or cold, fire resisting to the highest degree, of extreme durability, and possessing considerable elasticity are immune from cracking or scaling under frost or extremes of temperature.

'Fibrent' slates are extremely light, weighing when fixed about 20 lbs. to the square yard, thus great economy may be effected in general roof construction, rafters being spaced at a minimum of 24 ins. to maximum 30 ins. They may be fixed direct to light battens or rough boarding by means of two nails and a copper disc rivet, and, invariably, a perfectly satisfactory, inexpensive, and effective roof covering is the result. 'Fibrent' has been used upon a number of modern factories where slating is the roof covering, with very satisfactory results.

'Fibrent' Ladies' and Duchess Slates.

Sizes 15¾ ins. by 7⅞ ins., and 24 ins. by 12 ins. are fixed for straight cover similar to ordinary natural slates; they can be used with satisfactory results for all classes of private dwelling houses, the cost of up-keep being reduced to a minimum. A guarantee against disintegration is given for a period of twenty years, that is to say at the end of twenty years there will be no signs of disintegration.

'Fibrent' slates are manufactured in colours of grey, red, purple, blue-black, and antique brown.

Underlining.

Felt laid upon boarding under slates effectually keeps out the wet, and is a non-conductor of heat and sound; it should be battened over to preserve it.

Willesden Paper (*The Willesden Paper & Canvas Works, Ltd.*) is an excellent material for underlining. It is a good non-conductor, does not perish when shut up beneath slates or tiles, is waterproof, and will not allow rain or snow to penetrate, has no smell, even under the greatest heat, and lies very close on the roof.

See also 'Serval' Asbestos Felt, p. 325.

Tiling.

Roofing tiles may be *plain* (i.e. rectangular and almost flat), or they may be wavy, as *pantiles*, or they may have grooves or other structures which enable them to *interlock* with on another. Interlocking tiles are much heavier than plain tiles, but form a more weatherproof roof. (See also p. 428.)

The unit of measure for roofing tiles (when fixed) is a 'square' having an area of 100 sq. ft. Thus, if 4 in. is to be exposed on a *plain tile*, 600 such tiles will form a 'square,' but in flat roofing, where the overlap is much less, only 210 tiles will be required for a square. 1 square weighs about 15 cwt. **100 plain tile laths 5 ft. long = 1 bundle.**

For *pantiles*, 180 to a 10-inch gauge, 164 to an 11-inch gauge, and 150 to a 12-inch gauge. The lap of the tile over the one below it will be just the difference between its length and the length of the gauge. 1 square weighs about 8 cwt. 12 pantile laths 10 feet long = 1 bundle. Fir tiling laths are 1½ inch by 1 inch; and oak tiling laths, 1¼ inch by ¼ in. 1 square of tiling requires 1 bundle of laths, 150 nails, 1 peck of tile-pins, and 3 hods of mortar.

Wall and *Floor Tiles* are usually glazed; they are not very suitable for external use.

'Turner's Trafford Tiles.'

(Patent No. 111,172/17.)

(*Asbestos Cement Building Products, Ltd., Trafford Park, Manchester.*)

These tiles (fig. 8) are claimed to be the original tiles to embrace scientific principles in the design of corrugated asbestos cement sheets. They are light in weight, fire-resisting, possess great strength, and require no special knowledge to lay. Supplied with ready mitred corners and close-fitting lap joints, they save time in fixing and provide absolute weather protection. Purlins may be paced out to 3 ft. 6 in. centres, thus economising on roof structure. On steel structures they are fixed direct to purlins and fastened with hook bolts or straps. On timber, use galvanised

driving screws or roofing nails, with compressed asbestos or lead washers under the head. The *original* weather-resisting qualities of 'Turner's Trafford Tiles' are guaranteed for twenty years.

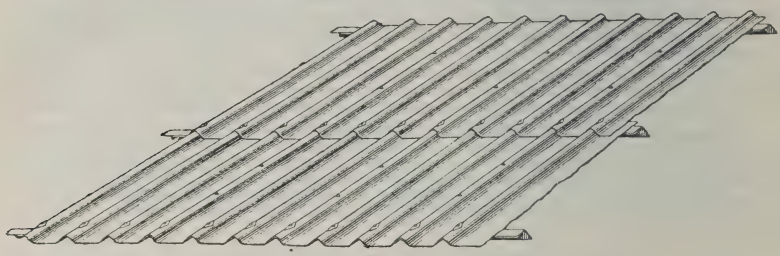


FIG. 8.

Standard size: 4 ft. × 3 ft. 8 ins. × ¼ in.; gross area, 14 ft. 8 ins.
Approximate weight per tile—38 lbs.; 59 tiles to the ton.
Quantity required to cover 100 sq. ft.—with end or horizontal overlap of 6 ins.—8·57 tiles.

'POILITE' STANDARD TILES.

(*Asbestos Cement Building Products, Ltd., Trafford Park, Manchester.*)

Poilite tiles for diagonal fixing (available in red, blue, grey, and russet-brown colours), size 15⅝ ins. by 15⅝ ins. are made in two varieties, viz. the Standard (or C4) pattern and the Honeycomb (or F5) pattern. The former, when laid, shows diagonal lines on the roof, while the latter, owing to the lower point being cut off, produces a design of honeycomb form.

'POILITE' PANTILES.

(*Asbestos Cement Building Products, Ltd., Trafford Park, Manchester.*)

The tiles 15⅝ ins. by 13½ ins. (actual cover of 11⅛ ins. by 11⅛ ins.), are light for transport and fixing, and reduce cost of sub-structure, are rapidly fixed, improve with age, and are not affected by atmospheric changes.

Number of tiles required per square, net, is 108, but, including eaves and cuttings, 111 to 113.
Weight of tiles per 1,000, about 29 cwt.
These tiles are made in grey and russet-brown colours.

'POILITE' ROMAN TILES.

(*Asbestos Cement Building Products, Ltd., Trafford Park, Manchester.*)

'Poilite' tiles, Roman type, 4 ft. by 1 ft. 8½ ins., are made in grey, red, and russet-brown colours.

Glazed tiles can be supplied for use with Poilite Roman tiles.

Glass and Glazing.

Sheet glass is that which is in ordinary use for windows, &c., and the usual glazing qualities are known as *best*, *seconds*, and *thirds*; a commoner quality, known as *fourths* or *coarse*, is also made, but is not much used.

The weights per foot super. are 15 oz., 21 oz., 26 oz., 36 oz., and 42 oz., the latter being nearly ¼ in. thick. As a rule, every $\frac{1}{16}$ in. may be taken as 13 oz. to the foot super.

In addition to the usual glazing qualities there are two others, called *picture qualities*, *A* and *B*, made in the 15 oz. and 21 oz., and one other, *C*, in the 26 oz. and 32 oz. thicknesses.

Sheet glass is sold by the foot super. of the required quality and weight, in *crates* of stock sizes for cutting up, or *squares* cut to order.

The ordinary stock sheets do not exceed in area 22 ft. super., in length 85 in., or in breadth 49 in., nor are they less than 11 ft. in area, 45 in. in length, or 34 in. in breadth.

The largest sizes that can be made in the various substances of sheet glass are as follows, *but the extreme limits of length and width cannot be combined in the same sheet*:—

	Extreme length.	Extreme width.	Extreme area.
15 oz.	60 in.	40 in.	15 ft.
21 „	90 „	50 „	26 „
26 „	90 „	50 „	25 „
32 „	85 „	48 „	21 „
36 „	70 „	44 „	17 „
42 „	70 „	44 „	15 „

The extreme area, taken in connection with the extreme length or width required in any particular case, will indicate approximately the corresponding limit of width or length.

Sheet glass obscured or frosted so as to appear like ground glass is made from any description of the third quality glass.

Sheet glass can be polished on face, bent to any required curve, or ground on the edges, as may be required.

Fluted sheet is glass rolled to form flutes or corrugations on both sides; which, while it secures privacy without obstructing the light, makes it much stronger than ordinary ground or obscured glass. It is made from 15 oz. to 32 oz. in weight, and in squares 60 in. in length and 38 in. in breadth. It can also be obscured if required.

White dappled is sheet glass about 21 oz. to the foot, the surfaces being mottled or undulatory, as its name implies. *White muffled* sheet is somewhat similar.

Patent plate is merely a superior kind of sheet glass, and is now not much used. It can be distinguished from British polished plate, by a more wavy appearance of the surface, as well as by the air bubbles, which are *oval* in patent plate and *circular* in British plate. It is made in four thicknesses—viz. $\frac{1}{16}$, $\frac{1}{12}$, $\frac{1}{10}$, and $\frac{1}{8}$ in.—and three qualities—bests, seconds, thirds—in each. $\frac{1}{16}$ in. and $\frac{1}{12}$ in. are generally used for picture purposes, and $\frac{1}{10}$ in. and $\frac{1}{8}$ in. for coach plates. It can be supplied *smoothed* or *finely ground*, in sheets 50 in. in length or 36 in. in width.

Polished or *British polished plate* is the best article made, and is kept in stock sheets up to 100 ft. super., or 160 in. long or 104 in. wide, but can be made in larger sheets if specially ordered. The usual thickness is $\frac{1}{4}$ in., but it can be obtained $\frac{1}{8}$ in. thick, varying by $\frac{1}{16}$ in. up to 1 in. in thickness. It is manufactured in three qualities, known as *ordinary* and *best glazing*, for windows, etc., and *silvering quality*, for mirrors, etc. It can be bent to any desired curve, and the edges can be ground or polished. If scratched the surface can be repolished or resilvered at any glass works.

Rough cast plate is the cheapest quality of plate glass, and is made in plates even exceeding 100 ft. super., but the stock sizes run up to about 60 ft. superficial. Its thickness ranges from $\frac{1}{8}$ in. to 1 in. It can be bent to any curve, or ground on one or both sides if required. From its strength it is very useful for risers to steps, glazing pavement lights, and similar rough work where light has to be transmitted.

Plain and *fluted rolled rough plate* is principally used for railway roofs, factories, workshops, etc., and other purposes where the object is to intercept the vision but not the light. The *plain* is rolled with fine lines on the surface; the *fluted* is either of the small pattern, with about 11 flutes to the inch, or the large pattern, with about 4 flutes to the inch. It is made from $\frac{1}{8}$ in. to $\frac{3}{8}$ in. thick, in stock sizes not exceeding 25 ft. super., or 130 in. in length, or 46 in. in breadth. It can be bent to curves, or obscured on the rough side.

Diamond and *quarry rolled rough plate* is principally used for churches, chapels, schools, lecture halls, etc. The diamond rough plate has a bright and brilliant appearance. The pattern is of a lozenge shape, and in relief upon the body of the glass. The quarry rolled is in imitation of, and is used as a substitute for, leaded quarries, and is made in either *large* or *small* patterns. It is made from $\frac{3}{8}$ in. to $\frac{1}{2}$ in. thick, in sheets from 20 in. to 34 in. wide and up to 100 in. long.

Chequered rolled plate is similar to *diamond*, but the pattern is square. It is principally used for screens, door panels, etc.

Rolled plate corrugated one side is about $\frac{1}{4}$ in. thick, and is made in sheets up to about 110 in. long and about 36 in. wide. It is used for glazing in any position where extra light is required and opaqueness a necessity, and is suitable, when silvered, for reflectors.

Figured rolled glass is made in a variety of patterns known by distinctive names, such as 'Muranese,' 'shell,' 'Arabesque,' etc., and is made either white or in various tints.

Patent wire rolled glass consists of a sheet of glass, about $\frac{1}{4}$ in. thick, having wire netting embedded in its centre. It is extremely useful for many purposes.

Rolled cathedral glass weighs about 21 oz. to 26 oz. to the foot super., but can be made thicker if required. It is made either in white, pot metal colours, or cathedral tints. This last term is applied to light tints of no positive colours, which are suitable for glazing the windows of churches throughout.

Coloured and *stained glass* is made from 15 oz. to 32 oz. in weight, and in sheets not above 48 in. long or 34 in. wide, in a variety of colours, which are known as *flashed*, in which the body is of ordinary sheet glass thinly covered on one surface only with coloured glass, or *pot metals*, in which the glass is coloured throughout. This glass can also be *sanded* or *obscured*.

CONSTANTS OF LABOUR FOR GLAZIER.

		Hours of a glazier.
Crown glass stopped in new sashes	per ft. super.	·20
" " old sashes	"	·60
Sheet glass stopped in large squares in new sashes . .	"	·15
" " " old sashes . .	"	·40
Hacking out glass, including painting putty . . .	"	·30
Cleaning windows, both sides	"	·04

(J. T. Rea.)

Putty, Paint, and Solder.

Glazier's putty is composed of whiting and linseed oil (see p. 418).

In order to prevent the constant expansion and contraction of large panes of glass, especially in tap-lights, from loosening the putty, tallow may be mixed with it, forming *thermoplastic putty*, which, by retaining its pliability, gives and takes with the glass.

The priming and paint used by the glazier are the same as used by the painter.

The solder used for lead glazing is the plumber's *fine solder*, consisting of equal parts of lead and tin.

To remove old putty, apply a hot iron, to soften it, or paint it with nitric acid or hydrochloric acid, and, after about an hour, remove the putty with a knife.

Painting.

All surfaces to be painted should be smooth, clean, and well dusted. The first operation in painting resinous woods, such as fir, is *killing the knots*, or treating them so as to prevent the resin or turpentine from exuding and ruining the paint. This may be done by painting the knots over with hot lime, and, when dry, ironing them with a hot iron, and then painting them smooth; or leaving the lime on for about 24 hours, scraping it off, and then painting with red and white lead and linseed oil, and, when dry, pumice-stoning the knots smooth. But these processes take time, to obviate which a liquid called *patent knotting*, chiefly shellac and naphtha, is much used. It forms a skin over the knots, and by drying in about 5 minutes allows of the painting going on at once. *Red-lead knotting*—made by grinding red lead in water mixed with strong glue size, and used warm—is, however, considered preferable to patent knotting, and dries in about 10 minutes.

The first coat of paint is called the *priming coat*. This in ordinary lead painting is composed of red lead, or red and a little white lead, mixed with raw linseed oil, a little litharge, and burnt copperas ground very fine in turpentine as a drier, the object being to fill the pores of the wood with oil before applying the colouring coats, which would otherwise be sucked up and wasted by the wood.

After the priming comes the stopping, or filling up all nail-holes, etc., with putty, which if applied before the priming, would become loose from the wood drawing the oil out of the putty, and allowing it to shrink.

After all defects have been made good, and the work has been rubbed smooth, the colouring coats are applied, the second coat, or first after priming, being diluted with about one-third turps, the third coat having the oil and turps in equal proportions, and the fourth coat one-third oil to two-thirds turps. If not to be painted white, the second coat should approximate to the required colour.

When, as in drawing-room work, it is desirable to avoid the glossy surface of oil paint, a fifth coat, for which all turps and no oil is used, is added, and is called a *flatting* coat, and should be laid on before the last coat has become hard. It will not admit of washing, as oil paint will, and therfore is not suitable for outdoor work.

Varnish adds greatly to both the appearance and durability of paint. A priming coat followed by a dark coat, such as a chocolate or purple-brown, and finished off with a coat of common varnish, is cheaper than, and as durable as, 4 coats of colour; it looks better, is more rapidly executed, and stands washing well. A single coat of gum-lac or good resin, dissolved in linseed oil to about the consistency of cream, and coloured as may be required, also forms a good and durable covering.

Walls may also be coloured and varnished thus: First, apply at boiling heat 2 coats of whiting mixed with strong glue size, then fill up defects with mastic in water, rule smooth with pumice stone, and cover with 2 coats of coloured varnish, the first coat mixed with one-fourth of the required colour, and the last coat with only half as much colour. The colour should be ground very fine, and the varnish should be white spirit varnish.

Plaster, being very absorbent, is often primed with glue size and then covered with 4 coats of colour (if the ordinary lead paints are used), and a flatting coat can be added if desired; but the best plan is to apply a priming coat of boiled oil quite warm, and when dry and hard add a thin coat of weak size tinged with red lead, in order to stop all absorption and give the work a uniform appearance, and finish off with 2 coats of oil paint, and a flatting coat if required; or 2 coats of coloured varnish can be applied as described above.

In painting on plaster, in order to guard against damp injuring the paint, it is safer to distemper the walls for the first two years, and then wash it off and paint, taking care that the walls are perfectly dry. If the distemper is not greasy or dirty, it is better to merely brush it well down with a dry brush, and paint over it without any washing.

With the ordinary paints, new wood or ironwork requires 4 coats, including the priming coat, but exclusive of any flatting coat, and old paint should have 2 coats for inside and 3 for outside work.

Painting Fresh Cemented Surfaces.

The McNicholl process seems to be the best treatment yet devised for coating a fresh cemented surface to enable it to be painted. It consists of zinc sulphate, 5 lbs. to the gallon of soft water. The zinc sulphate forms a chemical compound that prevents the alkali from attacking the paint coating, afterwards applied. This process is also good for interior walls, over Keene's or Portland cement. When it is dry apply a coat of either pure raw linseed oil or an oil paint, thinned out with benzene. (*Contract Record.*)

VALUE OF PAINTS.

Details.	White lead.	Zinc white.	Red lead.	Red oxide.
Covering capacity in sq. yds. per cwt	806	1,411	594	1,083
Price per cwt. in shillings	32	36	32	28
Cost, in shillings, per 100 sq. ft.	·44	·28	·60	·28
Times painted in 20 years	5	5	3	7
Cost, in shillings, per 100 sq. ft. for 20 years	2·20	1·40	1·80	1·96
Relative economic value on 20 years' basis, the highest value being represented by 100	64	100	77	71

(*Cruickshank Smith.*)

VALUING AND ESTIMATING PAINTERS' WORK.

The rule in measuring painters' work is to take wherever the brush goes, by the *yard superficial* in the case of large surfaces, such as walls, doors, etc.; by the *foot run* in the case of narrow surfaces *cut-in* on both edges, as in skirtings, where the paint has to be run to a line both at top and bottom, to avoid painting the floor or wall; and by the *number* in the case of separate pieces, such as chimneypieces, newels, balusters, brackets, gratings, etc. In iron railings and other open work each side is measured by the yard superficial, as if it were a plain surface, and if ornamental each side is doubled, or a fair extra allowance made according to the nature of the work.

The outsides of ordinary-size frames, being *cut* to the wall, the insides, as well as when there are no linings, as in stores, riding-schools, etc., are taken by the number. In cases of sash-frame with mullions or transoms each division is taken as one side of a frame.

The sashes themselves are paid for by the *dozen squares*, the price being for one side only, the number being doubled if painted at the same cost on both sides.

Letters and figures are paid for by the *inch in height* as plain or ornamental.

Common colours include *lampblack*, *red* and *white lead*, *Venetian red*, *ambers*, and all other *common ochres*, such as *greys*, *buffs*, *stones*, *etc.* *Superior* or *ornamental colours* include *bright yellows*, *warm tints*, *blues*, *mineral greens*, *etc.* Some colours, such as *verditer*, *pea-greens*, *rich reds*, *pinks*, and *bright blues*, are charged at a higher rate still, as *delicate tints*.

The following are the yards super which a fixed amount of material will cover in each coat:—

First coat, or priming	10 lbs. white lead 1 oz. red lead	2 ozs. litharge* 4 pints linseed oil	63 superficial yards.
Second coat	10 lbs. white lead 2 ozs. litharge	2½ pints linseed oil* 1½ pint spirits of turpentine	100 superficial yards.
Third and fourth coats	10 lbs. white lead 2 ozs. litharge*	2 pints linseed oil 2 pints spirits of turpentine	113 superficial yards.

For outside work, when exposed to the sun, more turps should be used to prevent the paint from blistering, and only *boiled* linseed oil should be used. For inside work, raw linseed oil may be used, but the less oil the less gloss. For the flatting coat, the colour being ground in oil, only turps is added.

For coloured paints the last two coats have the colour added to the composition in the proportion of 1 to 2 lbs. for every 10 yards of surface to be painted; and the quantity of white lead is reduced in proportion.

One pint of varnish will cover about 16 yards with 1 coat.

For paints for various purposes see Section XLII., 'Workshop Recipes.'

SPRAY PAINTING AND WHITEWASHING.

Tests carried out on paint-spraying machines have shown that spraying requires approximately 10 per cent. more paint than brushing when the surface being treated is an iron roof, while brushing requires approximately 200 per cent. more labour than spraying. On exterior brick walls, spraying requires 7 per cent. more paint than brushing, while brushing requires 109 per cent. more labour than spraying. On interior ceilings and walls of plaster, spraying requires 40 per cent. more paint than brushing, but gives a better hiding in one coat. On similar work, brushing requires 160 per cent. more labour than spraying, and gives a poorer hiding in one coat. Although it has not been possible as yet to arrive at a definite conclusion confirming the relative durability of spraying as compared with hand-painting, it is claimed that the investigations so far made tend to support the belief that the spraying work will last longer.

* Or ½ oz. of burnt white vitriol and 1½ oz. of litharge.

CONSTANTS OF LABOUR FOR PAINTING WORK.

		Hours.
Knotting	per yard super.	·05 painter.
Stopping	,,	·05 ,,
First or priming coat on wood	,,	·16 ,,
Second and following coats, each	,,	·14 ,,
First coat on iron	,,	·25 ,,
Second and following coats, each	,,	·22 ,,
Add if painting done from a ladder	,,	·10 ,,
Iron bars, fillets, etc., one coat	,,	·06 ,,
Sash squares, each side, first coat	per dozen	·50 ,,
,, ,, ,, second coat	,,	·40 ,,
Tarring, first coat on wood	per yard super.	·25 labourer.
,, second and following coats	,,	·20 ,,
,, first coat on iron	,,	·28 ,,
,, second ,,	,,	·21 ,,

(*J. T. Rea.*)

PUTTY.

Mix 50 lbs. of whiting, 4 lbs. of white lead, and 1 gallon of raw linseed oil, preferably in a suitable mill, though hand-mixing is quite satisfactory, if well done. To make the putty set more quickly, add 2 lbs. of 'patent dryer,' or replace part of the raw oil by 'boiled' linseed oil. The white lead may be omitted if desired.

See also p. 416.

Aluminium Paint.

Very few paints are really waterproof, and wooden objects in particular, if subjected to continuous moisture under the protection of ordinary paint, gradually become sodden and the paint tends to peel away; even the paint itself appears to absorb a certain amount of moisture to form a 'bloom' or discoloration of the surface of the paint. The nature of the thin metallic layer beneath the surface of aluminium paint makes it practically moisture proof, and this property is made use of for many purposes, but more particularly for the painting of wood exposed to the atmosphere. Tests have shown that on this account aluminium paint is the best undercoat for woodwork, even when other paints of different colours are to be superimposed. The addition of aluminium powder to ordinary paint increases its waterproofing qualities, but the best results are obtained with at least one coat of pure aluminium paint. This waterproofing quality is also made use of in painting the wooden patterns used in foundry work, as it prevents the patterns taking up moisture from the wet sand and thereby warping. The high covering or concealing power of aluminium paint makes it useful as a primer or undercoat, more particularly when painting over bituminous compositions or creosote compounds. Aluminium paint is mechanically stronger than ordinary paint. Being chemically inactive to the atmosphere, deterioration of the surface can only take place through a slow abrasion by rain and grit.

Distemper.

Distempers are wall-coverings which are intermediate in nature between paint and whitewash, being more adherent than the latter, but less impervious than the former. They are usually made of casein and alkali or lime, and are slightly cementitious in character, so that they 'set,' do not 'rub off,' and may be cleaned by washing with a sponge and water. Distempers may be in the form of a powder, or paste, which must be mixed with water (sometimes with a special 'petrifying liquid') so as to form a thick cream which is readily applied with a brush. Distempers may be obtained in various colours.

A good distemper produces a dull flat finish, but the surface may be stippled if desired. Most distempers are for interior work, but some of them can be used for outdoor work over brick, stucco, cement, stone, plaster, wood, iron, or other usual building material, provided a little pale boiled linseed-oil is added before thinning as above.

Whitewash.

Whitewash is a mixture of whiting, water, and size, of the consistency of cream, which is used for covering walls and ceilings. It will not stand rain or rubbing, but it is cheap and effective in situations where it will not be disturbed.

The *whiting* should be bought ready made; it is usually prepared by purifying natural chalk, but some other forms of calcium carbonate, recovered as a by-product from various industrial processes, are equally suitable.

The *size* is powdered or flake glue, but the term is sometimes applied to a jelly made by soaking such glue in water overnight, so as to cause it to swell, and it is then melted by adding hot water and heating gently. About 1 lb. of glue should make a gallon of size. It is usually more satisfactory to buy the glue and produce the size than to purchase the size ready made. The size is added to make the particles of whiting adhere to each other and to the wall or ceiling.

A very good whitewash contains 10 lbs. of whiting, 1 lb. of flake or powdered glue, and 1 gallon of water, but this is regarded by some users as being too 'rich' and unnecessarily expensive. In that case it is diluted by adding more water.

Limewash

is used for common walls or ceilings where for sanitary reasons a frequent application is necessary.

It will not stand rain, and adheres badly to very smooth or porous surfaces. It is cheap, and is made by slaking lump lime with about one-third of its weight of water, and then adding enough water to make a 'milk.' Hydrated lime is easier to use, and for most purposes is equally effective. The use of hot, freshly slaked lime is sometimes preferred, as it is supposed to have antiseptic and disinfectant properties. The proportions of lime and water are not important: about 5 lbs. of lime to 1 gallon are ample, and much less lime is often used. Some people consider that limewash is improved by adding 1 lb. of tallow (free from salt) to every bushel of lime.

Paperhanging.

Approximate Number of Pieces of English Wall-paper required for any Room.

(Allowance to be made for doors and windows.)

Measurement in Feet Round the Walls.	Height of Room, in Feet, from Skirting to Cornice.									
	6	7	8	9	10	11	12	13	14	15
	Number of Pieces Required.									
32	4	4	5	5	6	6	7	7	8	8
36	4	5	5	6	6	7	7	8	9	9
40	4	5	6	6	7	8	8	9	9	10
44	5	5	6	7	8	8	9	10	10	11
48	5	6	7	7	8	9	10	10	11	12
52	6	6	7	8	9	10	10	11	12	13
56	6	7	8	8	9	10	11	12	13	14
60	6	7	8	9	10	11	12	13	14	15
64	7	8	9	10	11	12	13	14	15	16
68	7	8	9	10	11	12	13	15	16	17
72	7	9	10	11	12	13	14	15	17	18
76	8	9	10	11	13	14	15	16	17	19
80	8	9	11	11	13	15	16	17	18	20
84	9	10	11	12	14	15	17	18	19	21
88	9	10	12	12	14	16	17	19	20	22
92	9	11	12	13	15	17	18	19	21	22
96	10	11	13	13	16	17	19	20	22	23
100	10	12	13	15	16	18	20	21	24	24

Paper for wall decoration is sold by the *piece*. A piece of English paper is supposed to be 12 yds. long and 20 ins. wide when hung; therefore each yard run will contain 5 ft. super, and each piece 60 ft. super; consequently the feet super of surface to be papered divided by 60 will give the number of pieces of paper required, any odd number of yards being taken as a piece. With common patterns 1 piece in 10 should be added to make up for waste in cutting and fitting the patterns in, whilst in large patterns 1 in 7 would not be too much. In all cases a liberal allowance should be made, as it is useful for repairs, the same pattern not being always procurable when required.

A gallon of paste will hang 5 pieces of English paper, a quart of flour being required for each gallon of paste; sometimes a little alum is used to increase its strength, but makes it harder and prevents it spreading so well. One gallon of boiling water to 2 lbs. of flour and 1 oz. of alum is a common proportion in making paste. The thinner the paste can be made, the better; and it should be well strained before using.

Before papering, the walls should be stopped, pumiced, and sized.

In paperhanging, the trimmed edges of the paper should face the light, to prevent any shadow marking the joints, and should be truly plumb; the pattern should accurately join.

In all cases of re-papering walls, the old paper should be stripped off and the walls scraped, washed, and stopped, previous to hanging the new paper. Papering-over old paper must never be allowed; many rooms which smell close and unhealthy owe it all to several thicknesses of old paper and paste decomposing on the walls. Disease and death have been clearly traced home to the omitting to strip off the old paper before re-papering.

Flock papers when old, instead of being stripped off, can be painted; they will absorb three or four coats of paint; but it produces a very good effect, from the embossed pattern standing out on the surface.

Plumbing

Plumbing, as the word implies, is the art of working in lead—the principal operations being the covering of roofs and flats, laying gutters, covering hips, ridges, and valleys, fixing water

pipes with all necessary taps, making and fixing certain cisterns and reservoirs, water-closets, washing appliances, etc. Plumbers also undertake fixing gas fittings and pipes related thereto (though this is really smith-work), and also work in sheet-zinc; they often fix rain-water pipes and sometimes drain-pipes, though these are really 'builder's work.'

The plumber's chief jointing material is solder, but in roofing its use should be avoided as far as possible, turned-over joints being far preferable.

Lead in gutters should turn up under slates, etc., from 7 ins. to 9 ins., according to pitch of roofs, and against walls from 5 ins. to 8 ins., and be covered with a flashing, or 'apron,' the upper edge of which is let into a mortar-joint a little above the gutter lead, the lower edge overlapping the latter at least 3 ins.

No sheet of lead should be laid in a greater length than 10 ft. or 12 ft. without a drip or vertical overlap at least 2 ins. deep, to allow of expansion.

Gutters should have a fall of at least 1 in. in 10 ft., better 1½ in. Flashings should have 3-in. laps, ridge and hip joints 4-in. laps, valley and roof joints 5-in. laps.

The thickness of the lead used for roofing should not be less than 7 lbs. per foot super for roofs, flats, gutters, bottoms of cisterns, and sinks; 6 lbs. per foot super for hips, ridges, small gutters, valleys, and sides of valleys; and 5 lbs. per foot super for flashings.

For soil pipes it is well to use lead weighing 8 lbs. per foot super.

Zinc Roofing.

The gauges of zinc recommended for roofing purposes are Nos. 13, 14, 15, and 16, the weights per sq. ft. of each gauge being respectively 16 ozs. 15 drs., 18 ozs. 12 drs., 21 ozs. 12 drs., and 24 ozs. 12 drs. The sheets usually measure 8 ft. × 3 ft.

Rain-water Pipes.

These are made of cast iron, generally in 6-ft. lengths. The round pipes vary in diameter from 1½ in. to 8 ins. The square pipes are made from 2½ ins. × 2 ins., up to 6 ins. × 6 ins. The 3-in. round pipe is a size largely used.

'Everite' Rain-water Pipes.

(*Asbestos Cement Building Products, Ltd., Trafford Park, Manchester.*)

Everite rain-water pipes (made of asbestos-cement), approved by the Ministry of Health for State-aided housing schemes, are stronger than cast iron and require no painting; their upkeep cost is insignificant. These consist of the following, all made to 2-in., 2½-in., 3-in., 3½-in., 4-in., 4½-in., 5-in., and 6-in. diameters:

Rain-water pipes; heads; shoes; bends or elbows, square or obtuse; plinth bends; offsets; swannecks; 'T' and 'Y' pieces; branch pieces, single and double.

'Everite' Eaves Gutters.

Everite rain-water gutters consist of the following, all made to 4-in., 4½-in., 5-in., and 6-in. diameters:

Half-round gutters; half-round nozzles, angles and 'T' pieces; half-round drop ends; half-round union clips; half-round stop ends.

Corrosion of Zinc and Lead Water Pipes.

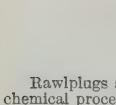
Zinc water pipes in contact with mortar containing gypsum become corroded and eaten through in places. The most active agent of destruction is sulphate of lime. Zinc is energetically attacked by mortar containing gypsum in the sand, but only feebly by mortar containing limestone sand alone, or with an addition of gypsum. Lead, on the other hand, is energetically attacked by limestone containing gypsum, but only feebly by gypsum in sand. Both metals are strongly attacked by pure cement, but the destructive action is lessened in progressive measure by the addition of sand. Hence zinc should be put in contact only with limestone (carbonate) mortar, and lead only with clean sand. Lead is also attacked by acid soils, and should be protected from these by means of drain pipes or troughs.

Wall Plugs.

Rawlplugs.

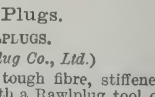
(*The Rawlplug Co., Ltd.*)

Rawlplugs are small patent tubes made of tough fibre, stiffened and treated by a special chemical process. A small hole is first made with a Rawlplug tool or a drill and the Rawlplug



Section showing plug in position ready for screw.

Fig. 9.



Screw entering, showing expansion of plug against sides of hole.

Fig. 10.

(which should be an exact fit) inserted. On turning a screw into the Rawlplug the latter expands

and grips the wall firmly, at the same time the screw is held in position by a perfect thread which is automatically formed in the fibre.

Figs. 8 and 9 show the arrangement.

The 'Metlex' Wall Plug.

(Metallic Wall Plugs, Ltd.)

The 'Metlex' Wall Plug is an all-metal wall plug for use with wood screws in very hard materials.

The arrangement, which consists of a metal wrap, is shown in fig. 11. One end, *a*, of the plug is coloured red to indicate that it is the outer end; at the opposite end, *b*, the longitudinal overlap of the wrap is depressed to facilitate its insertion into the plug hole. When the plug has been driven into the hole, this lip ensures a tight fit, and prevents it from rotating. The first turning motion of the screw enlarges the plug slightly, but powerfully, so as to provide the hole with what is virtually a solid metal lining, on which the screw now cuts its own thread, and develops a very strong grip.

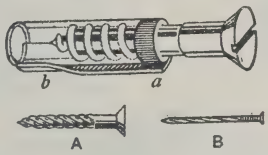


FIG. 11.

A and B show a 'Metlex' Drive Screw and a 'Metlex' Screw Nail. These can be driven into the plug by means of a hammer, no screw-driver being required. They may be hammered through wooden fixtures without previous drilling of the fixture.

Fixing Bolts.

(1)

A device for this purpose, fig. 12, comprises an expanding washer of curved and truncated conical form, with slots cut around its periphery, as shown at *A*. The washers are stamped cold from steel sheet, and as a result of the cold-working have considerable stiffness, while the slots around the periphery permit of easy expansion by drifting. With the aid of a drift and a few smart

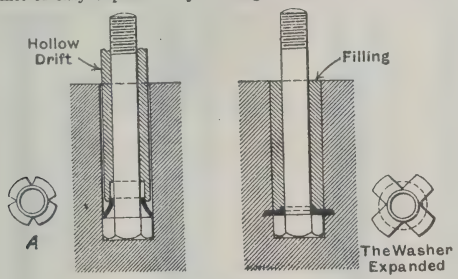


FIG. 12.

blows with a hammer the washer is easily expanded and flattened out to a sufficiently large diameter as to form a substantial shoulder that is embedded, as shown. The excess space in the bolt hole may be caulked with oakum, gaskin, old lead pipe, or yarn. The arrangement will sustain a load quite proportionate to the strength of the wall or floor in which it is fitted.

(J. T. Towlson, 'Machinery,' Nov. 8, 1923.)

(2)

The Rawlplug Bolt Anchor.

(The Rawlplug Co., Ltd.)

The 'Rawlplug Bolt Anchor' is constructed from specially chilled iron in combination with lead.

Fig. 13 shows the arrangement. The lower end, A, of the anchor consists of a truncated cone of chilled iron, attached to a cylinder of lead; the iron cone is divided into segments, which are shaped in such a way that the edges are sharp at their greatest diameter.

The method of use is, first to prepare a hole with the aid of a 'Rawlplug Boring Tool,' then (B) place a bolt in the hole, head foremost, place a thick iron washer on the bolt, pressing it snugly

against the bolt-head, and, finally, follow with the anchor, placed cone-shaped, head foremost, so that when in position it rests against the washer.

The space around the bolt is then caulked, which is carried out with the aid of the Rawlplug caulking tool and a heavy hammer.

The action of caulking has the effect of not only wedging the lead firmly against the bolt and the walls of the hole, but also of causing the hard iron cone to fracture at the segments and force the cutting edges into the wall of the hole.

The Rawlplug Bolt Anchor is made in sizes ranging from $\frac{3}{8}$ in. to $1\frac{1}{4}$ in., and the depth to which they should be fixed varies according to the kind of material in which the fixing is to be

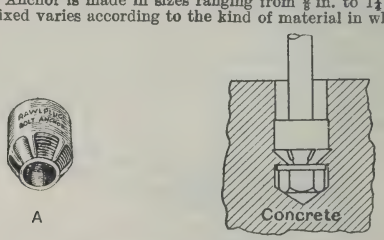


FIG. 13.

made. Should the material be hard and unyielding, a hole of lesser depth is required than when fixing in weak concrete. In good concrete for small sizes a depth of about 4 ins. is sufficient, whereas a depth of from 6 ins. to 9 ins. is recommended for the fixing of large size bolts. To facilitate caulking, and to enable this operation to be carried out more effectively, the lead cylinder of the larger size bolt anchors is furnished in two sections.

(3)

A (fig. 14) is an ordinary bolt, with head slightly tapered on both sides; B is a piece of ordinary gas barrel cut with a hack saw to the shape shown; C is the bolt in position inside the rube ready for fixing in position; and D is a 'jumper' for cutting the hole and is made out of a piece of the same barrel as B.

The teeth at F on D (as shown in the fig.) are V-shaped, roughly filed.

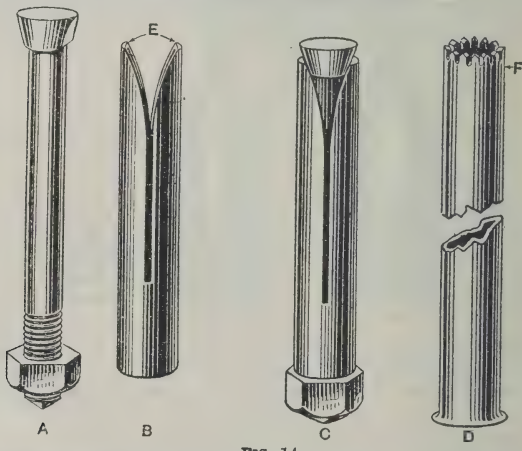


FIG. 14.

When the complete fitting C is placed inside the hole, and the work that it is desired to hold is inserted on the bolt underneath the nut, very great pressure can be exerted, as the tighter the nut is screwed down the further the jaws open at E (B) and the tighter the grip becomes.

Many thousand bolts with the fitting have stood the test of carrying loads under all conditions. At a large generating station the bolts, fixed in stone concrete and brickwork, have carried a 24-in. steam header subject to heat moisture and vibration.

(*London Association of Foremen Engineers' Magazine.*)

SECTION X

CLAYS AND ALLIED PRODUCTS USED BY ENGINEERS.

CLAYS AND EARTHS—PUDDLE—BRICKS—TILES—BLOCKS AND TERRA-COTTA—STONEWARE, SANITARY WARE, AND DRAIN PIPES — ELECTRICAL WARE — REFRACTORY MATERIALS — FIREBRICKS — FURNACE LININGS — RETORTS — CRUCIBLES — HEAT-INSULATING MATERIALS—FIRE-RESISTING CEMENTS.

(pp. 425–435)

Contributed by Alfred B. Searle
(Consulting Advisor on Clay Products).

HIGH GRADE REFRACTORIES

FROM our wide range of refractories we are in a position to supply the proper refractory to suit the requirements of most industries. Our range includes Silica bricks containing 95% Silica, Semi-Silica Bricks, ordinary Firebricks and also Firebricks of the High Alumina type containing over 43% Alumina. We can supply Silica Cement, Ganister, Fireclays and Refractory Cements to suit every purpose.

Our various works are of modern design and embody all the best developments and technique known to the refractories industry. Every precaution is taken at all stages in the manufacture to maintain a constant high standard of quality. Careful making and thoroughness of burn ensure accuracy of shape and size and constancy of volume at working temperatures.

It is our desire to co-operate with users to get the maximum efficiency and most economical results, and our representatives are always pleased to call and discuss problems.

We have a large and well-equipped laboratory, and the services of our technical staff are always available.

Write for our leaflets, quoting Department 'E.'

JOHN G. STEIN & CO. LTD.

BONNYBRIDGE - - SCOTLAND

Telegrams ; 'Stein' Bonnybridge.

THE LARGEST MAKERS IN THE BRITISH EMPIRE.

SECTION X

CLAYS AND ALLIED PRODUCTS USED BY ENGINEERS.

CLAYS AND EARTHS—PUDDLE—BRICKS—TILES—BLOCKS AND TERRA-COTTA—STONEWARE, SANITARY WARE, AND DRAIN PIPES — ELECTRICAL WARE — REFRACTORY MATERIALS —FIREBRICKS — FURNACE LININGS — RETORTS—CRUCIBLES — HEAT-INSULATING MATERIALS — FIRE-RESISTING CEMENTS.

Contributed by Alfred B. Searle
(Consulting Advisor on Clay Products).

CLAYS AND EARTHS.

For engineering purposes, clays and earths may be regarded as (1) substances which must be excavated and transported from their present situation; (2) the raw materials from which earthwork, puddle, etc., are made; and (3) the raw materials used in the production of bricks, drain pipes and many other articles employed by engineers. The term 'clay' in the broadest possible sense is applied to a material which is tenacious and plastic when wet. Such clays may contain only a small proportion of true clay, together with a large proportion of inert material such as sand. For many engineering purposes a perfectly pure clay would be useless, as it might be of so fine a texture and so highly plastic that it would shrink excessively on drying, and it would be extremely difficult to make into bricks or other desired articles. When made, these articles would be deficient in certain important properties, such as resistance to crushing, impermeability to water, etc.

The term 'earth' is usually applied to a friable material partaking of the general properties of soil, though not necessarily of any agricultural value. It usually consists chiefly of inert material, the particles of which are united by a very variable proportion of clay, humus, or other adhesive. Hence the 'earths' are often regarded as 'clays' of low plasticity, but both these terms are used very loosely by many engineers and no really satisfactory definition of them has yet been formulated.

In selecting a 'clay' or 'earth' for any given purpose, the engineer must, therefore, decide what properties are essential to his requirements and must endeavour to find a natural material possessing these properties, or to prepare a mixture of two or more materials which suit his purpose.

The chemical and physical properties of 'clays' depend chiefly on the composition and plasticity of the material, it being obvious that a 'clay' containing 50 per cent. of quartz-grains must behave very differently from one which consists wholly of 'true clay.' The *plasticity* of clays varies with the nature of the clay and the amount of water present. Dry clays are friable and devoid of plasticity; on mixing clay with an equal weight of water a liquid slurry or slip is formed which is not really plastic, but becomes so when a suitable proportion of the water has been removed by evaporation or otherwise. No generally accepted explanation of the cause of plasticity has been published; this property may be due to the presence of colloidal particles which may be derived from the clay itself or from impurities which have become associated with it. The purest clays—e.g. 'china clay'—are very feebly plastic; the highly plastic ball clays which are supposed to have been derived from them are rich in carbonaceous matter, to which their plasticity is sometimes attributed. The clays commonly used for bricks, pipes, terra-cotta, etc., are never pure; they are naturally occurring complex mixtures possessing the requisite properties to make them useful for specific purposes.

It is a great mistake to suppose that any plastic 'clay' can be made into bricks, or that any 'system' of manufacture is equally suitable for all 'clays.' On the contrary, the manufacture

of clay products is in reality so complex (notwithstanding its apparent simplicity) and the ordinary manufacturer depends on the simultaneous occurrence of so many fortuitous circumstances that it is not surprising that many firms fail to manufacture clay goods of uniform quality.

BRICKS.

Many kinds of bricks are in use. The most important are: *Common Bricks*, used for ordinary building purposes; *Facing Bricks*, used for external work where appearance is important; *Paviours*, used for yards, footpaths, and other locations where a hard-wearing and impervious brick is required (for *Road Bricks*, see p. 739); *Blue Bricks*, which are similar in many respects to 'paviours' and are employed where exceptional strength and impermeability are important; *Stock Bricks*, which are practically the same as 'common bricks,' but have usually been burned in a clamp instead of in a kiln; they have proved exceptionally resistant to the climatic conditions of London, but are seldom, if ever, used in other large industrial centres (in some localities the term 'stock bricks' relates to 'facing bricks').

Bricks are also known according to the manner of their production. Thus, *Malms* contain a large proportion of chalk, either naturally or artificially mixed with the clay; *Wire-cut Bricks* are made by cutting up a column of clay issuing from a machine, by means of wires; *Pressed Bricks* have been made in presses; *Semi-plastic* or *Stiff Plastic Bricks* have been made from a paste which is stiffer and less plastic than that used for hand-moulded or wire-cut bricks; *Semi-dry Bricks* are made by compressing the powdered clay in powerful presses instead of first converting it into a paste.* *Place Bricks*, *Grizzles*, and *Chuffs* are inferior bricks which are soft through insufficient burning; *Crozzles* are excessively hard, misshapen bricks which have been partially melted and overheated. *Rubbers* and *Cutters* are soft, sandy bricks.

Glazed Bricks are used where a smooth and impervious surface is required (see 'Glazed Tiles').

Sand-lime Bricks † are made of sand mixed with about 6 per cent. of lime and matured in an atmosphere of steam under pressure. They have never become very popular, chiefly on account of the abundance of clay bricks and stone, but in localities where sand is plentiful and clay scarce they appear to be an excellent substitute for clay bricks.

Several attempts have been made by various Government officials to extend the use of sand-lime bricks, but such efforts have proved abortive, as such bricks are much more difficult to make at a profit, and there is a larger proportion of unsaleable waste than with clay bricks. Some of the statements respecting the speed of production and cost for manufacture of sand-lime bricks, published in certain Government publications, are by no means correct if applied to average conditions.

Cement-concrete Bricks.†—These bricks have attracted attention, but are not so suitable as larger blocks of concrete for general purposes, as the cost of production is relatively high. Such bricks are made in precisely the same manner as concrete blocks—i.e. from a mixture of Portland cement with six times its volume of one of the following aggregates:—sand, destructor, furnace-, or boiler-clinker, ashes, quarry waste, and broken bricks or stone. The use of these bricks appears to be limited in practice to districts where clay bricks cannot be obtained at a reasonable price, and even then concrete blocks or mass concrete would be much cheaper, as the manufacture of such bricks is not nearly so simple as a description of the process may imply.

Engineering Bricks is the term usually applied to bricks of an exceptionally dense and vitreous character, of which the best known are the *Staffordshire* blue bricks. These owe their unusual strength and impermeability to the fact that they have been burned under such conditions that the particles composing them are united very firmly by means of a film of clinker surrounding each. For purposes where a somewhat weaker brick has ample strength, well-made *pressed* or *stiff plastic* bricks are usually satisfactory and much less costly, but they are more porous and therefore less suitable for supporting heavy masses of material on a boggy or damp subsoil. Bricks for *boilers* should be accurate in size and shape so as to enable the thinnest possible joints to be used. In order to prevent undue leakage they should be glazed or painted with tar or other impervious material. For boiler *blocks* see Section XXX, Part II.

For *firebozes*, *furnaces*, etc., firebricks and blocks are employed.

Colliery Bricks are usually of very poor quality, due to their being made from unsuitable material, or from good clay which has been spoiled by mixing it with rubbish.

There is serious need for great improvements in the manufacture of bricks in many collieries, and those firms who have called in a competent and independent advisor have found that it has paid them well to follow his advice. The services of unemployed foremen, machinery manufacturers, and kiln builders, in this connection, are seldom satisfactory, as their recommendations are unduly biassed.

---

* For further details respecting the manufacture of bricks see 'Modern Brickmaking,' by A. B. Searle (Scott, Greenwood & Son, London).

† See 'Bricks and Artificial Stones of Non-Plastic Materials,' by A. B. Searle (J. & A. Churchill, London).

Crushing Strength of Bricks.

Opinions differ greatly as to the respective values of different kinds of bricks, and as the crushing strength of bricks taken from the same works at different times varies 25 per cent. and even more, it is clear that no comparison of crushing strength tests can be of much value in deciding between various kinds of bricks. The chief use of the tests is in fixing a minimum crushing strength which shall ensure the safety of the structure, but as the strength of even the commonest bricks is much greater than that of average brickwork (the mortar being usually weak), it is difficult to stipulate any satisfactory minimum strength for bricks.

The following figures are the averages of a large number of bricks, but it must be clearly understood that bricks from any given works may be either much stronger or much weaker than the ones mentioned.

Description.	Crushing Commences.	Ultimate Crushing.
	Tons per square foot.	Tons per square foot.
Common stocks	40–60	80–120
London best stocks	80–140	100–180
Gault white, wire-cut	40–120	135–200
Aylesford, red pressed	70	140
Manchester, common red	75	120
,, wire-cut	85	260
Accrington, plastic *	690	1005
Leicester, wire-cut	115–250	230–340
Rugby, common red	160	190
Fletton	125–200	170–250
Notts.	60–300	120–400
Ruabon	360	—
Staffordshire blue bricks	250–450	400–750
Terra-cotta blocks	—	100–260
Glazed bricks	70–165	165–175
Stourbridge fire bricks	90–160	110–210
Sand-lime bricks	150–200	180–230

The crushing strength of brickwork is shown on pp. 175, 376.

Sizes of Bricks.

In many countries there is no official standard; the following are average sizes:—

Standard Sizes of Bricks in different Countries.†

Country.	Length, inches.	Breadth, inches.	Thickness, inches.
Great Britain	9	$4\frac{3}{4}$	$2\frac{5}{8}$ ‡
Germany	10	$4\frac{2}{4}$	$2\frac{5}{8}$
Austria	$11\frac{5}{8}$	$5\frac{5}{8}$	$2\frac{5}{8}$
France	$8\frac{1}{2}$	$4\frac{1}{8}$	$2\frac{3}{8}$
Italy	9	$5\frac{1}{2}$	$2\frac{3}{8}$
Holland and Belgium	$9\frac{5}{8}$	$4\frac{3}{4}$	$2\frac{3}{8}$
Switzerland	10	$4\frac{3}{4}$	$2\frac{3}{8}$
Russia (varies greatly)	$11\frac{1}{2}$	$5\frac{1}{8}$	$2\frac{3}{8}$
United States	8	4	$2\frac{3}{8}$

Specification for Bricks.

The sizes of bricks vary in different parts of Great Britain, but there are many advantages in adopting the Specification of the Royal Institute of British Architects as follows:

The Royal Institute of British Architects' standard size lies between—length, $8\frac{7}{8}$ ins. and 9 ins.; breadth, $4\frac{5}{16}$ ins. and $4\frac{3}{8}$ ins.; thickness, $2\frac{3}{8}$ ins. and $2\frac{11}{16}$ ins.

The method of measuring the bricks as set out in the standard specification is as follows:

(1) The length of the brick should be double the width plus the thickness of one vertical joint.

(2) Brickwork should measure four courses of bricks and four joints to a foot.

* These figures supplied by The Accrington Brick and Tile Co., 1926.

† See also p. 366.

‡ In Scotland, $2\frac{1}{16}$ in. or 3 in.

Joints should be ¼ in. thick, and an extra $\frac{1}{10}$th, making $\frac{5}{10}$th, for the bed joints to cover irregularities in the bricks. This gives a standard length of 9¼ ins. centre to centre of joints.

The bricks, laid dry, to be measured in the following manner:

A. Eight stretchers laid square end and splay end in contact in a straight line, to measure 72 ins.

B. Eight headers laid side to side, frog upwards, in a straight line, to measure 35 ins.

C. Eight bricks, the first brick frog downwards and then alternately frog to frog and back to back, to measure 21½ ins.

A margin of 1 in. less will be allowed as to A and ½ in. less as to B and C.

TESTS OF BRICKS.

All dry bricks should *ring clearly* when two, held in the hands, are struck sharply together. If a dull sound is emitted the bricks are either saturated with water or they have not been sufficiently well burned.

The *porosity* of bricks is usually tested by soaking them in water, wiping dry and noting the increase in weight. Whilst many good building bricks will absorb one-fifth of their weight of water, this is no criterion of the value from an engineering standpoint; on the contrary, engineers usually prefer bricks with low porosity on account of their greater strength.

Rattler Test.—In the United States, and to a much smaller extent in this country, engineering bricks are tested for toughness and resistance to abrasion by placing a weighed quantity of them in a cylinder with steel balls and rotating the cylinder a thousand times in about half-an-hour.* The contents of the cylinder are then removed, all pieces of brick larger than 1 in. diameter are picked out and reweighed. The loss in weight is regarded as a measure of the inferiority of the bricks. The rattler test is particularly applicable to bricks used for road making (Section XV.) and for lining furnaces.

Lime-Sand Bricks.

Bricks made of sand with about 8 per cent. of lime and hardened by exposure to steam under pressure have long been used in Germany and in this country, though they are by no means well known. They are white, cream, or grey in colour, and have a crushing strength of 200 tons per sq. ft. They are, when properly made, as durable as good sandstone, but poorly made bricks are not suitable for damp situations. The cost is about the same as clay bricks.

Concrete Bricks.

Bricks made of 6 parts of sand and 1 part of Portland cement are cheap to produce when the sand is on the building site. Ordinarily they cost more than clay bricks. They are made in plain moulds, and must be kept wet for a fortnight. Concrete bricks are as durable as sandstone; they are suitably made in isolated places where sand is available and carriage is costly.

TILES.

Roofing Tiles are made of materials similar to those used for bricks, but of a somewhat finer texture. No official specifications have been published, but it is recognised that roofing tiles should be light, tough, and ring clearly when struck, be impervious to water in the sense that even under a continuous supply of water no dripping will occur from the underside of the tile when placed at the same angle as when in use on a roof. It is desirable that tiles should be moderately porous, so as to avoid undue condensation of their underfaces. (See also p. 413.)

Glazed Tiles are used partly for decorative purposes, but chiefly for sanitary reasons. The glaze should be sufficiently hard to resist all reasonable abrasion and it should not craze (forming hair-like cracks) after prolonged use. For chemical laboratories and other special purposes a felspathic glaze burned at about 1300° C. must be used in order that it may be sufficiently resistant to corrosive fluids, etc. (see 'Stoneware,' p. 429).

Paving Tiles, like paviour bricks, should be very hard and resistant to abrasion. To prevent animals slipping on them, the surface may be channelled with small grooves forming a diamond pattern.

Paving tiles are usually made of the same material and in the same manner as paving or engineering bricks.

BLOCKS AND TERRA-COTTA.

The chief requirements of *blocks* made of clay and allied materials are accuracy in shape and size, and ample strength to carry the superimposed loads. For external work, appearance is also important.

---

* The weight of the balls, the number of revolutions, and the size of the rejected pieces, differ with various engineers.

Terra-cotta is largely used in conjunction with structural steel work, the latter taking the bulk of the stresses and the terra-cotta forming a sort of filler and cover of great resistance to climatic conditions and of pleasing appearance. There is no necessity to impose any conditions as to its strength, as in a properly designed structure it would not be exposed to any severe stresses.

Blocks for boilers are made of special shapes. They must be sufficiently heat-resisting not to be affected by the conditions to which they are subject when in use. They must be very accurate in shape and size or there will be a serious leakage into the side flues. (Section XXX., Part II.)

HOLLOW BLOCKS.

For many years in France and in some of the Colonies, but more recently in this country, the advantages which hollow blocks possess over building bricks have been realised. Such blocks are only one-third of the weight of the same number of bricks; they can be laid about four times as rapidly, and they are of ample strength for all purposes for which ordinary bricks are used. In addition, they automatically form a cavity wall, with all its advantages of warmth

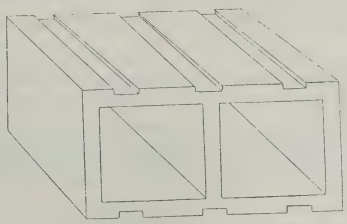


FIG. 1.

and soundproofness. Their great strength is really due to the fact that the bricks in a solid mass of brickwork have about ten times the crushing strength of the brickwork as a whole. Consequently a large proportion of the solid interior is quite useless and only adds to the cost of material, carriage, and labour; these costs are saved when hollow blocks are used.

Hollow building blocks are made in various shapes and sizes, according to the purposes for which they are to be used, and numerous patents have been granted in connection with them.

The *clays* used for making hollow blocks are the same as those employed in the manufacture of terra-cotta and roofing tiles. Such clays must vitrify sufficiently to form a strong, weather-resisting mass when fired. They must retain their shape perfectly at any temperature to which they may be exposed in the kiln, and whenever possible they should have a pleasing colour so as to avoid the necessity of 'rough casting.' Suitable clays and shales are found in various parts of the country, particularly in the Midlands. The lower grades of fireclays found in the various coalfields are also suitable, provided their buff colour is not regarded as objectionable.

The *process* of manufacture consists in preparing a clay paste of suitable consistency and extruding it through a die or mouthpiece, the interior of which is fitted with a number of cores, corresponding to the hollow portions of the blocks, fig. 1. The extruded column of clay is then cut by wires into suitable lengths, and the resulting blocks are dried and burned like bricks or architectural terra-cotta.

No standard specifications for hollow blocks have as yet been issued, but it is obvious that if the crushing strength of such blocks when they are laid in the same position as that in which they will be used, is greater than the normal strength of brickwork, such blocks will have ample strength. In most cases the blocks themselves have more than twice the strength of normal brickwork.

FOSALSIL.

This material (see p. 435) is used for the purpose of hollow blocks suitable for partitions; it possesses great mechanical strength and high fire resistance. It is supplied by *Messrs. J. H. Sankey & Son, Ltd.*, 7 & 8 Norfolk Street, Strand, London, W.C. 2.

STONEWARE, SANITARY WARE, AND DRAIN PIPES.

Stoneware, as used by engineers, is chiefly employed in chemical works in the construction of appliances for dealing with corrosive fluids both at low and high temperatures.

The conditions under which it is used are exceptionally severe and yet are so varied that no widely acceptable specifications have been found possible. For chemical and industrial purposes, stoneware should consist of an impervious body of great crushing strength and with a minimum sensitiveness to sudden changes in temperature. The ideal is closely approached by the porcelain crucibles used in chemical laboratories, but this is too costly and too fragile

for most commercial purposes and various coarser substitutes are therefore employed. To a very large extent the manufacturers of stoneware appliances may be relied upon to supply a satisfactory product if they are fully informed as to the conditions to which it will be subjected. Where troubles arise, they are often due to an engineer requiring a piece of stoneware of a design or shape which is incompatible with the nature of the material used.

Sanitary Ware is not subjected to the same conditions as stoneware, and may therefore consist of a porous body covered with a hard felspathic glaze, though an impervious body is preferable where expense is no object. It is highly important that the glaze should be of a suitable character, earthenware and lead glazes seldom being sufficiently durable. In designing sanitary ware it is desirable to work in close co-operation with the manufacturer, as in this way much delay in production and a considerable saving in cost may be effected, and the ware is likely to be more satisfactory. As far as possible, joints and sharp arrises or corners should be avoided.

Drain Pipes are frequently made of a porous body which is covered with a salt-glaze, but for the best work a vitreous or impervious body should be used. If the glaze is defective or damaged an impervious body will maintain the efficiency of the pipes, but a porous body may permit seepage of a disagreeable or even harmful character.

For 'The British Standard Specification for Salt-Glazed Pipes' see Section XXVI.

ELECTRICAL WARE.

Electrical Ware is chiefly used in the form of conduits, insulators, and fittings. The conduits are usually made of coarse stoneware and are similar in many respects to salt-glazed drain pipes (*supra*).

Other electrical ware is made of fine stoneware or porcelain. Insulators must have a fine and uniform texture and their fractured surface should be free from holes and fissures; they must be able to resist the passage of electric currents of the desired voltage without appreciable leakage, and should be tested at a considerably higher voltage to ensure this.

REFRACTORY MATERIALS.*

The term '*Refractory*' is applied to various heat-resisting materials, such as firebricks, crucibles, and furnace linings. There is no official definition of this term, but it is becoming increasingly customary to regard 1580° C. (2876° F.) as a minimum temperature, materials which suffer deformation when heated rather slowly to this temperature being excluded from what are generally regarded as refractory materials.

As the rate of heating has an important influence on the temperature at which deformation occurs, it is necessary, for comparative purposes, to specify the conditions of heating.

MELTING POINTS OF REFRACTORY MATERIALS.

	°C.		°C.
Alumina (pure)	2010†	Ganister	1700 to 1800
Bauxite	1820†	Kaolin	1835 to 1740†
Bauxite brick	1565 to 1785†	Magnesia brick	2165
Bauxite clay	1795†	Silica—Pure	1750†
Chromite	2180†	Pure (fine grains)	1650
Chromite brick	2050†	Pure with 14·5 per cent. alumina	1690
Fireclay brick—Highest grade	1843	Pure with 63·0 per cent. alumina	1790
General grade	1680 (average)	Brick	1700 to 1750
General grade (average of 41 samples)†	1649		

The term 'melting point' is misleading when applied to refractory materials, as these are seldom, if ever, melted, and the term is generally used to indicate the temperature at which a small pyramid or 'cone' of the material bends over until its apex is on a level with its base, or, alternatively, at which the angular edges of the pyramid begin to lose their angularity.

FIREBRICKS.*

Fireclay bricks are made of clays derived chiefly from the coal measures. These clays are crushed to powder mixed with water and made into bricks either by hand-moulding or machines of the stiff-plastic or semi-dry type; the bricks are then dried and are afterwards burned in kilns. The temperature attained in burning should be at least as high as that to which the bricks will be subjected when in use, as otherwise shrinkage and other defects may occur. It is seldom that bricks burned at such high temperatures are obtained unless they are definitely specified; and for some purposes, such as lining steel-melting furnaces, users must be content with bricks fired at a lower temperature than that at which they will be employed. The highest temperature ordinarily attained in burning firebricks is about 1350° C.

---

* See also *Refractory Materials, their Manufacture and Uses* (2nd edn.), by A. B. Searle (Griffin & Co., Ltd., London).

† Kanolt. The temperatures are those at which the materials were first seen to flow easily.

Firebricks are also made of silica (in which case they are known as *Ganister* or *Silica Bricks*), alumina (*Bauxite* or *Semi-bauxite Bricks*), magnesia (*Magnesia* or *Magnesite Bricks*), chrome iron ore (*Chromite Bricks*), sillimanite (*Sillimanite Bricks*), zirconia (*Zirconia Bricks*).

Firebricks made of clay or silica are acid in character and are readily attacked by the limestone used as a flux in steel furnaces.

The materials of which the bricks are made may be classified into three groups:

(*a*) Aluminous clays.
(*b*) Siliceous clays.
(*c*) Silica rock (sometimes erroneously termed 'silica clays').

The *Aluminous clays* include the Stourbridge, Yorkshire, Scotch, Welsh, and other Coal Measure fireclays, which contain, on an average, about 65 per cent. of silica. The *Siliceous clays* include those which contain 80-92 per cent. of silica, such as Ewell clay. *Silica rock* contains more than 92 per cent. of silica, such as Dinas rock and ganister. The composition of some of these materials is shown in the following table*:

	Stourbridge Clay.	Ewell Clay.	Dinas Clay.	Black Ganister.†
Silica	64·62	87.57	97.60	98·5
Alumina	21·65	3·60	0·50	0·3
Ferric oxide	1·48	4·80	1·50	1·3
Lime	1·88	0·74	0·20	0·2
Magnesia	0·62	0·41	—	—
Potash	—	0·69	0·09	trace
Soda	—	0·10	0·03	trace
Moisture	9·62	—	—	—
Combined water	—	2·09	—	—

Ordinary fireclays contain 58 to 75 per cent. of silica, 25 to 36 per cent. of alumina, and ½ to 2 per cent. of iron oxide. Good silica bricks contain upwards of 95 per cent. of silica, little or no alumina, and under 2 per cent. of lime. Ganister bricks contain about 90 per cent. of silica, 5 per cent. of alumina, and less than 2 per cent. of iron oxide. Magnesia bricks consist almost wholly of magnesia, unless about 5 per cent. of clay or iron oxide has been used to bind the particles together.

Basic Bricks are made of magnesia, whilst *Neutral Bricks* are made of bauxite, sillimanite, zirconia, chromite, or carbon; the two latter are chiefly used as an intermediate course between acid and basic bricks in furnace linings. The materials used for silica and neutral firebricks usually need a 'binder' such as fireclay or lime, but the clays are self-binding. Firebricks made of sillimanite appear to be specially suitable for use at very high temperatures. As natural sillimanite ($Al_2O_3.SiO_2$) is a rare mineral which has to be imported from Northern India, an artificial material made by heating fireclay and bauxite in a blast-furnace is frequently used instead; the artificial product is not true sillimanite, but more closely resembles millite ($3Al_2O_3.2SiO_2$). Those made of zirconia have proved disappointing—chiefly on account of the impurities in the zirconia used.

The impurities in fireclays, i.e. those constituents other than the silica and alumina, have a most harmful effect on the refractoriness of the material owing to their low fusing point. These constituents should, if possible, never be allowed to exceed more than 4 to 5 per cent. Much must, of course, depend upon the type of clay, and it is possible to point to certain substances containing so much as 4 per cent. of iron alone, and yet they are highly refractory. Green clay when fired is subject to considerable shrinkage, and it is for this reason that chamotte, or 'grog,' is added in varying proportions before the article is placed in the kiln. This 'grog' is usually prepared from a clay which originated in the same seam as the clay with which it is mixed. It should be prepared specially for the purpose, and should not consist of the crushings from defective and broken articles which have passed through the kiln. The quantity of 'grog' added varies, of course, with the material required and the kind of clay made use of. For firebricks and smaller articles it is usually added in a proportion of 15 to 20 per cent. For larger articles, such as retorts, the amount is increased to 30 to 40 per cent.

The deformation or softening points of firebricks vary considerably. The following are fairly typical:—

Fireclay bricks, 1610°–1770° C.; Silica bricks, 1670°–1790° C.; Magnesia bricks, 1730°–2000° C.

* Analyses of these and other heat-resisting materials will be found in *Refractory Materials*, by A. B. Searle (Griffin & Co., Ltd., London).

† The Dinas material is not a true clay, but a silica rock composed of almost pure quartz; ganister is similar, but consists of much smaller particles.

Specification for Fireclay Bricks.

The following paragraphs contain the salient features of the Specifications for Firebricks issued by the Institution of Gas Engineers (1911, revised 1912).

(1) *Refractoriness.*—Two grades of material are covered by the specification :—

(*i*) The material which shows no sign of fusion when heated to a temperature of not less than Seger Cone 30 (about 1670°C.).

(*ii*) Material which shows no sign of fusion when heated to a temperature of not less than Seger Cone 26 (about 1580°C.).

(2) *Surfaces and Texture.*—The material shall be evenly burnt throughout and the texture regular, containing no holes or flaws. All surfaces shall be reasonably true and free from flaws or winding.

(3) *Contraction or Expansion.*—A test piece, when heated to a temperature of Seger Cone 14 for two hours, shall not show more than the following linear contraction or expansion :—

No. 1 grade, 1 per cent. ; No. 2 grade, 1·25 per cent.

After the test temperature has been reached, the furnace shall be maintained constant at that temperature throughout the testing period.

The test piece shall be 4½ inches long and 4½ inches wide, the ends being ground flat, and the contraction being measured by means of vernier calipers reading to 0·1 mm., a suitable mark being made on the test piece, so that the calipers may be placed in the same position before and after firing.

(4) *Variations from Specified Measurements.*—In the case of ordinary bricks, 9 inches by 4½ inches by 3 inches or 2½ inches thick, there shall not be more than ± 1½ per cent. variation in length, nor more than ± 2½ per cent. variation in width or thickness, and in all cases the bricks shall work out their own bond, with not more than ⅛ inch allowance for joint. In the case of special bricks, blocks or tiles, there shall not be more than ± 2 per cent. variation from any of the specified dimensions.

(5) *Crushing Strength.*—The material shall be capable of withstanding a crushing strain of not less than 1,800 lbs. per square inch when applied to whole bricks placed with their long side vertical between the jaws of the machine, giving a vertical thrust.

Specification for Silica Bricks.

(Institution of Gas Engineers, 1912.)

The material covered by this specification is divided into two classes :—

(*i*) That containing 92 per cent. and upwards of silica, and hereinafter called 'silica' material.

(*ii*) That containing from 80 to 92 per cent. of silica, and hereinafter called 'siliceous.'

(1) *Refractoriness.*—Test pieces of the material shall show no sign of fusion when heated to the following temperatures :—

'Silica' material not less than Seger Cone 32 (about 1710°C.).

'Siliceous' material not less than Seger Cone 29 (about 1650°C.).

The test shall be carried out in an oxidising atmosphere, the temperature of the furnace being increased at the rate of about 50°C. per five minutes.

(2) *Surface and Texture.*—The material should be evenly burned throughout, and the texture regular, with no holes or flaws. All surfaces shall be reasonably true and free from flaws or winding.

(3) *Contraction and Expansion.*—A test piece, when heated in a gas muffle or electric furnace to a temperature of Seger Cone 14 for two hours, shall not show on cooling more than 0·75 per cent. linear contraction or expansion. The test piece shall be at least 4½ inches long by 4½ inches wide, the ends being ground flat, and the contraction or expansion measured by means of vernier calipers reading to 0·1 mm., a suitable mark being made on the test piece, so that the calipers may be placed in the same position before and after firing. The test samples shall not be taken out of the middle of a brick or tile, but shall be representative of the whole.

(4) *Variations from Measurements.*—In the case of ordinary bricks, 9 inches by 4½ inches by 3 inches or 2½ inches thick, there shall not be more than ± 1½ per cent. variation in length, nor more than ± 2½ per cent. variation in width or thickness ; and in all cases the bricks shall work out their own bond. Special bricks, blocks, or tiles shall not have more than ± 2 per cent. variation from any of the specified dimensions.

Specifications for Refractory Mortar or Cement.

(Institution of Gas Engineers, 1911; revised 1912.)

Cementing clay or fireclay mortar shall be machine-ground, and, at the discretion of the manufacturer, may contain a suitable percentage of fine grog ; but in all cases the cement-clay shall be quite suitable for the purpose of binding together the bricks, blocks or tiles for which it is supplied, and shall be capable of withstanding the same test for refractoriness.

This specification applies equally to cement or mortar used for bricks made of either fireclay or silica.

Firebricks used in Gasworks.

The firebricks used in gasworks should conform to the specifications of the Institution of Gas Engineers (p. 432). They are usually, but by no means invariably, made of the materials mentioned on p. 431.

BRICKS OR BLOCKS FOR FURNACE LININGS.*

The bricks or blocks used for lining furnaces should be selected according to the conditions to which they will be subjected. It is almost impossible to obtain a material which is wholly efficient in every respect, and some compromise is essential. Thus, great resistance to abrasion can only be obtained by increasing the amount of vitreous material and thereby lowering the refractoriness of the bricks. Answers to the following questions will usually provide most of the information required :—

(*a*) What is the highest temperature to which the brick will be exposed (excessive or only moderate), and will it be constant or fluctuating? (*b*) Will the material come into contact with any flame, and will the latter be of an 'oxidising' or 'reducing' character? (*c*) Is mechanical strength required? (*d*) Is heat to be conducted or retained? (*e*) Is the brick expected to withstand sudden and repeated changes in temperature? (*f*) Will the brick come into contact with fluid slags or glass, or with flue-dust, cement or other fluxes? (*g*) Must the brick resist blows, shocks and other unavoidable strains? (*h*) What is the nature of the fuel used and is it fed by hand or mechanically? (*i*) Is the electrical conductivity important?

To meet (*a*), refractoriness is of much importance, but, in most cases, other factors deserve even greater attention. For (*b*) and (*c*) the texture of the surface of the brick must be right or the brick will not be durable, no matter how great its refractoriness. If mechanical strength is needed, the brick cannot be as refractory as otherwise, for great strength is dependent on the presence of a large amount of bond, which is necessarily less refractory than the particles to be bound together. For (*d*), it will be found that the conduction of heat depends chiefly on the density of the material if made of clay or silica and on its chemical composition if made of graphite, plumbago or other carbonaceous material. Graphite, fused alumina and zirconia are good heat conductors, whilst clay, silica and magnesia are good heat retainers. For (*e*) a lean, coarse and porous material should be used, whilst for (*f*) a fat, close material is required, i.e. exactly opposite to that for (*e*), and for (*g*) a hard-burned material which has not been heated more than once is preferable; broken bricks and other refractory articles should not be used as grog in this case, and the mass should not be either too lean or too fine-grained.

Consideration of the above points will show that there is some difficulty in deciding upon the correct description of the material for use in the ordinary type of industrial furnace. Where bricks of high refractoriness are required to withstand the temperature employed, heat must be retained and radiation reduced to a minimum, though most of these requirements are satisfied by using an open-grained, porous material. Still greater difficulty arises in dealing with furnaces in which fluxes are brought into contact with the firebricks, and in these the material must be close-grained, even if this is accompanied by the loss of some heat resistance.

Where great resistance to fluxes (flue-dust, lime and other bases) is required, the refractoriness of the brick is of secondary importance to the closeness of texture. Under such conditions, a porous, highly refractory brick will be far less durable than a dense brick of much lower fusing point. The fluxes act less rapidly on a comparatively smooth, dense surface than on a more porous one, so that the 'life' of the former brick will be longer than the latter, notwithstanding the fact that its actual heat resistance may be lower. For this reason, many fireclays which are of No. 2 quality as regards heat resistance are preferable for use in some furnaces and for flue work generally.

In some heating appliances, the draught and the atmosphere to which the refractory material is exposed have a far greater influence on the durability of the latter than is generally supposed. Unless properly managed, such furnaces may cause firebricks to be destroyed in a few weeks, whereas under proper conditions of draught they would last almost as many years. This is a matter of which firebrick manufacturers have no special knowledge and most builders and users of furnaces are equally ignorant. It is, in fact, one which can only be properly investigated by a specialist in refractory materials.

RAMMED FURNACE LININGS.

Furnaces are sometimes lined with a stiff paste which is rammed into position. The composition of such rammed linings varies with the purpose for which the furnace is used; a well-known material consists of ganister to which a little lime has been added and the mixture then made into a very stiff paste with water.

Various kinds of sand, with or without sufficient clay or other binding agent to make the mass adhesive, are largely used for lining and repairing furnaces. (See also 'Refractory Cements,' p. 435.)

* For a much fuller description of the properties required in bricks and blocks used for various furnaces, kilns, etc., see *Refractory Materials*, by A. B. Searle (Griffin & Co., Ltd., London).

GAS RETORTS.

The chief forms of gas retorts are briefly described in Section XXXVII. They are usually made of a mixture of fireclay and 'grog,' the latter consisting of fireclay which has previously been burned and reduced to a very coarse powder. The grog should not consist of particles of uniform size, but should be graded so as to produce a material capable of withstanding sudden changes in temperature. The proportions of grog and clay must be adapted to the nature of the clay and the size and shape of the retort. The retorts require to be dried very slowly and carefully, and to be burned at a sufficiently high temperature to prevent them shrinking unduly when in use.

The following paragraphs contain the salient features of the Specification for Gas Retorts issued by the Institution of Gas Engineers in 1911, revised 1912.

SPECIFICATION FOR GAS RETORTS.

(Institution of Gas Engineers, 1911; revised 1912.)

(1) *Constituents.*—The retorts or retort tiles shall be made of sufficiently seasoned raw clay and clean-burnt 'grog.' No 'grog' shall be used which will pass through a test sieve having 16 meshes to the linear inch.

Standard sieves* to be used.

(2) *Refractoriness.*—A piece of the material shall show no signs of fusion when heated to a temperature not less than Seger Cone 28 (about 1630°C.), the heat being increased at a rate of about 50°C. per five minutes in an oxidising atmosphere.

The material is to be chipped to the form and size of a Seger Cone, and tested against standard Seger Cones (small size).

(3) *Surfaces and Texture.*—All surfaces shall be reasonably true and free from flaws or winding, and after burning no 'washing' shall be done without the consent of the engineer (or purchaser). The texture throughout shall be even and regular, containing no holes or flaws, and the 'apparent porosity' shall not be less than 18 per cent. by volume.

(4) *Contraction.*—The material shall be evenly burnt throughout and contain no black core. A test piece, when heated to a temperature of Seger Cone 14 for two hours, shall not show, when cold, more than 1½ per cent. contraction or expansion. The test piece shall be 4½ inches by 4½ inches, the ends being ground flat, and the contraction being measured by means of vernier calipers, reading to 0·1 mm., a suitable mark being made on the test piece, so that the calipers may be placed in the same position before and after firing.

Retorts made of *silica bricks and blocks* have, in recent years, achieved great success, as they transmit the heat better than fireclay. For these the Specification for Silica Bricks (p. 432) is suitable.

CRUCIBLES.

Crucibles are made of (*i*) fireclay to which a suitable proportion of sand has been added, (*ii*) graphite or plumbago with a sufficient amount of clay or other binder, and (*iii*) various other materials of a special character such as magnesia, zirconia, etc. Crucibles must be sufficiently refractory to withstand the heat to which they are subjected, and whilst at the highest temperature at which they are used they must be sufficiently strong to be lifted out of the furnace, with their contents. Equally important is the resistance offered to the action of their contents, and as no crucible is equally suitable for all purposes careful selection is necessary.

A common test for the ability of a crucible to withstand corrosion is to half fill it with copper and to add a little borax. As the copper melts, part of it will be oxidised, the borax will absorb the oxide and will rapidly corrode the crucible, unless the latter is of excellent quality.

Another useful test consists in melting red lead and copper oxide in the crucible to be tested. Most crucibles will be perforated by the fused mixture after a few minutes' heating, but a satisfactory crucible will stand prolonged heating.

Silica is more readily corroded than clay or grog, but graphite is not affected at all. Hence, for highly corrosive contents, grog or graphite crucibles and not siliceous ones should be used.

Resistance to sudden changes in temperature is an essential property of a good crucible. How necessary is this resistance may be realised from the fact that a crucible may be withdrawn from a furnace at 1800°C., with its contents of 80 lbs. of steel at 1500°C. The steel is at once poured out and replaced by air at a temperature of only 20°C. Yet, notwithstanding the severity of such conditions, crucibles are regularly used for this purpose. On the other hand, many clay crucibles which have been used once and allowed to become cold cannot be used again, as they crack when reheated.

Further information will be found in *Refractory Materials*, by A. B. Searle (Griffin & Co., Ltd., London).

* It is necessary to observe that in the standard 16-mesh sieve the wires and spaces are of equal diameter, so that such a sieve only allows pieces less than $\frac{1}{33}$ inch linear dimension to pass through it. Sieves with spaces and wires of any other dimensions will not give results in conformity with the specification.

ABRASIVE WHEELS AND BLOCKS.

The vitrified bonds used in the manufacture of some abrasive wheels, blocks, etc., are largely composed of clay. For further details see Section XXI., Part II.

HEAT-INSULATING MATERIALS.

Attention is being increasingly paid to the saving of fuel by retaining heat in structures from which it ordinarily escapes very rapidly. Thus the loss of steam and of heat in hot gases passing through pipes, cylinders, etc., may be greatly reduced by coating the pipes or cylinders with a highly porous material, such as Kieselguhr, light magnesia, felt, cork, or various fibrous materials. The choice of a material must depend partly on the temperatures to which it will be raised when in use; for instance, felt and fibrous vegetable materials should not be raised to more than 220° F., but Kieselguhr can be used at a bright red heat.

A considerable saving in fuel may be effected in kilns, furnaces, and the like by inserting a 'wall' of insulating bricks between the firebrick lining and the exterior. Such bricks should not be placed where their temperature will exceed 2000° F., or they will not long remain effective. They are most efficient at temperatures below 1800° F. They must be protected from damp, which would fill the pores, and so are not suitable for exterior work which is exposed to the weather.

These porous materials depend for their efficacy on air being a valuable thermal insulator, especially if it is quite stationary.

The difference between the thermal conductivity of diatomaceous earth, cork, and charcoal is small, but the earth can be used at much higher temperatures.

Heat-insulating bricks for kilns, furnaces, etc., are supplied by Newell's Ltd., Newcastle-upon-Tyne, and by J. H. Sankey & Son, Ltd., 7 & 8 Norfolk Street, Strand, London, W.C. 2; in addition to bricks, the latter supply the loose material and also slabs of all sizes.

'FOSALSIL.'

(Moler.)

(*J. H. Sankey & Son, Ltd., 7 & 8, Norfolk Street, Strand, London, W.C.* 2.)

For the conservation of heat, still air—after a vacuum—is the best heat insulator, and there occurs in nature a material composed of minute cells, each separately sealed, that makes the theory of 'still air' practical. This material is known as 'Moler,' 'Fossil Meal,' 'Kieselguhr,' and 'Diatomaceous Earth,' and is utilised in the material (of English manufacture) known as 'Fosalsil' (Moler).

The difference between the thermal conductivity of diatomaceous earth and cork or charcoal is very small, but owing to the fact that diatomaceous earth has very great mechanical strength and a high fire resistance, it can be used where cork would be useless.

'Fosalsil' is supplied by Messrs. Sankey as clay—both raw and calcined—and in bricks and slabs of all sizes.

CEMENTS.

Many cements are clay products of a more or less definite character. Thus, *Portland cement* is made by heating a carefully proportioned mixture of clay and limestone or chalk until they combine and the mass sinters. *Roman cement* and hydraulic lime are crude materials of the same character, in which a natural material is used containing the clay and lime compounds in suitable proportions (see Section IX, p. 401).

FIRE-RESISTING OR REFRACTORY CEMENTS.

These cements—usually sold under a fancy name—are used for patching furnaces, and other appliances subjected to heat. The precise nature of most of these cements is regarded as a trade secret, but many of them are composed of fireclay, ground firebricks, and Portland cement or water-glass. These materials, when properly selected and mixed in the proper proportions, form a paste which adheres readily to cold brickwork, and when the latter is heated the nature of the cement is changed and a strong heat-resisting material is produced.

The commonest cements for use with various bricks consist of a slip made from the same material as that of the bricks. Thus, a mixture of fireclay and brick-dust (grog) is used with fireclay bricks, whilst a mixture of sand and sodium silicate is often used for bonding silica bricks.

Ground ganister or a mixture of silica rock and fireclay is largely used for lining furnaces.

Special cements are used in certain cases; fused alumina and carborundum cements are sometimes used in retorts and muffles, a small proportion of clay being used as a bonding agent. Powdered zirconia, ground so as to pass through a 200-mesh sieve, is very effective at exceptionally high temperatures, whilst other special mixtures may also be employed where a mixture of unusual composition is needed.

SECTION XI

REINFORCED CONCRETE CONSTRUCTION

(pp. 439–478.)

(Contributed by Alexander Drew.)

SECTION XI

REINFORCED CONCRETE CONSTRUCTION.

(Contributed by Alexander Drew.)

REPORT OF JOINT COMMITTEE OF ROYAL INSTITUTE OF BRITISH ARCHITECTS.*

(*Abstract.*)

Good workmanship and materials are essential in reinforced concrete.

It is essential that the workmen employed should be skilled in this class of construction.

Very careful superintendence is required during the execution of the work in regard to—

(*a*) The quality, testing, and mixing of the materials.

(*b*) The sizes and positions of the reinforcements.

(*c*) The construction and removal of centering.

(*d*) The laying of the material in place, and the thorough punning of the concrete to ensure solidity and freedom from voids.

If the metal skeleton is properly coated with cement, and the concrete is solid and free from voids, there is no reason to fear decay of the reinforcement in concrete of stone, gravel, cinder, coke-breeze, etc., made with clean, fresh water.

Fire Resistance.

Experiment and actual experience of fires show that concrete in which limestone is used for the aggregate is disintegrated, crumbles and loses coherence when subjected to very fierce fires, and that concretes of gravel or sandstones also suffer, but in a rather less degree. The metal reinforcement in such cases generally retains the mass in position, but the strength of the part is so much diminished that it must be renewed. Concrete in which coke-breeze, cinders, or slag forms the aggregate is only superficially injured, does not lose its strength, and in general may be repaired. Concrete of broken brick suffers more than cinder concrete and less than gravel or stone concrete.

Rigidly attached web members, loose stirrups, bent-up rods, or similar means of connecting the metal in the lower or tension sides of beams or floor slabs (which sides suffer most injury in case of fire) with the upper or compression sides of beams or slabs not usually injured are very desirable.

In all ordinary cases a cover of ½ inch on slabs and 1 inch on beams is sufficient. It is undesirable to make the covering thicker. All angles should be rounded or splayed to prevent spalling off under heat.

More perfect protection to the structure is required under very high temperature, and in the most severe conditions it is desirable to cover the concrete structure with fire-resisting plastering which may be easily renewed. Columns may be covered with coke-breeze concrete, terra-cotta, or other fire-resisting facing.

Materials.

Cement.—Only Portland cement complying with the requirements of the specification adopted by the British Engineering Standards Committee should be employed; in general the slow-setting quality should be used.

Sand.—The sand should be composed of hard grains of various sizes up to particles which will pass a ¼-inch square mesh, but of which at least 75 per cent. should pass ⅛-inch square mesh.

* By permission.

R.I.B.A. Joint Committee's Report—*continued.*

Fine sand alone is not so suitable, but the finer the sand the greater is the quantity of cement required for equal strength of mortar. It should be clean and free from ligneous, organic, or earthy matter.

Aggregate.—The aggregate, consisting of gravel, hard stone, or other suitable material,* should be clean and angular, varied in size as much as possible between the limits of size allowed for the work. In all cases material which passes a sieve of $\frac{1}{4}$-inch square mesh should be reckoned as sand. The maximum allowable size is usually $\frac{3}{4}$ inch. The maximum limit must always be such that the aggregate can pass between the reinforcing bars and between these and the centering. The sand should be separated from the gravel or broken stone by screening before the materials are measured.

Proportions of the Concrete.—For ordinary work a proportion of 1 part cement to 2 parts sand will be found to give a strong, practically watertight mortar, but where special watertightness or strength is required the proportion of cement must be increased.

The weight of a cubic foot of cement for the purpose of proportioning the amount of cement to be added may be taken at 90 lbs.

Metal.—The metal used should be steel having the following qualities :—

(*a*) An ultimate strength of not less than 60,000 lbs. per square inch.

(*b*) A yield point of not less than 32,000 lbs. per square inch.

(*c*) It must stand bending cold 180° to a diameter of the thickness of pieces tested without fracture on outside of bent portion.

(*d*) In the case of round bars the elongation should not be less than 22 per cent., measured on a gauge-length of eight diameters. In the case of bars over one inch in diameter the elongation may be measured on a gauge-length of four diameters, and should then be not less than 27 per cent.

For other sectional material the tensile and elongation tests should be those prescribed in the British Standard Specification for Structural Steel. If hard or special steel is used, it must be on the architect's or engineer's responsibility and to his specification.

Before use in the work the metal must be clean and free from scale or loose rust. It should not be oiled or painted.

Welding should in general be forbidden ; if it is found necessary, it should be at points where the metal is least stressed, and it should never be allowed without the special sanction of the architect or engineer responsible for the design.

The reinforcement ought to be placed and kept exactly in the positions marked on the drawings, and, apart from any consideration of fire resistance, ought not to be nearer the surface of the concrete at any point than 1 in. in beams and pillars, and $\frac{1}{2}$ in. in floor slabs or other thin structures.

Mixing.—When the materials are mixed by hand they are to be turned over and thoroughly mixed on a clean platform until the colour of the cement is uniformly distributed over the aggregate. Whenever practicable the concrete should be mixed by machinery.

Laying.—The thickness of loose concrete that is to be punned should not exceed 3 ins. before punning, especially in the vicinity of the reinforcing metal. Special care is to be taken to ensure perfect contact between the concrete and the reinforcement, and the punning is to be continued till the concrete is thoroughly consolidated. Each section of concreting should be as far as possible completed in one operation† ; when this is impracticable, and work has to be recommenced on a recently laid surface, it is necessary to wet the surface ; and where it has hardened it must be hacked off, swept clean, and covered with a layer of cement mortar $\frac{1}{2}$ in. thick, composed of equal parts of cement and sand. Work should not be carried on when the temperature is below 34° Fahr. The concrete when laid should be protected from the action of frost, and shielded against too rapid drying from exposure to the sun's rays or winds, and kept well wetted. All shaking and jarring must be avoided after setting has begun. The efficiency of the structure depends chiefly on the care with which the laying is done.

Centering or Casing.—The centering must be of such dimensions, and so constructed, as to remain rigid and unyielding during the laying and punning of the concrete. It must be so arranged as to permit of easing and removal without jarring the concrete. Provision should be made wherever practicable for splaying or rounding the angles of the concrete. Timber when used for centering may be advantageously limewashed before the concrete is deposited.

---

* Coke-breeze, pan-breeze or boiler-ashes ought not to be used for reinforced concrete. It is advisable not to use clinker or slag, unless the material is selected with great care.

† In particular the full thickness of floor slab should be laid in one operation.

R.I.B.A. Joint Committee's Report—*continued.*

Striking of Centres.—The casing for columns, for the sides of beams, and for the soffits of floor slabs not more than 4 ft. span must not be removed under eight days; soffits of beams and of floors of greater span should remain up for at least fourteen days, and for large span arches for at least twenty-eight days.

Testing.—In the case of concrete made in proportions of 1 cement, 2 sand, 4 hard stone, the strength should not be less than 1,800 lbs. per square inch in compression after 28 days.

Loading tests on the structure itself should not be made until at least two months have elapsed since the laying of the concrete. The test load should not exceed one and a half times the accidental load. Consideration must also be given to the action of the adjoining parts of the structure in cases of partial loading. In no case should any test load be allowed which would cause the stress in any part of the reinforcement to exceed two-thirds of that at which the steel reaches its elastic limit.

Methods of Calculation.

Data.

Loads.—The weight of the concrete and steel structure may be taken at 150 lbs. per cubic foot.

In structures subjected to very varying loads and more or less vibration and shock, as, for instance, the floors of public halls, factories, or workshops, the allowance for shock may be taken equal to half the accidental load. In structures subjected to considerable vibration and shock, such as floors carrying machinery, the roofs of vaults under passage ways and courtyards, the allowance for shock may be taken equal to the accidental load.

In the case of columns or piers in buildings, which support three or more floors, the load at different levels may be estimated in this way:—For the part of the roof or top floor supported, the full accidental load assumed for the floor and roof is to be taken. For the next floor below the top floor 10 per cent. less than the accidental load assumed for that floor. For the next floor 20 per cent. less, and so on to the floor at which the reduction amounts to 50 per cent. of the assumed load on the floor. For all lower floors the accidental load on the columns may be taken at 50 per cent. of the loads assumed in calculating those floors.

In the case of many warehouses and buildings containing heavy machinery it is desirable not to make any reduction of the actual loads.

Stresses in Beams.

(*a*) The coefficient of elasticity in compression of stone or gravel concrete, not weaker than 1 : 2 : 4, is treated as constant and taken at one-fifteenth of the coefficient of elasticity of steel.

	Lbs. per Sq. In.
Coefficient for concrete	$E_c = 2{,}000{,}000$
„ „ steel	$E_s = 30{,}000{,}000$

$$\frac{E_s}{E_c} = 15.$$

It follows that at any given distance from the neutral axis, the stress per square inch on steel will be fifteen times as great as on concrete.

(*b*) The resistance of concrete to tension is neglected, and the steel reinforcement is assumed to carry all the tension.

(*c*) The stress on the steel reinforcement is taken as uniform on a cross-section, and that on the concrete as uniformly varying.

Working Stresses.

If the concrete is of such a quality that its crushing strength is 1,800 lbs. per square inch after twenty-eight days, and the steel has a tenacity of not less than 60,000 lbs. per square inch, the following stresses may be allowed:—

	Lbs. per Sq. In.
Concrete in compression in beams subjected to bending	600
Concrete in columns under simple compression	600
Concrete in shear in beams	60
Adhesion * of concrete to metal	100
Steel in tension	16,000
Steel in shear	12,000
Steel in compression	15 times the stress in the surrounding concrete.

* It is desirable that the reinforcing rods should be so designed that the adhesion is sufficient to resist the shear between the metal and concrete. Precautions should in every case be taken by splitting or bending the rod ends, or otherwise to provide additional security against the sliding of the rods in the concrete.

R.I.B.A. Joint Committee's Report—*continued.*

When the proportions of the concrete differ from those stated above, the stress allowed in compression on the concrete may be taken at one-third the crushing stress of the cubes at twenty-eight days as determined above.

If stronger steel is used, the allowable tensile stress may be taken at one-half the stress at the yield point of the steel, but in no case should it exceed 20,000 lbs. per square inch.

Beams of Rectangular and T Section.

In the calculation of all beams the area upon which the ratio of tensile reinforcement is taken is considered as a rectangle of breadth equal to the greatest breadth of the beam and of depth equal to the greatest effective depth of the beam.

In designing beams where the rib is monolithic with a slab, the beam may be considered to be of T form. The slab must first be calculated and designed, having its own reinforcing bars transverse to the rib. The whole of the slab cannot in general be considered to form part of the upper flange of the T beams. The width b_s of the upper flange may be assumed to be not greater than one-third the span of the beams, or more than three-fourths of the distance from centre to centre of the reinforcing ribs or more than fifteen times the thickness of slab. The width b_r of the rib should not be less than one-sixth of the width b_s of the flange.

Shear Reinforcement.

It is always desirable to provide reinforcement to resist the shearing and diagonal tension stresses in reinforced concrete beams. The diagonal tension stresses depend on the vertical and horizontal shear and also on the longitudinal tension at the point considered. As the longitudinal tension in the concrete at any given point is very uncertain, the amount and direction of the diagonal tension cannot be exactly determined.

It is the general practice to determine the necessary reinforcement by taking the vertical and horizontal shearing only into consideration.

Experience shows that:

(*a*) In general, floor slabs require no special reinforcement against shearing and that the bending up of alternate bars near the end is sufficient.

(*b*) In beams, especially in T beams, shearing reinforcement should be provided at distances apart not exceeding the depth of the beam.

(*c*) It is desirable to bend up one or more of the bars of the tension reinforcement near the supports. When bent at an angle of about 45° the effect of this may be taken into account in the manner set out above; when bent at a small angle to the horizontal the effect is very indeterminate.

(*d*) As the resistance of the shear members to the pull depends on the adhesion, and the anchorage at the ends, it is desirable to use bars of small diameter, and to anchor the stirrups at both their ends. In all cases the stirrups must be taken well beyond the centre of compression.

Slabs Supported or Fixed on more than Two Sides.

It does not appear that there is either a satisfactory theory or trustworthy experiment from which the strength of rectangular slabs supported or fixed on all four edges can be determined.

Pillars and Pieces under Direct Thrust.

Definitions.

The length is to be measured between lateral supports (neglecting ordinary bracketing).

The effective diameter of a pillar means the least width and should be measured to the outside of the outermost vertical reinforcement.

The effective area of a pillar means the area contained by the outermost lateral reinforcement, and should be measured to the outside of the outermost vertical reinforcement.

Loading and Length of Pillars.

If the load is strictly axial the stress is uniform on all cross-sections.

Lateral bending of the pillar as a whole is not to be feared, provided—

(*a*) That the ratio of length to least outside diameter does not exceed 18.

(*b*) That the stress on the concrete does not exceed the permissible working stress for the given pillar.

(*c*) That the load be central.

(*d*) That the pillar be laterally supported at the top and base.

Construction.

Lateral reinforcement properly disposed raises the ultimate strength and increases the security against sudden failure, by preventing the lateral expansion of the concrete and the sudden disruption of the pillar.

R.I.B.A. JOINT COMMITTEE'S REPORT—*continued.*

Practical considerations lead to the addition of longitudinal bars, and the formation of an enveloping network of steel.

The total cross-sectional area of the vertical reinforcement should never be less than 0·8 per cent. of the area of the hooped core.

There should be at least six vertical bars when curvilinear laterals are used, and four for square pillars having rectilinear laterals.

In the case of rectangular pillars in which the ratio between the greater and the lesser width (measured to the outside of the vertical bars) exceeds one and a half, the cross-section of the pillars should be subdivided by cross ties; and the number of vertical bars should be such that the distance between the vertical bars along the longer side of the rectangle should not exceed the distance between the bars along the shorter side of the rectangle.

The most efficient disposition of the lateral reinforcement would appear to be in the form of a cylindrical helix, the pitch or distance between the coils being small enough to resist the lateral expansion of the concrete.

Jointed circular hoops as ordinarily made are apparently not quite so efficient.

Rectilinear ties are still less adapted to resist the lateral or radial expansion of a highly stressed core.

The volume of curvilinear laterals should never be less than 0·5 per cent. of the volume of hooped core.

The diameter of rectilinear laterals should not be less than $\frac{3}{16}$ inch.

Pillars Eccentrically Loaded.

If a pillar, initially straight, is loaded eccentrically, as when a beam rests on a bracket attached to the pillar, it may be regarded as fixed at the base and free at the loaded end.

It is desirable in pillars that there should be no tension, and generally when the vertical load is considerable there is none. Cases in which the eccentricity is so great that there is tension must be treated by the methods applicable to beams if it is made a condition that the steel carries all the tension.

Long Pillars Axially Loaded.

For columns more than 18 diameters in length there is risk of lateral buckling of the column as a whole. The strength of such pillars would be best calculated by the Gordon-Rankine formulæ.

LONDON COUNTY COUNCIL (GENERAL POWERS) ACT, 1909.

REINFORCED CONCRETE REGULATIONS—(MADE JULY 1915).*

The following brief summary is intended merely to indicate the scope, and give the principal approved data.

Nothing in these Regulations interferes with the application of the formulæ, etc., for the designing of reinforced concrete structures given in the following pages.

Part I. defines reinforced concrete and indicates the application of the same to buildings whose skeleton framework, together with any party walls, is capable of safely and independently sustaining the whole of the dead and superimposed loads.

Floors, main stairs, and landings must be of incombustible material; internal stairs and landings may be of fire-resisting materials; and roofs may be of wood framing, boards, and battens.

No part of the reinforcing metal is to be used for conducting electrical currents.

Notices to be served on, and plans and calculations to be provided to, the District Surveyor are detailed.

---

* Where fuller information is desired, the complete Regulations should be referred to. A reprint of the Regulations, with useful notes, illustrations, and worked examples, has been published by B. T. Batsford, Ltd., 94 High Holborn, London.

Part II.—Superimposed Loads.

On Floors: Equivalent dead loads.

For a floor intended to be used for the purpose of	Equiv. dead load in lbs. per sq. ft.	For a floor intended to be used for the purpose of	Equiv. dead load in lbs. per sq. ft.
Domestic buildings not hereinafter specified	70	Music halls	112
Other similar purposes		Public assembly	
Asylum wards	84	Public concert rooms	
Common lodging-house bedrooms		Public library reading rooms	
Hospital wards		Retail shops	
Hotel bedrooms		Theatres	
Workhouse wards		Workshops	
Other similar purposes		Other similar purposes	
Counting houses	100	Ball rooms	150
Offices		Drill rooms	
Other similar purposes		Other similar purposes	
Art galleries	112	Similar floors subject to vibration	
Chapels		Book stores at libraries	224
Churches		Museums	
Classrooms in school buildings, not being dwelling houses so used		For every floor in a building of the warehouse class, except such floors as are intended to be used for any of the purposes aforesaid—not less than	
Lecture rooms			
Meeting rooms			

On Stairs and Landings: Minimum to be equal to a distributed dead load of 120 lbs. per square foot, but each step must be capable of safely supporting a concentrated load of 300 lbs. at any point.

On Roofs of greater upward inclination than 20° with the horizontal, a superimposed load (including wind pressure and weight of ice) of '28 lbs. per square foot of sloping surface, normal to such sloping surface, on either side of such roof.' On all other roofs, '56 lbs. per square foot measured on a horizontal plane.'

On floors, stairs, landings, or roofs, other than those indicated, the actual superimposed load must be provided for, 'with an equivalent margin of safety.'

For Rolling Loads actuated by mechanical power, take an equivalent static load equal to 1½ times the actual rolling load and provide for the stresses due to every position of such load.

Partitions and other Structures on floors or roofs whose weight exceeds the superimposed load must be specially provided for.

For calculating the Resistance Moment, the angle of dispersion of a point load through hard filling and concrete may be taken at not more than 45° from the vertical.

For Buildings of more than two storeys in height the loads on foundations, pillars, and walls may be reduced, as in Steel Regulations (para. 19, page 374).

Wind Pressure to be equal to a static horizontal pressure of not less than 20 lbs. per square foot of the whole projected surface normal to the wind direction. On panels of external walls allow for not less than 30 lbs. per square foot 'assumed as acting uniformly over the area of one panel from either side.'

For all structures or attachments to buildings (including towers or other parts extending above the roof, etc.), allow for not less than 40 lbs. per square foot of whole projected surface normal to direction of every wind.

Weight of Reinforced Concrete to be taken at not less than 144 lbs. per cubic foot measured over finished surfaces.

The span of a beam shall not be greater than twenty times its effective depth, unless the maximum tensile or compression stresses are correspondingly reduced below the maximum stresses recognised in these regulations.

The length of a Cantilever shall not exceed five times its effective depth at the bearing.

Bending Moments.—For the purpose of calculation the effective span of a beam or slab shall be taken as the clear span plus the effective depth, or the span to centres of bearings, whichever is the lesser.

[The values for the bending moments due to various loadings may be obtained graphically by means of the diagrams on pp. 212, 213, except for type loading (h), where these

regulations call for equal maximum moments at centre and at ends (to obtain the former by diagram, make value of $A=\frac{1}{6}$ Wl).]

For Continuous Beams or Slabs the maximum positive and negative bending moments in each span may be calculated for all necessary arrangements of loading to same and to adjoining spans. Or the approximate maximum bending moments may be taken as follows:—Near middle of end span, at support next to end support, and at other interior supports, $=\frac{1}{10}$ total load × effective span; at middle of interior spans, $=\frac{1}{13}$ total load × effective span.

For Slabs reinforced with mesh reinforcement, or reinforced in two directions at right angles to each other, the bending moments in each direction may be obtained from the table on page 463 (Reinforced Concrete Construction Section), the accepted values being those marked 'Grashof and Rankine.'

For all loadings other than those indicated, the actual bending moments shall be calculated 'so as to give an equivalent margin of safety.'

The permissible working stresses in concrete, except as further provided for in pillars, shall not exceed the following:—

Stresses in Concrete.	Proportion by Volume.											
	Cement.	Sand.	Coarse Material.	Cement.	Sand.	Coarse Material.	Cement.	Sand.	Coarse Material.	Cement.	Sand.	Coarse Material.
	1	2	4	1·2	2	4	1·5	2	4	2	2	4
	V=6			V=5			V=4			V=3		
	Lbs. per Sq. In.			Lbs. per Sq. In.			Lbs. per Sq. In.			Lbs. per Sq. In.		
Direct compressive stress.	600			650			700			750		
Extreme flexural compressive stress in beams	600			650			700			750		
Shearing stress	60			65			70			75		
Grip or adhesion between concrete and steel bars hooked at both ends	100			100			100			100		
Grip or adhesion between concrete and steel bars otherwise effectively anchored at the ends	60			60			60			60		
Tensile stress	Nil			Nil			Nil			Nil		

If V is less than 3 the stresses in the last column shall not be exceeded.

The permissible working stress in steel shall not exceed the following:—

Stress in mild steel complying with the British Standards Specification	Pounds per square inch.
Compressive stress	m times the stress in the concrete immediately surrounding the steel.
Tensile stress	16,000.

Eccentric loading stresses must be provided for.

For Hooks or Anchors to ends of tensile and shear reinforcement, and for Grip Lengths, refer to Regulations.

The elastic modulus of steel in tension or compression shall be taken at 30,000,000 lbs. per square inch.

The modular ratio for steel and concrete shall be taken as follows:—

For beams . . . . . $mc=9{,}000$ or $m=\frac{9{,}000}{c}$

For pillars with the minimum of hooping or binding . $mc=9{,}000$ or $m=\frac{9{,}000}{c}$

For pillars with more than the minimum of hooping or binding . . . . . $mi = 9{,}000$ or $m = \frac{9{,}000}{i}$,

where

c = permissible *compressive* working stress given in preceding table;

i = *increased* stress allowed in pillars with more than the minimum of hooping or binding (see PART IV.).

Part III.

The term 'beam' shall include any beam, girder, lintel, bressummer, or cantilever, or any other similar member carrying transverse loads.

Longitudinal bars must be at least ¼ inch in diameter, all other reinforcing bars being not less than ⅛ inch in diameter.

Distance apart of bars at least 1 inch horizontally and ½ inch vertically, except at joints or crossings of bars.

Tensile reinforcement to be not more than 6 inches apart.

Compression reinforcement must not be stressed to more than m times the stress in the immediately surrounding concrete, and be anchored by bars extending through a depth at least equal to the resistance moment arm, and spaced not further apart than this resistance moment arm or than sixteen times the least diameter of the anchored bar.

Where all compression stress is taken by steel rods, these may be stressed to 16,000 lbs. per square inch, and be anchored laterally and vertically not further apart than 6 inches or than eight times the diameter of the anchored bars; the anchors to be passed round or hooked over both the compressive and the tensile reinforcement.

If hooping is adopted, the compression stress for ¼ span from each end may be taken as that for the core of a similarly hooped pillar; the concrete outside the hooping must not be taken into account.

Shear and Web Reinforcement.—Regulations are provided as to the value of concrete alone, and with the addition of rods; and as to spacing, and anchoring of shear rods to the main reinforcing bars, etc.

Splays at ends of beams intended to increase the resistance moment must not be calculated at a greater angle than 30° from the horizontal.

Brackets acting as cantilevers must be suitably reinforced.

Slabs.—The effective depth must not be less than 3 inches, and ¼ inch to be the least diameter or thickness of reinforcing bars. Any tension cross wires, in mesh or similar reinforcement, to be at least $\frac{1}{10}$ inch in diameter or thickness. Tension bars to be at least 1 inch apart, except at joints or crossings. Mesh reinforcement to be of such dimensions as to allow the concrete to freely pass through the same.

Tension bars or strands to be not further apart than 12 inches, or more than twice the effective depth. When reinforced in one direction only, distributing bars must be provided on top and at right angles to the main bars; these distributing bars to be not more than 18 inches apart, and have an aggregate cross-section of at least 0·08 per cent. of effective cross-section of the slab; or the diameter of each distributing bar must be at least $\frac{1}{11}$th of the effective depth of the slab, and spaced not more than four times the effective depth apart. Wiring or binding shall not be regarded as reinforcement.

Resistance Moments.—Tee beams.—The breadth of the flange shall not be more than ¼ of the effective span, or the distance between the centres of the ribs of the tee beams, or twelve times the thickness of the slab, whichever is the least.

Ell beams.—The breadth of flange shall not be more than four times the thickness of the slab, except where the moments due to the eccentricity of the compression are provided for, when six times the slab thickness is permitted.

The breadth of the rib of a tee or ell beam shall be not less than ⅓ depth of rib below the slab.

Buckling in rectangular beams to be provided for when length exceeds twenty times the least width, or thirty times the least width over the outermost longitudinal reinforcement.

A beam supported at its end by a transverse beam must have special provision made as to connecting together, and resisting shear and torsion.

Notes must be added to drawings or diagrams of beams or pillars showing the loads which have been provided for.

Complete formulæ are given for rectangular and tee beams, but these are similar to the data given on pp. 459, 460; 461 (Reinforced Concrete Construction), to which reference may be made.

Part IV.

Pillars shall include piers, posts, columns, detached supports, or any other vertical compression member; struts any compression member at any angle

In calculations for strength, the maximum ratio of length to gyration radius, or of length to effective diameter, shall be taken, the length being measured between the lateral supports irrespective of any splays other than allowed for beams, and the effective diameter to be that over the outermost vertical rods.

Vertical and lateral reinforcement must be provided to all pillars, four full length verticals for rectangular laterals (binding) not less than $\frac{3}{16}$ inch in diameter, and six verticals to curvilinear laterals not less than $\frac{1}{4}$ inch diameter.

Pitch of laterals (binding) shall not exceed $\frac{5}{6}$ effective diameter at any part in length, or sixteen times the diameter of the least vertical. For a length of one and a half times effective diameter at each end, laterals to be spaced not more than $\frac{3}{4}$ effective diameter. All laterals to be firmly secured at each end. The volume of laterals to be not less than 0·5 per cent. of the volume of the hooped core.

Verticals to be not less than $\frac{1}{2}$ inch or more than 2 inches in diameter, and have a minimum total cross-section of 1·0 per cent. of the volume of the hooped core. Joints in verticals to be made at or adjacent to points of lateral support, and lapped at least twenty-four times the diameter of the upper rod.

In Rectangular Pillars, where the ratio between the greater and the lesser diameter exceeds $1\frac{1}{2}$, the cross-section shall be subdivided by cross ties, and the distance apart of verticals in the longer side be not greater than that in the shorter side.

(*As the rules and tables given for calculating the strength of pillars are largely empirical, they are included here in full.*)

Notation for pillars, struts and other compression members.

113.—

A = effective *area* of the pillar, *i.e.* the *area* bounded by the lateral reinforcement (binding) measured to the inside of the hooping.

A_b = cross-sectional *area* of one bar of the *binding* or lateral reinforcement.

A_v = *area* of the *vertical* reinforcement.

c = permissible direct *compressive* stress.

d = effective *diameter*.

f = *form* factor, depending upon form or type of binding or laterals (para. 118).

g = *gyration* radius of the area $[A + (m-1)\,A_v]$ about an axis passing through the centroid of the area.

i = *increased* stress permissible in the core of a pillar suitably hooped

l = actual *length* of the pillar.

m = *modular* ratio $= \frac{E_s}{E_c}$.

P = *permissible* load or *pressure* on pillars with both ends fixed and with a ratio of virtual length to gyration radius not exceeding 45 (para. 122).

p = *percentage* of volume of binding or lateral reinforcement with respect to the volume of hooped core in any given length of pillar $= 100\ V_v$.

p_b = *pitch* of the *binding* or lateral reinforcement.

s = *spacing* factor depending upon the spacing or pitch of the binding or laterals (para. 118).

V_v = *volume* ratio or *ratio* of the *volume* of binding or lateral reinforcement to the volume of the hooped core in any given length of pillar (paras. 116, 117).

ι = *virtual* length of the pillar, strut or other compression member for different conditions of the ends (paras. 122, 124).

114.—A pillar, or other strut, shall be deemed to have fixed ends when the ends are sufficiently secured to other parts of the construction having such rigidity as will maintain the axis at the ends in its original position and direction under all loads less than the crippling load.

115.—In a pillar with fixed ends the stress in the concrete in the area bounded by the lateral reinforcement shall not exceed

c, in a pillar with the minimum of hooping or binding,

i, in a pillar with more than the minimum of hooping or binding, where

$$i = c\,[1 + f\,s\,V_v]$$

116.—The required ratio of the volume of binding or lateral reinforcement to the volume of the hooped core may be obtained from the equation—

$$V_v = \frac{i-c}{c\,f\,s}.$$

117.—The value of V_v actually provided, in either round or square pillars, may be ascertained by direct measurement or from the equation—

$$V_v = \frac{4\,A}{d\,p_b}.$$

118.—

Table showing the value of f_s.

Form of Lateral Reinforcement or Binding.	Form Factor *f*.	Spacing of Laterals throughout the length of the Pillar (in terms of diam. of hooped core).	Spacing Factor *s*.	Value of f_s.
Helical (curvilinear on plan) . .	1·0	0·2*d*. or less	32	32
Ditto ditto . .	1·0	0·3*d*.	24	24
Ditto ditto . .	1·0	0·4*d*.	16	16
Circular hoops	0·75	0·2*d*. or less	32	24
Ditto	0·75	0·3*d*.	24	18
Ditto	0·75	0·4*d*.	16	12
Rectilinear	0·5	0·2*d*. or less	32	16
Ditto	0·5	0·3*d*.	24	12
Ditto	0·5	0·4*d*.	16	8
Ditto	0·5	0·5*d*.	8	4
Ditto	0·5	0·6*d*.	0	0

119.—Notwithstanding any other provision in these regulations the increased stress in the concrete of pillars shall not exceed one-third of the ultimate compressive resistance at four months as given in subsequent table.

120.—The working stress in the vertical reinforcement shall not exceed m times the stress in the concrete.

121.—The permissible load or pressure on pillars with both ends fixed and with ratios not exceeding those in the first column of figures in table below shall be obtained from either of the equations—

For pillars with the minimum of hooping or binding

$$P = c\,[A + (m-1)\,A_v]$$

For pillars with more than the minimum of hooping or binding—

$$P = i\,[A + (m-1)\,A_v]$$

122.—For pillars the permissible load or pressure shall be obtained from the following table :

For the maximum ratio of *virtual* length to *gyration* radius	$\frac{v}{g}$ =	45	54	63	72	81	90
For the maximum ratio of *virtual* length to effective *diameter* for square pillars	$\frac{v}{d}$ =	15	18	21	24	27	30
For the maximum ratio of *virtual* length to effective *diameter* for round pillars	$\frac{v}{d}$ =	12	15	18	21	24	27
Permissible load		P	0·8 P	0·6 P	0·4 P	0·2 P	Nil

For other ratios higher than those in the first column of figures the stress shall be proportional to the above.

123.—Struts shall be subject to all the rules applying to pillars, and the combined stresses at any part shall not exceed the permissible stresses for pillars of like ratios.

124.—For the different conditions of the ends of pillars and other struts mentioned in the

following table, the permissible load shall be ascertained by using the corresponding value of the virtual length in such table in conjunction with paras. 121 and 122:

Condition of Axis at Ends.	Virtual Length, v.
Both ends fixed in position and direction	$v=l$
One end fixed in position and direction and . . . . . / One end fixed in position but not in direction	$v=1{\cdot}4l$
Both ends fixed in position but not in direction	$v=2l$
One end fixed in position and direction and . . . . . / One end not fixed in position and direction	$v=4l$

125.—In the case of arches or other similar constructions the combined stresses at any part shall not exceed the permissible stresses set out in Table of Working Stresses for Beams.

Part V.

External enclosure walls of reinforced concrete formed between reinforced concrete framework must be designed for any loads they may have to carry, but must not be less than 4 inches thick. Where they support vertical loads or resist lateral pressures they must be of the necessary thickness to keep the stresses within the prescribed limits. For enclosure walls of brickwork, stonework, or plain concrete, the topmost 20 feet of their height must be not less than 8½ inches thick, and 13 inches thick for all lower portions. If the walls are formed of hollow blocks, the following minimum dimensions are required. Aggregate thickness, including any cavity, 8½ inches; sides, 2 inches; ends and any interior partitions, 1 inch; clear unsupported span of any part of side, end, or interior partition, four times the least thickness of such part; and aggregate thickness of material, measured at right angles to face of wall, 6 inches.

Any less thickness of walls prescribed under the London Building Act, 1894, may be adopted; but any external wall or portion of same, not supported or secured by the reinforced concrete framework, must comply with the requirements of the above Act.

All walls and facing materials shall be incombustible, and securely connected to the reinforced concrete framework.

Each panel in any external wall must be designed to safely resist a horizontal pressure equal to a static pressure of at least 30 lbs. per square foot, assumed to act uniformly over the area of one panel from either side.

The aggregate area of openings in external walls shall not exceed in a wall of any storey above the ground storey two-thirds of the whole area of such wall, nor the aggregate width three-fourths of the whole length of such wall.

Party and division walls in reinforced concrete shall be not less than 8 inches thick, but party walls between a reinforced concrete building and a steel framed building, or a building constructed in accordance with rules of the London Building Act, 1894, shall in no part be less than 13 inches thick.

All brickwork, stonework, and plain concrete shall be executed in Portland cement mortar.

The pressure on any brickwork supporting reinforced concrete work shall not exceed the values given under Steel Regulations, page 376.

The height of such brickwork with proper lateral supports must not exceed twelve times its least thickness; if without proper lateral supports, six times its least thickness. This thickness must in no case be less than 13½ inches.

Part VI.

The pressure of foundations on the natural soil, or on plain concrete, shall not exceed the values given under Steel Regulations, page 375.

Part VII.

Protection (measured from outer surface of concrete to outer surface of reinforcement) shall not be less than the following values:—Beyond the anchored ends of hooked or upward bent bars, 2 inches, or twice the normal diameter of the bar; vertical bars in pillars, 1½ inches, or the diameter of the vertical bar; longitudinal bars in beams, 1 inch, or the diameter of the longitudinal bar; tensile, compressive, shear, or any other slab reinforcement, ¼ inch, or the diameter of bar to be covered; and any reinforcement in any other member, 1 inch, or the diameter of the bar to be covered.

Part VIII.

Materials.—The Cement shall be Portland cement of slow setting quality and to British Standard Specification, and one cubic foot of same shall be assumed as weighing 90 lbs. The Sand shall be clean and gritty, of hard silicious grains, and free from clay, animal, vegetable, or bituminous matter. It must be capable of passing through a mesh $\frac{3}{16}$ inch square, measured in the clear, and must be separated from the coarse materials before measuring. The coarse material (which includes all the ingredients in the concrete except the sand, cement, and water) shall consist of clean Thames or pit ballast or gravel,* hard stone, such as granite, basalt, trap rock, or other hard and equally suitable material, and free from clay or any animal, vegetable or bituminous matter. If necessary this material shall be thoroughly washed.

The following materials shall not be used with the sand or coarse material in the composition of concrete:

(*a*) Coal residues, including clinkers, cinders, ashes, coke breeze, pan breeze, slag, and other similar material.

(*b*) Blast furnace slag, copper slag, forge breeze, dross, and other similar material.

(*c*) Sulphates, including plaster of Paris, and other similar material.

(*d*) Limestones, magnesian limestones, marbles, and other calcium carbonates.

The coarse material must pass through a mesh $\frac{3}{4}$ inch square, measured in the clear, and be retained on a mesh $\frac{3}{16}$ inch square, measured in the clear; it shall be varied as much as possible between these limits. If the material is of a porous nature, it must be thoroughly wetted before mixing with the other materials.

The volume of mortar shall be in excess of that required to fill the voids in the coarse material. The volume of sand shall not be more than twice the volume of cement, nor the coarse material twice that of the sand.

The concrete shall be composed of cement, sand and coarse material in one of the following or intermediate proportions, and the ultimate compressive resistance shall not be less than that specified for the proportion adopted:

Proportion by volume.			V	*Ultimate* compressive resistance in pounds per square inch.		Value of one-fourth of u_4
Cement.	Sand.	Coarse material.		1 month after mixing, $=u_1$	or, 4 months after mixing, $=u_4$	
1	2	4	6	1,600	2,400	600
1·2	2	4	5	1,800	2,600	650
1·5	2	4	4	2,000	2,800	700
2	2	4	3	2,200	3,000	750

Test cubes shall be not less than 4 inches each way in dimensions, or test cylinders not less than 6 inches each way; the preparing, setting, maturing, and testing of such pieces should, as far as possible, conform to the actual conditions under which the work would be carried out.

All three materials shall be thoroughly mixed dry in batches, and again mixed after wetting. Salt water shall not be used for the concrete.

The concrete shall be placed in its final position before initial set has taken place. For beams, pillars and walls the thickness of the layers of loose concrete shall not exceed three inches before ramming.

As soon as possible after mixing, the concrete shall be properly rammed into the moulds, so as to secure a compact mass, without voids and of the greatest possible density for the proportions used.

All metal reinforcement shall be of steel to the British Standard Specification for structural steel for bridges and general building construction (see page 185).

All reinforcement metal shall be cleaned of all scale, dust and loose rust, immediately before depositing the concrete.

No welding shall be employed in any tensile reinforcement.

The builder or other person directing the work shall furnish the District Surveyor with reasonable proof as to the quality of materials to be used in such construction, and shall make any tests which shall be reasonably necessary.

If testing during the construction or within two months after the completion of the reinforced concrete construction is found necessary by reason of any sign of weakness or faulty work

* For further information on gravel, see p. 389.

appearing, the builder or other person directing the work shall make such tests, and, if these show faulty work, it shall be reconstructed and reinstated in accordance with these regulations.

The total deflection of beams or slabs freely supported and carrying the uniformly distributed working load shall not exceed 1/600th of the span, when the span is twenty times the effective depth, and shall be in proportion for other ratios of span to depth, and for other conditions of ends and stress and loading.

The superimposed test load shall be one and a half times the superimposed load.

Loading tests shall not be made earlier than 90 days after laying the concrete.

Part IX.

The formwork or centering shall be of such dimensions and so constructed as to remain rigid during the laying, ramming, and setting of the concrete.

The vertical strutting shall be maintained continuous through the lower storeys to the foundations or to other floors or beams which are sufficiently set to afford the required safe support, and all formwork or centering shall be removed without shock or vibration.

Before the formwork or centering under any beam or floor slab is removed, the pillars below such beam or floor slab shall be partially stripped to allow of complete examination on all sides.

Part X.

All reinforcement shall be placed and maintained in the position shown on the drawings, and the concreting in any member shall be carried out as continuously as possible.

Where work has to be recommenced on a surface which has hardened, such surface shall be well hacked, swept clean, thoroughly wetted, and covered with mortar composed of equal volumes of cement and sand.

Concrete laid during dry weather shall be protected against too rapid drying. During the first week of hardening it shall be kept damp by wet sacking or other methods, or by watering daily (Sundays and holidays included).

Concrete shall not be laid when the temperature is below 4° C. (39° F.), and shall be protected when necessary.

Concrete or mortar which has been frozen shall not be used.

The concrete or mortar in any member which has been exposed to frost during the first week of hardening shall be removed, or such members shall be tested as previously provided.

No cutting for piping or any other purpose shall be done which would reduce the strength of any part of the structure below the standard required by these regulations.

Blocks or strips of hard wood, coke breeze, or other equally fire-resisting material may be embedded in the concrete and used solely for fixing purposes, provided that they do not reduce the strength of any part of the structure below the standard required by these regulations, and the area of such blocks or strips at any given cross-section is not included in the calculated compression area of any beam, slab, pillar, or other constructional member.

Soft wood or other equally combustible material shall not be embedded in the concrete.

Wood or other combustible material may be placed on or over the surface of the concrete, provided that any voids or hollow spaces between the combustible and incombustible materials be filled up with incombustible materials.

SYSTEMS OF REINFORCED CONCRETE CONSTRUCTION.

Although there exists a considerable number of so-called independent systems of reinforced concrete construction, the principles underlying the same are in every way similar, and it is only in the manner in which these principles are carried out that any difference arises. The several systems can be readily divided into groups, and the following notes will be sufficient here to indicate wherein one differs from another :—

Systems which make use of Plain Steel Rod Reinforcing Bars.

In the Hennebique system, the main bars are set parallel to the lower or tension face of the beam, some being, where necessary, bent or cranked upwards towards the upper face (as where the beam is continuous over a support). In addition to these bars, flat stirrups are hooked at intervals over the lower bars, and carried upwards towards the upper part of the beam ; these bind the concrete and steels securely together, and assist largely in resisting the shearing stresses. For columns or piles, the rods are grouped symmetrically about the centre of the pile, and are tied together at intervals by means of straps or wire lacing.

The Coignet system is in many respects very similar to the Hennebique, except that generally a number of smaller rods are used, these being in turn bent upwards as they approach the ends of the beam. A continuous top bar is also provided, to which the bent-up members are securely tied or laced. Any additional stirrups or binding ties which are necessary are formed of small

section round bars, varying from $\frac{3}{16}$ in. to $\frac{1}{2}$ in. diameter; these are laced or tied between the lower or main tension bars and the upper bar.

In the Considère system, the portions of any structure under compression are provided with a close-set spiral of steel wire or small-section steel rod, which, by surrounding the concrete and preventing its displacement, enables the enclosed portion to resist safely very high compressive stresses.

A number of other systems in which plain bars are used have recently appeared, the difference between them being in the manner in which the several bars are connected together, and the method of providing for shearing stresses, and for connecting the shear bars to the main bars. Of these might be mentioned the types supplied by the British Reinforced Concrete Engineering Company, the Lewis Construction Company, etc.

Systems in which Bars of Special Sections are utilised.

The Trussed Concrete Steel Company own the 'Kahn' bar and the 'Rib' bar. The former consists of a square or other section bar provided with two projecting wings which can be sheared and bent up to form stirrups or shear members. As these members form part of the main bar there is no possibility of displacement or slipping occurring between the shear and the main bars. From this method of construction it will be obvious, however, that the length and spacing of these shear bars are limited, but in practice it is very seldom that this is in any way objectionable. The rib bar is of round section provided with four ribs or projections running the full length of the bar; cross ribs are set at frequent intervals between the longitudinal ribs, so as to form pockets or recesses into which the concrete can key.

The Indented Bar and Concrete Engineering Company, Ltd., possess a square and a round bar provided throughout their full length with a series of depressions and ridges all round; it is claimed that by arranging the depressions and ridges on opposite sides of the bar the full section of the bar is retained throughout its length. By this formation a mechanical bond or key with the concrete is provided which it is claimed greatly increases the key between the bar and the concrete.

In the Perfector System a round bar with a continuous rib or fin is used, this fin being provided with perforations through which the shear members, etc., are passed.

Other bars providing a mechanical bond are sometimes used, notably some American patents such as the 'Ransome,' which consists of a twisted square rod, etc.

Systems in which the Reinforcement consists of a Metallic Meshwork.

The Expanded Metal Company supply sheets formed from steel sheets of various thicknesses which, by special machinery, have been slit or slotted and expanded to form a metallic network. By embedding this network in the correct position in a concrete beam or floor slab, the necessary metal to resist the tension stresses is provided in a satisfactory and simple manner. For detailed information regarding this material and system, see p. 334.

The Lock Woven Mesh System consists of a mechanically woven fabric made of steel wires crossing at right angles and secured at the intersections.

Reinforced Concrete or Fire-Resisting Floors.

Numerous systems of these are on the market, but they can be grouped together into types, of which the following may be noted :—

(1) A series of hollow or solid blocks of fireclay, concrete, or other material are laid at short distances apart, and the space between same filled up with concrete or cement grouting, with steel rods embedded in same. In this way a series of reinforced concrete beams are formed surrounding the blocks. Where the area to be covered is approximately square, or the proportion of length to breadth does not much exceed 2 to 3, these reinforced concrete beams can be formed on all four sides of the blocks; but where hollow blocks are used, the open ends must be closed up, or provision made to prevent the concrete finding access into the hollows which are intended to lighten the weight of the floor. Of this type may be mentioned the 'Kleine,' the 'Fram,' Diespeker's 'Bigspan,' etc.

(2) Hollow blocks, usually of arched formation, and of a width or span of from 2 to 3 feet, or over, are laid between and supported by lines of rolled steel beams, or of reinforced concrete beams, these latter being made either at works, and delivered and erected complete, or formed *in situ* during the construction of the floor. A level floor surface is obtained by a light filling of breeze concrete over the blocks. Where steel beams or finished reinforced concrete beams are used, the floor can be erected without the use of centering. The dispensing with large masses of wet concrete minimises all trouble due to dripping from the new floor, and allows it to be used immediately after it has been erected. Floors of this nature are manufactured by Messrs. D. G. Somerville & Co., the Armoured Tubular Floor Company, etc.

A modification of this floor consists in the use of half-arch shaped blocks, which are set into the webs of steel beams, and brought together at the centre where they are lightly grouted at their meeting edges. A levelling-up covering of breeze concrete completes the floor. This type is represented by the 'Fram' arched floor, the 'Carlisle' tubular arch floor, etc.

(3) Concrete tubes of various sections and of considerable lengths may be formed, and reinforced where necessary with steel rods or wire. By placing these close together, it is only necessary to grout in between the joints in the several tubes to complete the floor ready for any top finish; the advantage of a dry laid floor is thus secured. The 'Siegwart,' etc., provide this type of floor.

(4) Floors may be formed of solid concrete slabbing reinforced with plain or special section bars as required, and supported by rolled-steel beams or reinforced concrete beams. Such types are supplied by any reinforced concrete contractor.

(5) Floors provided with a continuous mesh reinforcement :—The floors provided by the Expanded Metal Company and by the Lock Woven Mesh Company may be taken as examples of this class.

Other types of floors are to be met with, each possessing some particular qualification or arrangement peculiar to itself.

DESIGNING REINFORCED CONCRETE STRUCTURES.

Concrete is very strong under compression, but weak under tension. Good average concrete will safely resist a compressive stress of from 600 to 700 lbs. per square inch of section; its tensile strength does not exceed about one-tenth of this value, and may altogether disappear through the formation of minute cracks.

Concrete and steel possess practically the same coefficient of expansion, and therefore expand and contract together under changes of temperature, without developing internal stresses.

Concrete adheres strongly to a clean surface of steel; the average safe adhesion can generally be taken as about 100 lbs. per super. inch. It is thus possible to form a practical and economical combination of concrete and steel, the former material being utilised to resist the compressive stresses, with the necessary steel bars embedded at the points where tensile stresses occur. In certain cases, where the concrete alone cannot provide all the necessary compressive resistance, steel rods are added to assist the concrete; but from an economical point of view this should be avoided whenever possible, for the following reason :—

Steel possesses, on an average, about fifteen times the elasticity of concrete; where steel bars are embedded in concrete to assist in resisting compression, it follows that when the concrete around the bars has been stressed up to its limit, the steel has been compressed to fifteen times that of the concrete only, and is still far from its limit. Notwithstanding this want of economy, it is frequently necessary to utilise steel in this way, as, for instance, in the case of columns, arches, etc., and in beams when sufficient depth, or area of concrete in the compression flange, cannot be obtained.

Steel is very perfectly protected from deterioration by a covering of concrete, and many instances could be quoted to show that a covering of less than 1 inch of good concrete is sufficient to preserve steel almost indefinitely.

In many cases reinforced concrete forms an almost ideal method of construction, particularly where great strength and durability are essential. The strength of the concrete increases with age, and there is no cost of upkeep, as in the case of steelwork, which requires regular inspection, painting, etc.

All structures can have their several members resolved into three types or into combinations of these three types—namely, beams or girders, struts or stanchions, and ties. Beams or girders support loads which act at right angles to their length, and tend to cause bending or deflection of the member. In struts or stanchions the loads act in the direction of the length, and tend to compress the members; in long and slim compression members, however, stresses similar to those in beams may also arise, through bending or side-yielding when the load exceeds a particular value. In ties the load acts in the direction of their length, as in struts, but here the tendency is to produce tension or stretching in the member.

Composite members may be represented by struts or ties which are not placed vertically, and consequently tend to bend by their own weight, and to this latter extent act as beams. Arches also are generally subjected to bending stresses in addition to that of compression.

Beams or Girders.

The bending moment and shearing force values are calculated exactly as in the case of ordinary beams, and reference may be made to pages 212 and 213 for information on this point. Owing to the composite nature of a reinforced concrete beam, however, the determination of the

internal stresses, or of the scantlings of any beam necessary to limit the tension and compression stresses to some specific maximum, are distinctly more complex than in the case of beams formed of a homogeneous material.

The following assumptions, which are practically correct within the usual limits of safe stress and ordinary conditions, are generally admitted, and form the basis on which the formulæ are built up :—

(*a*) The coefficient of elasticity in compression of good concrete is constant throughout the working limit of stress, and for an average 1 : 2 : 4 mixture may be taken as $\frac{1}{15}$ that of the steel reinforcement. For any distance from the neutral axis, therefore, the stress per square inch on the steel is fifteen times that on the concrete.

(*b*) The tensional resistance of concrete is ignored.

(*c*) The stress on the concrete varies uniformly from zero at the neutral axis to a maximum at outer edge ; the stress on the area of the steel is constant.

(*d*) The bond, or adhesion, between the concrete and the steel is sufficient to transmit any stress arising between these two materials.

Considering, in the first instance, a beam formed of some homogeneous material, supported at each end, and loaded so that it tends to deflect, the compressive stress at the upper edge of the beam will be exactly equal to the tensile stress at the lower edge, and these values will gradually diminish towards the centre of the depth, where they will altogether disappear ; this plane of no stress is called the neutral axis.

Assuming that the stresses vary uniformly, fig. 1 may be taken to represent the stresses in any section of the beam, the horizontal distance between the vertical line and the inclined line at any point above the neutral axis will be proportionate to the compressive stresses at the given point; similarly, the horizontal lengths below the neutral axis will be proportionate to the tensile stress.

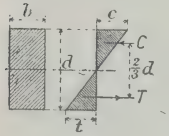


FIG. 1.

The total resistance to compression or extension would be represented by the area of either triangle. The resultant of either of these stresses would act through the centre of gravity of these triangles, or at a distance equal to $\frac{1}{3}$ the perpendicular depth from the base, giving the distance between the two resultants, or the lever arm of the resistance couples, as equal to $\frac{2}{3}d$.

The corresponding diagram for a simple reinforced concrete beam would be that of fig. 2. Here the compressive resistance, which occurs above the neutral axis, is represented, as before, by a triangle; while the tensile resistance, being that of the steel reinforcement only, is represented by a line (the tensile strength of the concrete is ignored). The position of the resultant of the compressive resistance would pass through the centre of gravity of the triangle, or at $\frac{1}{3}n$ from the upper surface ; the lever arm in this case would equal $d - \frac{n}{3}$.

Where double reinforcement is adopted—that is to say, where a certain amount of steel is introduced into the upper or compression area of the beam to assist the concrete (fig. 3)—the total compressive resistance would, in addition to that of the concrete, include the compressive resistance of the steel : the combined resultant of these two resistances would equal the sum of the two, and would act at some point between the two independent resistances, the exact position depending on the relative values of these two resistances.

In the case of T beams, or those in which the upper or compression flange is formed in part of the flooring slabs with which the beam proper is combined, two cases arise : the one in which the distance from the neutral axis to top of beam (n) is *not greater* than the thickness of the floor slab (d_s), and the other in which this value (n) is *greater* than the thickness of the slab (d_s). Since the compressive stress is assumed as being resisted by the concrete alone, the first case naturally falls under the same condition as that of a simple beam (fig. 2), except that the breadth of the beam must now be taken, not as the breadth of the reinforced beam only (b_r), but must include a certain portion of the slab at each side of the main beam, or b_s.

The difference between the first and second cases referred to will be obvious by a comparison between figs. 4 and 5. In the former the compression resistance is represented by a triangle, with a resultant acting $\frac{1}{3}n$ from the upper surface; while in the latter case the compressive resistance is represented by the area of a trapezoid, the point of action being through the centre of gravity of this figure. It should be noted that in this latter case the compressive value of the concrete in the main beam between the bottom of slab and the neutral axis is ignored, as its value is very slight, and the formulæ are much simplified by doing this.

Two extensions of the above two cases are shown in figs. 6 and 7, where double reinforcement is adopted ; the treatment in these two cases is modified generally, as previously noted when referring to figs. 2 and 3 for rectangular beams.

The following series of formulæ for the several types of single and double reinforced beams just referred to will enable the scantlings for any such members to be readily calculated or the designs for these checked with the minimum of trouble.

Rectangular Beams with Single Reinforcement.

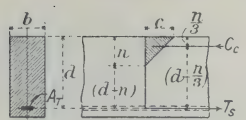


FIG. 2.

$$n=\frac{1}{1+\frac{t}{cm}} \quad . \quad . \quad . \quad . \quad (1)$$

$$r=\frac{1}{\frac{2t}{c}\left(1+\frac{t}{cm}\right)} \quad . \quad . \quad (2)$$

$$t=c\frac{(1-n)m}{n} \quad . \quad . \quad . \quad (3)$$

$$c=t\frac{n}{(1-n)m} \quad . \quad . \quad . \quad . \quad (4)$$

$$A_T=\frac{B}{td\left(1-\frac{n}{3}\right)} \quad . \quad . \quad . \quad (5)$$

$$n=\sqrt{(mr)^2+2mr}-mr \quad . \quad . \quad (6)$$

$$R=bd^2\left[tr\left(1-\frac{n}{3}\right)\right] \quad . \quad . \quad (7)$$

$$R=bd^2\left[\tfrac{1}{2}cn\left(1-\frac{n}{3}\right)\right] \quad . \quad . \quad (8)$$

SYMBOLS.

n = Ratio of distance of neutral axis from compression edge of beam to effective depth.
r = Ratio of area of steel to area of concrete = A_T/bd.
t = Tensile stress intensity on steel.
c = Compressive stress intensity on concrete.
m = Modulus ratio = E_s/E_c.
A_T = Area of tensile reinforcement.
B = Bending moment of external forces.
R = Resistance moment of internal stresses.
b = Breadth of beam.
d = Effective depth of beam.

DESIGNING.—Decide values for m, t, c, and trial dimensions for b and d. Calculate n by (1) and r by (2); then value of A_T by (5). R is got from (7) or (8). If any other value of r is used than is given by (2), calculate n by (6) and R by (7) and (8), the lesser of these two values being adopted.

CHECKING A DESIGN.—Calculate r ($=A_T/bd$), and with this determine n from (6); test t and c by (3) and (4). If t exceeds the safe stress, get R from (7); if c exceeds the safe stress, get R from (8), in each case using the maximum safe stress for t or c.

Note.—Where more than one bar is used to form the reinforcement, measure d to the centre of gravity of the group.

Usual Dimensions.—d = one-twelfth to one-sixteenth of span; b = one-half to two-thirds d.

Rectangular Beams with Double Reinforcement.

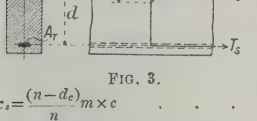


FIG. 3.

$$c_s=\frac{(n-d_c)}{n}m\times c \quad . \quad . \quad . \quad (9)$$

$$c_s=\frac{(n-d_c)}{(1-n)}t \quad . \quad . \quad . \quad (10)$$

$$A_c=\frac{B-\left[\tfrac{1}{2}cnbd^2\left(1-\frac{n}{3}\right)\right]}{c_s d(1-d_c)} \quad . \quad . \quad (11)$$

$$A_T=\frac{B-\left[A_c c_s d\left(\frac{n}{3}-d_c\right)\right]}{dt\left(1-\frac{n}{3}\right)} \quad . \quad . \quad (12)$$

$$n=\sqrt{m^2(r_t+r_c)^2+2m(r_t+r_c d_c)}-m(r_t+r_c) \quad (13)$$

$$R=bd^2\left[tr_t\left(1-\frac{n}{3}\right)+c_s r_c\left(\frac{n}{3}-d_c\right)\right] \quad (14)$$

$$R=bd^2\left[\tfrac{1}{2}cn\left(1-\frac{n}{3}\right)+c_s r_c(1-d_c)\right] \quad 15)$$

ADDITIONAL SYMBOLS.

c_s = Compressive stress intensity on steel.
d_c = Ratio of depth to centre of compression reinforcement to effective depth.
A_c = Area of compression reinforcement.
r_t = Ratio of area of steel in tension to area bd.
r_c = Ratio of area of steel in compression to area bd.

DESIGNING.—Calculate n by (1), then c_s by (9) or (10); the area of Compression Steel is got from (11), and the area of Tension Steel from (12). The Resistance Moment is got from (14) or (15).

CHECKING A DESIGN.—Calculate r and r_c from scantlings given, and n from (13). c_s is got from (9) and (10), the smaller value being adopted. R is obtained from (14) and (15), the former assuming the concrete, and the latter the steel as stressed to their respective safe limits; the least value of R from (14) and (15) must consequently be taken as the maximum safe R.

See Diagrams on pages 460–461.

T Beams with Single Reinforcement.

Class I.—n not greater than d_s.

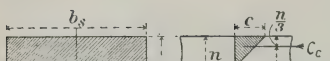


FIG. 4.

$$n=\frac{1}{1+\frac{t}{cm}} \quad . \quad . \quad . \quad . \quad (16)$$

$$r=\frac{b_s}{\frac{2t}{c}\left(1+\frac{t}{cm}\right)b_r} \quad . \quad . \quad . \quad . \quad (17)$$

$$t=c\frac{(1-n)m}{n} \quad . \quad . \quad . \quad . \quad (18)$$

$$c=t\frac{n}{(1-n)m} \quad . \quad . \quad . \quad . \quad (19)$$

$$A_T=\frac{B}{td\left(1-\frac{n}{3}\right)} \quad . \quad . \quad . \quad . \quad (20)$$

$$n=\sqrt{\left(mr\frac{b_r}{b_s}\right)^2+2mr\frac{b_r}{b_s}}-mr\frac{b_r}{b_s} \quad . \quad (21)$$

$$R=b_rd^2\left[tr\left(1-\frac{n}{3}\right)\right] \quad . \quad . \quad . \quad (22)$$

$$R=b_sd^2\left[\tfrac{1}{2}cn\left(1-\frac{n}{3}\right)\right] \quad . \quad . \quad (23)$$

Symbols.

n = Ratio of distance of neutral axis from compression edge of beam to effective depth.
r = Ratio of area of steel to area (b_rd).
t = Tensile stress intensity on steel.
c = Compressive stress intensity on concrete.
m = Modulus ratio = E_s/E_c.
A_T = Area of tensile reinforcement.
B = Bending moment of external forces.
R = Resistance moment of internal stresses.
b_s = Breadth of T slab.
b_r = Breadth of rib.
d = Effective depth of beam.

Designing.—Decide values for m, t, c, and trial dimensions for b_r, b_s, and d. Calculate n by (16) and r by (17), then value of A_T by (20). R is got from (22) or (23). If any other value of r is used than is given by (17), calculate n by (21) and R by (22) and (23), the lesser of these two values being adopted.

Checking a Design.—Calculate r ($=A_T/b_rd$), and with this determine n from (21). Test t and c by (18) and (19). If t exceeds the safe stress, get R from (22), if c exceeds the safe stress get R from (23), in each case using the maximum safe stress for c or t.

T Beams with Single Reinforcement.

Class II.—n greater than d_s.

FIG. 5.

$$r=\frac{nd_s\left(1-\frac{d_s}{2n}\right)b_s}{(1-n)mb_r} \quad . \quad . \quad . \quad . \quad (24)$$

$$A_T=\frac{B}{td\left(1-\frac{3nd_s-2d_s^2}{6n-3d_s}\right)} \quad . \quad . \quad (25)$$

$$=\frac{d^2r\frac{b_s}{b_r}+2mr}{\frac{b_s}{b_r}+2mr} \quad . \quad . \quad . \quad . \quad (26)$$

$$R=bd^2\left[\left(1-\frac{d_s}{2n}\right)cd_s\left(1-\frac{3nd_s-2d_s^2}{6n-3d_s}\right)\right] \quad (27)$$

$$R=b_rd^2\left[tr\left(1-\frac{3nd_s-2d_s^2}{6n-3d_s}\right)\right] \quad . \quad . \quad (28)$$

Additional Symbol.

d_s = Ratio of depth or thickness of slab to effective depth of beam.

Designing.—Decide values for m, t, c, and trial dimensions for b_r, b_s, and d. Calculate n by (16) and r by (24), then value of A_T by (25). R is got from (27) or (28). If any other value of r is used than is given by (24), calculate n by (26), and R by (27) and (28), the lesser of these two values being adopted.

Checking a Design.—Calculate r ($=A_T/b_rd$), and with this determine n from (26). Test t and c by (18) and (19). If t exceeds the safe stress get R from (28), if c exceeds the safe stress get R from (27), in each case using the maximum safe stress for c or t.

T Beams with Double Reinforcement.

Class I.—n not greater than d_s.

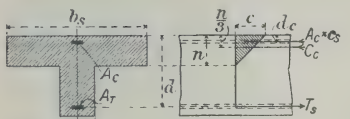


FIG. 6.

SYMBOLS.

n = Ratio of distance of neutral axis from compression edge of beam to effective depth.
t = Tensile stress intensity on steel.
c = Compressive stress intensity on concrete.
c_s = Compressive stress intensity on steel.
m = Modulus ratio = E_s/E_c.
A_T = Area of tensile reinforcement.
A_c = Area of compression reinforcement.
b_s = Breadth of T slab.
b_r = Breadth of rib.
d = Effective depth of beam.
d_c = Ratio of depth to centre of compression reinforcement to effective depth of beam.
r_t = Ratio of area of steel in tension to area $b_r d$.
r_c = Ratio of area of steel in compression to area $b_r d$.
B = Bending moment of external forces.
R = Resistance moment of internal stresses.

$$n=\frac{1}{1+\frac{t}{cm}} \quad . \quad . \quad . \quad . \quad (29)$$

$$t=c\frac{(1-n)m}{n} \quad . \quad . \quad . \quad . \quad (30)$$

$$c=t\frac{n}{(1-n)m} \quad . \quad . \quad . \quad (31)$$

$$c_s=\frac{(n-d_c)m\times c}{n} \quad . \quad . \quad . \quad . \quad (32)$$

$$c_s=\frac{(n-d_c)_t}{(1-n)} \quad . \quad . \quad . \quad . \quad (33)$$

$$A_c=\frac{B-\left[\frac{1}{2}cnb_sd^2\left(1-\frac{n}{3}\right)\right]}{c_sd(1-d_c)} \quad . \quad . \quad (34)$$

$$A_T=\frac{B-\left[A_cc_sd\left(\frac{n}{3}-d_c\right)\right]}{td\left(1-\frac{n}{3}\right)} \quad . \quad . \quad (35)$$

$$n=\sqrt{\left[\frac{m(r_t+r_c)b_r}{b_s}\right]^2+\left[\frac{2m(r_t+r_cd_c)b_r}{b_s}\right]}-\left[\frac{m(r_t+r_c)b_r}{b_s}\right] \quad (36)$$

$$R=b_rd^2\left[tr_t\left(1-\frac{n}{3}\right)+c_sr_c\left(\frac{n}{3}-d_c\right)\right] \quad . \quad (37)$$

$$R=d^2\left[\tfrac{1}{2}cnb_s\left(1-\tfrac{n}{3}\right)+c_sr_cb_r(1-d_c)\right] \quad (38)$$

DESIGNING.—Calculate n by (29), then c_s by (32) or (33), the area of Compression Steel is got from (34), and the area of Tension Steel from (35). The Resistance Moment is got from (37) or (38).

CHECKING A DESIGN.—Calculate r_t and r_c from scantlings given, and n from (36). c_s is got from (32) and (33), the smaller value being adopted. R is obtained from (37) and (38), the former assuming the Steel and the latter the Concrete as stressed to their respective safe limits. The least value of R from (37) and (38) must consequently be taken as the maximum safe R.

T Beams with Double Reinforcement.

Class II.—n greater than d_s.

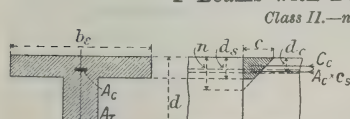


FIG. 7.

ADDITIONAL SYMBOL.

d_s = Ratio of depth or thickness of slab to effective depth of beam.

$$A_c=\frac{B-\left[\left(1-\frac{d}{2n}\right)cd_sb_sd^2\left(1-\frac{3nd_s-2d_s^2}{6n-3d_s}\right)\right]}{c_sd(1-d_c)} \quad (39)$$

$$A=\frac{B-\left[A_cc_sd\left(\frac{3nd_s-2d_s^2}{6n-3d_s}-d_c\right)\right]}{td\left(1-\frac{3nd_s-2d_s^2}{6n-3d_s}\right)} \quad . \quad (40)$$

$$n=\frac{d_s^2b_s+2m(r_t+r_cd_c)b_r}{2d_sb_s+2m(r_t+r)b} \quad . \quad . \quad (41)$$

$$R=d^2\left[\left(1-\frac{d_s}{2n}\right)cd_sb_s\left(1-\frac{3nd_s-2d_s^2}{6n-3d_s}\right)+c_sr_c(1-d_c)r\right] \quad . \quad (42)$$

$$R=b_rd^2\left[tr_t\left(1-\frac{3nd_s-2d_s^2}{6n-3d_s}\right)+c_sr_c\left(\frac{3nd_s-2d_s^2}{6n-3d_s}-d_c\right)\right] \quad . \quad (43)$$

DESIGNING.—Calculate n from (29), then c_s by (32) or (33), the area of Compression Steel is got from (39) and the area of Tension Steel from (40). The Resistance Moment is got from (42) or (43).

CHECKING A DESIGN.—Calculate r_t and r_c from scantlings given, and n from (41), c_s is got from (32) and (33), the smaller value being adopted. R is obtained from (42) and (43), the former assuming the Concrete and the latter the Steel as stressed to their respective safe limits. The least value of R from (42) and (43) must consequently be taken as the maximum safe R.

Note.—In many cases of double reinforcement the compression reinforcement can be placed at the centre of gravity of the compression area; the formulæ for R could then be somewhat simplified.

Shearing Stresses and Diagonal Tension in Beams.

These stresses and the methods of resisting them require careful consideration in all cases. At any point in a beam, the horizontal and vertical shearing stresses are of equal value, and these stresses may be looked upon as the connecting link between successive increments of tension in the reinforcement and corresponding increment of compression in the concrete; also, in the case of double reinforcement, in connecting up the increments of tension and compression in the two forms of reinforcement. The tendency of the longitudinal shearing stress is to destroy the adhesion between the concrete and steel, and it is necessary to calculate this value to ensure that no slipping or failure takes place from this cause.

If S represents the total vertical shear, which is equal to the rate of change of the bending moment from point to point along the beam; s_b the shearing stress between concrete and steel; O the periphery of reinforcement, whether one bar or a number of bars; the shearing stress round the bars per unit surface of reinforcement is approximately equal to :—

$$s_b = \frac{S}{O \times 0{\cdot}86d} \quad . \quad . \quad . \quad . \quad . \quad . \quad (51),$$

and this same stress per unit length may be written—

$$s_b O = \frac{S}{0{\cdot}86d} \quad . \quad . \quad . \quad . \quad . \quad . \quad (52).$$

The fraction $0{\cdot}86d$ in the above equations represents the approximate average length of the lever arm, or distance between the centre of tension in the steel and the centre of compression in the concrete; and these equations may therefore be used to give practical values for rectangular or T beams, either with single or double reinforcement.

If s represents the unit horizontal shearing stress in the concrete, the shearing stress per unit length on any width b of a beam is as follows :—

$$s \times b = \frac{S}{0{\cdot}86d} \quad . \quad . \quad . \quad . \quad . \quad . \quad (53).$$

This last formula not only gives the value of the horizontal shearing stress, but also that of the vertical shearing stress at the same point, since both these stresses are always of equal value.

Failure by *diagonal tension* is the cause of numerous mishaps, which are frequently attributed to shear. The maximum diagonal tension in a reinforced concrete beam is largely governed by the amount of the horizontal tension in the concrete as well as that of the shearing stress at the same point. From practical experiments and data it would appear that, with average concrete, beams with no web reinforcement are liable to fail by diagonal shear when the vertical shearing stress intensity, at point of failure, averages about 120 lbs. per square inch. Accepting a factor of safety of 4, the shearing stress intensity given by formula (53) should not exceed 30 lbs. per square inch. With the same factor of safety, the shearing stress calculated by this formula in beams with web reinforcement may be taken at 120 lbs. per square inch.

Web Reinforcement.

This generally consists either of vertical or diagonal stirrups, alone or in conjunction with inclined bars, these latter generally consisting of a portion of the main reinforcing bars bent upwards as required.

The sectional area of steel to withstand the total shearing force at any vertical section, when the shear bar is inclined at an angle ϕ with the horizontal, should be as follows :—

$$\text{Sectional area required} = \frac{S}{t \times \sin \phi} \quad . \quad . \quad . \quad . \quad (54),$$

t being the tensile strength per square inch of the steel.

The resistance to shear and to diagonal tension in a T beam is practically similar to that for rectangular beams, if the breadth in the former be taken as the breadth of the rib (b_r), this value being compared with the breadth (b) of the rectangular beam.

The breadth of rib in T beams (b_r) will usually be found sufficient to resist any horizontal shear when of a size to give sufficient practical space for embedding the steel reinforcing bars, but to allow for any excessive shear, and for diagonal tension, it is well to keep this breadth of ample size.

Inverted Beams.

Where a beam is continuous over one or more intermediate supports, or where a single span beam is securely fixed or built in at its ends, the effect on the continuous beam over the supports or at the ends of the single span beam is to produce reverse bending, and consequently an inversion of the stresses; the upper portion now being in tension, while the lower portion is in compression. To meet this change in stress, it is usual to bend up a number of the lower reinforcing bars as they approach the ends, and for partial fixing this may be sufficient, but it will generally be found that additional reverse stress bars (sometimes called 'continuity bars') must be added to provide the full area of reinforcement necessary at the upper portion of the beam.

In the case of inverted T beams, it will usually be found that the neutral axis falls outside the slab, and it may therefore be calculated as for an ordinary rectangular beam, of a breadth (b) equal to the breadth of rib (b_r).

Floor Slabs.

By imagining any slab to be broken up into a series of parallel strips, each say 12 ins. wide, it is only necessary to treat each strip as an independent beam, and determine the thickness and the amount of reinforcement according to the particular conditions of loading, etc. The amount of reinforcement thus found will give that required per 12 ins. of width, and may be arranged in the form of a series of small rods set at the required spacing to give this calculated area.

When slabs are reinforced in one direction only, it is generally assumed that the two beams upon which the ends of the reinforcing bars rest will carry the complete load, but should the slab be of approximately square shape, and the reinforcement arranged in two directions, all four beams upon which the slab rests are assumed as taking a share of the total load. In the case of a square slab supported on all four sides, it will be sufficiently accurate to calculate the total load as distributed equally over all four girders. If the slab is of rectangular form, the short or end girders may be calculated as each supporting a triangular area formed by the beam and by lines drawn at 45° from its corners; the long or side girders each supporting one-half the remainder of the slab. This will be obvious from an examination of figs. 8 and 9, where an indication is given of the manner in which two plain concrete slabs failed when tested to destruction.

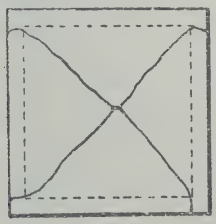


FIG. 8.

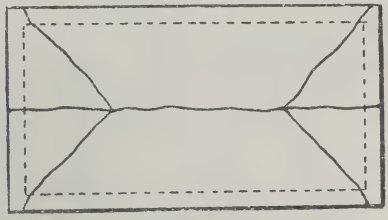


FIG. 9.

There is no satisfactory theory as to the stresses in slabs supported on all sides, nor does there seem to be any single generally accepted rule for determining these stresses. The table below gives a series of multipliers for adjusting the bending moment stresses for a slab as calculated in the direction of its length and its breadth; these multipliers are based respectively on the theory developed by Professors Grashof and Rankine, and on the instructions issued by the French Government to the Ingénieurs des Ponts et Chaussées. A comparison of the two series will show a marked divergence of opinion. It is recommended that for ordinary requirements the first rule be adopted, but where the work is carried out under very careful supervision, and first-class material and workmanship are ensured, the second rule may be applied.

Rectangular Slabs supported on four sides.

Value of $\frac{L}{B}$	Grashof and Rankine.		French Government.	
	$s=\frac{L^4}{L^4+B^4}$	$l=\frac{B^4}{L^4+B^4}$	$s=\frac{1}{1+2\frac{B^4}{L^4}}$	$l=\frac{1}{1+2\frac{L^4}{B^4}}$
1·1	·500	·500	·333	·333
1·1·1	·594	·406	·423	·254
1·1·2	·675	·325	·509	·194
1·1·3	·741	·259	·588	·149
1·1·4	·793	·206	·658	·115
1·1·5	·835	·165	·717	·090
1·1·6	·868	·132	·766	·071
1·1·7	·893	·107	·807	·056
1·1·8	·913	·087	·840	·045
1·1·9	·929	·071	·867	·037
1·2·0	·941	·059	·889	·030

A glance at this table will show that when the proportion of length to breadth approaches that of 1 to 2, the effect of the end supporting becomes negligible, and beyond this limit slabs should

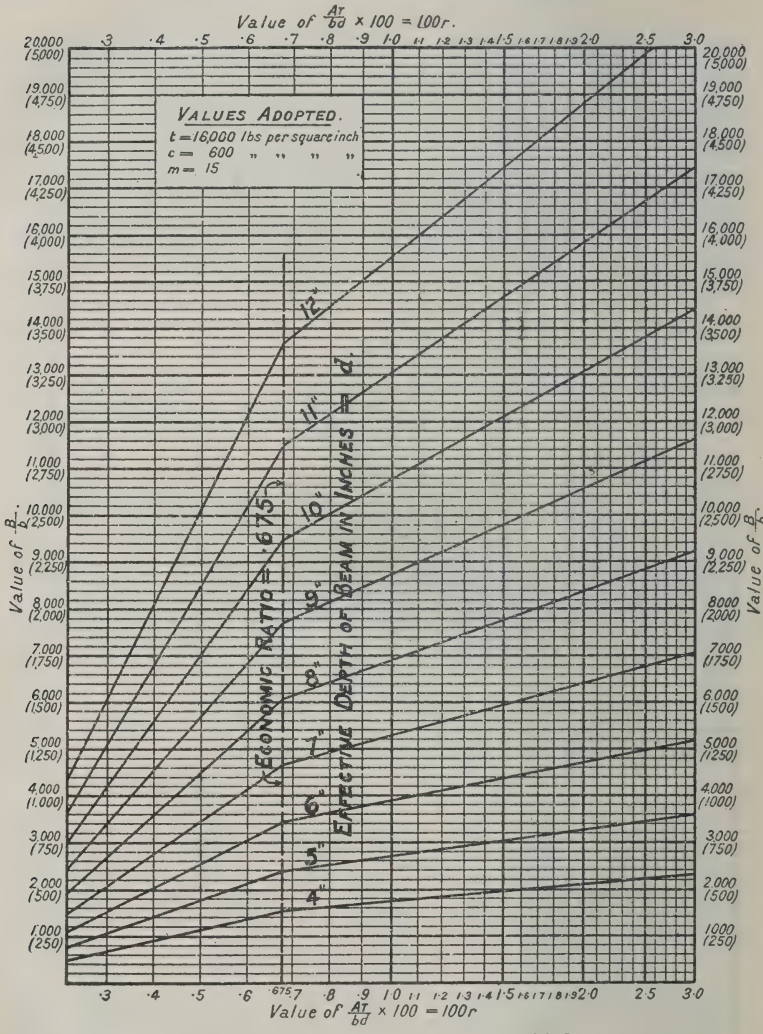
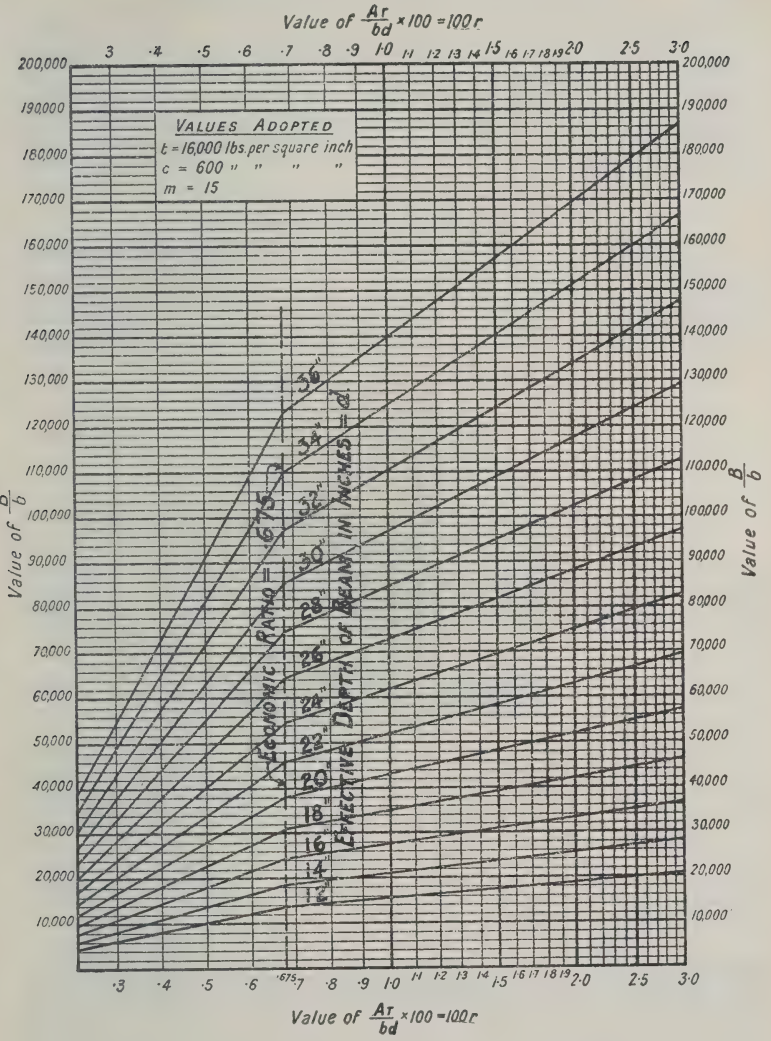


DIAGRAM OF $\frac{B}{b}$ VALUES FOR VARIOUS RATIOS OF REINFORCEMENT.

(Figures in brackets give $\frac{B}{b}$ Values for Beams of one-half the above depths.)



RECTANGULAR BEAMS— SINGLE REINFORCED.

DIAGRAM OF $\frac{B}{b}$ VALUES FOR VARIOUS RATIOS OF REINFORCEMENT.

be single reinforced, and treated as if supported on two sides only The use of the foregoing table is as follows :—

In any floor slab of length L and breadth B, calculate the bending moment first for a beam of span B under the total load from the slab, and as supported or fixed at the sides only. Multiply the bending moment thus found by the multiplier s in the table corresponding to the value L/B to determine the actual bending moment on the long axis of the slab. By a similar operation, with a span L, and the use of the multiplier l, determine the actual bending moment on the short axis. By means of the formulæ already given for rectangular beams, the necessary thickness and amount of reinforcement can be determined, and it remains only to distribute this total area of reinforcement in the form of small bars arranged at the necessary spacing throughout the full length and breadth of the slab.

Deflection of Reinforced Concrete Beams.

An ingenious method of calculating the approximate deflection of a reinforced concrete beam or slab was suggested by Mr. Eli White in the 'Engineering Record' for November 9, 1907. The following explanations are based on this method :—

The deflection of a reinforced concrete beam being proportionate to the elongation of the steel tensile reinforcement, this deflection will be similar in amount, for similar tensile stress intensities, to that of a symmetrical beam, whose flanges are each of an area equal to the tensile reinforcement in the reinforced concrete beam, and are placed at a distance apart equal to twice that of

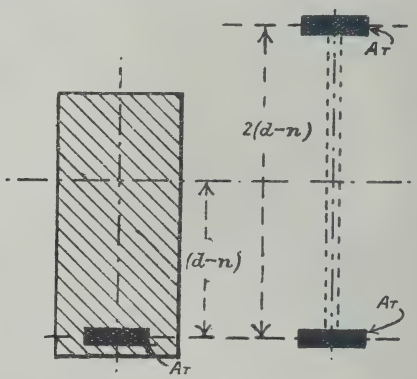


FIG. 10.

the distance between the tensile steel and the neutral axis in the reinforced concrete beam Fig. 10 indicates to the left a sketch section of a reinforced concrete beam, with tensile reinforcement equal to A_T, and the sketch to the right indicates the section of an imaginary symmetrical steel beam, whose two flanges are each equal in area to A_T and are placed at $2(d-n)$ apart. If t = tensile stress intensity on steel, the

$$\text{Resistance Moment of this symmetrical beam} = 2(d-n) \times A_T \times t. \quad . \quad (55).$$

When any beam is carrying a load, it is obvious that the resistance moment must be equal to the bending moment of the external force acting on the beam.

For this symmetrical beam, the

$$\text{Approximate Moment of Inertia value} = 2A_T \times (d-n)^2 \quad . \quad . \quad (56),$$

and taking E at 30,000,000 lbs., the deflection formulæ given in the tables on pages 212 and 213 can be applied to calculate the approximate deflection in any single or double reinforced concrete beam, the correct value for $(d-n)$ being used in the above formulæ.

It should be noted that in the above formulæ the tensile resistance of the concrete has been entirely neglected, and that the formulæ for deflection on pages 212 and 213 are those for beams of uniform section (reinforced concrete beams only more or less approach this condition) ; for these reasons the calculated deflection will generally be distinctly higher than the actual values, which may at times be found as much as 10 to 15 per cent. less than given by these calculations.

The tables on pages 212 and 213 may prove of service in determining *graphically* the bending moment and shearing stress values for various types of loading ; and since these diagrams show clearly the *change of values* they can frequently be used in arranging the main reinforcement and the points where these bars may be bent up to resist shear ; independent shear members can also be set out by the same means.

COLUMNS, PILES, AND WALLS.

SYMBOLS USED IN THE FOLLOWING FORMULÆ.

A = Sectional Area of Concrete in Column.

A_E = Equivalent Area of Concrete to that of Concrete and Steel Longitudinals.

A_L = Sectional Area of Steel Longitudinal Reinforcement.

A_H = „ „ „ Hooping.

W = Safe Load on Short Column.

W_I = „ „ Long „

C_u = Ultimate Compressive Stress on Concrete at 90 days.

c = Safe Stress per square inch on Concrete.

p = Percentage of Longitudinals to Concrete = $A_L \div A$.

l = Length of Column in inches.

I_c = Moment of Inertia of Concrete about Neutral Axis of Column.

I_L = „ „ of Longitudinals „ „ „

G_E = Radius of Gyration of A_E.

b = Smaller Side of Rectangle, or Side of Square.

b_y = Larger „ „

d = Diameter of Round Column, or of Round Rods.

y = Distance of Longitudinal from Neutral Axis, in inches.

x = „ Eccentric Load from Centre of Column.

r_h = Ratio of Volume of Hooping to Volume of Concrete.

Short Columns Reinforced with Longitudinal Bars and Cross Ties.

Load Applied at Centre.

In ordinary cases, the Longitudinal Reinforcement takes the form of round or square steel rods of comparatively small area placed symmetrically about the cross-section of the column; these Longitudinals are held or tied securely together by Horizontal Bindings, which should not be set further apart than twenty to twenty-four times the least diameter of the Longitudinals. This spacing of Bindings should, as a rule, be reduced towards the end of the columns, where in the majority of cases failure first occurs.

When the length does not exceed about 18 times the least diameter or width, and the Maximum Stress per sq. in. on the concrete is limited to 600 lbs., the column may be expected to yield by crushing. Since the Modulus of Elasticity of the Steel Longitudinals is assumed as 15 times that of the concrete, the Maximum Stress on the steel will be 15 times that of the concrete only; 1 sq. in. of steel is, therefore, equal to 15 sq. ins. of concrete. It is sometimes convenient to consider a reinforced column as consisting of concrete only; in which case the sectional area of the concrete would be increased by 14 times the sectional area of the steel reinforcement to give this equivalent area. This may be written :—

$$A_E = A + 14A_L \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (57)$$

Thus, the Safe Load for a short column would be as follows :—

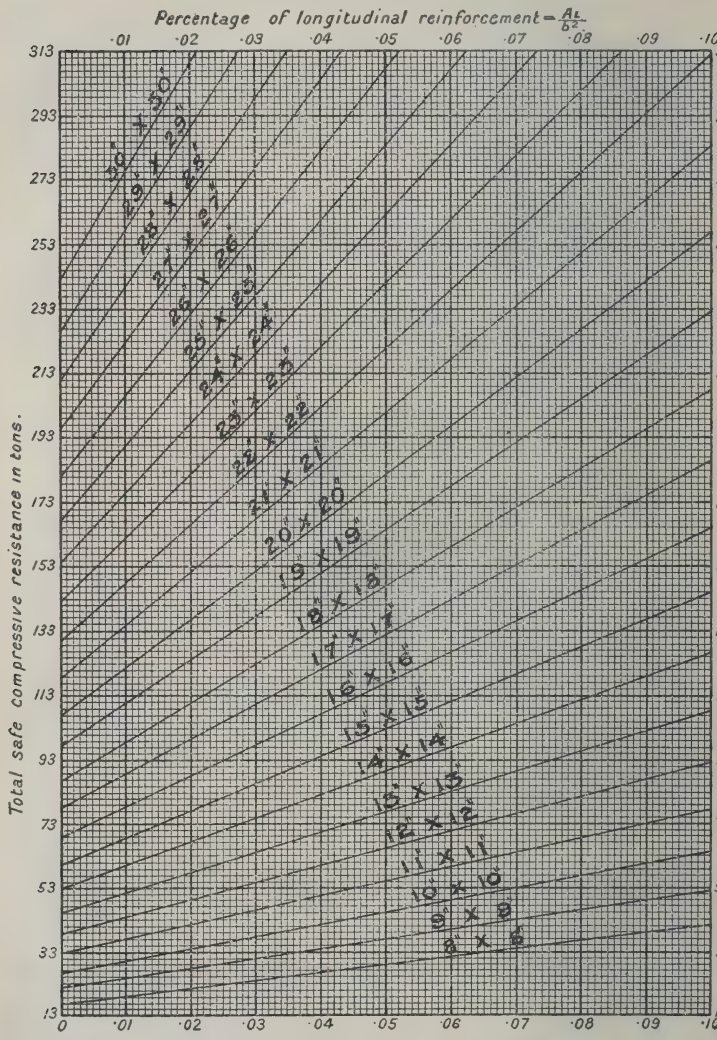
$$W = c\,(A + 14A_L) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (58)$$

It is sometimes convenient to consider the area of the steel longitudinals as a certain percentage of the total area of the concrete, and when this is done, we have the following equation :—

$$\text{Safe Compressive Stress per square inch} = c\,(1 + 14\,p) \quad . \quad . \quad (59)$$

The following table gives the Safe Stress Values for varying percentages of Steel, and may be used directly when designing Short Columns :—

Percentage of Reinforcement.	Safe Stress, when $c = 600$.	Percentage of Reinforcement.	Safe Stress, when $c = 600$.
$\frac{1}{2}$	642 lbs. per sq. in.	4	936 lbs. per sq. in.
$\frac{3}{4}$	663 „ „	5	1,020 „ „
1	684 „ „	6	1,104 „ „
$1\frac{1}{2}$	726 „ „	7	1,188 „ „
2	768 „ „	8	1,272 „ „
$2\frac{1}{2}$	801 „ „	9	1,356 „ „
3	852 „ „	10	1,440 „ „



TOTAL SAFE LOADS FOR SQUARE COLUMNS, WITH LONGITUDINAL REINFORCEMENT ONLY

High Columns Reinforced with Longitudinal Bars and Cross Ties.

Load applied at Centre.

When the length exceeds about 18 times the least diameter or width there is a risk of Deflection or Lateral Buckling of the column as a whole. It is necessary, therefore, to take into account this tendency, and what is known as Rankine's formula may be used for this purpose. Here the strength calculated as for a short column is reduced in proportion to the ratio of length of the column to its least Radius of Gyration. The formulæ for three forms of end fixings are as under :—

$$\text{Fixed Ends} \quad W_1 = \frac{W}{1 + \frac{l^2}{34000\,(G_E{}^2)}} \qquad (60)$$

$$\text{One End fixed, the other free} \quad W_1 = \frac{W}{1 + \frac{l^2}{25500\,(G_E{}^2)}} \qquad (61)$$

$$\text{Free Ends} \quad W_1 = \frac{W}{1 + \frac{l^2}{17000\,(G_E{}^2)}} \qquad (62)$$

$$\text{Where } G_E{}^2 = \frac{I_c + \cdot 14\, I_L}{A_E} \qquad (63)$$

The values for I_c may be taken as follows :—

Any Rectangular Section (b = small side)	$= \cdot 0833 b^3 b_y$	(64)
„ Hexagonal Section (b_x = rad. of circumscribed circle)	$= \cdot 5413 b_x{}^4$	(65)
„ Octagonal Section („ „ „)	$= \cdot 638 b_x{}^4$	(66)
„ Circular Section (d = diameter)	$= \cdot 0491 d^4$	(67)

If all Longitudinals are of small size and the same section,

$$I_L = \frac{A_L\, \Sigma y^2}{\text{Number of Longitudinals}} \qquad (68)$$

Piles, Walls, etc.

In the case of Piles, the formula given for short columns may be utilised, but it would be well to reduce the Maximum Compressive Stress to 400 lbs. per sq. in., or under, and to close up the spacing of the Cross Ties, particularly towards the ends, owing to the heavy stress which arises during driving. For unsupported lengths of Piles, where they exceed in length 18 times their diameter, the formula for High Columns would be adopted.

In the case of walls, these may be assumed as broken up into a series of independent columns set close together, and their strength calculated as for short or long columns, the particular formula depending on the relation between height and the thickness of wall.

Columns with Eccentric Loads.

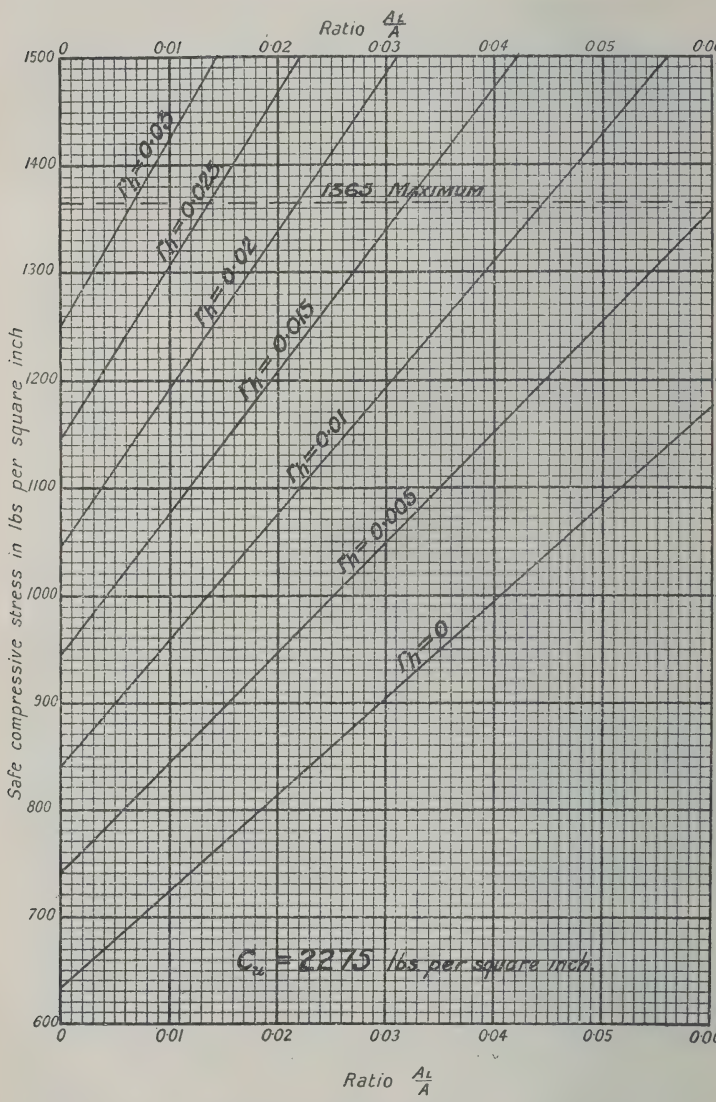
When a load is applied at a distance 'x' from the centre of a column, the stress produced may be considered as that due to two independent actions : *first*, a load equal to the Total Eccentric Load applied at the centre of the column, and giving rise to a Uniformly Distributed Compressive Stress, and *second*, a Couple, with a Moment equal to the Eccentric Load $\times\, x$. This gives rise to a bending tendency, the effect of which may be calculated by considering the column as a Double Reinforced Beam, acted on by a Bending Moment equal in value to this couple. The Maximum Compressive Stress per sq. in. got by this calculation must be added to the Uniform Compressive Stress first got, to arrive at the Maximum Compressive Stress on column at edge nearest the load. At edge furthest from the load the Stress will equal the difference between the Tensile Stress per sq. in. got when calculating as for a beam, and the Uniform Compressive Stress. When the Eccentricity is considerable, the compression at furthest edge may be changed into tension. To resist any such tension, the Steel Longitudinals at that side should alone be calculated on.

Columns with Longitudinal Bars and Steel Hooping.

This type of column was carefully investigated by M. Considère, who showed that by confining the Concrete within a close set Steel Hooping, so as to prevent it bulging, its resistance to Compression was very greatly increased. From experiments made, this Engineer recommended that the hooping should in all cases be circular, that the Longitudinal Rods should be not less than 6 in number, and that the total volume of the Longitudinals should not be less than $\frac{1}{100}$th of the volume of the Concrete. It will be noted that these proportions are given in volumes, not in areas so as to take into account any necessary pitch of the spiral hooping.

The diameter of the column should be assumed as that within the spiral hooping, and, taking as a unit a length equal to the diameter, we have :—

Unit Volume of Concrete	$= \cdot 7854 d^2$	(69)
Unit Volume of Longitudinals	= total area of Longitudinals $\times\, d$	(70)
Unit Volume of Hooping	= Sectional area of coil $\times 3 \cdot 14 d \times \frac{d}{\text{pitch}}$	(71)



SAFE STRESS PER SQUARE INCH FOR CIRCULAR HOOPED COLUMNS.

The following is the formulæ adopted by the French Government Commission on Reinforced Concrete :—

Safe compressive resistance $= c(0{\cdot}7854d^2 + 14A_L)$ . . . . . (72)

For pitches of hooping between $\frac{1}{5}d$ and $\frac{1}{8}d$, $c = 0{\cdot}28\ C_u\ (1 + 32r_h)$ . . . (73)

The value of c should not exceed $0{\cdot}6\ C_u$.

The value of C_u for the usual 1 : 2 : 4 mixture of Concrete may be taken as 2,275 lbs. per sq. in.

The approximate diameters for the hooping may be taken as follows :—

For a pitch of $\frac{1}{5}d$ diameter $= {\cdot}030d$ . . . . . (74)

" " $\frac{1}{6}d$ " $= {\cdot}028d$ . . . . . (75)

" " $\frac{1}{7}d$ " $= {\cdot}026d$ . . . . . (76)

" " $\frac{1}{8}d$ " $= {\cdot}024d$ . . . . . (77)

PIPES, CIRCULAR RESERVOIRS OR BINS, etc.

SYMBOLS USED IN THE FOLLOWING FORMULÆ.

A_h = Sectional area of hoop reinforcement.
C = Direct compressive stress on the shell.
b_r = Breadth of ring under consideration, in inches
d = Diameter of pipe or reservoir, in inches.
t = Tensile stress in lbs. on steel reinforcement.
p_1 = Pressure in lbs. per sq. in. on shell.
H_w = Head of water, in feet.

When under Internal Pressure.

(Concrete neglected.)

Consider any circumferential strip or ring, then :—

$$A_h = \frac{p_1 \times d \times b_r}{2t} \quad . \quad . \quad . \quad . \quad (78)$$

The total area thus found is divided up into any suitable number of hoops or rings.

Where water pressure has to be resisted :—

$$A_h = \frac{0{\cdot}216H_w \times d \times b_r}{t} \quad . \quad . \quad . \quad (79)$$

In addition to the hoop or spiral reinforcement determined above, longitudinal bars are necessary to strengthen the concrete shell between each hoop to resist the pressure they have to sustain ; the shell here acts as a fixed ended beam or slab of a span equal to the distance between the hoops, and the scantlings calculated by the formulæ previously given for such a type of beam.

Practical considerations generally decide the thickness of the concrete shell, which is usually $1\frac{1}{2}$ to 2 ins. thick for pipes up to 9 or 10 ins. diameter, and seldom exceeds 3 to $3\frac{1}{2}$ ins. even for large diameter pipes. In vertical reservoirs or bins the thickness towards the base must also be such as will safely carry the weight of the shell.

NOTE.—The required reinforcement for circular bins or silos may be determined by calculating 'p' from formulæ (82) or (83) on next page, for two levels say 12 ins. apart ; the difference of the two values thus got will give the pressure per lin. foot and per one foot in height between these levels. Dividing this value by 12×12 (144) will give the average pressure per sq. in., and formula (78) above will at once decide the necessary hoop reinforcement.

When under External Pressure.

(Concrete and hooping both effective.)

$$C = \frac{p_1 d}{2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (80)$$

The safe stress which the shell will sustain depends on the proportion of the steel hooping, and is given in table at foot of page 463. A few trials with varying thicknesses and proportions, or reinforcements may be required. The minimum practical thickness of shell is that necessary to give complete protection to the steel rods, and to permit of safe transport and handling.

Longitudinal reinforcement is provided under the same conditions as in the last case.

STORAGE BINS OR SILOS.

SYMBOLS USED IN THE FOLLOWING FORMULÆ.

h = Depth of bin in feet, or height of material above any point.
b = Breadth of rectangular bin, in feet.
l = Length of rectangular bin, in feet.
d = Diameter of circular bin, in feet.
p = Pressure against sides of bin in lbs. per lin. foot of horizontal perimeter.
w = Weight per cubic foot of material in bin.
μ = Coefficient of friction within the material stored.
μ_1 = Coefficient of friction between material and sides of bin.
(For values of the last three items, see table below.)

Bins can be conveniently divided into two classes, as follows :—

First.—Shallow Bins, where the plane of rupture (or the plane along which the material would tend to slide if the sides yielded) passes out through the material itself before it meets the opposite side of the bin, and

Second.—Deep Bins, where the plane of rupture meets the opposite side of the bin within the mass of the contained material.

The particular depth forming the dividing point between the above two classes is obtained by the following formula.

$$h = b \times \left[\mu + \sqrt{\mu \frac{1+\mu^2}{\mu+\mu_1}}\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (81)$$

All depths less than the above value would indicate shallow bins, all greater depths would be classed as deep bins.

For square bins :—

$$\textit{First Class.}—p = \tfrac{1}{2} w h^2 \left[\frac{1}{\sqrt{\mu(\mu+\mu_1)} + \sqrt{1+\mu^2}}\right]^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (82)$$

$$\textit{Second Class.}—p = \tfrac{1}{2} w b^2 \left[\frac{\sqrt{\frac{2h}{b}(\mu+\mu_1) + (1-\mu\mu_1)} - \sqrt{1+\mu^2}}{\mu+\mu_1}\right]^2 \quad . \quad (83)$$

The total pressure on bottom of bin =

(Total weight of material) $-$ ($\mu_1 \times p \times$ perimeter of bin in feet) . . (84)

The maximum pressure on the bottom of a bin occurs when

$$h = \frac{b}{2}\left[\frac{1+\mu^2}{\mu+\mu_1}\left(\frac{4\mu_1}{3\mu_1-\mu}\right)^2 - \frac{1-\mu\mu_1}{\mu+\mu_1}\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (85)$$

For Rectangular Bins the values given by formulæ (82) and (83) will be different for the sides and the ends, since in the latter case the value l would replace that of b.

For Circular Bins substitute d for b in formulæ (81), (83), and (85).

These formulæ are not applicable to the inclined sides of Hopper bins, nor to the conical or spherical bottoms of shallow bins.

Nature of Grain, etc.	Weight per Cubic Foot in Pounds.	Coefficients of Friction.	
		Grain on Grain.	Grain on Cement.
Wheat	53	0·466	0·444
Barley	43	0·509	0·452
Oats	32	0·532	0·466
Maize	48	0·521	0·423
Beans	50	0·613	0·442
Peas	54	0·477	0·296
Tares	53	0·554	0·394
Linseed	45	0·456	0·414
Cement	100	0·554	0·316
Sand	90	0·674	0·577
Bituminous Coal	50	0·700	0·700
Anthracite Coal (small)	52	0·510	0·510
Ashes	40	0·839	0·839
Coke	28	0·839	0·839

DOMES AND CONICAL COVERINGS.

The following graphic method of determining the stresses in domes is only correct for thin coverings of true spherical or conical section, and formed of a material of uniform weight per unit of surface, and capable of resisting both tension and compression. The majority of reinforced concrete domes approximate to these conditions.

For a detailed explanation of the following statements and formulæ, &c., a reference may be made to a Paper by Mr. W. Dunn, Assoc.M.Inst.C.E., F.R.I.B.A., published in the 'Transactions of the Royal Institute of British Architects,' May 21, 1904.

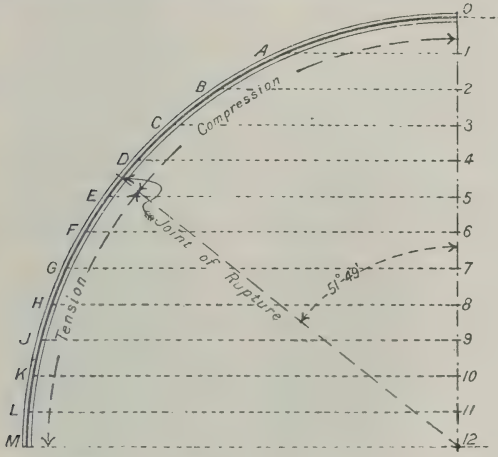


FIG. 11.

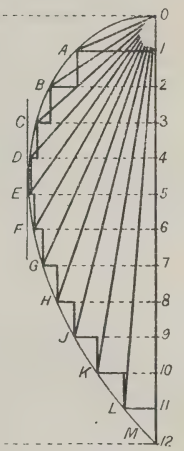


FIG. 12.

To construct Figs. 11 and 12. Divide centre line of dome (fig. 11) into any convenient number of parts, and draw the horizontal lines to shell, as 1A, 2B, &c. Draw load line 0–12 (fig. 12) and from O draw radial lines parallel to tangents on the curve at points A, B, &c., till they meet the corresponding horizontal lines; from these meeting points verticals are drawn to the adjacent horizontal lines to complete the zigzag lines shown. If correctly set out, the outer meeting points will lie on a regular curve as shown.

Select a scale such that the length 0–12 corresponds with the total weight of the dome; then the several heights between any two horizontal lines can be scaled off direct to give the weight of the portion between these two lines. To same scale, the radial lengths from O (fig. 12) to any point A, B, C, &c., indicate the total pressure on any horizontal section; the horizontal lengths A1, B2, &c., to same scale, give the total horizontal or radial thrust; the shorter full lines drawn from A, B, &c., give the variation in the radial thrust between the adjoining section lines.

To determine the hoop stress, or that acting at the vertical face of any section, the radial trust must be divided by $6 \cdot 28 \left(= \frac{\text{Radius}}{\text{Circumference}} \right)$, or by choosing a scale such that the height $0\text{–}12 = \frac{1}{6 \cdot 28} \times$ total weight of dome, the horizontal lengths to this scale will give the radial thrust direct.

The pressure per lineal unit on any horizontal section is equal to the length of the radial line from O to the particular section measured by the modified scale just mentioned, divided by the radius length at this section.

The position of the joint of rupture is at an angle of about $51\frac{1}{2}°$ with the vertical, or where the tangent to the curve in fig. 12 is parallel to the loadline 0–12; this is also the point of no hoop stress.

The addition of a lantern, or other load, at apex of dome will raise the point O in load line, and also the joint of rupture; the formation of a central opening will depress the point O to level of edge of opening, and lower the joint of rupture.

This graphic method will similarly apply to conical coverings.

ARCHES.

The designing of concrete arched bridges as a rule should be undertaken only by experts, as the accurate determination of the stresses, and the economical arrangement of concrete and steel to resist those stresses, calls for a large amount of skill and experience. The following diagrams and brief explanations will serve to indicate the manner in which such arches can be designed, or the strength of existing arches calculated.

Arched bridges may be divided into three general types :—

1. That in which a continuous arch is provided, without joints or hinges, and connected rigidly to the abutments.

2. Where the arch is hinged or jointed at each abutment, but is continuous between these two points.

3. Where the arch is hinged or jointed not only at the abutments, but also at the centre.

The existence of a hinge or joint implies that any stress in the arch ring must pass through the centre of the joint, and it is necessary to keep this in view when drawing the equilibrium polygon, and to arrange that this polygon passes through the centre of all joints.

Fig. 13 indicates an arched bridge of the first type, viz., that in which no joints occur, and consequently the position of the stresses in the ring is not restricted, as it would be by the introduction of joints or hinges. The complete width of the bridge may be considered as divided up into a series of strips, each say 12 inches wide, and attention given to one strip only. In the event of any irregularity in the loading of one strip more than the other, as for instance the outer edge of the arch which carries the parapet wall, such a strip would be considered independently. The complete span would be divided into a number of sections as indicated by dotted lines, and the weight or load on each section carefully calculated ; these weights can be taken as acting each through the centre of gravity of the section.

Assuming in the first place that the total dead load is practically uniformly distributed over the length of the bridge, so that the weight or load at each section is equal, the shorter line of loads A, B, . . . K, to the right of the illustration can be set up, and a polar point, or pole, V, chosen. Since the load is uniformly distributed, the vertical reaction at each abutment will be equal to one half the total load, and as the closing line of the equilibrium polygon must be in the same line as that joining the two joints in the abutments where it is desired that the resultant pressures should occur, and in this case these two points are on the same horizontal line, a horizontal line would be drawn from the centre of the line of loads and the point V chosen on this line, at such a distance from the lead line as to limit the height or rise of the equilibrium polygon drawn on the arch ring, so that it may coincide, as nearly as possible, with the centre line of the arch. The approach of this polar point to the load line tends to raise the centre of the equilibrium polygon, while a movement in the opposite direction will depress the centre.

If the loading is not uniform, as for example where the live load covers half the length of the bridge, the load line would take the form somewhat as that indicated by the longer line A, B . . . K, and to determine in the first place the vertical reaction at each abutment, a focal point W is chosen at random, and an equilibrium polygon drawn, as shown below the arch ; the closing line of this latter, marked *a*, gives the direction in which a line must be drawn from the point W in the polar diagram, so that its intersection with the load line will divide this latter into two portions, the vertical height, A O, giving to scale the vertical reaction at the left abutment, or the A end of arch, while the lower length, O K, gives the corresponding value for the opposite abutment. Having found this point O, a horizontal line is drawn from O on which the new focal point X must lie, to ensure the starting and finishing point of polar diagram occurring at the required points on the abutments. The position of X must be chosen, by one or two trials if necessary, so that the equilibrium polygon line, as before, is as near as possible to the centre line of the arch ring.

If the ring is formed of stone or brick, and no tension can be resisted, it is essential that all equilibrium polygons, for any possible combination of loads, should lie within the middle third of the depth of the arch ring ; if a line of any of the polygons extends beyond this limit, tension stresses are likely to develop.

The length of the outer rays or lines on the polar diagram will give the values of the total resultant pressures at each abutment, and assuming that the abutments are of sufficient strength and stability to give this required resistance, the values calculated for the stresses in the arch ring by this method are correct. Should, however, the abutments tend to yield, the amount of horizontal reaction is limited in value, and the distance of the point X from the load line must not indicate a horizontal stress greater than the abutments can offer ; the equilibrium polygon drawn under these conditions would in all probability pass above the centre of the arch ring, and thereby indicate that excessive bending stresses arise in the ring, tending to cause failure or collapse through the arch sinking downwards. If, by any means the abutments are forced together, or if the ring of the arch tends to increase in length, as by a rise of temperature, the polar point would tend to move further from the load line, and the equilibrium polygon would tend to fall below the centre of the arch ring, indicating the tendency of the ring to be forced upwards, heavy stresses being again produced.

In the case of types Nos. 2 and 3 the positions of the centre of the hinges or joints limit the position at these points of the equilibrium polygon, and in Fig. 14 a diagram is shown of a

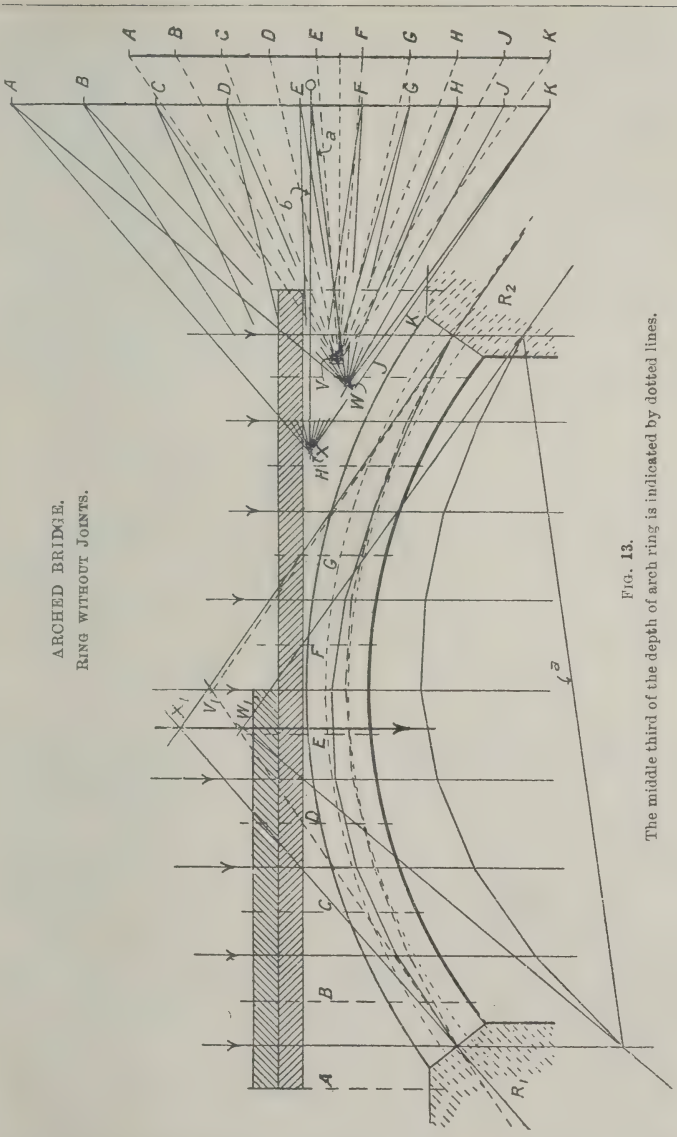
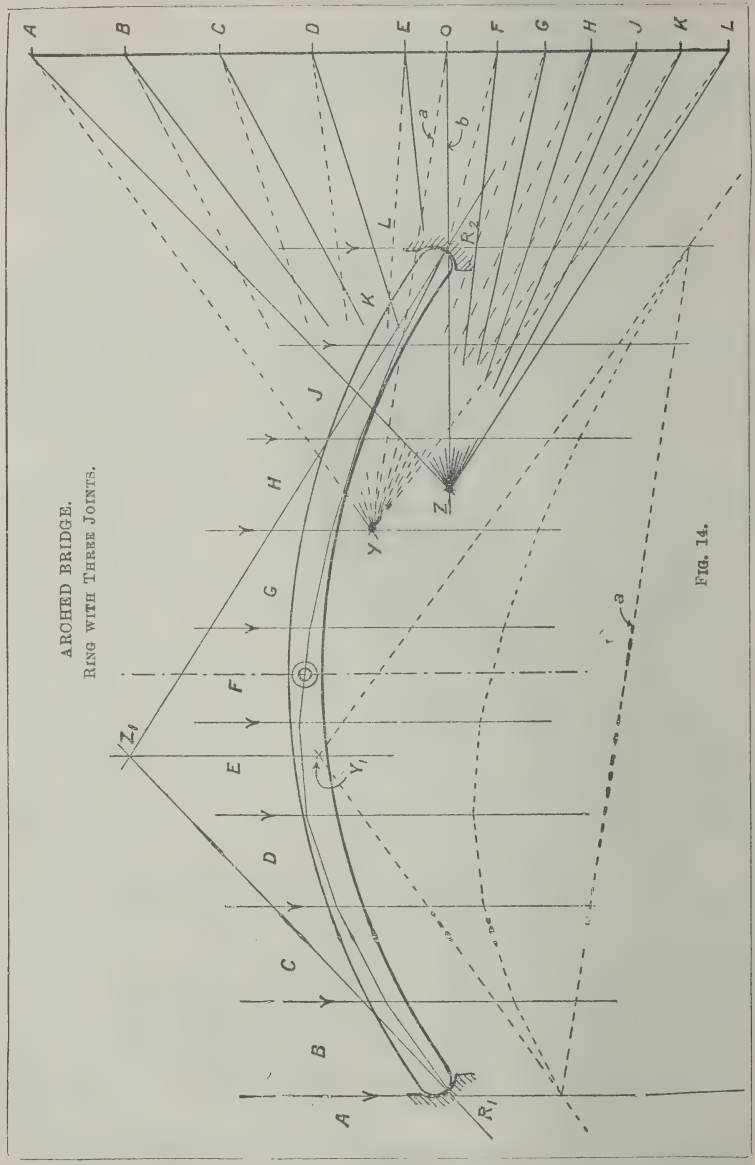


Fig. 13.

The middle third of the depth of arch ring is indicated by dotted lines.



ARCHED BRIDGE.
RING WITH THREE JOINTS.

FIG. 14.

three-hinge arch sustaining an unequally distributed load. After drawing the load line A, B. . . . L, a polar point Y is chosen, and the equilibrium polygon, shown in dotted lines, is completed, the closing line *a* determining the point O in the line of loads which divides this line into two sections, giving the vertical loads on the two abutments. A line from this point O drawn parallel to a line joining the centres of the hinges at the abutments, will give the line *b*, and on this line a point Z is chosen, such that the equilibrium polygon shown in full lines on diagram will pass through the centre hinge. If this centre hinge does not exist, as in type No. 2, the same process would be gone through, but here the aim would be to keep the equilibrium line as near the centre of the depth of the arch ring as possible.

The value of hinges or joints in the arch ring is that definite points are determined at the abutments through which the total thrust must pass, as also does a hinge in the centre of the rib; another marked advantage of a centre hinge is that a slight yielding of the abutment would imply only a drop in the centre of the ring, but would not induce the excessive bending stress such as would arise in the case of a ring without a centre hinge. Again, a slight lengthening in the arch due to a rise of temperature would not produce excessive stress, owing to the fact that the arch would be enabled to rise more or less freely in the centre.

In addition to drawing equilibrium polygons for the total dead load only, it is advisable in the majority of cases to draw similar polygons for live loads extending over say $\frac{1}{3}$, $\frac{1}{2}$, and $\frac{2}{3}$ of the complete span, and also for the live load confined to the centre of the bridge, and extending for one-fourth of the span on each side of the centre. No other arrangement of the live load is likely to give higher maximum stresses than got from these positions.

The next step necessary is to determine the actual stresses in the arch ring itself, and this may be done as follows:—

Figs. 15 and 16 indicate to a larger scale a portion of the arch shown in fig. 14 about

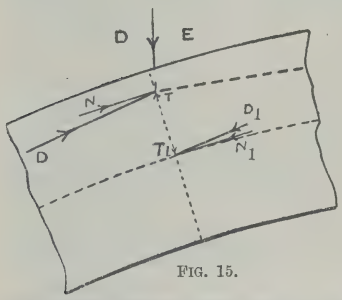


FIG. 15.

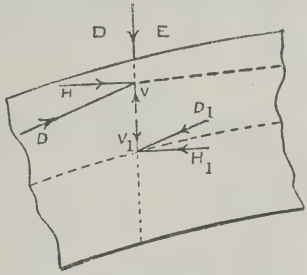


FIG. 16.

the point of application of the load D E. Taking in the first instance fig. 15, the value of the force or thrust, represented by the equilibrium polygon line 'D,' can be obtained by scaling the length of the line Z D in the polar diagram; the effect of this eccentric load on the cross-section of the rib is equal to that of a single load of this value acting at the centre of gravity of the section (and producing a uniformly distributed compressive stress over the whole section), along with a couple value equal to the load or thrust, multiplied by its normal distance from the neutral axis of rib; this couple produces a bending tendency, and gives rise to tensile stress at the distant edge and compressive stress at the near edge of the section. The combined result is an increase of compressive stress at the near edge, and a reduction at the far edge. As previously mentioned, should the force act without the middle third of the depth of the ring, the tensile stress due to the couple will be in excess of the uniformly distributed compressive stress, and tension at the distant edge will arise.

Considering the bending moment effect of the couple referred to above, D, D_1, fig. 15, indicate the two forces forming the couple; these are of value equal to the thrust indicated by the equilibrium polygon, the one acting along the polygon line, and the other at the neutral axis of the arch ring. Each of these forces may be resolved into two forces without altering the effect on the rib; thus D may be resolved into a force N acting normal to the radius line (shown by a dotted line), and a small force T acting in the direction of the radius line. Similarly D, can be resolved into two forces N, and T_1, and as T and T, are of equal value, but act in opposite directions, these will neutralise or balance one another, with the result that the two forces N, N_1, acting normal to the radius line, will produce the same effect as the two original forces D, D_1. It may be pointed out that the normal force N alone serves to produce direct stress in the arch rib, the smaller resolved force T producing shearing stress; this shearing stress, however, is always very small, and may frequently be neglected.

In calculating the bending moment by means of those resolved forces it is necessary to determine the value of N for each point under consideration; these calculations, however, can be simplified by modifying this treatment, as indicated in fig. 16. Since the effect of the force D

is not altered by resolving it into any two forces, a vertical line may be chosen, and D resolved into force H acting horizontally, and V acting vertically; D_1 would be similarly resolved into H_1 and V_1. A glance at the polar diagram will show that at any point in the equilibrium polygon the horizontal component of the thrust is constant, and can be determined by scaling the length Z O; from this it will be evident that the values H, H_1 in fig. 16 will be constant for any section of the arch, and since V, V_1 neutralise one another, the constant forces H, H_1 remain. For any section of the ring, therefore, it is only necessary to carefully measure or calculate the vertical distance between polygon line and centre of gravity of the arch ring, and to multiply this by the constant horizontal thrust H, to determine the bending moment value at the point under consideration; the addition of the uniformly distributed load value will give the complete stress throughout this section.

Thus, if the thrust H, fig. 16, is 40,000 lbs., the vertical distance between H and the neutral axis 8 ins., and the dimensions of the arch ring under consideration are 20 ins. deep × 12 ins. wide; the uniformly distributed compressive stress throughout the section will be $\frac{40,000}{20\times12}=167$ lbs. per sq. ft.; the B M $=40,000\times8=320,000$ inch lbs., and the maximum compression and tension stresses at upper and lower edges of the section $=\frac{320,000\times10\times12}{12\times20^3}=400$ lbs. This gives the total maximum compression at the upper edge $=167+400=567$ lbs. per sq. in., and the total tension at lower edge $=400-167=233$ lbs. per sq. in., necessitating the addition of steel reinforcement at the tension portion.

Temperature stresses due to variations in the temperature of the arch may be considerable, and should be carefully examined; a rise of temperature (by causing expansion) tends to increase the compression stress, while a fall has the opposite effect. The result is very similar to that of increasing or reducing the horizontal thrust (or the distance Z O in the polar diagram) throughout the arch. Owing to the mass of the materials, and its non-conducting nature, the range of temperature in concrete arches will probably not exceed 15° to 20° Fahr. between summer and winter seasons.

Specification for Reinforced Concrete Work.

The following sketch specification may be of assistance. It can be modified to any required extent to meet special conditions.

(Date) ———— Specification for Reinforced Concrete Work to be executed in erecting New Premises at ———— for Messrs. ————

Engineer. Architect.

General Description and Conditions.—The Structure will be erected in strict accordance with the Engineer's Drawings Nos. ..... and ..... and to any further Detail Drawings to be afterwards provided.

The Structure will consist of (*give general description*). The Foundations shall be taken out to (*the clay, ballast?*). The Maximum Load at this point must not exceed ..... tons per square foot. The average depth of foundations may be assumed as ..... below datum level.

The whole of the work included in this Contract shall be constructed of the best materials and workmanship, and shall be carried out to the entire approval of the Consulting Engineer, and be completed to his satisfaction.

No variations or alterations from Specification or Drawings shall be permitted unless same are approved in writing by the Consulting Engineer.

Immediately on acceptance of his tender, the Contractor must provide a complete and detailed Bill of Quantities (priced in ink) for this work. Should any variations or alterations occur during the carrying out of the work, the variations shall be measured up and priced according to the Bill of Quantities, and any necessary additions or deductions thereafter made from the quoted lump sum price. Any materials or labour required during the progress of the work which are not indicated in the Bill of Quantities shall be paid for at Rates or Prices to be agreed with the Engineer.

The Contractor will be responsible for meeting all requirements of the London County Council, or other Local Authorities, with regard to the mode of construction or the carrying out of same, and shall at once notify the Consulting Engineer should such requirements necessitate any modification in his designs or details.

The Contractor will be held responsible for setting out the Works, and for amending any errors due to inaccuracy, or other cause, at his own cost.

The Contractor must provide for these Works all labour, tools, tackle, boards, horsing, centering, scaffolding, cartage, water, etc.

The Contractor must provide and maintain proper sheds, latrines, and other accommodation for the use of workmen, and proper sheds and shelters for the protection of materials.

The Contractor will be held responsible for all losses, accidents, and damages that may happen to the Works, adjoining properties, or persons during the progress, and caused by the execution of the Works.

The Contractor will be bound to complete all the Works contained in this Specification, etc., within ..... months of the date of signing the Contract or obtaining possession of site, under the penalty of ..... per week or part of a week as liquidated and agreed damages, except the delay in the opinion of the Engineer arises from causes beyond the control of the Contractors, in which case the Engineer shall grant an extension of time in writing.

The Contractor must allow for making good at his own expense any defect or defects which may become noticeable in the Works during a period of twelve months after certified completion.

Samples.—Samples of all materials must be submitted to the Consulting Engineer, and approved by him in writing before same are used. All materials rejected must be removed immediately from the vicinity of the Works.

Cement.—The Cement used must be the best quality of British Portland Cement, of an approved make, and guaranteed to conform to the Specification of the British Standards Committee for slow-setting cement. Before being used all Cement must be tested, as required by the Engineer, at the Contractor's expense. The Cement should be delivered to the Site in sealed bags bearing the maker's name and the weight of the cement contained.

Cement shall be stored on the Site in such a manner as to permit of easy access for proper inspection and identification. The building used for same must be raised from the ground, and properly weatherproofed to protect the cement.

Sand.—The Sand used shall be clean, sharp, and coarse, composed of grains of various sizes up to particles which shall pass a $\frac{1}{4}$-inch square mesh, but of which at least 75 per cent. shall pass a $\frac{1}{8}$-inch square mesh. It shall be free from all ligneous, organic, or earthy matter.

Aggregate.—This must be composed of Thames ballast; screened, broken stone of a hard close-grained quality; or gravel, cleaned free from organic matter, and angular, varied in size as much as possible, and to pass through a $\frac{3}{4}$-inch ring, but not through a $\frac{1}{4}$-inch ring.

Proportions of Concrete.—(*Here specify the composition of concrete, which for usual requirements, may be* 1 *part cement*, 2 *sand*, 4 *aggregate. Foundations are frequently made of a weaker mixture, say* 1 : 3 : 6.) These proportions are approximately correct, but in all cases sufficient sand must be added to fill the interstices. The proportioning should be done in an approved manner, as by means of a bottomless box, to ensure each of the materials being accurately measured.

Mixing of Concrete.—The concrete must be thoroughly well mixed in a suitable machine, sufficient clean water being added to convert the whole into a semi-liquid mass, which will quiver under the blows of the rammer. The water for each batch of concrete must be measured in a small tank, so as to ensure the same amount being used each time. When consent is given to mix any of the concrete by hand, it shall be done as follows:—The necessary quantity of the three materials—coarse aggregate, sand, and cement—shall be measured separately, 5 per cent. more cement being added in addition to the above specified amount. The materials should first be thoroughly mixed dry, then the necessary amount of water added, and the mixing continued till the concrete shows a uniform colour. As far as possible, the mixture shall always be brought to the same degree of wetness. A competent foreman must be in constant attendance at the mixer, to give his approval of every batch which leaves the machine.

Allowable Working Stresses.—(*Here specify maximum stresses desired, or state requirements to be those embodied in Report of R.I.B.A. Joint Committee.*)

Steel Reinforcement.—All steel used should have a tensile strength of not less than 60,000 lbs. per square inch of section, and an elastic limit of not less than 50 per cent., nor more than 60 per cent. of the ultimate tensile strength, with an elongation of not less than 20 per cent. in a length of 8 ins. Any bar must stand bending cold round a bar of its own diameter, through an angle of 180°, and should close down upon itself without fracture on the outside of the bent portion. All welding will be prohibited, unless where absolutely necessary, and must be done only with the sanction of the Consulting Engineer.

All bars shall be free from scale, flaws, or other imperfections, and from oil, but a slight coating of rust is not objectionable. Wherever cinder concrete is used, the bars before being placed in position should receive a coating of neat cement grouting.

Any temporarily protruding bars which shall be exposed to the weather for any period must be protected from rusting by a thin coat of cement grout.

Centering.—All centering must be of dressed timber with close joints, and must be true to line, and sufficiently braced and strutted to be rigid and free from measurable deflection when carrying the dead load of the concrete which is to be placed upon it. A slight camber of $\frac{1}{4}$ in. per 10 ft. of span should be given to all beams, &c. It must also be without appreciable deflection when men are working upon it, and barrows, &c., are being wheeled over it. The joints must be tight, so as to prevent leakage of the liquid cement. The centering shall be arranged so that the sides of the columns can first be removed, then that at the side of beams, and below floor slabs, and finally that below the beams themselves. The time of removal of these several portions shall be arranged with the Consulting Engineer.

Placing of Reinforcement.—All bars must be accurately placed in the exact position shown on the detail drawings, and great care must be taken that they are not displaced during the process of packing the concrete around them. Where, for unavoidable reasons, it is necessary to stop the concreting work, care must be taken to place in position all the bars which will be embedded in the portion of the concrete work which is being executed.

Placing of Concrete.—Before any concrete is put in position a careful inspection of the centering must be made, to ascertain that no dirt, shavings, loose stones, etc., have been allowed to remain in or about the centering, and the woodwork should be well watered to avoid any danger of its sucking up too much moisture from the concrete.

All concrete shall be placed in its final position in the work as soon as possible after mixing. In no case must more than half-an-hour be allowed to elapse before this is done. The concrete must be placed gently into position, in layers not exceeding 6 inches thick, and must not be tipped or dropped from a height. It must then be thoroughly rammed into position, great care being taken to see that the steel reinforcement is thoroughly surrounded by the liquid cement, and that no voids or cavities are left. This can be ensured only by repeated ramming with a suitable tool.

In hot weather the concrete shall be continually moistened for at least a week after placing in position.

In beams and slabs, if any difficulty is found in getting the concrete to pass through the bars which have been placed in position in the bottom of the timber centering, the concrete at the bottom of the beam should be made of finer and more liquid consistency than that used elsewhere, so that it may freely pass between the bars.

Cessation and Recommencement of Work.—Stoppage of work in laying concrete in slabs, beams, etc., will be allowed only at the centre of the span of the slab, or a beam, and the joint with the new concrete should be a vertical plane at right angles to the direction of the span of beam or slab. No joint must be made towards the ends of beams or slabs, or close to a concentrated load.

Wherever a stop has been made in concreting, before recommencing the work, the surface of the existing concrete must be carefully washed with a stiff brush to remove loose particles and dust, and a thick grout of neat cement must be poured over it before the concrete is rammed against it.

Frosty Weather.—No concreting work should be done when the temperature is near or below freezing point. If a frost comes on shortly after any concreting work has been completed, this should be very carefully covered over with sawdust, straw, or sacking, and the centering must be allowed to remain up for at least as long a time after the complete disappearance of the frost as it would have been left in position if no frost had occurred.

Finish.—All concrete roofs and gutters must be laid with the necessary fall, and finished for asphalte, all as may be directed.

All floors shall be finished. (*Here state whether cement, granolithic, preparation for wood blocking, or other covering is desired.*)

The top surface of all outside walls to be weathered.

Tests.—The Maker's Certificate must be provided, when required, with each lot of steel reinforcing bars supplied, but in addition to this the Consulting Engineer shall choose whatever test pieces he may desire from the material delivered, and shall have same tested by Messrs. Kirkaldy, or other firm, and should the result not comply with Specification the right is reserved to condemn all bars included in same consignment.

The cement shall be tested as required, and must comply with the Specification. Cubes of concrete from the material used shall be made as and when directed, and these shall be tested for crushing strength by Messrs. Kirkaldy, or other firm.

The Consulting Engineer shall select at least one bay in each floor, or the equivalent, and these bays shall be tested to $1\frac{1}{2}$ times the safe superimposed load, by means of bricks, cement bags, or in such other manner as the Engineer may decide. Should the deflection under this test exceed $\frac{1}{360}$th part of the span, or should the beam or floor slab fail to regain its normal position after the removal of the load, or show any indication whatever of weakness, the building shall be considered unsafe, and the contractor shall carry out at his own expense such additional work as the Engineer may consider necessary to make the building secure.

The cost of all such testing shall be borne by the Contractor.

Practical Notes.

Wood Strips.

Particularly long lengths of wood for nailing strips in concrete beams should be avoided; they have been known to absorb the moisture from the concrete, and cause serious cracks in the beams. Short plugs are the safest, and where a nailing strip must be used flush with beams, a moulded impress of the wood strip should be made and the strip then removed, and only fixed when the concrete beam is thoroughly dry, it being then secured by means of the plugs. (*Horobin.*)

Durability.

Reinforced concrete will last as long as plain concrete in any situation provided that certain special precautions are taken during its construction. The precautions to be taken are as follows: The materials (cement, sand, and stone) must be of good quality. They must be most carefully and thoroughly mixed and scientifically proportioned, so as to be practically water-proof and air-proof. The mixture must be fairly wet, and must be well punned into position so as to minimise voids. The aggregate should be as non-porous as possible, and any aggregate which

is known to have a chemical action on steel should be avoided. The aggregate should all pass through a $\frac{3}{4}$ in. mesh. The concrete covering should in no case be less than $\frac{1}{4}$ in., and it is suggested that if round or square bars be used the covering should not be less than the diameter of the bar. In structures exposed to the action of water or damp air the thickness of covering should be increased at least 50 per cent., or the size of the aggregate should be reduced so as to ensure a dense skin. In the case of structures exposed to very severe conditions the concrete might be covered with some impervious coating as an extra precaution. The reinforcement should be so arranged that there shall be sufficient space between one piece and its neighbour to allow the concrete to pass and to completely surround every part of the steel. All steel should be firmly supported during the ramming of the concrete, so as to avoid displacement. It should not be oiled or painted, and thick rust should be scraped and brushed off before placing. The scantling of the various members of the structure should be sufficient to prevent excessive deflection. If electric mains are laid down very great care must be taken that no current is allowed to pass through the reinforced concrete. Fresh water should be used in mixing, and aggregates charged with salt should be washed. These recommendations have regard only to the prevention of corrosion of steel and not to fire-resistance or any other property of reinforced concrete. (*Committee of the Concrete Institute.*)

Causes of Failures in Concrete Structures.

Owing to the failure of several reinforced concrete floors in the United States within ten or a dozen years of their construction, Prof. H. J. M. Creighton, of Swarthmore College, examined a large number of reinforced concrete structures in which cracks were developing, and found that in every case the cracks ran along the reinforcing rods, due to the deteriorating action of salt and brine on the concrete. Solutions of the chlorides reacted with the lime and the silicates in the concrete, and penetrating to the iron of the reinforcement converted it into oxide and hydrate, which occupy more space than the metal and force the concrete apart. It was therefore necessary to waterproof reinforced concrete structures which will be in contact with brine, to cease to use in the concrete beach gravel which has not been thoroughly washed with fresh water, and never to add salt to the concrete to prevent it freezing during building operations in cold weather.

Cracking.

According to a recent report, the cracking of reinforced concrete structures is markedly prevalent in the Philippine Islands. A study of this trouble has shown that not a single structure showing rusted steel has been free from salt, the percentage of which varies considerably. In view of this, engineers in the Philippines have been advised that not only is the use of salt water dangerous in concrete structures, but that beach sand and beach gravel should be employed only after having been thoroughly washed with fresh water. The foregoing, and many other similar pieces of evidence, indicate that salt and brine exert a deteriorating action on concrete.

Specification Notes.

Specifications governing concrete construction generally require that 1–2–4 concrete be used in the best class of reinforced concrete construction. In other reinforced concrete work, and in some mass concrete work, 1–3–5 concrete is specified, while in other mass work, such as dams, gravity retaining walls, etc., 1–3–6 concrete is generally required.

TABLE GIVING THE SECTIONAL AREAS OF **Round and Square Rods** AT VARIOUS DISTANCES APART, IN 12-INCH WIDTH OF SLAB.

Diameter in Ins.	Circumference in Ins.	Area in Sq. Ins.	Distance in Inches from Centre to Centre of **Round** Rods.										
			$2\frac{1}{2}$	3	4	5	6	7	8	9	10	11	12
$\frac{1}{4}$	0·785	0·049	0·24	0·20	0·15	0·12	0·10	0·08	0·07	0·06	0·06	0·05	0·05
$\frac{5}{16}$	0·982	0·077	0·37	0·31	0·23	0·18	0·15	0·13	0·11	0·10	0·09	0·08	0·08
$\frac{3}{8}$	1·178	0·110	0·53	0·44	0·33	0·26	0·22	0·19	0·16	0·15	0·13	0·12	0·11
$\frac{7}{16}$	1·374	0·150	0·72	0·60	0·45	0·36	0·30	0·25	0·22	0·20	0·18	0·16	0·15
$\frac{1}{2}$	1·571	0·196	0·94	0·78	0·59	0·47	0·39	0·33	0·29	0·26	0·23	0·21	0·20
$\frac{9}{16}$	1·767	0·248	1·19	0·99	0·74	0·59	0·50	0·42	0·37	0·33	0·30	0·27	0·25
$\frac{5}{8}$	1·964	0·307	1·47	1·23	0·92	0·74	0·61	0·52	0·46	0·41	0·37	0·33	0·31
$\frac{11}{16}$	2·160	0·371	1·78	1·48	1·11	0·89	0·74	0·63	0·56	0·49	0·44	0·40	0·37
$\frac{3}{4}$	2·356	0·442	2·12	1·77	1·32	1·06	0·88	0·75	0·66	0·59	0·53	0·48	0·44
$\frac{13}{16}$	2·552	0·518	2·49	2·07	1·55	1·24	1·04	0·88	0·78	0·69	0·62	0·56	0·52

TABLE GIVING THE SECTIONAL AREAS OF **Round and Square Rods** AT VARIOUS DISTANCES APART, IN 12-INCH WIDTH OF SLAB—*continued.*

Diameter in Ins.	Circumference in Ins.	Area in Sq. Ins.	Distance in Inches from Centre to Centre of **Round** Rods—*continued.*										
			2½	3	4	5	6	7	8	9	10	11	12
⅞	2·743	0·601	2·89	2·40	1·80	1·44	1·20	1·02	0·90	0·80	0·72	0·65	0·60
15/16	2·945	0·690	3·31	2·76	2·07	1·66	1·38	1·17	1·03	0·92	0·83	0·75	0·69
1	3·142	0·785	3·77	3·14	2·36	1·88	1·57	1·33	1·18	1·05	0·94	0·86	0·78
1 1/16	3·338	0·887	—	3·55	2·66	2·13	1·77	1·51	1·33	1·18	1·06	0·97	0·89
1⅛	3·534	0·994	—	3·98	2·97	2·39	1·99	1·69	1·49	1·32	1·19	1·08	0·99
1 3/16	3·731	1·107	—	—	3·32	2·66	2·21	1·88	1·66	1·48	1·33	1·21	1·11
1¼	3·927	1·227	—	—	3·68	2·94	2·45	2·09	1·84	1·64	1·47	1·34	1·23
1 5/16	4·123	1·353	—	—	—	3·25	2·71	2·30	2·03	1·80	1·62	1·47	1·35
1⅜	4·320	1·485	—	—	—	3·56	2·97	2·52	2·23	1·98	1·78	1·62	1·48
1 7/16	4·516	1·623	—	—	—	—	3·25	2·76	2·43	2·16	1·95	1·77	1·62
1½	4·712	1·767	—	—	—	—	3·53	3·00	2·65	2·36	2·12	1·93	1·77

Side in Ins.	Perimeter in Ins.	Area in Sq. Ins.	Distance in Inches from Centre to Centre of **Square** Rods.										
			2½	3	4	5	6	7	8	9	10	11	12
¼	1·00	0·063	0·30	0·25	0·19	0·15	0·13	0·11	0·09	0·08	0·07	0·07	0·06
5/16	1·25	0·098	0·47	0·39	0·29	0·23	0·20	0·17	0·15	0·13	0·12	0·11	0·10
⅜	1·50	0·141	0·68	0·56	0·42	0·34	0·28	0·24	0·21	0·19	0·17	0·15	0·14
7/16	1·75	0·192	0·92	0·77	0·58	0·46	0·38	0·33	0·29	0·26	0·23	0·21	0·19
½	2·00	0·250	1·20	1·00	0·75	0·60	0·50	0·43	0·37	0·33	0·30	0·27	0·25
9/16	2·25	0·317	1·52	1·27	0·95	0·76	0·63	0·54	0·47	0·42	0·38	0·34	0·32
⅝	2·50	0·392	1·88	1·57	1·77	0·94	0·78	0·67	0·59	0·52	0·47	0·43	0·39
11/16	2·75	0·474	2·27	1·90	1·42	1·14	0·95	0·81	0·71	0·63	0·57	0·52	0·47
¾	3·00	0·564	2·71	2·26	1·69	1·35	1·13	0·96	0·84	0·75	0·68	0·61	0·56
13/16	3·25	0·661	3·17	2·64	1·98	1·59	1·32	1·13	0·99	0·88	0·79	0·72	0·66
⅞	3·50	0·767	3·68	3·07	2·30	1·84	1·53	1·31	1·15	1·02	0·92	0·84	0·77
15/16	3·75	0·881	4·23	3·52	2·64	2·11	1·76	1·51	1·32	1·17	1·03	0·96	0·88
1	4·00	1·000	4·80	4·00	3·00	2·40	2·00	1·71	1·50	1·33	1·20	1·09	1·00
1⅛	4·50	1·266	6·08	5·06	3·80	3·04	2·53	2·16	1·90	1·69	1·52	1·38	1·27
1¼	5·00	1·575	7·56	6·30	4·72	3·78	3·15	2·69	2·36	2·10	1·89	1·72	1·57
1⅜	5·50	1·905	9·14	7·62	5·71	4·57	3·81	3·26	2·86	2·54	2·29	2·08	1·90
1½	6·00	2·253	10·81	9·01	6·76	5·41	4·51	3·85	3·38	3·00	2·70	2·45	2·25

REINFORCED CONCRETE FOR FACTORY WORK.

One of the objections to the employment of reinforced concrete for factory work has been the difficulty experienced in attaching shaft hangers, pipe lays, and other fixtures, because of the hardness of the material. *Huntley & Sparks, Ltd.*, De Burgh Road, S.W. 19, claim to have solved the problem by employing standard pressed steel fittings called RIGIFIX bolt-hanger sockets and slotted inserts. These are embedded in the beams and columns of the building during course of construction. The bolt-hanger sockets are placed at fixed points to receive a ⅝-in. or ¾-in. bolt. The inserts are supplied in 12 in., 18 in., and 36 in. lengths, the runway being provided with a slot to receive a ⅝-in. or ¾-in. bolt, which can be moved to any point on the length. Many modern factories have adopted these fitments.

SECTION XII

EARTHWORK – EXCAVATION – FOUNDATIONS – PILING – SCAFFOLDING – STAGINGS – SHORING – DAMS – RETAINING-WALLS – CHIMNEYS – STONE AND BRICK ARCHES – PIERS – BRIDGES

(pp. 481 to 526)

(With contributions by Reginald Ryves, Assoc.M.Inst.C.E.)

SECTION XII

EARTHWORK — EXCAVATION — FOUNDATIONS — PILING — SCAFFOLDING — STAGINGS — SHORING — DAMS — RETAINING-WALLS — CHIMNEYS — STONE AND BRICK ARCHES—PIERS—BRIDGES.

EARTHWORK.

To determine the Volume of Earth in a Railway Cutting.

First Method.

Divide the length of the cutting into an *even* number of equal parts, and take the areas of the transverse sections at the points of division. Add together the areas of the extreme sections, twice the sum of the areas of all the intermediate sections of an *odd* order, and four times the sum of the areas of all the intermediate sections of an *even* order, and multiply by the distance between two consecutive sections; the product will be three times the volume.

Second Method.

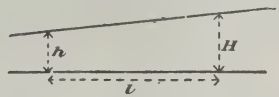


FIG. 1.

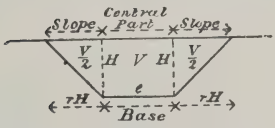


FIG. 2.

H, h=heights of section, in feet, at each end of a length, l, in feet (fig. 1).

rH=breadth of each slope at one end (fig. 2).

rh=breadth of each slope at other end.

(r=ratio of breadth of each slope to height=$\frac{\text{breadth}}{\text{height}}$).

e=base.

$\frac{A}{2}$=area of each slope at one end.

$\frac{a}{2}$=area of each slope at other end.

V=cubic contents of both slopes=$\frac{1}{6}(A+\sqrt{Aa}+a)\times l=\frac{1}{6}r(H^2+Hh+h^2)\times l$.

V=cubic contents of centre=$\frac{1}{2}(H+h)e\times l$.

Q=cubic contents of length, l, in feet=V+V.

Third Method.

If the transverse section is on an incline, instead of horizontal, as in figs. 3 and 4, then, if A, A′, a, and a' are the respective areas of the ends of the slopes,

$$V = \tfrac{1}{6}(A + \sqrt{Aa} + a) \times l + \tfrac{1}{6}(A' + \sqrt{A'a'} + a') \times l. \qquad V = \tfrac{1}{2}(H + h)e \times l.$$

The longitudinal section of the line of railway, road, or canal, having been drawn, and the gradients laid off, the quantities of earthwork may be calculated as follows :—

If the ground is tolerably even, record the heights (or depths in cuttings) on the section at every 100 feet, or some other convenient distance. The sectional areas at each of these recorded heights (or depths) may then be calculated.

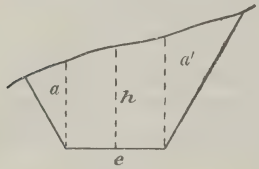


FIG. 3.

Let
A = area of section, $s = s$ to 1 slope,
h = height or depth of section,
F = width at formation-level,

all expressed in feet ; then, if the slopes on both sides are the same, as is usually the case,

$$A = h(sh + F).$$

By this formula the areas of the first three sections of a series increasing in height regularly are calculated, and from them the remainder are at once written down by the method of second differences, as follows :

Assume heights to be 1, 2, 3, 4, 5 feet,

$$F = 10, \quad s = 2 \text{ to } 1.$$

Then

	Difference.	Second Difference.
A for 1 ft. = 12		
„ 2 „ = 28	16	4
„ 3 „ = 48	20	
„ 4 „ = 72..	$\frac{4}{24}$	—
„ 5 „ = 100..	$\frac{4}{28}$	—

and so on.

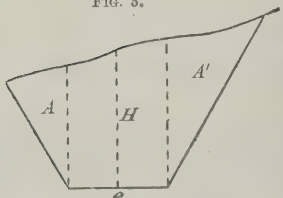


FIG. 4.

If the cross-slope of the ground is not much, and the areas of the sections do not differ very widely, the quantity of earthwork may be found by taking the mean of the two areas and multiplying by the distance between them, thus :

A′ = area of first section ; A″ = area of second section ; d = distance apart.

$$\text{Cubic contents} = d\,\frac{A' + A''}{2} \text{ feet.}$$

When the cross-slope is considerable, and the heights of the section differ widely, cross-sections must be taken at each place, and their areas calculated independently, and the cubic contents should then be calculated by the *First Method.*

Earthwork Tables.*

(*H. R. Kempe.*)

The tables on pages 485 to 490 are for the purpose of facilitating the calculation of the volumes of earthwork.

V = cubic contents, in *yards*, of both slopes.
V = cubic contents, in *yards*, of centre part.

V is calculated from the formula

$$V = \tfrac{1}{3}r\,(H^2 + Hh + h^2) \times l,$$

r being = 1, l = 1 chain = 66 feet, and H and h being in *feet* ; so that

$$V = \tfrac{1}{3}(H^2 + Hh + h^2)\,66 \div 27 = \cdot 814815\ (H^2 + Hh + h^2) \text{ cubic yards.}$$

V is calculated from the formula

$$V = \tfrac{1}{2}(H + h)e \times l,$$

e being = 1 foot ; so that

$$V = \tfrac{1}{2}(H + h)\,66 \div 27 = 1 \cdot 22222\ (H + h) \text{ cubic yards.}$$

For any slope other than 1 to 1 the values of V in Table I. must be multiplied by r, that is, by $\frac{\text{breadth of slope}}{\text{height of slope}}$. Table II. gives the values of V and V for slopes of 1 to 1½.

* Revised 1920.

EXAMPLE.—The length of a cutting is 3 chains, the width 10 feet, the $\frac{\text{breadth of slope}}{\text{height of slope}} = 2$, the height at one end 20 feet, and at the other end 15 feet; what is the cubic content of the cutting?

From table (page 486), V = 753·70, $\sqrt{}$ = 42·78: then,

total cubic content = (753·70 × 2 + 42·78 × 10)3 = 5805·60 cubic yards.

Excavation.

A good excavator will dig and throw into a barrow in a day of 10 hours:

In common ground, from 8 to 10 cubic yards. In stiff clay or firm gravel, 6 cubic yards. In hard ground, where picking is required, from 3 to 5 cubic yards.

One excavator to 5 or 6 feet breadth of face of a cutting is as near as they should be.

A shoveller will throw each shovelful of earth from 6 to 10 feet horizontally, or from 4 to 5 feet vertically upwards.

PROPORTION OF EXCAVATORS, SHOVELLERS, AND WHEELERS (100 to 120 ft.).

Material.	Excavators.	Shovellers.	Wheelers.
Loose sand and vegetable moulds	0	1	1
Compact earth	½	1	1
Ordinary clay	½ to 1	1	1
Hard clay	1½ to 2	1	1

The rate at which a cutting may be expected to advance will be, for each line of wheelers or for each shoveller, in one rank:

20 cubic yards of loose sand or mould a day. 16 cubic yards of clay or compact earth a day.

Excavating Machinery.

Dry land excavating machinery may be divided into two principal types, namely power shovels—sometimes known as power navvies—and dragline excavators, the essential difference being that the former is equipped with a rigid handle carrying a bucket, which is forced upwards through the soil by a positive control, the motion being, therefore, upwards and away from the machine, while the dragline excavator, on the contrary, has its bucket suspended from a wire rope at the end of a long jib, and drags this bucket through the earth towards the machine by means of another wire rope. Thus the shovel is principally used for excavation against a bank or face of material, while the dragline digs from below its own level, and is therefore much used on canal excavation, very frequently excavating from below water level.

Modern steam shovels and dragline excavators range in size from machines of some 20 tons, with ¼ cubic yard bucket, to immense excavators weighing up to 350 tons, and equipped with an 8 cubic yard bucket.

The larger types of shovel and dragline may be driven by steam or electricity, and the smaller types—up to about 1¾ cubic yard capacity bucket—by steam, electric, petrol, or Diesel engine.

Modern practice is to equip all the smaller and some of the larger types with caterpillar traction in place of the old railway or road wheels.

Particulars of several sizes of shovels and draglines are given below.

RUSTON EXCAVATORS.

(Ruston & Hornsby, Ltd., Lincoln.)

These excavators are divided into three groups or types:

(1) Crane Navvies. (2) Dragline Excavators. (3) Railroad type Shovels.

Items Nos. 1 and 2 are of the full circle type, and in the case of No. 3, the swing of the jib is restricted to approximately 180°.

CRANE NAVVIES.

Size of Machine.	Jib Centres.	Bucket Capacity.	Nett Weight.		Approximate Output per Day of 10 Working Hours.
			Rail Wheels.	Caterpillars.	
	Ft. Ins.	Cu. Yds.	Tons.	Tons.	Cu. Yds.
No. 4	16 9	⅛	11¾	14	300 to 400
No. 6	20 0	⅞	21	25	400 to 600
No. 10	29 0	1¼	36	46	800 to 900
No. 15	28 0	2¼	48	62	1,200 to 1,400
No. 20	31 0	2¾	61	82	1,500 to 1,700
No. 135	60 0	2¾	150	—	1,400 to 1,600
No. 300	80 0	6	320	—	3,000 to 4,000

DRAGLINES.

Size of Machine.	Jib Centres.	Bucket Capacity.	Nett Weight.		Approximate Output per Day of 10 Working Hours.
			Rail Wheels.	Caterpillars.	
	Ft. Ins.	Cu. Yds.	Tons.	Tons.	Cu. Yds.
No. 4 . .	22 0	½	11¾	14	200 to 300
No. 6 . .	30 0	¾	20½	25½	360 to 550
No. 10 . .	40 0	1¼	34½	44½	700 to 800
	60 0	2¼			
No. 75 .	70 0	1¾	75	93	600 to 1,100
	80 0	1½			
	70 0	4			
No. 135 .	80 0	3½	142	—	1,000 to 2,000
	90 0	3			
	100 0	2¾			
	90 0	6			
No. 180 .	100 0	5	195	—	1,100 to 3,000
	120 0	4			
	140 0	3			
No. 300 . .	100 to 160 ft.	10 to 5	316	—	1,700 to 4,000

The railroad shovels are made in two sizes, Nos. 30 and 40, with buckets of 3½ and 4½ cubic yards capacity.

Output.—The output figures given represent average outputs when machines are working in easy material such as sand, loose earth, etc., with skilled operating crew and adequate wagon service.

When working in medium clay, the output will be reduced by approximately 20 per cent., and when excavating stone, outputs can only be determined after an inspection of the actual conditions.

Varying jib lengths and bucket capacities can be fitted to the Nos. 20, 135, and 300 crane navvies and Nos. 75, 135, 180, and 300 draglines, *e.g.* on the No. 300 dragline machine, in place of the 120 ft. jib with 8 cubic yard bucket there can be supplied a 160 ft. jib and a 5 cubic yard bucket.

Sizes Nos. 6 and 10 are universal machines, *i.e.* convertible from crane navvies into dragline excavators and grabbing cranes, or from dragline excavators into crane navvies and grabbing cranes. These machines can also be used as travelling cranes and pile drivers.

Costs.—The cost of excavation per cubic yard varies according to the output, nature of material, depth of face, local fuel and labour costs.

The saving possible by mechanical excavation over hand excavation is definite in most cases, and it is possible to get excavation costs down to 1½d. per cubic yard, when working in easy material and where the disposal arrangements can handle the full output of the machine.

With moderate outputs and working in clay, the cost per cubic yard, including full depreciation and maintenance charges, would range from 3*d.* to 6*d.*

Operating Crew.—The crew required to operate 'Ruston' excavators comprise one driver and a youth as fireman; and in the case of caterpillar machines, one or two men in the bottom. With rail wheel machines up to the No. 20, two to four men in the bottom are called for, and on the larger machines, six men in the bottom.

The dragline excavator can be used for under-water excavation, and has successfully carried out the cutting of new waterways and the cleaning out of existing waterways. In India, 'Ruston' No. 300 draglines are now engaged in excavating waterways having a bottom width of 280 ft.

In addition to steam operation, 'Ruston' excavators can be supplied for electrical and internal combustion engine drive.

See also Excavating (Mechanical)—'Cost of Engineering Works,' Section XLV.

Shrinkage of Materials.

In estimating the relative amounts of excavation and embankment required, allowance must be made for difference in the spaces occupied by the material before excavation and after it is settled in embankment. The shrinkage of the different materials is about as follows:—

Gravel	8 per cent.	Loam and light sandy earths .	12 per cent.
Gravel and sand . . .	9 per cent.	Loose vegetable soil . .	15 per cent.
Clay and clay earths . .	10 per cent.	Puddled clay	25 per cent.

Rock, on the other hand, increases in volume by being broken up, and does not settle again into less than its original bulk. The increase may be taken at 50 per cent. Thus, an excavation of loam measuring 1,000 cubic yards will form only about 880 cubic yards of embankment, or an embankment of 1,000 cubic yards will require about 1,120 cubic yards measured in excavation to make it. A rock excavation measuring 1,000 cubic yards will make from 1,500 to 1,700 cubic yards of embankment, depending upon the size of the fragments.

TABLE I. FOR CALCULATING EARTHWORK.

Slopes **1** to **1**. Base = 1 foot. $l = 66$ feet = 1 chain. H and h in feet. V and V in cubic yards.

H	h = 0		1		2		3		4		5		6		7		8		9	
	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V
1	·81	1·22	2·44	2·44																
2	3·26	2·44	5·70	3·67	9·78	4·89														
3	7·33	3·67	10·59	4·89	15·48	6·11	22·00	7·33												
4	13·04	4·89	17·11	6·11	22·81	7·33	30·15	8·56	39·11	9·78										
5	20·37	6·11	25·26	7·33	31·78	8·56	39·92	9·78	49·70	11·00	61·11	12·22								
6	29·33	7·33	35·04	8·56	42·87	9·78	51·33	11·00	61·92	12·22	74·15	13·44	88·00	14·67						
7	39·93	8·56	46·45	9·78	54·59	11·00	64·37	12·22	75·78	13·44	88·81	14·67	103·48	15·89	119·78	17·11				
8	52·15	9·78	59·48	11·00	68·44	12·22	79·04	13·44	91·26	14·67	106·11	15·89	120·59	17·11	137·70	18·33	156·44	19·56		
9	66·00	11·00	74·15	12·22	83·98	13·44	95·38	14·67	106·37	15·89	118·04	17·11	139·33	18·33	157·26	19·56	176·81	20·78	198·00	22·00
10	81·48	12·22	90·44	13·44	101·04	14·67	112·26	15·89	127·11	17·11	142·59	18·33	159·70	19·56	178·44	20·78	198·81	22·00	220·81	23·22
11	98·59	13·44	108·37	14·67	119·78	15·89	132·81	17·11	147·48	18·33	163·78	19·56	181·70	20·78	201·26	22·00	222·44	23·22	245·26	24·44
12	117·33	14·67	127·93	15·89	140·15	17·11	154·00	18·33	169·48	19·56	186·59	20·78	205·33	22·00	225·70	23·22	247·70	24·44	271·33	25·67
13	137·70	15·89	149·11	17·11	162·15	18·33	176·81	19·56	193·11	20·78	211·04	22·00	230·59	23·22	251·78	24·44	274·59	25·67	299·04	26·89
14	159·70	17·11	171·98	18·33	185·78	19·56	201·26	20·78	218·37	22·00	237·11	23·22	257·48	24·44	279·48	25·67	303·11	26·89	328·37	28·11
15	183·33	18·33	196·37	19·56	211·04	20·78	227·33	22·00	245·26	23·22	264·81	24·44	286·00	25·67	308·81	26·89	333·26	28·11	359·33	29·33
16	208·59	19·56	222·44	20·78	237·93	22·00	255·04	23·22	273·78	24·44	294·15	25·67	316·15	26·89	339·78	28·11	365·08	29·33	391·92	30·56
17	235·48	20·78	250·15	22·00	266·44	23·22	284·37	24·44	303·92	25·67	325·11	26·89	347·92	28·11	372·37	29·33	398·44	30·56	426·15	31·78
18	264·00	22·00	280·48	23·22	296·59	24·44	315·33	25·67	335·70	26·89	357·70	28·11	381·33	29·33	406·59	30·56	433·48	31·78	462·00	33·00
19	294·15	23·22	310·45	24·44	328·37	25·67	347·92	26·89	369·11	28·11	391·92	29·33	416·37	30·56	442·44	31·78	470·15	33·00	499·48	34·22
20	325·93	24·44	343·04	25·67	361·78	26·89	382·15	28·11	404·15	29·33	427·78	30·56	453·04	31·78	479·92	33·00	508·44	34·22	538·59	35·44
21	359·38	25·67	377·26	26·89	396·81	28·11	418·00	29·33	440·81	30·56	465·26	31·78	491·33	33·00	519·04	34·22	548·37	35·44	579·33	36·67
22	394·37	26·89	413·11	28·11	433·48	29·33	455·48	30·56	479·11	31·78	504·37	33·00	531·26	34·22	559·78	35·44	589·92	36·67	621·70	37·89
23	431·04	28·11	450·59	29·33	471·78	30·56	494·59	31·78	519·04	33·00	545·11	34·22	572·81	35·44	602·15	36·67	633·11	37·89	665·70	39·11
24	469·33	29·33	489·70	30·56	511·70	31·78	535·33	33·00	560·59	34·22	587·48	35·44	616·00	36·67	646·15	37·89	677·92	39·11	711·33	40·33
25	509·26	30·56	530·45	31·78	553·26	33·00	577·70	34·22	603·78	35·44	631·48	36·67	660·81	37·89	691·78	39·11	724·37	40·33	758·59	41·56
26	550·81	31·78	572·81	33·00	596·44	34·22	621·70	35·44	648·59	36·67	677·11	37·89	707·26	39·11	739·04	40·33	772·44	41·56	807·48	42·78
27	594·00	33·00	616·81	34·22	641·26	35·44	667·33	36·67	695·04	37·89	724·37	39·11	755·33	40·33	787·92	41·56	822·15	42·78	858·00	44·00
28	638·81	34·22	662·45	35·44	687·70	36·67	714·59	37·89	743·11	39·11	773·26	40·33	805·04	41·56	838·44	42·78	873·48	44·00	910·15	45·22
29	685·26	35·44	709·70	36·67	735·78	37·89	763·48	39·11	792·81	40·33	823·78	41·56	856·37	42·78	890·59	44·00	926·44	45·22	963·92	46·44
30	733·33	36·67	758·59	37·89	785·48	39·11	814·00	40·33	844·15	41·56	875·92	42·78	909·33	44·00	944·87	45·22	981·04	46·44	1019·33	47·67
31	783·04	37·89	809·11	39·11	836·81	40·33	866·15	41·56	897·11	42·78	929·70	44·00	963·92	45·22	999·78	46·44	1037·26	47·67	1076·37	48·89
32	834·37	39·11	861·26	40·33	889·78	41·56	919·92	42·78	951·70	44·00	986·11	45·22	1020·15	46·44	1056·82	47·67	1095·11	48·89	1135·04	50·11
33	887·33	40·33	915·04	41·56	944·37	42·78	975·33	44·00	1007·92	45·22	1042·15	46·44	1078·00	47·67	1115·48	48·89	1154·59	50·11	1195·33	51·33
34	941·98	41·56	970·45	42·78	1000·59	44·00	1032·37	45·22	1065·78	46·44	1100·81	47·67	1137·48	48·89	1175·78	50·11	1215·70	51·33	1257·26	52·56
35	998·15	42·78	1027·48	44·00	1058·44	45·22	1091·04	46·44	1125·26	47·67	1161·11	48·89	1198·59	50·11	1237·70	51·33	1278·44	52·56	1320·81	53·78
36	1056·00	44·00	1086·15	45·22	1117·98	46·44	1151·33	47·67	1186·37	48·89	1223·04	50·11	1261·33	51·33	1301·26	52·56	1342·82	53·78	1386·00	55·00
37	1115·48	45·22	1146·45	46·44	1179·04	47·67	1213·26	48·89	1249·11	50·11	1286·59	51·33	1325·70	52·56	1366·44	53·78	1408·82	55·00	1452·81	56·22
38	1176·59	46·44	1208·37	47·67	1241·78	48·89	1276·81	50·11	1313·48	51·33	1351·78	52·56	1391·70	53·78	1433·26	55·00	1476·44	56·22	1521·26	57·44
39	1239·33	47·67	1271·93	48·89	1306·15	50·11	1342·00	51·33	1379·48	52·56	1418·59	53·78	1459·33	55·00	1501·70	56·22	1545·70	57·44	1591·33	58·67
40	1303·70	48·89	1337·11	50·11	1372·15	51·33	1408·81	52·56	1447·11	53·78	1487·04	55·00	1528·59	56·22	1571·78	57·44	1616·59	58·67	1663·03	59·89

TABLE I. FOR CALCULATING EARTHWORK—(*continued*).

Slopes 1 to 1. Base = 1 foot. l = 66 feet = 1 chain. H and h in feet. V and V in cubic yards.

H	h = 10		11		12		13		14		15		16		17		18		19	
	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V
10	244·44	24·44																		
11	269·70	25·67	295·78	26·89																
12	296·59	26·89	323·48	28·11	352·00	29·33														
13	325·11	28·11	352·81	29·33	382·15	30·56	413·11	31·78												
14	355·26	29·33	388·78	30·56	418·92	31·78	445·70	33·00	479·11	34·22										
15	387·04	30·56	416·37	31·78	447·33	33·00	479·92	34·22	514·15	35·44	550·00	36·67								
16	420·44	31·78	450·59	33·00	482·37	34·22	515·78	35·44	550·82	36·67	587·48	37·89	625·78	39·11						
17	455·48	33·00	486·44	34·22	519·04	35·44	553·26	36·67	589·11	37·89	626·59	39·11	665·70	40·33	706·44	41·56				
18	492·15	34·22	528·92	35·44	557·33	36·67	592·37	37·89	629·04	39·11	667·38	40·33	707·26	41·56	748·81	42·78	792·00	44·00		
19	530·44	35·44	568·04	36·67	597·26	37·89	633·11	39·11	670·59	40·33	709·70	41·56	750·44	42·78	792·81	44·00	836·81	45·22	882·44	46·44
20	570·37	36·67	608·78	37·89	638·81	39·11	675·48	40·33	713·78	41·56	753·70	42·78	795·26	44·00	838·44	45·22	883·26	46·44	929·70	47·67
21	611·92	37·89	646·15	39·11	682·00	40·33	719·48	41·56	758·59	42·78	799·33	44·00	841·70	45·22	885·70	46·44	981·38	47·67	978·59	48·89
22	655·11	39·11	690·15	40·33	726·81	41·56	765·11	42·78	805·04	44·00	846·59	45·22	889·78	46·44	934·59	47·67	981·04	48·89	1029·11	50·11
23	699·92	40·33	735·78	41·56	773·26	42·78	812·37	44·00	853·11	45·22	895·48	46·44	939·48	47·67	985·11	48·89	1032·37	50·11	1081·26	51·33
24	746·87	41·56	788·04	42·78	821·33	44·00	861·26	45·22	902·81	46·44	946·00	47·67	990·81	48·89	1037·26	50·11	1085·33	51·33	1135·04	52·56
25	794·44	42·78	831·92	44·00	871·04	45·22	911·78	46·44	954·15	47·67	998·15	48·89	1043·78	50·11	1091·04	51·33	1139·92	52·56	1190·44	53·78
26	844·15	44·00	882·44	45·22	922·37	46·44	963·92	47·67	1007·11	48·89	1051·92	50·11	1098·37	51·33	1146·44	52·56	1196·15	53·78	1247·48	55·00
27	896·48	45·22	934·59	46·44	975·33	47·67	1017·70	48·89	1061·70	50·11	1107·38	51·33	1154·59	52·56	1203·48	53·78	1254·00	55·00	1306·15	56·22
28	948·44	46·44	988·37	47·67	1029·92	48·89	1073·11	50·11	1117·92	51·33	1164·37	52·56	1212·44	53·78	1262·15	55·00	1313·48	56·22	1366·44	57·44
29	1003·04	47·67	1043·78	48·89	1086·15	50·11	1130·15	51·33	1175·78	52·56	1223·04	53·78	1271·92	55·00	1322·44	56·22	1374·59	57·44	1428·37	58·67
30	1059·26	48·89	1100·82	50·11	1144·00	51·33	1188·82	52·56	1235·26	53·78	1283·33	55·00	1333·04	56·22	1384·37	57·44	1437·33	58·67	1491·92	59·89
31	1117·11	50·11	1159·48	51·33	1203·48	52·56	1249·11	53·78	1296·37	55·00	1345·26	56·22	1395·78	57·44	1447·92	58·67	1501·70	59·89	1557·11	61·11
32	1176·59	51·33	1219·78	52·56	1264·59	53·78	1311·04	55·00	1359·11	56·22	1408·82	57·44	1460·15	58·67	1513·11	59·89	1567·70	61·11	1623·92	62·33
33	1237·70	52·56	1281·70	53·78	1327·33	55·00	1374·59	56·22	1423·48	57·44	1474·00	58·67	1526·15	59·89	1579·92	61·11	1635·33	62·33	1692·37	63·56
34	1300·44	53·78	1345·26	55·00	1391·70	56·22	1439·78	57·44	1489·48	58·67	1540·82	59·89	1593·78	61·11	1648·37	62·33	1704·59	63·56	1762·44	64·78
35	1364·82	55·00	1410·44	56·22	1457·70	57·44	1506·59	58·67	1557·11	59·89	1609·26	61·11	1663·04	62·33	1718·44	63·56	1775·48	64·78	1834·15	66·00
36	1430·82	56·22	1477·26	57·44	1525·33	58·67	1575·04	59·89	1626·37	61·11	1679·33	62·33	1738·92	63·56	1790·15	64·78	1848·00	66·00	1907·48	67·22
37	1498·44	57·44	1545·70	58·67	1594·59	59·89	1645·11	61·11	1697·26	62·33	1751·04	63·56	1806·44	64·78	1863·48	66·00	1922·15	67·22	1982·44	68·44
38	1567·70	58·67	1615·78	59·89	1665·48	61·11	1716·82	62·33	1769·78	63·56	1824·37	64·78	1880·59	66·00	1938·44	67·22	1997·92	68·44	2059·04	69·67
39	1638·59	59·89	1687·48	61·11	1736·00	62·33	1790·15	63·56	1843·92	64·78	1899·38	66·00	1956·37	67·22	2015·04	68·44	2075·33	69·67	2137·26	70·89
40	1711·11	61·11	1760·89	62·33	1812·15	63·56	1865·11	64·78	1919·70	66·00	1975·92	67·22	2033·78	68·44	2093·26	69·67	2154·37	70·89	2217·11	72·11

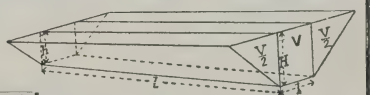
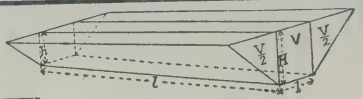
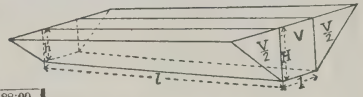


TABLE I. FOR CALCULATING EARTHWORK—(*continued*).

Slopes 1 to 1. Base = 1 foot. l = 66 feet = 1 chain. H and h in feet. V and V in cubic yards.



H	h = 20		21		22		23		24		25		26		27		28		29	
	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V
20	977·78	48·89																		
21	1027·48	50·11	1078·00	51·33																
22	1078·82	51·33	1130·15	52·56	1183·11	53·78														
23	1131·78	52·56	1183·92	53·78	1237·70	55·00	1293·11	56·22												
24	1186·37	53·78	1239·33	55·00	1293·92	56·22	1350·15	57·44	1408·00	58·67										
25	1242·59	55·00	1296·37	56·22	1351·78	57·44	1408·82	58·67	1467·48	59·89	1527·78	61·11								
26	1300·44	56·22	1355·04	57·44	1411·26	58·67	1469·11	59·89	1528·59	61·11	1589·70	62·33	1652·44	63·56						
27	1359·92	57·44	1415·33	58·67	1472·37	59·89	1531·04	61·11	1591·33	62·33	1653·26	63·56	1716·82	64·78	1782·82	66·00				
28	1421·04	58·67	1477·26	59·89	1535·11	61·11	1594·59	62·33	1655·70	63·56	1718·44	64·78	1782·82	66·00	1848·82	67·22	1916·44	68·44		
29	1488·78	59·89	1540·81	61·11	1599·48	62·33	1659·78	63·56	1721·70	64·78	1785·26	66·00	1850·44	67·22	1917·26	68·44	1985·70	69·67	2055·78	70·89
30	1548·15	61·11	1606·00	62·33	1665·48	63·56	1726·59	64·78	1789·33	66·00	1853·70	67·22	1919·70	68·44	1987·33	69·67	2056·59	70·89	2127·48	72·11
31	1614·15	62·33	1673·82	63·56	1733·11	64·78	1795·04	66·00	1858·59	67·22	1923·78	68·44	1990·59	69·67	2059·04	70·89	2129·11	72·11	2200·82	73·33
32	1681·78	63·56	1741·26	64·78	1802·37	66·00	1865·11	67·22	1929·48	68·44	1995·48	69·67	2063·11	70·89	2132·37	72·11	2205·26	73·33	2275·78	74·56
33	1751·04	64·78	1811·33	66·00	1873·26	67·22	1936·82	68·44	2002·00	69·67	2068·82	70·89	2137·26	72·11	2207·33	73·33	2279·04	74·56	2352·37	75·78
34	1821·92	66·00	1883·04	67·22	1945·78	68·44	2010·15	69·67	2076·15	70·89	2143·78	72·11	2213·04	73·33	2283·93	74·56	2356·44	75·78	2430·59	77·00
35	1894·44	67·22	1956·37	68·44	2019·92	69·67	2085·11	70·89	2151·93	72·11	2220·37	73·33	2290·44	74·56	2362·15	75·78	2435·48	77·00	2510·45	78·22
36	1968·59	68·44	2031·33	69·67	2095·70	70·89	2161·70	72·11	2229·33	73·33	2298·59	74·56	2369·48	75·78	2442·00	77·00	2516·15	78·22	2591·93	79·44
37	2044·37	69·67	2107·92	70·89	2173·11	72·11	2239·93	73·33	2308·37	74·56	2378·44	75·78	2450·15	77·00	2523·48	78·22	2598·45	79·44	2675·04	80·67
38	2121·78	70·89	2186·15	72·11	2252·15	73·33	2319·78	74·56	2389·04	75·78	2459·93	77·00	2532·45	78·22	2606·59	79·44	2682·37	80·67	2759·78	81·89
39	2200·82	72·11	2266·00	73·33	2332·82	74·56	2401·26	75·78	2471·33	77·00	2543·04	78·22	2616·37	79·44	2691·33	80·67	2767·93	81·89	2846·15	83·11
40	2281·48	73·33	2347·48	74·56	2415·11	75·78	2484·37	77·00	2555·26	78·22	2627·78	79·44	2701·93	80·67	2777·70	81·89	2855·11	83·11	2934·15	84·33



H	h = 30		31		32		33		34		35		36		37		38		39	
	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V
30	2200·00	73·33																		
31	2274·15	74·56	2349·11	75·78																
32	2349·93	75·78	2425·70	77·00	2503·11	78·22														
33	2427·33	77·00	2503·93	78·22	2582·15	79·44	2662·00	80·67												
34	2506·37	78·22	2588·78	79·44	2662·82	80·67	2743·48	81·89	2825·78	83·11										
35	2587·04	79·44	2665·26	80·67	2745·11	81·89	2826·59	83·11	2909·70	84·33	2994·45	85·56								
36	2669·33	80·67	2748·37	81·89	2829·04	83·11	2911·33	84·33	2995·26	85·56	3080·82	86·78	3168·00	88·00						
37	2753·26	81·89	2833·11	83·11	2914·59	84·33	2997·70	85·56	3082·45	86·78	3168·89	88·00	3256·82	89·22	3346·45	90·44				
38	2838·81	83·11	2919·48	84·33	3001·78	85·56	3085·70	86·78	3171·26	88·00	3258·45	89·22	3347·26	90·44	3437·70	91·67	3529·78	92·89		
39	2926·00	84·33	3007·48	85·56	3090·59	86·78	3175·33	88·00	3261·70	89·22	3349·70	90·44	3439·33	91·67	3530·59	92·89	3623·48	94·11	3718·00	95·33
40	3014·81	85·56	3097·11	86·78	3181·04	88·00	3266·59	89·22	3353·78	90·44	3442·59	91·67	3533·04	92·89	3625·11	94·11	3718·81	95·33	3814·15	96·56

Table II. for Calculating Earthwork.

Slopes 1½ to 1. Base = 1 foot. l = 66 feet = 1 chain. H and h in feet. V and V in cubic yards.

H	h = 0		1		2		3		4		5		6		7		8		9	
	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V
1	1·22	1·22	3·67	2·44																
2	4·89	2·44	8·56	3·67	14·67	4·89														
3	11·00	3·67	15·89	4·89	23·22	6·11	33·00	7·33												
4	19·56	4·89	25·67	6·11	34·22	7·33	45·22	8·56	58·67	9·78										
5	30·56	6·11	37·89	7·33	47·67	8·56	59·89	9·78	74·56	11·00	91·67	12·22								
6	44·00	7·33	52·56	8·56	63·56	9·78	77·00	11·00	92·89	12·22	111·22	13·44	132·00	14·67						
7	59·89	8·56	69·67	9·78	81·89	11·00	95·56	12·22	113·67	13·44	133·22	14·67	155·22	15·89	179·67	17·11				
8	78·22	9·78	89·22	11·00	102·67	12·22	118·56	13·44	136·89	14·67	157·67	15·89	180·89	17·11	206·56	18·33	234·67	19·56		
9	99·00	11·00	111·22	12·22	125·89	13·44	143·00	14·67	162·56	15·89	184·56	17·11	209·00	18·33	235·89	19·56	265·22	20·78	297·00	22·00
10	122·22	12·22	135·67	13·44	151·56	14·67	169·89	15·89	190·67	17·11	213·89	18·33	239·56	19·56	267·67	20·78	298·22	22·00	332·22	23·22
11	147·89	13·44	162·56	14·67	179·67	15·89	199·22	17·11	221·22	18·33	245·67	19·56	272·56	20·78	301·89	22·00	333·67	23·22	367·89	24·44
12	176·00	14·67	191·89	15·89	210·22	17·11	231·00	18·33	254·22	19·56	279·89	20·78	306·00	22·00	338·56	23·22	371·56	24·44	407·00	25·67
13	206·56	15·89	223·67	17·11	243·22	18·33	265·22	19·56	289·67	20·78	316·56	22·00	345·89	23·22	377·67	24·44	411·89	25·67	448·56	26·89
14	239·56	17·11	257·89	18·33	278·67	19·56	301·89	20·78	327·56	22·00	355·67	23·22	386·22	24·44	419·22	25·67	454·67	26·89	492·56	28·11
15	275·00	18·33	294·56	19·56	316·56	20·78	341·00	22·00	367·89	23·22	397·22	24·44	429·00	25·67	463·22	26·89	499·89	28·11	539·00	29·33
16	312·89	19·56	333·67	20·78	356·89	22·00	382·56	23·22	410·67	24·44	441·22	25·67	474·22	26·89	509·67	28·11	547·54	29·33	587·89	30·56
17	353·22	20·78	375·22	22·00	399·67	23·22	426·56	24·44	455·89	25·67	487·67	26·89	521·88	28·11	558·56	29·33	597·67	30·56	639·22	31·78
18	396·00	22·00	419·22	23·22	444·89	24·44	473·00	25·67	503·56	26·89	536·56	28·11	572·00	29·33	609·89	30·56	650·22	31·78	693·00	33·00
19	441·22	23·22	465·67	24·44	492·56	25·67	521·89	26·89	553·67	28·11	587·89	29·33	624·56	30·56	663·67	31·78	705·22	33·00	749·22	34·22
20	488·89	24·44	514·56	25·67	542·67	26·89	573·22	28·11	606·22	29·33	641·67	30·56	679·56	31·78	719·89	33·00	762·67	34·22	807·89	35·44
21	539·00	25·67	565·89	26·89	595·22	28·11	627·00	29·33	661·22	30·56	697·89	31·78	737·00	33·00	778·56	34·22	822·56	35·44	869·00	36·67
22	591·56	26·89	619·67	28·11	650·22	29·33	683·22	30·56	718·67	31·78	756·56	33·00	796·89	34·22	839·67	35·44	884·89	36·67	932·56	37·89
23	646·56	28·11	675·89	29·33	707·67	30·56	741·89	31·78	778·56	33·00	817·67	34·22	859·22	35·44	903·22	36·67	949·67	37·89	998·56	39·11
24	704·00	29·33	734·56	30·56	767·56	31·78	808·00	33·00	840·89	34·22	881·22	35·44	924·00	36·67	969·22	37·89	1016·88	39·11	1067·00	40·33
25	763·89	30·56	795·67	31·78	829·89	33·00	866·56	34·22	905·67	35·44	947·22	36·67	991·22	37·89	1037·67	39·11	1086·56	40·33	1137·89	41·56
26	826·22	31·78	859·22	33·00	894·67	34·22	932·56	35·44	972·89	36·67	1015·67	37·89	1060·89	39·11	1108·56	40·33	1158·67	41·56	1211·22	42·78
27	891·00	33·00	925·22	34·22	961·89	35·44	1001·00	36·67	1042·56	37·89	1086·56	39·11	1138·00	40·33	1181·88	41·56	1233·22	42·78	1287·00	44·00
28	958·22	34·22	998·67	35·44	1031·56	36·67	1071·89	37·89	1114·67	39·11	1159·89	40·33	1207·56	41·56	1257·67	42·78	1310·22	44·00	1365·22	45·22
29	1027·89	35·44	1064·56	36·67	1103·67	37·89	1145·22	39·11	1189·22	40·33	1235·67	41·56	1284·56	42·78	1335·89	44·00	1389·67	45·22	1445·89	46·44
30	1100·00	36·67	1137·89	37·89	1178·22	39·11	1221·00	40·33	1266·22	41·56	1313·89	42·78	1364·00	44·00	1416·56	45·22	1471·56	46·44	1529·00	47·67
31	1174·56	37·89	1213·67	39·11	1255·22	40·33	1299·22	41·56	1345·67	42·78	1394·56	44·00	1445·89	45·22	1499·67	46·44	1555·89	47·67	1614·56	48·89
32	1251·56	39·11	1291·89	40·33	1334·67	41·56	1379·89	42·78	1427·55	44·00	1477·67	45·22	1530·22	46·44	1586·22	47·67	1642·67	48·89	1697·56	50·11
33	1331·00	40·33	1372·56	41·56	1416·56	42·78	1468·00	44·00	1511·89	45·22	1568·22	46·44	1617·00	47·67	1678·22	48·89	1731·89	50·11	1793·00	51·33
34	1412·89	41·56	1455·67	42·78	1500·89	44·00	1548·56	45·22	1598·67	46·44	1651·22	47·67	1701·22	48·89	1763·67	50·11	1823·56	51·33	1885·89	52·56
35	1497·22	42·78	1541·22	44·00	1587·67	45·22	1636·56	46·44	1687·89	47·67	1741·67	48·89	1797·89	50·11	1856·58	51·33	1917·67	52·56	1981·22	53·78
36	1584·00	44·00	1629·22	45·22	1676·89	46·44	1727·00	47·67	1779·56	48·89	1834·56	50·11	1892·00	51·33	1951·89	52·56	2014·22	53·78	2079·00	55·00
37	1672·22	45·22	1719·67	46·44	1768·56	47·67	1819·89	48·89	1878·67	50·11	1929·89	51·33	1988·56	52·56	2049·67	53·78	2113·22	55·00	2179·22	56·22
38	1764·89	46·44	1812·56	47·67	1862·67	48·89	1915·22	50·11	1970·22	51·33	2027·66	52·56	2087·56	53·78	2149·89	55·00	2214·67	56·22	2281·89	57·44
39	1859·00	47·67	1907·89	48·89	1959·22	50·11	2013·00	51·33	2069·22	52·56	2127·89	53·78	2189·00	55·00	2252·56	56·22	2318·56	57·44	2387·00	58·67
40	1955·55	48·89	2005·67	50·11	2058·22	51·33	2113·22	52·56	2170·67	53·78	2230·56	55·00	2292·89	56·22	2357·67	57·44	2424·89	58·67	2494·56	59·89

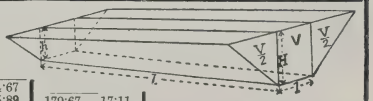
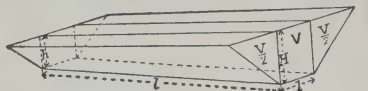


TABLE II. FOR CALCULATING EARTHWORK—(*continued*).

Slopes $1\frac{1}{2}$ to 1. Base=1 foot. l=66 feet=1 chain. H and h in feet. V and V in cubic yards.



H	h = 10		11		12		13		14		15		16		17		18		19	
	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V
10	366·67	24·44																		
11	404·56	25·67	443·67	26·89																
12	444·89	26·89	485·22	28·11	528·00	29·33														
13	487·67	28·11	529·22	29·33	574·22	30·56	619·67	31·78												
14	532·89	29·33	575·67	30·56	620·89	31·78	668·56	33·00	718·67	34·22										
15	580·56	30·56	624·56	31·78	671·00	33·00	719·89	34·22	771·22	35·44	825·00	36·67								
16	630·67	31·78	675·89	33·00	723·56	34·22	773·67	35·44	826·22	36·67	881·22	37·89	938·67	39·11						
17	683·22	33·00	729·67	34·22	778·56	35·44	829·89	36·67	883·67	37·89	939·88	39·11	998·56	40·33	1059·67	41·56				
18	738·22	34·22	785·89	35·44	835·00	36·67	888·56	37·89	948·56	39·11	1001·00	40·33	1060·89	41·56	1123·22	42·78	1188·00	44·00		
19	795·67	35·44	844·56	36·67	895·89	37·89	948·67	39·11	1005·89	40·33	1064·56	41·56	1125·67	42·78	1189·22	44·00	1255·22	45·22	1328·67	46·44
20	855·56	36·67	905·67	37·89	958·22	39·11	1013·22	40·33	1070·67	41·56	1130·56	42·78	1192·89	44·00	1257·67	45·22	1324·89	46·44	1494·56	47·67
21	917·89	37·89	969·22	39·11	1023·00	40·33	1079·22	41·56	1137·89	42·78	1199·00	44·00	1262·56	45·22	1328·56	46·44	1397·00	47·67	1567·89	48·89
22	982·67	39·11	1035·22	40·33	1090·22	41·56	1147·67	42·78	1207·56	44·00	1269·89	45·22	1334·67	46·44	1401·89	47·67	1471·56	48·89	1648·67	50·11
23	1049·89	40·33	1103·67	41·56	1159·89	42·78	1218·56	44·00	1279·67	45·22	1343·22	46·44	1409·22	47·67	1477·67	48·89	1548·56	50·11	1721·89	51·33
24	1119·56	41·56	1174·56	42·78	1232·00	44·00	1291·89	45·22	1354·22	46·44	1419·00	47·67	1486·22	48·89	1555·89	50·11	1628·00	51·33	1802·56	52·56
25	1191·67	42·78	1247·89	44·00	1306·56	45·22	1367·67	46·44	1431·22	47·67	1497·22	48·89	1565·67	50·11	1636·56	51·33	1708·89	52·56	1885·67	53·78
26	1266·22	44·00	1323·67	45·22	1383·56	46·44	1445·89	47·67	1510·67	48·89	1577·89	50·11	1647·56	51·33	1719·67	52·56	1794·22	53·78	1971·22	55·00
27	1343·22	45·22	1401·80	46·44	1463·00	47·67	1525·56	48·89	1592·56	50·11	1661·00	51·33	1731·89	52·56	1805·22	53·78	1881·00	55·00	2059·22	56·22
28	1422·67	46·44	1482·56	47·67	1544·89	48·89	1609·67	50·11	1676·89	51·33	1746·56	52·56	1818·67	53·78	1893·22	55·00	1970·22	56·22	2149·67	57·44
29	1504·56	47·67	1565·67	48·89	1629·22	50·11	1695·22	51·33	1763·67	52·56	1834·56	53·78	1907·89	55·00	1983·67	56·22	2061·89	57·44	2242·56	58·67
30	1588·89	48·89	1651·22	50·11	1716·00	51·33	1788·22	52·56	1852·89	53·78	1925·00	55·00	1999·56	56·22	2076·56	57·44	2156·00	58·67	2337·89	59·89
31	1675·67	50·11	1739·22	51·33	1805·22	52·56	1873·67	53·78	1944·56	55·00	2017·89	56·22	2098·67	57·44	2171·89	58·61	2252·56	59·89	2435·67	61·11
32	1764·89	51·33	1829·67	52·56	1896·89	53·78	1966·56	55·00	2038·67	56·22	2113·22	57·44	2190·22	58·67	2269·67	59·89	2351·56	61·11	2536·89	62·33
33	1856·56	52·56	1922·56	53·78	1991·00	55·00	2061·89	56·22	2135·22	57·44	2211·00	58·67	2289·22	59·89	2369·89	61·11	2453·00	62·33	2638·56	63·56
34	1950·67	53·78	2017·89	55·00	2087·56	56·22	2159·67	57·44	2234·22	58·67	2311·22	59·89	2390·67	61·11	2472·56	62·33	2556·89	63·56	2743·67	64·78
35	2047·22	55·00	2115·67	56·22	2186·56	57·44	2259·89	58·67	2335·67	59·89	2413·89	61·11	2494·56	62·33	2577·67	63·56	2663·22	64·78	2851·22	66·00
36	2146·22	56·22	2215·89	57·44	2288·00	58·67	2362·56	59·89	2442·00	61·11	2519·00	62·33	2600·98	63·56	2685·22	64·78	2772·00	66·00	2961·22	67·22
37	2247·67	57·44	2318·56	58·67	2391·89	59·89	2467·67	61·11	2545·89	62·33	2626·56	63·56	2709·67	64·78	2795·22	66·00	2888·22	67·22	3073·67	68·44
38	2351·56	58·67	2423·67	59·89	2498·22	61·11	2575·22	62·33	2654·66	63·56	2736·56	64·78	2820·89	66·00	2907·67	67·22	2996·89	68·44	3189·56	69·67
39	2457·89	59·89	2531·22	61·11	2607·00	62·33	2685·22	63·56	2765·89	64·78	2849·00	66·00	2934·56	67·22	3022·56	68·44	3113·00	69·67	3305·89	70·89
40	2566·67	61·11	2641·22	62·33	2718·22	63·56	2797·67	64·78	2879·56	66·00	2963·89	67·22	3050·67	68·44	3139·89	69·67	3231·56	70·89	3425·67	72·11

TABLE II. FOR CALCULATING EARTHWORK—(*continued*).

Slopes 1½ to 1. Base=1 foot. *l*=66 feet=1 chain. H and *h* in feet. V and **V** in cubic yards.

H	*h* 20		21		22		23		24		25		26		27		28		29	
	V	**V**	V	**V**	V	**V**	V	**V**	V	**V**	V	**V**	V	**V**	V	**V**	V	**V**	V	**V**
20	1466·67	48·89																		
21	1541·22	50·11	1617·00	51·33																
22	1618·22	51·33	1695·22	52·56	1774·67	53·78														
23	1697·67	52·56	1775·89	53·78	1856·56	55·00	1939·67	56·22												
24	1779·56	53·78	1859·00	55·00	1940·89	56·22	2025·22	57·44	2112·00	58·67										
25	1863·89	55·00	1944·56	56·22	2027·67	57·44	2118·22	58·67	2201·22	59·89	2291·67	61·11								
26	1950·67	56·22	2032·56	57·44	2116·89	58·67	2203·67	59·89	2292·89	61·11	2384·56	62·33	2478·67	63·56						
27	2039·89	57·44	2123·00	58·67	2208·56	59·89	2296·56	61·11	2387·00	62·33	2479·89	63·56	2575·22	64·78	2673·00	66·00				
28	2131·56	58·67	2215·89	59·89	2302·67	61·11	2391·89	62·33	2483·56	63·56	2577·67	64·78	2674·22	66·00	2778·22	67·22	2874·66	68·44		
29	2225·67	59·89	2311·22	61·11	2399·22	62·33	2489·67	63·56	2582·56	64·78	2677·89	66·00	2775·67	67·22	2875·88	68·44	2978·55	69·67	3083·66	70·89
30	2322·22	61·11	2409·00	62·33	2498·22	63·56	2589·89	64·78	2684·00	66·00	2780·56	67·22	2879·56	68·44	2980·99	69·67	3084·88	70·89	3191·22	72·11
31	2421·22	62·33	2509·22	63·56	2599·67	64·78	2692·56	66·00	2787·89	67·22	2885·67	68·44	2985·89	69·67	3088·55	70·89	3198·66	72·11	3301·22	73·33
32	2522·67	63·56	2611·89	64·78	2703·56	66·00	2797·67	67·22	2894·22	68·44	2993·22	69·67	3094·67	70·89	3198·55	72·11	3304·88	73·33	3413·66	74·56
33	2626·56	64·78	2717·00	66·00	2809·89	67·22	2905·22	68·44	3003·00	69·67	3103·22	70·89	3205·89	72·11	3310·99	73·33	3418·55	74·56	3528·55	75·78
34	2782·89	66·00	2824·56	67·22	2918·67	68·44	3015·22	69·67	3114·22	70·89	3215·67	72·11	3319·55	73·33	3425·88	74·56	3534·66	75·78	3645·88	77·00
35	2841·67	67·22	2934·56	68·44	3029·89	69·67	3127·67	70·89	3227·89	72·11	3330·55	73·33	3435·67	74·56	3543·22	75·78	3653·22	77·00	3765·66	78·22
36	2952·89	68·44	3047·00	69·67	3148·56	70·89	3242·55	72·11	3344·00	73·33	3447·89	74·56	3554·22	75·78	3662·99	77·00	3774·22	78·22	3887·88	79·44
37	3066·56	69·67	3161·89	70·89	3259·67	72·11	3359·89	73·33	3462·56	74·56	3567·66	75·78	3675·22	77·00	3785·22	78·22	3897·66	79·44	4012·55	80·67
38	3182·67	70·89	3269·22	72·11	3378·22	73·33	3479·67	74·56	3583·56	75·78	3689·88	77·00	3798·67	78·22	3909·88	79·44	4023·55	80·67	4139·66	81·89
39	3301·22	72·11	3399·00	73·33	3499·22	74·56	3600·89	75·78	3707·00	77·00	3814·55	78·22	3924·55	79·44	4036·99	80·67	4151·88	81·89	4269·21	83·11
40	3422·22	73·33	3521·22	74·56	3622·67	75·78	3725·55	77·00	3832·89	78·22	3941·66	79·44	4052·89	80·67	4166·55	81·89	4282·66	83·11	4401·21	84·33

H	*h* 30		31		32		33		34		35		36		37		38		39	
	V	**V**	V	**V**	V	**V**	V	**V**	V	**V**	V	**V**	V	**V**	V	**V**	V	**V**	V	**V**
30	3300·00	73·33																		
31	3411·22	74·56	3523·67	75·78																
32	3524·89	75·78	3638·55	77·00	3754·67	78·22														
33	3640·99	77·00	3755·89	78·22	3873·22	79·44	3993·00	80·67												
34	3759·55	78·22	3875·67	79·44	3994·23	80·67	4115·22	81·89	4238·67	83·11										
35	3880·55	79·44	3997·89	80·67	4117·67	81·89	4239·89	83·11	4364·55	84·33	4491·67	85·56								
36	4003·96	80·67	4122·55	81·89	4243·55	83·11	4367·00	84·33	4492·89	85·56	4621·22	86·78	4752·00	88·00						
37	4129·89	81·89	4249·67	83·11	4371·89	84·33	4496·55	85·56	4623·67	86·78	4753·22	88·00	4885·22	89·22	5019·67	90·44				
38	4258·22	83·11	4379·22	84·33	4502·67	85·56	4628·55	86·78	4756·89	88·00	4887·67	89·22	5020·89	90·44	5156·55	91·67	5294·67	92·89		
39	4388·99	84·33	4511·22	85·56	4635·89	86·78	4763·00	88·00	4892·55	89·22	5024·55	90·44	5159·00	91·67	5295·89	92·89	5435·22	94·11	5577·00	95·33
40	4522·22	85·56	4645·67	86·78	4771·55	88·00	4899·89	89·22	5030·67	90·44	5163·89	91·67	5299·55	92·89	5437·67	94·11	5578·22	95·33	5721·22	96·56

Methods of getting Earth.

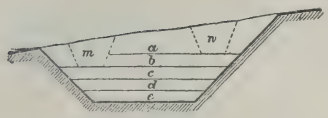


FIG. 5.

1. *Ordinary Method* (fig. 5).—Excavations carried out in layers, *a*, *b*, *c*, &c., each layer being opened out by cuts, as *m* and *n*, into which temporary ways or dobbin carts are brought. Suitable for easily-won earth and long cuttings, allowing the employment of many workmen, carts, and waggons at the same time.

2. *Method of Terracing.*—Figs. 6 and 7 show, in longitudinal section, the terraces, each of which has a transport-way, on a bench, on one or both sides of cutting, which is only removed on the completion of its terrace. Very great advantage, especially on sidelong ground, is often got by placing the temporary way alongside the cutting.

FIG. 6.

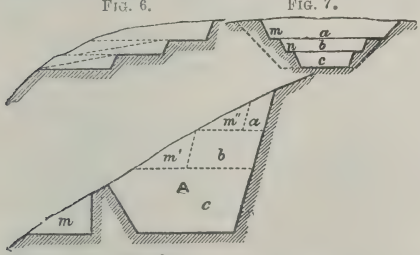


FIG. 8.

FIG. 7.

3. *Sidelong System.*—For cuttings on sidelong ground the principal temporary way runs parallel to the centre line, but at the side of cutting, which can be attacked at numerous points, and eventually along its whole length. In fig. 8, *m* represents the position of temporary way, and A the cutting. Different stages, *a*, *b*, *c*, with separate temporary ways, *m''*, *m'*, *m*, may often be advantageously employed, in which case it approximates to terracing. Steep sidelong cuttings chiefly occur in rock, where this system gives great advantages.

4. *Long-face System.*—A deep, narrow cutting, *g*, *z*, *n*, *m*, taken down to formation-level, if possible, is formed for the principal transport-way and drainage. This system is advantageous in rock strata sloping across the line, in which case the cutting will be on one side, as *m*, *n*, *p*, *o*, in fig. 9. In cases where a central cutting can be made the faces can be attacked on both sides, at different points, along the whole length, drainage and blasting being easily performed.

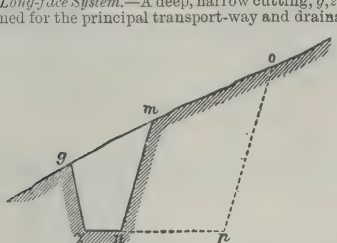


FIG. 9.

Central Lift, or English System.—A drift, *a*, *a* (figs. 10 and 11), is driven at formation-level about centre line of cutting, and a temporary way laid down in it; at the same time one or more shafts, *m*, *n*, *o*, *p*, are sunk down to it from above, and cut away in a funnel-shape, through which the stuff is shot direct into waggons standing below in the drift. This method is now widely used, with best results, both as regards time and cost. A deep cutting in lava, 500′ long × 60′ deep, containing over 124,000 cubic yards, with three shafts, was run to bank, 700′ distant, within six months, *i.e.*, at about 800 cubic yards a day. Compared with other methods it is quicker; the formation-level is at once defined; the temporary ways are shorter, and require less shifting; it is cheaper in loading waggons into which stuff falls; there is no up-throwing: and the drainage is readily done, and with the least disturbance of the work. The drift and shafts are the only source of expense, but their cost can be closely estimated.

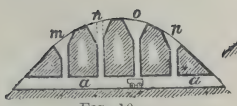


FIG. 10.

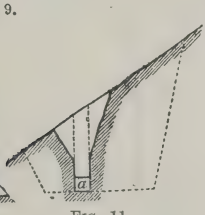


FIG. 11.

Transporting Earth.

The systems adopted comprise *barrow runs*, *horse and cart*, and *rail transport*; the latter include *hand trollies*, *horse waggons*, and *trucks drawn by locomotives or by ropes worked by stationary engines*.

Gradients in transport ways increase cost. The practical limits are: for barrows, 1 in 30; for horse and cart, 1 in 40; for temporary ways, 1 in 100.

Leading Material.

A higher up-throw, an unfavourable lifting, a repeated throwing or unnecessary moving of earth, add to cost. Good tools, well-formed barrows, carts, tip waggons, good temporary ways of rails, or planks for barrow runs, are essential. A barrow carrying 2 cubic feet without a plank run will carry 3 cubic feet with one. A tip cart carrying 8 to 10 cubic feet without, will carry 15 to 18 cubic feet with, a good temporary way.

Slopes of Excavations and Embankments.

Heights at which different soils will retain a vertical face in excavations for a short time without falling in :

	ft.	ft.		ft.	ft.
Clean dry sand and gravel from . .	0	to 1	Loamy soil, well drained . . .	5	to 10
Moist sand and ordinary surface mould	1	„ 3	Ordinary clay	9	„ 12

Slopes of railway cuttings are often—

In gravel, 1 to 1 when not more than 20′ deep	In clay, 2 to 1 when well drained
„ 1½ „ 1 when of greater depth	„ 3 or 4 to 1 when wet

Slopes of railway embankments are often—

In gravel, 1½ to 1 when not more than 30′ high	In clay, 2 to 1 when not more than 40′ high
„ 2 „ 1 when of greater height	„ 2½ „ 1 when of greater height

Chalk and rock will stand perpendicular, or with a slope of about ¼ base to 1 height.

In roads it is often better to make the sides of cuttings, especially on the south side, with greater slope than that they will stand at, to admit sun.

Earth in embankments should be laid in regular horizontal courses not more than 4′ thick. In high embankments they should be rather concave, to avoid slipping.

Form of Side Slopes.

The natural, strongest, and ultimate form of earth slopes is a concave curve, in which the flattest portion is at the bottom. This form is very rarely given to the slopes in constructing them ; in fact, the reverse is often the case, the slopes being made convex, thus saving excavation for the contractor, and inviting slips.

In cuttings exceeding 10 feet in depth the formation of concave slopes will materially aid in preventing slips, and in any case it will reduce the amount of material which will eventually have to be removed when cleaning up. Straight or convex slopes will continue to slip until the natural form is attained.

Table of Angles and Lengths of Slopes.

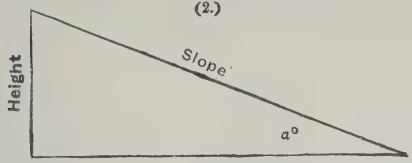
(1.)

Base of Slope is given in terms of the Height ; Length of Slope, in terms of the Base.

Angle.	Base of Slope (Height = 1).	Length of Slope (Base = 1).	Angle.	Base of Slope (Height = 1).	Length of Slope (Base = 1).	Angle.	Base of Slope (Height = 1).	Length of Slope (Base = 1).	Angle.	Base of Slope (Height = 1).	Length of Slope (Base = 1).
° ′			° ′			° ′			° ′		
45 0	1	1·414	8 0	7·1	1·009	2 52	20	1·0012	0 43	80	1·0000
40 0	1·2	1·305	7 7	8	1·007	2 18	25	1·0008	0 40	86	„
38 40	1¼	1·280	7 0	8·1	1·007	2 12	26	1·0007	0 38	90	„
35 0	1·4	1·220	6 20	9	1·006	2 7	27	1·0007	0 36	95	„
33 41	1⅓	1·201	6 0	9·5	1·005	2 3	28	1·0006	0 34	101	„
30 0	1·7	1·154	5 42	10	1·005	2 0	28·6	1·0006	0 23	150	„
29 44	1¾	1·151	5 11	11	1·0041	1 59	29	1·0006	0 17	202	„
26 34	2	1·118	5 0	11·4	1·0038	1 54	30	1·0005	0 14	245	„
25 0	2·1	1·103	4 46	12	1·0034	1 38	35	1·0004	0 12	286	„
20 0	2·7	1·064	4 23	13	1·0029	1 26	40	1·0003	0 10	343	„
18 26	3	1·054	4 5	14	1·0025	1 16	45	1·0002	0 9	381	„
15 0	3·7	1·035	4 0	14·3	1·0024	1 9	50	1·0002	0 8	429	„
14 2	4	1·030	3 49	15	1·0022	1 2	55	1·0001	0 7	491	„
11 18	5	1·019	3 34	16	1·0019	1 0	57·3	1·0001	0 6	572	„
10 0	5·7	1·015	3 22	17	1·0017	0 57	60	1·0001	0 4	859	„
9 27	6	1·013	3 11	18	1·0015	0 53	65	1·0001	0 2	1718	„
9 0	6·3	1·012	3 1	19	1·0013	0 49	70	1·0001	0 1	3437	„
8 8	7	1·010	3 0	19	1·0013	0 46	75	1·0000	—	—	„

Table of Angles and Lengths of Slopes.

(2.)



If Height = 1, then

$$\text{Base} = \cot a^\circ \; ; \; \text{Slope} = \sec a^\circ = \frac{1}{\cos a^\circ}.$$

Base and Slope in Terms of Angle.						Angle in Terms of Base and Slope.					
Angle.	Base of Slope (Height = 1).	Length of Slope (Base = 1).	Angle.	Base of Slope (Height = 1).	Length of Slope (Base = 1).	Angle.	Base of Slope (Height = 1).	Length of Slope (Base = 1).	Angle.	Base of Slope (Height = 1).	Length of Slope (Base = 1).
°			°			° ′			° ′		
90	·000	∞	45	1·000	1·414	89 25	·01	98·223	17 21	3·2	1·048
89	·017	57·299	44	1·036	1·390	88 17	·03	33·381	16 52	3·3	1·045
88	·035	28·654	43	1·072	1·367	87 8	·05	19·995	16 23	3·4	1·042
87	·052	19·107	42	1·111	1·346	86	·07	14·336	15 57	3·5	1·040
86	·070	14·336	41	1·150	1·325	84 51	·09	11·140	15 31	3·6	1·038
85	·087	11·474	40	1·192	1·305	84 17	·1	10·039	15 8	3·7	1·036
84	·105	9·567	39	1·235	1·287	81 28	·15	6·739	14 45	3·8	1·034
83	·123	8·206	38	1·280	1·269	78 41	·2	5·096	14 23	3·9	1·032
82	·141	7·185	37	1·327	1·252	75 58	·25	4·124	14 2	4·0	1·031
81	·158	6·392	36	1·376	1·236	73 18	·3	3·480	13 42	4·1	1·030
80	·176	5·759	35	1·428	1·221	70 43	·35	3·028	13 24	4·2	1·028
79	·194	5·241	34	1·483	1·206	68 12	·4	2·693	13 6	4·3	1 027
78	·213	4·810	33	1·540	1·192	65 47	·45	2·438	12 49	4·4	1·026
77	·231	4·445	32	1·600	1·179	63 26	·5	2·236	12 32	4·5	1·024
76	·249	4·134	31	1·664	1·167	61 11	·55	2·075	12 16	4·6	1·023
75	·268	3·864	30	1·732	1·155	59 2	·6	1·943	12 1	4·7	1·022
74	·287	3·628	29	1·804	1·143	56 59	·65	1·835	11 46	4·8	1·021
73	·306	3·420	28	1·881	1·133	55 1	·7	1·744	11 32	4·9	1·021
72	·325	3·236	27	1·963	1·122	53 8	·75	1·667	11 19	5	1·020
71	·344	3·072	26	2·050	1·113	51 20	·8	1·601	10 19	5·5	1·016
70	·364	2·924	25	2·145	1·103	49 38	·85	1·544	9 28	6	1·014
69	·384	2·790	24	2·246	1·095	48 1	·9	1·495	8 45	6·5	1·012
68	·404	2·669	23	2·356	1·086	46 28	·95	1·452	8 8	7	1·010
67	·424	2·559	22	2·475	1·079	45 0	1	1·414	7 8	8	1·008
66	·445	2·459	21	2·605	1·071	42 16	1·1	1·351	6 20	9	1·006
65	·466	2·366	20	2·747	1·064	39 48	1·2	1·302	5 43	10	1·005
64	·488	2·281	19	2·904	1·058	37 34	1·3	1·262	5 12	11	1·004
63	·510	2·203	18	3·078	1·051	35 32	1·4	1·229	4 46	12	1·003
62	·532	2·130	17	3·271	1·046	33 41	1·5	1·202	4 24	13	1·003
61	·554	2·063	16	3·487	1·040	32	1·6	1·179	4 5	14	1·003
60	·577	2·000	15	3·732	1·036	30 28	1·7	1·160	3 49	15	1·002
59	·601	1·942	14	4·011	1·031	29 8	1·8	1·144	3 35	16	1·002
58	·625	1·887	13	4·331	1·026	27 46	1·9	1·130	3 22	17	1·002
57	·649	1·836	12	4·705	1·022	26 33	2	1·118	3 11	18	1·002
56	·675	1·788	11	5·145	1·019	25 28	2·1	1·108	3 1	19	1·001
55	·700	1·743	10	5·671	1·015	24 27	2·2	1·099	2 52	20	1·001
54	·727	1·701	9	6·314	1·012	23 30	2·3	1·090	2 18	25	1·001
53	·754	1·662	8	7·115	1·010	22 37	2·4	1·083	1 55	30	1·001
52	·781	1·624	7	8·144	1·008	21 48	2·5	1 077	1 38	35	1·000
51	·810	1·589	6	9·514	1·006	21 2	2·6	1·071	1 26	40	1·000
50	·839	1·556	5	11·430	1·004	20 19	2·7	1·066	1 9	50	1·000
49	·869	1·524	4	14·301	1·002	19 39	2·8	1·062	58	60	1·000
48	·900	1·494	3	19·081	1·001	19 2	2·9	1·058	49	70	1·000
47	·933	1·466	2	28·636	1·001	18 26	3	1·054	43	80	1·000
46	·966	1·440	1	57·290	1·000	17 53	3·1	1·051	38	90	1·000

(*R. H. Kempe.*)

Angles of Repose of Soils.

Material.	Angle.	Ratio of Base of Slope to Height.
	°	
Clay, dry	29	1·8 to 1
„ damp, well drained	45	1 to 1
„ wet	16	3·5 to 1
Earth, dry	29	1·8 to 1
„ moist	45 to 49	1 to 1 to ·87 to 1
„ very wet	17	3·27 to 1
„ punned	66 to 74	·45 to 1 to ·28 to 1
Gravel, clean	48	·9 to 1
„ with sand	26	2 to 1
Sand, fine dry	37 to 31	1·3 to 1 to 1·6 to 1
„ wet	26	2 to 1
„ very wet	32	1·6 to 1
Shingle, loose	39	1·2 to 1
Peat	14 to 45	4 to 1 to 1 to 1

Weight of a Cubic Yard of different Soils.

	cwts.		cwts.		cwts.		cwts.
Dry peat	7½	Common earth	24	Common gravel	27	Grey chalk	36
Wet peat	15	Sandy loam	24	Wet sand	28	Sandstone	37
Top soil	20	Marl	26	Gravelly clay	30	Shale	39
Dry sand	22	Clay	27	Rough water gravel	34	Limestone	48

Loads of Carts and Waggons.

An ordinary cart, 6′ × 3½′ × 2½′ deep, holds 1⅜ cubic yards, equal 2½ tons of earth.
A dobbin cart „ ⅝ „ „ 1 „
An earth waggon „ 2 „ „ 3½ „
„ „ large „ 3 „ „ 4⅜ „
A wheelbarrow, navvy „ 1/10 „ „ 3/10 „
„ ordinary „ 3/20 „ „ 4/10 „

Carting is estimated at so much per cubic yard, by the run of 1 furlong (220 yards).

Equivalents of a Load of Different Materials.

A single load of sand, earth, rubbish = 1 cub. yd.
„ „ lime (a measure or hundred) = 1 cub. yd. = 21 striked bushels.
„ „ „ „ „ = 22 bushels.
„ „ squared or hewn timber, deals, &c. = 50 cub. ft.
„ „ unhewn timber = 40 „
„ „ bricks = 500 bricks.
„ „ tiles = 1,000 tiles.
„ „ water = 1 butt of 108 gallons.
„ „ light, bulky materials = 80 cub. ft.

Barrow and Cart Runs.

The barrows used usually hold about one-tenth cubic yard, and are mostly run on planks, which should not be narrower than 11″, and 3″ thick, nor, as a rule, have a greater inclination than 1 in 12, unless aided by ropes or winding machinery. Box-horses (fig. 12) are the best supports on an incline, either singly or resting one on another; they are mostly from 1½′ to 3′ square, made out of 11″ × 3″ planks. If a considerable height is required, trestles (fig. 13) are used, but box horses are stronger, and less likely to get out of repair.

A barrow run is 20 or 25 yards. Each foot of rise is considered equal to 6′ to 9′ additional run.

Wheeling is more economical than carting for distances under 100 yards.

An earth waggon holds as much as 20 or 30 wheelbarrows, and goes one-fifth faster—equal, therefore, to 24 or 36 barrows.

If loaded waggons have to be drawn up an ascent, and the temporary rails are in moderately good order, each foot of ascent may be considered equivalent to about 150 feet of additional horizontal distance.

BOXHORSE.

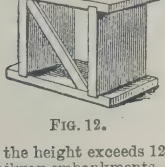


FIG. 12.

TRESTLE.

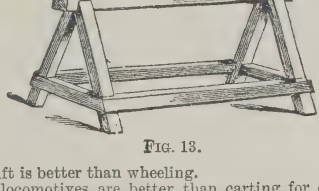


FIG. 13.

When the height exceeds 12 feet, a horse lift is better than wheeling.

For railway embankments and cuttings, locomotives are better than carting for distances over 1½ miles.

Tip Wagons.

End Tip Wagons (figs. 14 and 15) will hold 2 yds. filled up to top of sides, as with gravel, 2½ yds. when heaped up with clay, &c., the most economical size. The lowness of sides is of great advantage in filling.

The sides, one end, and bottom of body are of elm 1½″ thick (the earth being confined at other end by movable tailboard), bound together with iron straps at angles, and with upright

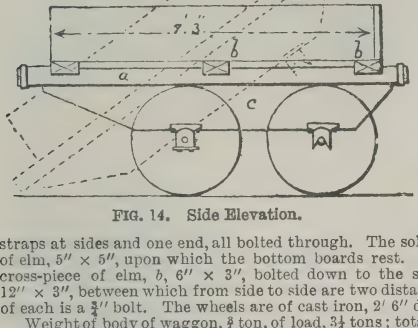


FIG. 14. Side Elevation.

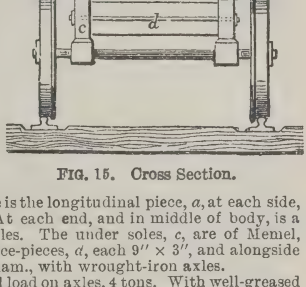


FIG. 15. Cross Section.

straps at sides and one end, all bolted through. The sole is the longitudinal piece, *a*, at each side, of elm, 5″ × 5″, upon which the bottom boards rest. At each end, and in middle of body, is a cross-piece of elm, *b*, 6″ × 3″, bolted down to the soles. The under soles, *c*, are of Memel, 12″ × 3″, between which from side to side are two distance-pieces, *d*, each 9″ × 3″, and alongside of each is a ¾″ bolt. The wheels are of cast iron, 2′ 6″ diam., with wrought-iron axles.

Weight of body of waggon, ¾ ton, of load, 3½ tons; total load on axles, 4 tons. With well-greased axles, resistance will be about 10 lbs. per ton, but may be as high as 20. Resistance, 80 lbs. per waggon. A horse can draw 150 lbs. 2½ miles an hour for eight hours; a horse can therefore draw two waggons.

Temporary contractor's rails weigh from 25 lbs. to 30 lbs. per yard.

It is generally better to throw earth away than to lead it three miles.

FOUNDATIONS.

In cases where there is any doubt as to the ground being of too soft a nature to bear the weight of the proposed structure, a trial pit should be sunk, or borings made with boring-rods; samples of the underlying strata may thus be brought up and examined.

Where a soft stratum overlies firm ground, the foundation should, if possible, be carried down to the firm ground; if this is not possible, piles may be driven to the solid ground and the building supported on them, or a number of piers of brickwork, masonry, or iron cylinders sunk.

Where ground is soft to a great depth, a wide trench filled with concrete may distribute the weight over a sufficiently large area to support it, or the trench may be filled with carefully rammed sand, prevented, by sheet piling or some other means, from escaping laterally. Another method is to drive a number of piles, which, by the friction of their sides, would bear the weight of the building.

Foundations on different Soils.

In any structure begun at different levels larger blocks should be used for the deeper parts, if the material admits of it, so as to reduce the number of mortar joints and the risks of unequal settlement.

With Sand or Gravel Foundation overlying Clay on a Slope, intercept water by drain on upper side of building.

Least Depth to escape Effects of Heat and Frost, from 3′ to 6′, according to climate. Not less than 3′ in England for ordinary soils, and 4′ for clay.

Rock.—Good foundation, but must be level; should be levelled, if necessary, in steps. The uneven parts should be filled up with large stones, firmly built with strong cement, or with concrete.

Gravel.—The best foundation.

Sand.—Good foundation, if dry and not liable to be washed away, but this easily occurs; drains with leaky joints may cause a subsidence, or any disturbance of the water-level in the stratum, whether by natural or artificial means, such as pumping operations connected with deep foundations, even at a great distance.

Clay.—Generally very treacherous and damp; the foundation must be deep.

Hard overlying Soft Ground.—If care is taken that the pressure per unit of area is not greater than the firm layer will bear, it may be wiser to build on it, sinking into it as little as possible.

Soft Ground overlying Hard Ground.—If the stratum of soft ground does not exceed 15′ to 20′ it will generally be cheaper to sink down to the firm ground; if not more than about 30′, drive piles or sink wells of masonry. If of indefinite depth, the platform must be supported by friction against the sides, and be therefore of considerable thickness.

Made Ground should never be trusted for the support of much weight, even though it may have lain undisturbed for years.

Chemical Treatment of Foundation Soils.—A method of increasing the bearing power of soft, sandy soils is said to have been employed with conspicuous success in connection with the construction of the Metropolitan Railway, Berlin. On a site composed of alluvial and diluvial sands, two solutions, one rich in silicic acid, the other containing soluble salts or acids, were successively injected under pressure into the soil. The first solution served solely to impregnate the soil, the second, by chemical reactions, causing solidification through the formation of silicates. The method is particularly applicable to soils containing acids and saline solutions, which exert a favourable influence on the chemical reactions involved. Its relative economic advantages have yet to be determined, the cost, principally attributable to the quantity of chemical material required, varying according to the site, and being comparatively lower with damp than with dry, sandy soils.

Load which Foundations will Bear.

Safe Load on Ordinary Foundations.

	Tons per sq. ft.		Tons per sq. ft.
1. Rock, moderately hard, strong as the strongest red brick	9·0	7. Loose beds with piling	1·82
2. Rock of the strength of good concrete	3·0	8. „ „ „ concrete	2·75
3. Rock very soft	1·8	9. Brick, stock, mortar	2·5
4. Moist clay and sand	1·36	10. „ „ Portland cement, 1 to 1	8·0
5. Coarse sand and dry clay	2·25	11. Concrete, Portland cement, 1 to 6	15
6. Firm stone on dry clay	3·18	12. Rubble on lias	4

Intensity of pressure on a rock foundation should at no point exceed one-eighth pressure which would crush the rock.

Buildings seldom press with a weight of more than 1 ton per square foot on foundations.

Safe Load on Materials.

	Cwts. per sq. in.	Tons per sq. ft.		Cwts. per sq. in.	Tons per sq. ft.
Portland cement, concrete, 5 to 1	2·0	15	Brick in cement	·8	5·75
Portland cement, concrete, 10 to 1	1·0	7·5	Granite	10·0	72
Mortar, ordinary	·5	3·5	Limestone	9·0	65
Brick in mortar	·5	3·3	Sandstone	5·0	38
			Rubble	·5	3·5

Safe Load on Bearing Surface for end of Girders.

	Tons per sq. ft.		Tons per sq. ft.		Tons per sq. ft.
Granite	15	Brick, blue, in cement	9	Brick, stock, in lime	4
Portland stone	12	Brick, stock, in cement	6	Earth, compact	2
Limestone	9	Brick, stock, in lias	5	„ made	1

Safe Load on Chimney Foundations.

	Tons per sq. ft.		Tons per sq. ft.
Granite	45	Concrete, 6 to 1	10
Brick, blue Staff., in cement	20	Clay	4

Safe Load on Stone Walls and Columns.

	Ashlar Walls, single bedstones. Columns, diameter = $\frac{1}{8}$ height. Lbs. per sq. in.	Block in course. Columns, diameter = $\frac{1}{12}$ height. Lbs. per sq. in.
Granite	712	570
Hard stone	356	280
Medium	214	142

Safe Load on Brick Walls and Columns.

	Walls not less than 18 ins. Columns, diameter < $\frac{1}{8}$ height. Lbs. per sq. in.	Walls under 18 ins. Columns, diameter < $\frac{1}{8}$ height. Lbs. per sq. in.	Columns, diameter = $\frac{1}{8}$ to $\frac{1}{12}$ height. Lbs. per sq. in.
Brick in mortar	72	36	—
,, ,, cement	108	72	—
,, ,, Portland cement	142	108	44
Rubble mortar	58	—	—
,, cement	72	—	—
Pressed bricks in mortar	128	114	108
,, ,, ,, cement	172	142	114
,, ,, ,, Portland cement	100	—	—

Safe Loads on Floors.

	Lbs. per sq. ft.		Lbs. per sq. ft.
Attics	31	Stairs	82
Dwelling-rooms	51	Workrooms	82
Library	72	Hay lofts	102
Dancing	72	Warehouses	112

Piled Foundations.

There are two kinds of piled foundations:—

1. *When the soil is soft to a great depth*, area should be enclosed with sheet piling before main piles are driven, to consolidate the ground better. When the tops of the piles are connected by timber, cross-pieces (12 ins. × 9 ins. for 12-in. piles) running across breadth of foundation are first notched on to heads of piles, and longitudinals, or *string pieces*, 12 ins. × 9 ins., running parallel to length of foundation, over them. On these latter, 3-in. or 4-in. planking (better diagonal) is laid to carry masonry. When upper stratum of ground is very soft, upper part of ground should be taken off, and filled in with stones or concrete or regular masonry. When of concrete, it should be 3 ft. thick for large buildings not excessively heavy.

2. *When there is hard ground below.*—In this case, as each pile resists partly by resistance to buckling as a column, the cross-section should be greater in proportion to the depth. Maximum depth may be put at 30 ft. The number of piles must be proportional to weight carried.

PILES.

Timber piles may be made of oak, elm, beech, larch, pine, or teak. Timber in sea-water is rapidly attacked by the wood borers *Teredo navalis* and *Limnoria terebrans*; creosoting will prevent their attack for a considerable time; it has also been suggested to cover piles with sheet copper or flat-headed copper nails. The head of the pile may be protected by a wrought-iron hoop, about 3 ins. × 1 in., to prevent splitting when being driven. The lower end should be pointed, the length of the point being from 1½ to twice the diameter, and where the ground is hard, protected with an iron shoe. Piles protected by shoes should have a blunt end 4 to 6 ins. in diameter. The shoes should have a solid point; the base being properly fitted to the blunt end of the pile prevents splitting by bolts which secure the shoes.

Concrete Piles.

Fig. 16 shows a 'Coignet' reinforced concrete pile. The reinforcement consists of longitudinal bars encircled by rings of round steel, with overlapping joints, spaced about 4 ins. apart.

For modern requirements concrete piles having a section less than 14 ins. square or equivalent area should never be used if over 30 ft. long; 12-in. square piles in many cases are quite large enough, but when it comes to actual practice it will be found that the extra cost of 14-in. piles

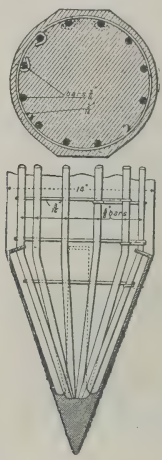


Fig. 16.

is more than compensated for by the extra convenience of handling. The pile shoe should be of uniform taper, the faces being at an angle of about 16° to the centre-line of the pile, with nose slightly blunted. The length of the cast-iron portion to be about half the total length of the taper, the area at the top of the cast-iron point being about 40 per cent. of the sectional area of the pile in the case of the 14-in. square piles. The vertical rods to rest on the cast-iron point and to be kept in place by the recess. The straight portions of the wrought-iron arms always to be embedded in the concrete of the pile and not kept flush or project from the body.

Piles made with a steel pipe driven to refusal and lined with concrete can develop not only the very considerable strength of the steel shell alone as an independent column, but the strength of the concrete core is materially increased by the shell acting as a continuous hooping, beside which there is an almost unlimited additional strength that can be obtained by the addition of vertical reinforcement bars. Such piles, says *Contract Record*, can endure very hard driving, can be thoroughly inspected inside, can have satisfactory bearing assured at the bottom, and can be reinforced to a high degree of strength comparable with massive structural steel columns. This reduces the number of piles and the cost of driving, and gives freedom from all uncertainty or possible deterioration from subterranean obstructions. The large factor of safety in the shell permits the safe application of working loads before the concrete filling is hard, which is often an important advantage.

The 'Simplex' System of Piling.

A cast-iron point is first driven to the required depth by means of a 16-in. steel pipe, $\frac{3}{4}$ in. or $\frac{1}{2}$ in. in thickness, stiffened with a collar at the top. The pipe is then filled with concrete and afterwards withdrawn slowly, the concrete filling adapting itself to irregularities in the ground. When more than one length of pipe is needed, welded joints may be preferred to riveted joints with cover plates. Instead of the permanent point or shoe an 'alligator' pile point may be used, consisting of a sleeve of annealed cast steel, about 17 ins. in diameter, and a pair of toothed jaws hinged to the sleeve in such a manner that when closed they form a flat cutting edge which penetrates the ground. When the pipe is drawn up a foot or more the jaws open, allowing the concrete to pass through to the bottom of the hole.

Moulded-in-Place Concrete Piles.

The Macarthur Compressed-Concrete Piles.

The moulded-in-place concrete pile has not to withstand handling before driving and, as it is formed in a mould which has taken the 'punishment' incidental to placing, does not require reinforcement, unless the pile is to be subjected to lateral stress, in which case only the minimum necessary to withstand such stresses need be placed.

The Macarthur piles are made in three different types, as shown by figs. 17-19.

In the first of these, fig. 17, the core and casing are driven into the ground, the core then being removed and the casing filled with concrete. The core is then replaced to compress the

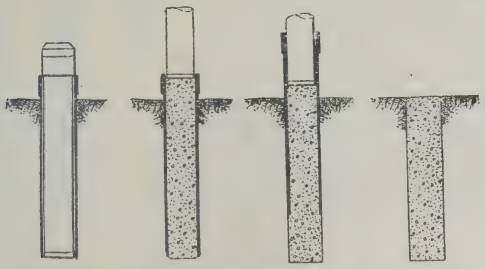


Fig 17.

concrete and the casing withdrawn, the core and hammer, weighing 7 tons or 8 tons, remaining on the concrete and forcing it into the space occupied by the casing and into any voids in the ground. Finally, a solid homogeneous mass of concrete is left in position in the ground.

The second type of pile employed is known as the compressed-concrete pedestal pile, and the method of placing piles of this type is shown in fig. 18. The core and casing are driven into the ground as shown on the left of fig. 139A, and, after removing the core, a charge of concrete is deposited in the bottom of the casing. After replacing the core, with the hammer

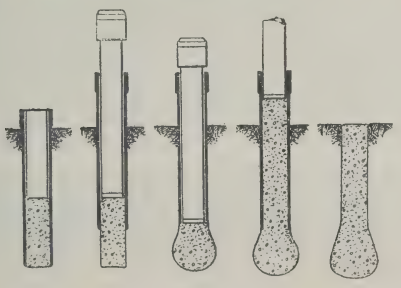


Fig. 18.

above it, on the concrete, the casing is withdrawn a distance of 18 ins. to 3 ft., and the charge of concrete is rammed out into the ground as shown. The core is then again removed, and the casing filled up with concrete, the casing being completely withdrawn as before with the core and hammer in position on the concrete. In this way a pile with an enlarged base is left in position, as shown on the right of fig. 18.

Where a fair stratum, insufficient to support an ordinary pile, lies between poor upper and lower strata, the compressed-concrete pedestal pile is used, but, under these conditions, one or more

piles should be load-tested to determine their carrying capacity. The piles, in such cases, are arbitrarily stopped in the fair stratum and the pedestal formed in the upper portion.

Another condition in which the pedestal proves valuable, is when the pile comes on to hard rock of a shelving formation. In such cases, the pedestal gives a full cross-section bearing on the rock, and also provides a very efficient means of eliminating any danger of side slip.

A further important field for the pedestal pile exists where it is necessary to hold down as well as to hold up a structure.

The third class of pile, of the composite compressed-concrete and timber type, is used for lengths exceeding 40 ft., and the method of placing these piles is shown in fig. 19. As before, the core and casing are first driven into the ground, and after withdrawing the core a 10-in. square timber pile is placed in the casing. This pile is then driven to the required depth, the casing acting as a guide. When this has been done the core is again withdrawn, and a batch of concrete is deposited in the casing, which is then withdrawn a certain distance after the core has been replaced on the concrete. The latter is then rammed so as to form a pedestal joint around the top of the timber pile as shown in the figure, and the pile is completed in a similar manner to that described above.

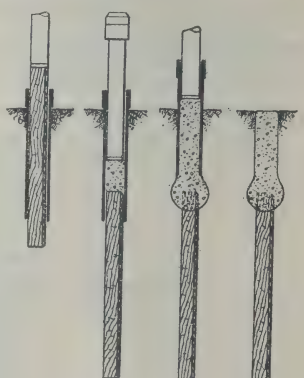


FIG. 19.

Loads Carried by Piles.

A timber or steel pile is usually of the same section throughout its length, and it is then only necessary to calculate the load borne by the greatest unbraced length of the pile, regarded as a strut or column. The part below ground may be regarded as held firmly or as quite free, according to the nature of the strata. Piles are often driven so closely together, or have to be of such strength for driving purposes, that they are more than strong enough for their loads. This also applies to piles which support their loads by skin friction only and are of ample section, or numerous, in order that the area subjected to skin friction may be great enough to support the loads. But it is often necessary for the sake of economy, especially when the piles are very long, to load them up to the safe limit.

The calculations of strength may then be made as follows:

(1) If the pile is unbraced above the ground, the length measured from a few feet below the surface of the ground to the head of the pile is treated as a long column, and the section at the mid-point must be sufficient (see Column formulæ and 'Reinforced Concrete').

(2) If it is braced, the length taken is, in the case of a uniform pile, the longest unbraced length; that is, from a few feet below the ground to the point of bracing, or from the point of bracing to the head of the pile.

(3) If the pile is not uniform, the strength of the section at the mid-point of each of these lengths may have to be calculated separately, each length being treated as a long column.

(4) Bracing must not be considered effective unless it is effective in both directions, except in the following case:

(5) When the dimension of the pile is greater at right angles to the bracing than it is in the direction of the bracing, the two part lengths must be calculated as struts liable to cripple in the direction of the smaller diameter, and the whole length as a strut liable to cripple only in the direction of the larger diameter. If the formula for a square section of a reinforced concrete pile be used, the permissible load, expressed as a proportion of the short column load, may be thus modified. In each case modify the factor with l^2 in it, by multiplying by $\frac{r^2}{r_1^2}$ for the smaller dimension, and by $\frac{r^2}{r_2^2}$ for the larger dimension; where r = radius of gyration of a square section of the same area, r_1 = the smaller actual radius of gyration, and r_2 the larger actual radius of gyration. In the case of reinforced concrete piles, use the sums of the squares of the radii of gyration of the concrete and of the steel as in the formula (see 'Reinforced Concrete'). In this case the outside dimension might be the same, but the reinforcement stronger in one direction than in the other.

(6) For the portion of the pile below the ground:—

If the pile is in firm ground, or passes through a sufficient number of firm layers so that it is effectively steadied, any section of the pile should have sufficient strength, calculated as a short

column; that is, it should take a load equal to the safe stress multiplied by the area. The load will be the whole load *less* an amount equal to the area of the pile above the section in question multiplied by the coefficient of friction. This will usually range from about 1 to 4 cwts. per square foot.

If the ground into which the pile is driven is soft and yielding, and is not regarded as effectively steadying the pile, the strengths at mid-points of each length $t_1, t_2, \ldots$ must be calculated as those of the mid-section of a long strut or column of that length. These lengths are measured from the head of the pile, or from the point of effective bracing, to successive points in the length of the pile, the last being measured to the tip. The load in each case is the whole load less the load carried by friction on the length of pile above the point in question. Unless the tip of the pile rests on a hard stratum, or is driven some way into a very firm stratum, the load vanishes before the point is reached, and very few calculations will actually be required.

(7) If the pile so rests upon a firm stratum that a large part of the load is carried from the lower end, the same procedure may be adopted, but the loads will, of course, be greater.

If the length of the pile below ground is small, and it is driven into or rests on a hard stratum, the whole length may have to be calculated as a strut carrying the whole load. Such length as is actually in firm ground should, however, be subtracted from the length for calculation.

In some cases the support given by the ground may not be regarded as equivalent to effective bracing, but may be considerable. The strength of any section may then be calculated in the first place as though the soil were quite firm and hard, giving a load W_1, and next as though the soil, though exerting the effect of skin friction, had no bracing effect, giving a load W_2. The actual load will be something between W_1 and W_2, depending upon the estimated value of the soil as a steadier of the pile. Even soft silt must be regarded as exerting a considerable steadying effect; but if there be progressive movement of silt, sand, or peaty ground, no steadying effect can be counted upon. Considerable movement, of course, demands the provision of special bracing, or, with short piles driven into a firm stratum, the calculation of a cantilever effect based upon the observed pressure as recorded by a torsion pressure meter or otherwise estimated.

Generally: to find the necessary strength at any section S, measure the distance upwards to the head or bracing, at A, and an equal distance downwards to a point B. Calculate the load P carried by skin friction above B, and (W — P) is the load carried by AB as a strut. The strength at S will be calculated as that of a short strut if most of the length BS be effectively steadied by the ground; if there be no such steadying the calculation will be that for a strut of length AB (but in no case any greater length). If the lower half of SB is in firm ground, or all of it in weak ground, the section may be calculated for a stress somewhere between that allowable as a short strut and that allowable on the length AB as a free strut.

The strength of the cross-sections of a pile does not change suddenly, and very few calculations need be made in actual practice; but any of the above may be necessary in a particular case.

The two most probable errors are, first, to calculate as free struts lengths which are really steadied by the ground, and, secondly, to neglect making for each section the reduction of load (regarding the section as the mid-point of a strut) due to skin friction from the surface of the ground to a point which is as far below the section as the bracing or head of the pile is above it. The load which the length in question itself carries by skin friction is imposed, upwards, upon some other mid-section nearer the head of the pile.

Loads on Reinforced Concrete Piles.

A well-designed reinforced concrete pile, 14 ins. by 14 ins. in section, will carry a load up to 100 tons (calculated). In actual practice such piles are carrying loads of 60 to 70 tons (at Bristol up to 65 tons; in the Thames, 15-in. × 15-in. piles, 70 tons).

Pile-sinking with the Water Jet.

In sinking piles the work may sometimes be much facilitated by the use of a water jet. In the case of hollow piles, the usual shoe may be replaced by a funnel-shaped end-piece, and the water pumped through the pile. In sinking cast-iron piles for bridge piers in New South Wales (*E. M. de Burgh*) it was found that a central nozzle with an aperture of 2 ins., and four holes of ½ in. diameter in the conical portion, gave good results in sand and soft clay, the piles being sunk through 14 ft. of sand in an average time of 6 minutes, and the actual rate of movement was about 8 ft. a minute. For sinking in sand not under water it was found expedient to flood the surface round the pile. In clay, the pile was loaded with 10 tons, and after 5 or 10 minutes

pumping the pile would sink 6 to 9 ins. At a depth of 11 ft. in the clay, progress was very slow and practically ceased at 13 ft., with the 10-ton load. These piles had flanged joints.

Timber piles can be similarly sunk, the pipe being taken down the side of the pile. On the work referred to above, timber piles, 14 ins. in diameter at one end and 9 ins. at the other, were sunk 16 ft. on an average; the average time taken being 4 minutes, and the water pressure 5 to 7 lbs. per sq. in. in sand and 30 to 40 lbs. per sq. in. in clay. The pipe (2-in.) was taken to a $1\frac{3}{4}$-in. augur hole in the point of the pile.

In driving steel sheet piling at Hodbarrow (*H.S. Bidwell*) a water jet was used in hard ground. Double ram pumps $5\frac{1}{2}$ ins. and 6 ins. in diameter were used, with 10 to 14 h.p. boiler capacity, the water being delivered by a 5-in. pipe, diminishing to 3-in. and to $1\frac{1}{2}$-in. From this a $1\frac{1}{4}$-in. vertical pipe led to two $\frac{3}{4}$-in. jets. The jet pipes were kept working up and down, with a block and tackle, about level with or slightly below the point of the pile. After three or four tides, the ground had recovered its normal consistency.

Pilnig Hammers.

SINGLE SEMI-AUTOMATIC DIRECT-ACTING STEAM PILING-HAMMERS.

(*The British Steel Piling Co.*)

This type of hammer, which is suitable for every description of pile driving, consists of a cylinder of cast iron raised by steam pressure and allowed to fall through gravity on the pile. The steam is admitted to the cylinder through a hollow piston rod by means of a piston valve, worked by a lever, at the upper end of this rod; steam being led to the valve by a flexible tube. The lower end of the piston rod, which is solid, passes through the bottom end of the cylinder, whilst the piston is between, the whole being made of one steel forging drawn down from an ingot and bored from the solid. The piston is fitted with rings, whilst the ports are placed on the upper face of the same and lead into the hollow piston rod. When the lever is pulled down, it allows the steam to pass down this hollow rod through the ports into the cylinder, when the cylinder immediately rises, the pressure being between the piston and the cover. As soon as the cylinder is raised to the required height, the lever is released and the piston in the valve forced back to the exhaust position, thus cutting off the incoming steam and allowing the steam in the cylinder to escape to the atmosphere. When the operation is repeated the pistons in the valve, having differential areas, the tendency is always to keep the valve at the exhaust position. The piston valve is of very heavy pattern and made of gun-metal throughout, and fitted with phosphor-bronze piston rings. These hammers are so simple in their construction, and so strongly built, that spare parts are rarely required. They do not wet the top of the pile and cannot overrun their stroke owing to a safety hole, placed in the lower end of the cylinder; whilst the whole weight of the hammer is always on the pile. The exhaust is particularly free, so that there is practically no cushioning in the cylinder and consequent loss of blow. One man can easily operate the hammer and maintain a speed of 50 to 60 blows per minute continuously. These hammers are made in sizes from 10 cwts. to 5 tons.

THE MCKIERNAN-TERRY PATENT DOUBLE-ACTING AUTOMATIC PILE-HAMMERS.

With these hammers 90 lbs. pressure gives a striking blow of 7,800 ft.-lbs. per min., at 140 strokes per min. A 40 h.p. boiler is required. The net weight of the hammer is 6,800 lbs., and the cylinder stroke 16 ins.

Safe Load on Piles.

Rankine gives safe load on piles as follows:—
For piles driven till they reach firm ground, 1,000 lbs. per sq. inch of area of head.
For piles standing in soft ground, by friction, 200 lbs. per square inch of area of head.

FORMULÆ FOR SAFE LOAD ON PILES.

(1.)

If P = safe load on a pile; h = height of fall in feet;
w = weight of ram; s = penetration per blow in inches
(P and w must be in the same units—i.e. both in lbs. or both in tons, for example),

$$P = \frac{2wh}{s+0{\cdot}1}.$$

(2.)

If P = safe load on pile; a = sectional area of pile;
w = weight of ram; E = modulus of elasticity;
h = height of fall; x = weight of pile;
s = depth driven by last blow; l = length of pile.

Rankine's formula,

$$P = \sqrt{\left(\frac{4Eawh}{l} + \frac{4E^2a^2s^2}{l^2}\right)} - \frac{2Eas}{l}.$$

Sanders' formula,

$$P = \frac{wh}{8s}.$$

Weisbach's formula, $$P = \frac{w^2 h}{s(w + x)}.$$

In Rankine's formula the safe load is taken to be about $\frac{1}{6}$ of the greatest load; if there is a great vibration, the factor of safety should be increased to 8 or 9; where there is no vibration, a factor of safety of 3 or even less may be adopted.

'Engineering News' (U.S.A.) formula, $$P = \frac{fwh}{p + 1}.$$

Where p is the average penetration in the last six blows; f a factor ranging from 1 to 12 and recommended to be taken as 2.

Screw Piles.

Screw Piles, figs. 20 and 21, are either of wood or of cast or wrought iron; they are seldom less than 6, or more than 18 inches, in diameter. They should be cylindrical in section to enable them to be more readily turned. They are fitted with a screw disc at the foot of the pile, similar to an auger, which usually does not make more than one and a half turns. This

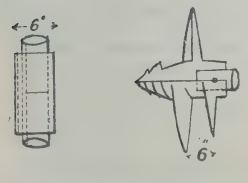


FIG. 20.—Wrought-iron pile with cast-iron disc.

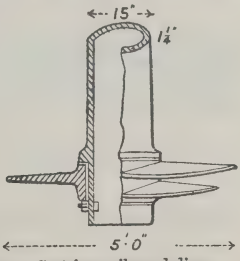


FIG. 21.—Cast-iron pile and disc.

disc is of cast-iron, which is made up to 6 feet in diameter. The piles are usually sunk by manual power acting on long levers fixed to a capstan head on the piles, so as to screw them down to a firm stratum. When screw piles have to be sunk through hard ground, a water jet may be used to facilitate sinking.

Cylinders.

Cylinders of cast-iron are sunk to a firm foundation by excavating inside and letting them sink by their own weight, or if that is not sufficient, adding additional weight. As the cylinder sinks, lengths are bolted on by means of flanges which are on the inside. When it is desirable to make the joints water-tight, a cloth strip saturated with thick paint is laid between the flanges.

Small cylinders from 4 to 10 feet in diameter are usually cast in lengths of from 5 to 10 feet, without any vertical joint, and from 1 to 1¼ inches in thickness with 1¾ or 2 inch internal flanges. The thickness in inches necessary to withstand the pressure of the water of small thin cylinders may be approximately found from the formula—

$$t = \cdot 001\, rh;$$

where r is the radius of cylinder, and h depth of water, both in feet, and assuming that the metal will stand with safety a compression of 5,000 lbs. per square inch. In no case should the thickness t be less than ½ inch. The inside of the cylinder is excavated by hand, or if the water cannot be kept out, by means of a grab, and is afterwards filled up with concrete or masonry.

At the St. Louis Bridge, U.S.A., the cylinders were 4, 6, and 8 feet in diameter, cast in 10-feet lengths, the metal being 1¼ inch. thick with 2¾-inch flanges.

The bottom cylinder of the Albert Bridge over the Thames was 21 feet in diameter and 4 feet 6 inches high, cast in one piece; on this was a reducing ring tapering to 15 feet in diameter; all the other rings were 15 feet in diameter and 6 feet high, also cast in one piece.

Brick or masonry cylinders built on an iron or wooden curb are frequently used instead of iron cylinders.

Pneumatic Process of Sinking Cylinders.

Cylinders and large caissons are often sunk by the pneumatic process. The cylinder is made as nearly as possible air-tight, and an air-lock or chamber, with two doors, one opening from the outer air and the other into the cylinder, is bolted on to it; both of these doors open downwards into the cylinder or against the pressure of the air. Air is pumped into the cylinder so as to drive the water out. The workmen then enter the air lock, the door communicating with the outer air is closed, and the compressed air allowed to enter from the cylinder; when the pressure has become equal the door into the cylinder is opened and the workmen descend. The excavated material is carried up through the air-lock, or blown up through a tube by air pressure. The

cylinder is buoyed up by the pressure of the air inside it, and when this is released it tends to sink. It has generally been considered that men cannot work in safety under a greater depth of water than 100 ft.; in the St. Louis Bridge 108 ft. was reached, and 147 ft. under mean low water is said to have been attained in the East River Gas Company's tunnel at New York.

Steel Sheet Piling.

BRITISH PRACTICE.

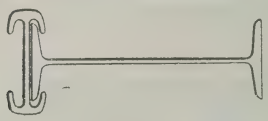


FIG. 22.—Universal Joist Section.

British standard joists are employed, having flanges 5 ins. wide and used in conjunction with a standard clutch or locking bar. The ranges of joists are 15 ins. by 5 ins., 12 ins. by 5 ins., 10 ins. by 5 ins., 8 ins. by 5 ins. and 6 ins. by 5 ins. The clutch is put on the flange of joist at works and the two driven as one. The section has great strength, and is very easily straightened for re-use and possesses high salvage value, as the joists can be used for constructional purposes. Supplied up to 65 ft. lengths.

Simplex Section, 12 ft.; 26·5 lbs. per sup. ft. Simplex Section, 8 ft.; 22 lbs. per sup. ft.

FIG. 23.

These sections consist of one member only and are used for lighter forms of construction. They can be obtained in lengths up to 40 ft.

With double-acting hammers, the cost of driving steel sheet piling, as against timber, is usually about one-fourth, and frequently as low as one-twentieth. In a recent example, a No. 6 double-acting hammer (McKiernan-Terry), at Larne, drove 845 12-in. 'Simplex' piles, in lengths varying from 33 ft. to 15 ft., in seventeen working days.

RANSOME TYPE 'D' STEEL SHEET PILING.

STANDARD SECTION.

Section.	Weight per ft. run.	Weight per sq. ft. Interlocked.	Sectional Area.	Moment of Inertia.	Radius of Gyration.	Section Modulus.	Dimension 'B.'
Standard	33 lbs.	28 lbs.	9·6 sq. in.	12·25 in.^{4}	1·12 in.	7·66 in.^{3}	$3\frac{3}{16}$ in.
	49 kgs. per metre	136·71 kgs. per sq. metre	6,200 sq. mm.	5,100,000 mm.^{4}	28·5 mm.	126,000 mm.^{3}	80·96 mm.
Uniform	31 lbs.	26·25 lbs.	9·12 sq. in.	11·9 in.^{4}	1·14 in.	7·32 in.^{3}	$3\frac{1}{8}$ in.
	46 kgs. per metre	128 kgs. per sq. metre	5,876 sq. mm.	4,953,000 mm.^{4}	28·9 mm.	119,000 mm.^{3}	79·37 mm.
Uniform light	26·88 lbs.	22·6 lbs.	7·76 sq. in.	7·152 in.^{4}	0·945 in.	5·031 in.^{3}	$2\frac{11}{16}$ in.
	40 kgs. per metre	110·4 kgs. per sq. metre	5010 sq. mm.	2,860,000 mm.^{4}	24 mm.	80,000 mm.^{3}	68·26 mm.

This section (fig. 24) is claimed to be the strongest British piling in proportion to its weight per sq. ft., the strength being due to the trough-shape, with a neutral axis running through the centre of the pile, as shown in the figure.

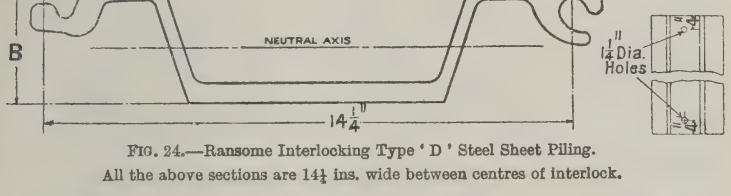


FIG. 24.—Ransome Interlocking Type ' D ' Steel Sheet Piling.

All the above sections are 14¼ ins. wide between centres of interlock.

The interlock is water-tight, without clay puddling, and it is sufficiently flexible to enable circular cofferdams to be built as small as 5 ft. 3 ins. in diameter, without using special piles. The safe working stress is 16,000 lbs. per sq. in.

The advantage claimed for this piling is that, in view of its large covering capacity in proportion to its weight, one-third less tons of the piling are required to cover a given area, so that about one-third less has, consequently, to be paid for freightage.

The piling is made of special toughened steel that penetrates the hardest ground without damage to the piling. It is specially easy to drive and extract with Ransome plant, and can be used repeatedly.

Some of the classes of work for which Ransome type ' D ' Piling is used are cofferdams, caissons, dams, retaining walls, lock, dock, harbour, breakwater, canal and reservoir work, bridge and other foundations, shaft-sinking, etc.

' SIDE-GROOVE ' STEEL PILING.

(Side Groove Steel Piling Supply Co., Ltd., 17 Victoria Street, Westminster, London, S.W. 1.)

These sections (fig. 25), known as ' Side-Groove Steel ' Piling (Annison's Patents), have an independent locking bar of heavy design, and consequent great tensional strength in the joints of the assembled piles. The area covered and the advance of the work per driven pile exceeds all other types; time and cost in driving them is therefore comparatively low. These sections are very adaptable, and special piles, such as corner-pieces, sections, etc., can readily be made. The heavier types can be rolled up to 61 ft., and the lighter types up to 40 ft. For permanent work, selection can be made from the thicker sizes listed, or the piles can be rolled at a small extra cost in special rust-resisting steel.

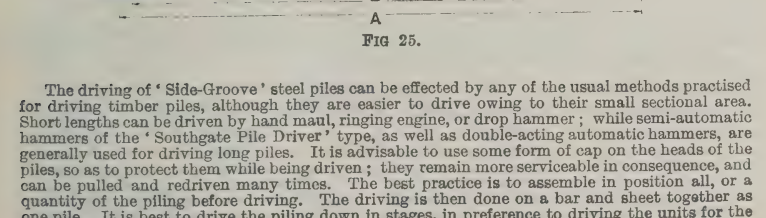


FIG 25.

The driving of ' Side-Groove ' steel piles can be effected by any of the usual methods practised for driving timber piles, although they are easier to drive owing to their small sectional area. Short lengths can be driven by hand maul, ringing engine, or drop hammer; while semi-automatic hammers of the ' Southgate Pile Driver ' type, as well as double-acting automatic hammers, are generally used for driving long piles. It is advisable to use some form of cap on the heads of the piles, so as to protect them while being driven; they remain more serviceable in consequence, and can be pulled and redriven many times. The best practice is to assemble in position all, or a quantity of the piling before driving. The driving is then done on a bar and sheet together as one pile. It is best to drive the piling down in stages, in preference to driving the units for the full length at one operation.

DIMENSIONS AND WEIGHTS OF 'SIDE-GROOVE' STEEL PILING.

Section of Piling.	Dimensions in inches.						Weight per sq. ft. of wall.
	A	B	C	D	E	F	
Heavy Section :							Lbs.
Sheet $\frac{1}{2}$ in. thick . .	$21\frac{3}{4}$	$20\frac{1}{8}$	$\frac{1}{2}$	$4\frac{9}{16}$	$4\frac{7}{8}$	$\frac{3}{4}$	44·7
,, $\frac{9}{16}$ in. ,, . .	$21\frac{3}{4}$	$20\frac{1}{8}$	$\frac{9}{16}$	$4\frac{9}{16}$	$4\frac{7}{8}$	$\frac{3}{4}$	47·1
,, $\frac{5}{8}$ in. ,, . .	$21\frac{3}{4}$	$20\frac{1}{2}$	$\frac{5}{8}$	$4\frac{9}{16}$	$4\frac{7}{8}$	$\frac{3}{4}$	49·5
Light Section :							
Sheet $\frac{13}{32}$ in. thick .	$21\frac{1}{2}$	$20\frac{7}{8}$	$\frac{13}{32}$	$2\frac{31}{32}$	$2\frac{3}{8}$	$\frac{11}{32}$	25·0
,, $\frac{1}{2}$ in. ,, . .	$21\frac{1}{2}$	$20\frac{7}{8}$	$\frac{1}{2}$	$2\frac{31}{32}$	$2\frac{3}{8}$	$\frac{11}{32}$	29·0

AMERICAN PRACTICE.

The following are three principal varieties. In the first form, (*a*), (*b*), and (*c*), fig. 26, the section is symmetrical with respect to a longitudinal axis, but unsymmetrical with respect to a transverse axis. In the second form, (*a*) and (*b*), fig. 27, the reverse is the case. In the third

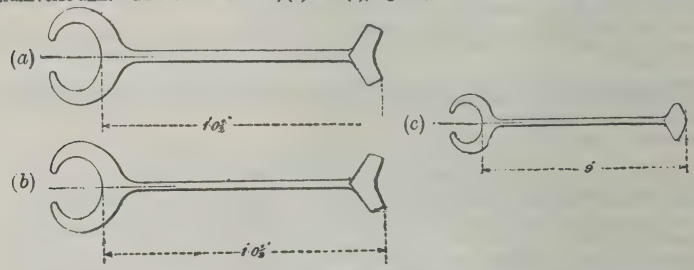


FIG. 26.

form, either of the two pieces together form a unit symmetrical to both axes. All the forms may be used over and over again as steel sheet piles.

Sheet piles of the rolled forms may be secured from the mills in lengths up to 45 ft. for one or two light sections and up to 70 ft., or perhaps more, for other weights.

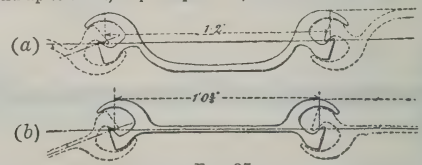


FIG. 27.

In fig. 28 is shown a simple steel plate bent round a vertical axle of curvature and given suitable interlocking hooks at the sides. The curvature adds to the rigidity. This piling is assembled with concavities and convexities alternating.

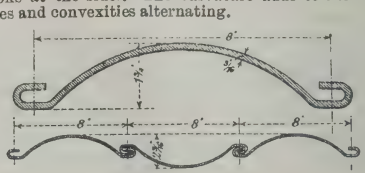


FIG 28.

Another type of steel-plate sheet piling has vertical corrugations, such as shown in fig. 29. These corrugations are made on radii of curvature of but an inch or so. The interlock is made by overlapping edges and securing one or both by vertical straps riveted on to the piling. Sometimes the straps are not continuous, but are broken up into short and separated lengths. In driving piling of these classes it is usual to hammer shut the interlock opening on the

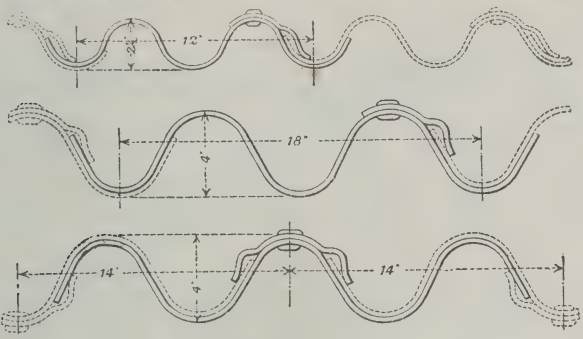


FIG. 29.

forward side at the bottom of the pile about to be driven. This tends to prevent rupture or distortion and to keep the interlock clear. When the next pile goes down it opens out the inter-locking space. The first two rolled forms are obtainable with ¼-in. webs. The second form is now obtainable with a web thickness of $\frac{7}{32}$ in. Plate steel sheet piling is obtainable in still thinner metal.

Driving.

The driving of the steel sheeting is generally effected by means of double-acting pile hammers, the McKiernan-Terry being a usual type. The blows are given in rapid succession.

Light-weight sheeting, and also heavy piling, may at times be very advantageously driven with a water jet. Such material as mud, soft clay, and clear sand will ordinarily respond to the jet. If, however, the material is hard clay or a sand containing a heavy percentage of gravel, the water jet is an inadvisable adjunct. A supply pipe of 1 in. diameter and a ¾-in. nozzle will be suitable for moderate weights of piling having a width of, say, 13 ins. or less.

Pulling.

It is quite customary to pull piles that have been used in temporary service. The cost of pulling is considerable. While, in general, it is not as great as the cost of driving, still it is to be rated as equal to a large percentage of the expense of driving the very same piles which it is proposed to pull out again. This percentage may rise as high as 75, or even higher. Where the piling has been driven with a wooden maul, or could have been so driven, a simple lever may suffice. This lever may have attached to it metal straps or other devices to facilitate connecting the pile head with the lever. Simple perforations in the top of the pile serve for the introduction of bolts or pins. Ordinary hoisting arrangements may suffice for these piles and for still heavier sheeting. Hydraulic jacks have been used for rather difficult pulling. The jacks may be used mainly for starting the units. By setting up jacks on the heads of the adjacent piles they may be brought to bear beneath a head block provided with a slot through which the intermediate pile may pass on its way up. Another method that has been used simply sets up a framework provided with a strong cross-piece, to which blocks and tackle may be rigged. The line may be operated by hand or by power, as circumstances may require. For pulling piles which had been put down to a considerable depth, a percussion-piston pile-driving hammer was hung upside down from a derrick boom and used thus to pull the piles. In pulling, the inverted hammer takes hold of the pile by means of the shackle, and steady tension is then exerted on the pile line, which pulls the shackle strap down upon the anvil block, and the hammer is placed in operation. Piles of 26 ft. penetration in clay, sand, and gravel were withdrawn in from 2 minutes to 10 minutes each, without distortion. The McKiernan-Terry pile hammers can also be used for pulling purposes.

Cost Data.

The following are the cost data relating to the driving of 19,654 lineal feet of 12-in. rolled piling of the first type.

The piling weighed 40 lbs. per lineal foot. The material penetrated was sand and clay, and the piles varied in length from 40 ft. to 60 ft. Twenty-one working days of 10 hours each were

consumed on the work. Maintenance and operation are put at $7 per day. Nine men, including a foreman, an engineer, and a fireman, constituted the crew. Their combined daily wage amounted to $28.50. The expense for 21 days, including a $60 item for handling and incidentals, totalled $805.50. The total cost per foot of penetration is given as $0.041. Estimated on the basis of the weight driven, the cost was $2.05 per ton of 2,000 lbs.

Caissons.

Very deep and large piers and abutments of bridges have recently been constructed with steel caissons braced internally so as to prevent them collapsing while sinking. These caissons are excavated and filled with concrete or masonry.

The foundations of the Hawksbury Bridge in Australia were sunk to a depth of 162 ft. below high-water level by dredging.

FRICTIONAL RESISTANCE OF PNEUMATIC FOUNDATIONS (SCHMOTT).

Tables of Coefficients of Friction.		Dry Material.		Wet Material.	
Material.	Soil.	First Movement.	During Motion.	First Movement.	During Motion.
Sheet-iron, without rivets .	Gravel and sand	·4015	·4583	·3348	·4409
Sheet-iron, with rivets .	,, ,,	·3965	·4911	·4677	·5481
Cast-iron, unplaned . .	,, ,,	·3677	·4668	·3646	·4963
Granite, roughly worked .	,, ,,	·4266	·5368	·4104	·4800
Pine, sawn	,, ,,	·4088	·5109	·4106	·4985
Sheet-iron, without rivets .	Sand	·5361	·6313	·3655	·3247
Sheet-iron, with rivets .	,,	·7269	·8391	·5156	·4977
Cast-iron, unplaned . .	,,	·5636	·6063	·4744	·3796
Granite, roughly worked .	,,	·6473	·7000	·4728	·5291
Pine, sawn	,,	·6633	·7340	5787	·4793

The foundation must be vertical, being kept in equilibrium by the friction of its surface.

According to Mr. F. W. Sweeney in the *Journal of the Western Society of Engineers*, experience n sinking concrete caissons for bridge foundations has shown that building the walls with a batter is not only of no particular advantage in reducing the sinking friction, but it is positively detrimental to the proper guiding of the caisson. It is considered the best practice to have a perfectly plumb wall. When the depth of water will permit, it is generally cheaper to build up a river caisson on shore in the form of a wooden shell, launch it, tow it into position between guide piles and sink it by filling with concrete.

Freezing Process of Sinking Foundations.

The Freezing Process has been used to sink large foundations in treacherous material, especially quicksand. Pipes 10-inch in diameter, open at the bottom, are sunk vertically about 3 feet apart round the ground to be excavated ; inside of these 8-inch pipes, closed at the bottom, are lowered, and again within the latter small pipes. The freezing material is pumped into the smaller pipes, and returns to the cooling tank through the large ones. A solid cylinder of frozen ground is gradually formed round the pipes ; these frozen cylinders unite, forming a solid ring round the place to be excavated. The material to be removed will, of course, be frozen solid, and therefore much more difficult to excavate.

DAMS.

If the depth of the water does not exceed 4 feet, the bottom moderately firm, and there is no current, a dam of clay puddle about 3 feet wide on top may be formed round the space to be enclosed ; before the dam is commenced care should be taken to remove all loose porous material and roots from the site, and, if possible, excavate a trench for its foundation. Long lengths of waterproof canvas, sewn together and weighted with a chain stitched to the ends so as to keep it in place, have also been used. The canvas should be supported by a framework of timber (see fig. 30). These frameworks are spaced about 8 feet apart and sheeted with boarding between. The ground should be dredged level to receive them.

Puddle Dams.

Coffer dams of puddle usually consist of two parallel rows of sheet piling, the intervening space between them being filled in with puddle; sometimes three or even four rows, with a

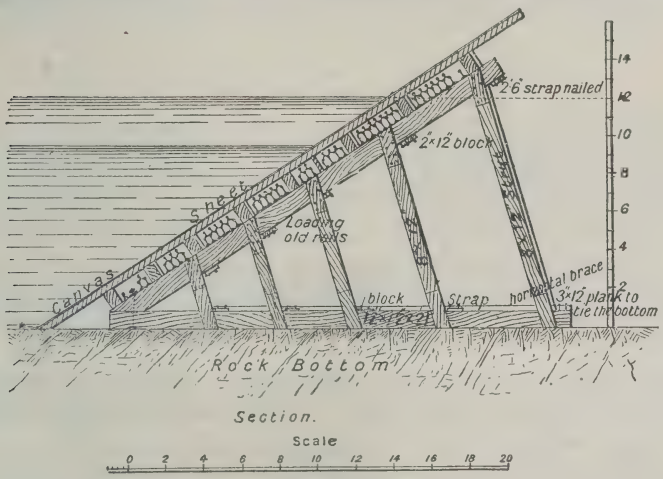


FIG. 30.

puddle wall between each, have been adopted in order to give greater stability and make the dam more watertight.

It is always desirable to make the puddle wall as thin as possible; it is seldom more than

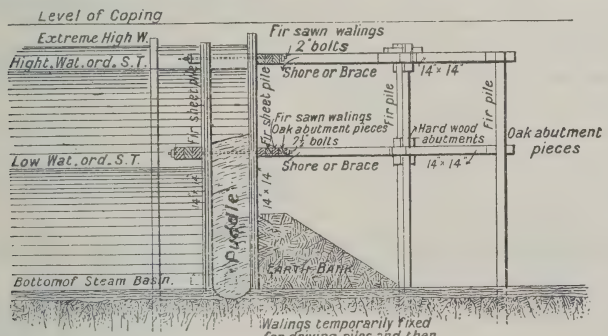


FIG. 31.

4½ or 5 feet in thickness. Bolts through coffer dams should be avoided as far as possible, as the puddle in settling leaves vacant spaces along them, causing leaks; to remedy this, iron discs through which they pass have been used; these discs are embedded midway in the puddle wall.

in coffer dams made for the construction of the East London Railway through the docks, through bolts were entirely dispensed with, their place being taken by buttress piles driven at intervals on the outside which withstood the pressure of the puddle.

Fig. 31. shows a coffer dam used at Portsmouth Dockyard extension. The shoring piles were driven 9 ft. centre to centre, and 12-in. by 12-in. struts, one horizontal and one inclined, were fixed to each.

Puddle.

(*Alfred B. Searle.*)

Puddle is a highly plastic paste made by mixing 'clay' with about one-fifth of its weight of water, the precise amount depending on the plasticity and dryness of the 'clay.' A perfectly pure 'clay' does not usually produce good puddle, as it is either deficient in plasticity (like china clay) or it shrinks excessively. The former defect can only be remedied by the use of a more highly plastic material; excessive shrinkage may be prevented by mixing a suitable proportion of sand with the clay. For the best results, a piece of puddle when dried should not shrink more than $1\frac{1}{2}$ in. per linear foot; on the other hand, it should not shrink less than $\frac{3}{4}$ in. per linear foot or it will probably not be sufficiently impermeable to water.

The materials used must be carefully selected and thoroughly well mixed; some contractors are very careless in this respect and will use anything which has been passed through a pugmill. The correct consistency for good puddle is that at which that paste can be squeezed easily in the hand without any appreciable quantity adhering to the latter when the pressure is released. Puddle made of the consistency of porridge is much too soft for best work, though it is better than an over-dry or badly mixed stiff paste.

Where possible, the materials used for making puddle should, before being mixed with water, be passed between a pair of rollers placed not more than $\frac{1}{2}$ in. apart, so as to crush any stones or coarse gravel present. It is also desirable that all roots and large stones should be picked out before sending the 'clay' to the mixer.

The usual method of preparing puddle is to mix the local 'clay' with sufficient water to form a soft paste, a pugmill being used for the mixing. The 'clay' may consist of a sandy loam with 50 per cent. of sand and, therefore, incapable of producing a mass impermeable to water except within extremely narrow limits of consistency, which are seldom investigated. On the other hand, it may be so rich in colloidal matter as to have an abnormally high shrinkage on drying; this is a very objectionable property, as puddle which shrinks greatly usually cracks and becomes unreliable. It may be prevented by using a plastic mixture of 'clay' and very fine sand which is well rammed into position, the proportion of sand being such that the puddle, when fully dried, has a shrinkage not exceeding 1 in. per linear foot. No definite figures can be equally suitable for all cases, but two parts of clay to one part of fine sand is often suitable, though much depends on the nature of the clay.

Too much water should not be used in making puddle; it is often better to use moist clay mixed with sand rather than to add sand to a clay-slurry in order to stiffen it. The stiffer puddle must be mixed mechanically in a powerful pugmill, and it must be rammed in position by mechanical rammers. Under these conditions it produces excellent results.

To produce a really satisfactory puddle, it is not sufficient to proceed along 'cookery-book' lines. The available materials should be mixed in different proportions, and each mixture should be tested in a hydraulic press to ascertain its permeability. A useful 'holder' for the puddle is a hollow cylinder made of Portland cement and sand, or even a 4-in. salt-glazed drain-pipe, into one end of which the puddle is placed. The cylinder or pipe is then placed vertically on a flat bed and the pressure applied. A little skill is required to make the puddle fit properly, but once this has been acquired, the test is easy to make.

The worst kind of puddle is that made by preparing a liquid slip, wash, or slurry of clay, and letting this settle. The sand and other non-plastic material normally present is removed in this method of preparing the slip, and the product has too great a shrinkage.

In trenches filled with properly made puddle, there appears to be no advantage in using a 'reinforcement' of strings of jute, as the plain puddle is quite impermeable when properly prepared. A mixture of wood-wool and clay makes an excellent puddle, but requires a machine for its production.

Permeability of Different Puddles.

Tests made under hydraulic pressure to determine the permeability of different puddles showed that it is possible to produce mixtures which, when only 6 ins. thick, will withstand 50 lbs. per sq. in. water-pressure for 5 hours without showing any signs of seepage. Puddles free from sand stood this test the best because of their greater density, but those containing sand are so much less liable to crack that, on the whole, they are the most suitable for general use.

Single Balk Dams.

Dams have also been most successfully constructed of single rows of sheet piling grooved and driven between waling pieces. Fig. 32 is the more usual arrangement; the piles are from 9 to 12 inches in thickness; a groove about 2 inches wide and 2 inches deep is cut out of each

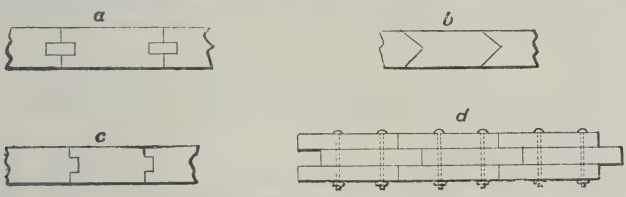


FIG. 32.

pile; into this is fitted a tongue, which is cut to the exact size to fit it. Sheet piles should be sawn die-square with the grooves perfectly straight and true. They should be of sufficient thickness to withstand the water pressure without bulging between the waling pieces. Small leaks at the joints may be stopped by caulking with oakum.

Cribs.

Timber cribs or boxes, which are usually made watertight and floated into position and then sunk, have been largely used in America for foundations of bridges on rock, or in places where there is no fear of undermining by the currents. They are filled with concrete.

Scaffolding.

Standards are upright poles, generally of spruce, 20′ to 50′ long, 5″ to 8″ diam. at butt; those under 2½″ at tips are termed *rickers*, and run about 22′ long. They are firmly fixed in the ground (or in tubs filled with earth, if excavation is inconvenient), 4′ 6″ from wall, 10′ to 12′ apart. When greater lengths are required, two are lashed together, tip to butt, and tightened with wedges. To these, on the side next the wall, are lashed horizontal *ledgers*, or *runners*, parallel to the wall at vertical intervals of 3′ 6″ to 5′. They support the *putlogs*, of squared timber (usually birch), 4″ × 3″, 6′ long, one end of which rests on the ledgers, the other in a hole in the wall left by omitting a brick header; they are about 3½′ or 4′ apart, and support the *scaffold boards*, usually 9″ × 1½″, sometimes 3″. Poles called 'braces,' lashed diagonally across every three or four standards, stiffen the scaffold. Masons often use a second row of standards and runners, close to the wall, to prevent the masonry being disfigured by putlog holes.

Fig. 33 (from Spon's 'Dict. of Eng.') shows the arrangement of a bricklayer's scaffold. When the scaffold has to be carried to a considerable height, other poles are lashed to the standards with ropes tightened by wedges. Poles are also lashed diagonally across every three or four standards in the shape of a St. Andrew's cross; these are called braces, and they serve to stiffen or brace the scaffold longitudinally.

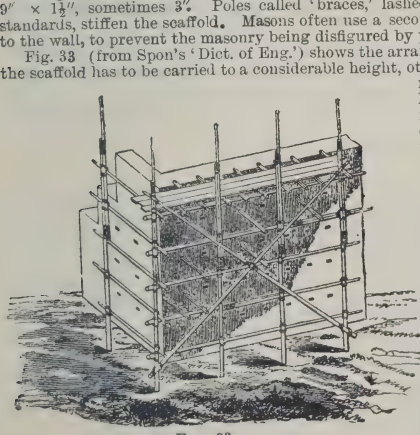


FIG. 33.

In buildings which do not admit of putlog holes in the walls, as where rubble stone or ashlar facing is used, and which do not require heavy machinery for hoisting, or strong timbers in the scaffold, two rows of standards with ledgers are used, one row being close to the wall, and the other at the usual distance, so that both ends of the putlogs may rest on the ledgers. Scaffolds such as we have just described are sometimes used for heights of 90′ or 100′ from the ground, as in building church steeples and similar work. In the erection of houses it is usual to construct a staging about 10′ square on the outside of the scaffolding, for the purpose of hoisting materials, and from which they are distributed for use.

This staging is usually formed with standards and ledgers in the same manner as the scaffold to which it is connected.

Scaffolding Knots.

The principles of a good knot are its facility in tying, its freedom from slipping, and its being easily untied. All knots will jam more or less when subject to a strain. The knots in fig. 34 are shown open before being drawn taut, in order to show the position of the parts. The names usually given, and their uses, are as follows :—

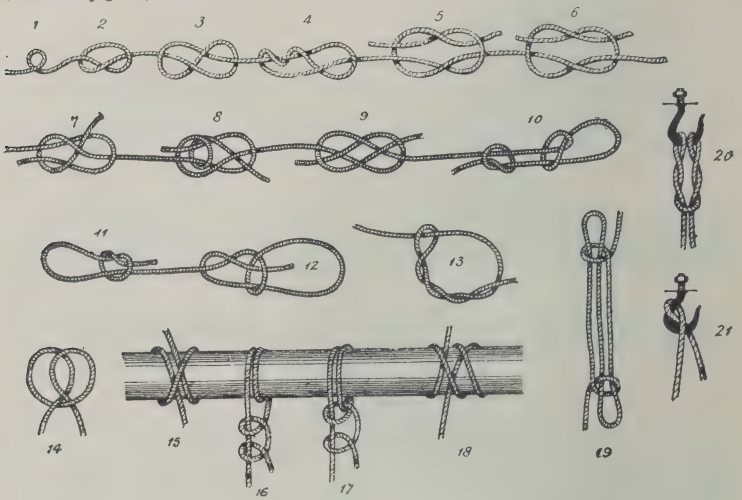


FIG. 34.

1. Bight of a rope.
2. Overhand, or thumb, knot, to prevent a rope running through the sheave of a block.
3. Figure of eight knot, used as No. 2.
4. Stevedore knot : is useful when the rope passes through an eye. It is easily untied after being strained.
5. Square, or reef, knot. This is a most useful knot for joining two ropes of same size. However tight it jams, it is easily 'upset' and undone.
6. Granny, or thief, knot. This should not be used, as it will jam tight but not slip (as erroneously supposed), will not 'upset,' and consequently is difficult to undo.
7. Single sheet bend, or weavers' knot, used principally for joining two ropes of unequal sizes more securely than a reef knot.
8. Double sheet bend, more secure than No. 7.
9. Carrick bend, for fastening the four guys to a derrick.
10. Flemish loop.
11. Slip knot.
12. Bowline, for making a loop that will not slip. After being strained, this knot is easily untied. Commence by making a bight in the rope, then put the end through the bight and under the standing part, pass the end again through the bight, and pull taut. This knot should be tied with facility by everyone who handles ropes.
13. Timber hitch ; the greater the strain the tighter it will hold.
14. Clove hitch, consisting of two half-hitches ; used chiefly to tie ledgers to standards. This is the most useful of all the knots used in scaffolding, on account of its simplicity and security.
15. Clove hitch, as No. 14, showing its application around a pole.
16. Round turn and two half-hitches, for securing a rope to a ledger or for fastening the guys of derricks, shear legs, &c.
17. Fisherman's bend, used when a thick rope, such as a fall, is made fast to a ring.
18. Rolling hitch, used in a variety of ways, chiefly in making fast one rope to another that is held taut.
19. Sheepshank, for shortening a rope when the ends are inaccessible.
20. Catspaw. An endless loop, and used where great power is required.
21. Blackwaller. Easily applied, but requires watching ; has a tendency to slip.

Shoring Buildings.

A plank (fig. 35), C, 9 ins. wide, 3 ins. thick, the length varying with height of building, is placed against upper part of wall to be supported. In this rectangular holes are cut to admit of timber 'needles,' D, from 4 ins. × 6 ins. × 6 ins., and about 12 ins. long. These are let into

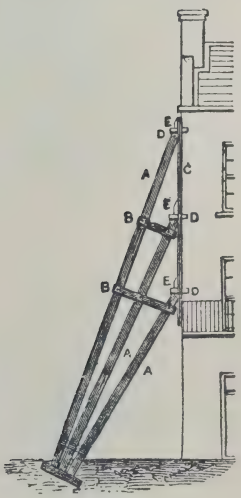
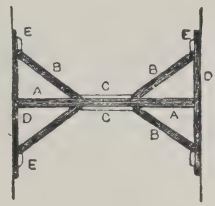


FIG. 35.

FIG. 36.



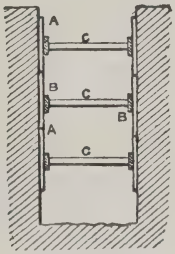


FIG. 37.

the wall about 4½ ins., and project beyond the plank about 4½ ins. to receive the ends of the 'shores,' A.

A cleat, E, is usually nailed to the plank on the upper side of each needle.

The struts, A, are from 6 ins. × 4 ins. in very small buildings to 12 ins. × 9 ins. or 12 ins. in very large; usually half timbers 12 ins. × 6 ins. They are fixed to a footing block, F.

The outer strut is called a 'top raker,' the inner the 'bottom shore,' the centre the 'middle raker.'

Short pieces, B, about 1 in. thick and from 6 ins. to 9 ins. wide are nailed on each side of the shores.

Struts between Houses.

A (fig. 36), horizontal strut, 6 ins. × 4 ins. to 9 ins. × 4 ins.
B, raking strut, 6 ins. × 4 ins.
C, straining-piece, 6 ins. × 3 ins.
D, plank, 9 ins. × 3 ins.
E, cleats.

The horizontal strut, A, is at height above ground equal to about two-thirds height from ground to eaves.

Timbering Trenches.

A (fig. 37), poling boards about 3 ft. long, 1 in. to 1½ in. thick, 7 ins. to 9 ins. wide, placed against the side as each successive depth of 3 ft. is excavated.

B, waling planks, 9 ins. × 3 ins., placed longitudinally against the centre of each row of poling boards.

C, struts from 4 ins. square to 6 ins. or 8 ins. square, according to their length and the pressures. Raking pieces are added in bad ground.

RETAINING-WALLS

Pressure of Earth.

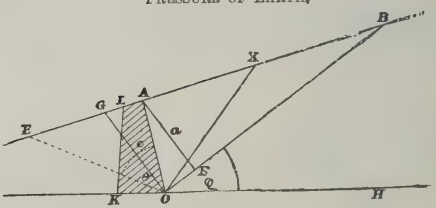


Fig. 38.

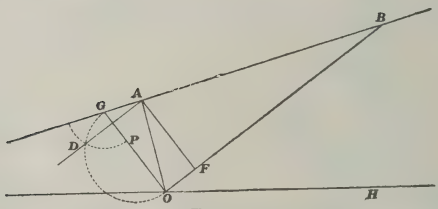


Fig. 39.

Let AB, fig. 38, be the surface of the ground, and OB the natural slope at an angle ϕ.

Draw OE, making an angle θ, this angle being equal to ϕ plus the angle of friction of earth against the back, AO, of the wall AOKL.

Take X so that EA × EB = $(EX)^2$.

Then AOX is the triangle of maximum pressure against the wall.

Draw AF (a) and GO (c) perpendicular to OB, then.

$$\text{Maximum thrust} = \tfrac{1}{2}w\,(c - \sqrt{c(c-a)})^2,$$

where w = weight of a cubic foot of earth.

Describe a semicircle on GO (figs. 38 and 39). Draw AD parallel to OB, cutting the semicircle at D. With G as centre and OD as radius describe a circle cutting GO at P, and join DP, then,

$$\text{Maximum thrust} = \frac{w}{2}\,(\text{OP})^2,$$ acting at $\tfrac{1}{3}$ AO, and making an angle θ with the normal to AO.

Walls for Railway Cuttings.

For walls over 18 ft. in height—

Thickness at bottom = $\frac{2}{3}$ height − 3 ft.

For walls less than 18 ft. high—

Thickness at bottom = $\frac{1}{3}$ height + 3 ft.

The two equations give the same result for a height of 18 ft. No allowance is necessary for a surcharge of less than 6 ft. (*Pennsylvania Railroad Co.*)

Pressure of Water.

The moment of pressure of water on a retaining wall per each foot of length of wall is $d^3 \times 10{\cdot}4$, where d is the depth of water in feet. This is the moment at 1 foot. The total pressure is $31{\cdot}2\,d^2$, assumed to be concentrated at one-third the depth from the bottom of the water. Hence the moment $10{\cdot}4\,d^3$. The weight of the wall may be assumed concentrated at its centre of gravity, and the effect of the pressure of the water is to turn the wall upon the point O (fig. 40). Calling H the height of the wall in feet, and t its thickness in inches, and assuming one cubic foot to weigh 120 pounds, the moment of stability is $\frac{\text{H}\,t^2}{2{\cdot}4}$ for rectangular walls, per foot of length; or $d^3 \times 10{\cdot}4$ must not exceed $\frac{\text{H}\,t^2}{2{\cdot}4}$; or $t^2 = \frac{25\,d^3}{\text{H}}$. This enables the thickness of a wall to be determined, that will exactly balance the pressure. Safety would be just secured by adding 50 per cent. to the thickness at the base, and subtracting the same amount from the top thickness, thus

giving the wall a sloping back as per dotted line, increasing the acting leverage of the C.G. (centre of gravity) of the section, so that the resultant pressure will now fall within the middle third of the wall thickness. This method is approximate and useful for small construction. Large masonry dams require more special consideration, and may be designed by calculations at each few feet of depth, designing the wall in a series of steps and drawing a final shape to enclose the whole in one harmonious section, which must also fulfil the conditions of strength proper to the material and crushing stress. Special care is necessary that water does not get in between bed joints, as it would greatly assist to turn a wall over.

It is usually considered desirable that the resultant of pressure on a wall should fall within the middle third of its thickness. Thus in fig. 41, if $a\,b$ to any scale represents the total pressure $31{\cdot}2\,d^2$ concentrated at the centre of pressure upon the face of a wall, and $b\,c$ to the same scale is the weight of the wall, the line $b\,c$ being drawn through the C.G. of the cross section, then if the diagonal of the rectangle $a\,b\,c\,d$ cuts the base of the wall within its middle third of thickness, the wall may be assumed fully safe. If the diagonal fall outside the middle third, the thickness of the wall must be increased. It may be made safe by building the wall higher, and thus loading it; but the danger of this method is that a greater area is exposed to wind pressure, and if margins of safety are narrow the addition may be worse than useless in small work. Where a tank wall forms a continuation of an earth retaining wall, for example, special care is necessary.

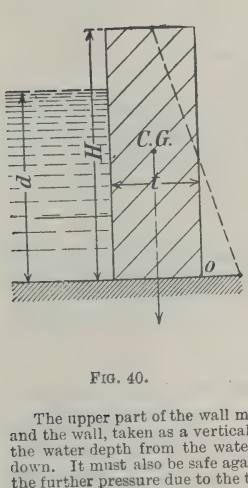


FIG. 40.

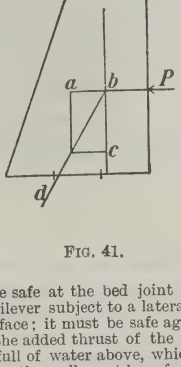


FIG. 41.

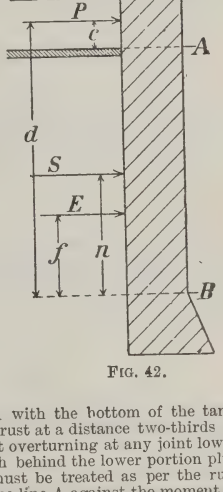


FIG. 42.

The upper part of the wall must be safe at the bed joint level with the bottom of the tank and the wall, taken as a vertical cantilever subject to a lateral thrust at a distance two-thirds of the water depth from the water surface; it must be safe against overturning at any joint lower down. It must also be safe against the added thrust of the earth behind the lower portion plus the further pressure due to the tank full of water above, which must be treated as per the rule on page 516, fig. 44. Thus in fig. 42 the wall must be safe on the line A against the moment of the water pressure P c. It must be safe at the line B against the moment (P $\times$ d) + the moment (E $\times$ f) + the moment (S $\times$ n). Here E is the horizontal component of the earth pressure, and f is one-third A B; S is the horizontal component of the surcharged load, and n is one-half of A B. The safety must be proved at sufficient other depths between B and A and below B. Economy is promoted by sloping the wall against the earth between the points A and B. The wall in the figure is palpably erroneous, being far too thin for its height; it might be made safe by a sufficiency of cross-ties buried in the concrete bottom of the tank, in which case the wall from A to B must be made safe by treating it as a beam exposed to a virtual load at S and E.

Equilibrium of Retaining-Walls.

By Rankine's theory the pressure of earth on a vertical plane, AB (fig. 43), may be considered as acting at a point, P, one-third AB from base A, and in a direction parallel to the surface. Let w = weight of earth in pounds per cubic ft., ϕ = angle of repose of earth, P = total pressure, h = height of wall in feet.

For a bank with horizontal top

$$P = \frac{wh^2}{2} \cdot \frac{1 - \sin\phi}{1 + \sin\phi}.$$

For bank with any surface slope, θ, of indefinite length

$$P = \frac{wh^2}{2} \cos\theta \frac{\cos\theta - \sqrt{\cos^2\theta - \cos^2\phi}}{\cos\theta + \sqrt{\cos^2\theta - \cos^2\phi}}.$$

For bank with maximum surface slope, ϕ, of indefinite length

$$P = \frac{wh^2}{2} \cos\phi.$$

For bank with surface slope of definite length an intermediate value is taken.

To resist the pressure there is weight of wall and of earth, ABD, resting on its back, which call W; this acts vertically from centre of gravity, G. The revetment may fail by overturning round f, by crushing at f, by sliding on Af.

Overturning round f.—On any scale make oa = W, ob = P; then oc is resultant (R), acting in direction oe at e. If e falls within base, the revetment will not overturn without crushing the toe at f; but e should fall within the centre third of Af.

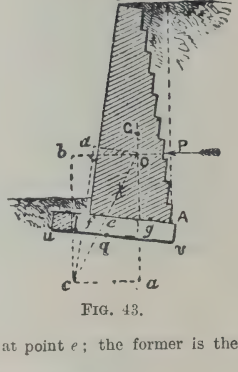


FIG. 43.

Crushing at f.—The resultant, R, must be resolved into two forces, cd (N), perpendicular to, and do parallel to, Af, acting at point e; the former is the crushing force.

Pressure per square inch at f on 1′ length of wall $= \frac{2N}{fg \times 12''}$, where $fg = 3fe$.

The pressure at f should not exceed power of material to resist crushing, divided by factor of safety, 4 to 8 (in practice seldom as high as 8).

Sliding on Af.—Force tending to produce sliding is od; force tending to resist it is N × coefficient of friction at the joint Af. od should not exceed N × ⅔ coefficient of friction.

Surcharged Wall.

If n = height of wall in feet, c = height of surcharge in feet, t = mean thickness of wall, in feet, to sustain horizontal bank; t' = mean thickness of wall, in feet, to sustain bank with indefinitely long natural slope, the factor of safety being about the same as for t; t'' = mean thickness in feet of surcharged wall; then

$$t'' = \frac{nt + 2ct'}{n + 2c}.$$

Effect of Weight of Buildings on Retaining-Walls.

Let w_1 = weight of building per unit of surface; the rest of the notation as at commencement. The total horizontal pressure, P, on revetment, due to the weight of building

$$= w_1 (h - x) \frac{1 - \sin\phi}{1 + \sin\phi},$$

acting at ½OB from O; and the total horizontal pressure, P_1, on revetment caused by the earth

$$= \frac{w(h - x)^2}{2} \cdot \frac{1 - \sin\phi}{1 + \sin\phi},$$

acting at ⅓OB from O.

If the front of the building were farther back from AO the effect would be less, but cannot be determined.

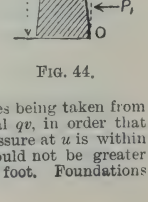


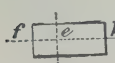
FIG. 44.

Foundations of Retaining-Walls.

Find the pressure perpendicular to uv (fig. 44), the weights and pressures being taken from point v instead of from A. If q is centre of pressure, qu should nearly equal qv, in order that the pressure may be uniformly distributed (this is unnecessary if the pressure at u is within what the earth can bear without yielding), and greatest pressure at u should not be greater than earth will bear, usually 1 to 1½ ton, or say 2,500 to 3,500 lbs. per square foot. Foundations to be carried deep enough to avoid effects of heat and frost.

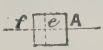
Prof. Rankine advises that the distance, fe, of centre of pressure from outside edge in buttresses, chimneys, walls, &c., should be as follows :—In a

Rectangle (solid).



fe = or $> \frac{1}{3} fA$.

Square (solid).



fe = or $> \frac{1}{3} fA$.

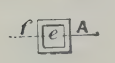
Ellipse (solid).

fe = or $> \frac{3}{8} fA$.

Circle (solid).

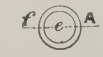
fe = or $> \frac{3}{8} fA$.

Hollow square (factory chimneys).



fe = or $> \frac{1}{6} fA$.

Circular ring (factory chimney).



fe = or $> \frac{1}{4} fA$.

Example of Brick Retaining-Wall.

Brickwork in mortar 100 lbs. per cubic ft., with a resistance to crushing of 500 lbs. per sq. in. and factor of safety of 8. Coefficient of friction for damp mortar, ·7. Weight of earth (w), 110 lbs. per cubic ft. ; angle of repose (ϕ), 37°. Pressure on foundation of sandy gravel not to exceed $1\frac{1}{2}$ ton per sq. ft. Height of wall (h) above footings, 18 ft. Wall sustains horizontal bank level with top. Top of wall two bricks thick, to allow of solid coping. Batter, $\frac{1}{12}$.

Assume for trial fA (fig. 154) $= \frac{1}{3}h = 6$ ft. The back of wall being built with offsets of $4\frac{1}{2}''$ at every 18″ in height, DB will = 3′. Foundations to be 3 ft. below ground.

Weight of wall above bed joint, fA = 6,750 lbs. per foot run of wall.
„ earth in front of AB = 3,025 „ „ „
Total weight on fA = 9,775 lbs. = W „ „

The wall is more likely to overturn at f than at any other point ; it will be sufficient, therefore, to inquire into the stability of that joint.

$$P = \frac{wh^2}{2} \frac{1 - \sin \phi}{1 + \sin \phi} = 4{,}594 \text{ lbs.},$$ which acts at P ; AP being one-third AB.

Drawing the parallelogram of forces, oc cuts Af at e, making $fe = 9\frac{1}{2}''$.

Resolving oc perpendicular and parallel to Af gives cd = 10,100 lbs.

Pressure per sq. in. at $f = \frac{2 \times 10100}{3 \times 9\frac{1}{2}'' \times 12''} = 59$ lbs.

This gives a factor of safety of 10, which is more than sufficient.

Angle ocd will be found to be 20°, tan 20° = ·36, which gives a factor of safety of 2, the coefficient of friction being ·7. The wall is therefore stable.

The pressure on foundations perpendicular to uv will be found to be 14,000 lbs., or $6\frac{1}{4}$ tons, acting at q. To equalise the pressure on the earth, let $qu = qv$, which will make $uv = 12\frac{1}{2}$ ft., giving a pressure of about $\frac{1}{2}$ ton per sq. foot ; but as the earth will bear $1\frac{1}{4}$ ton without yielding, a much less width will suffice.

Practical Rules for Retaining-Walls.

A mean thickness of one-fourth height with counterslope of one-fifth, built in steps, will not be far wrong for light soils. This gives a bottom thickness of about one-third height, a centre thickness of one-fourth height, and a top thickness of one-seventh height. A mean thickness one-third height will generally do for stiff clay.

The following rule has been extensively used for brick retaining-walls :—

Top thickness, 1 ft. $10\frac{1}{2}$ in. ; face batter of 1 in 6, and after every five courses from top increase thickness by half a brick ; front batter should not be more than one-sixth, or it will let in moisture

and encourage vegetable growth in the joints. Long walls should have counterforts at back. Walls retaining soils through which water freely passes, such as clean gravel and sand, should have a French drain of loosely packed rubble, &c., running along the back footings, from which good-sized weepholes, from 6 to 10 ft. apart, should lead through the base. With more retentive soils a French drain at least 9 or 12 inches wide should run nearly the whole way up the back of the wall. In some cases extra weepholes at higher levels may be advisable. The mouths of the weepholes should always be carefully protected by loose packing.

Trautwine's Rule for Retaining-Walls.

The horizontal thickness (*ab*, fig. 45) at the base of a vertical or nearly vertical retaining-wall, *cdba*, which sustains a backing of either sand, gravel, or earth, level with its top, *cd*, as in the figure should not be less than the following in railroad practice, when the foundations are not more than about 3 feet deep.

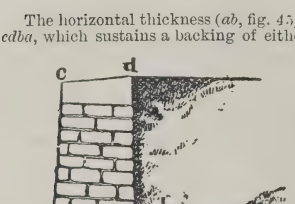


FIG. 45.

When the backing is deposited loosely, as usual, as when dumped from carts, cars, &c.—

Wall of cut stone or of first-class large-ranged rubble in mortar, *ab*. . . ·35 of its entire vertical height, *db*.

Wall of good common scabbled mortar-rubble or brick . . . . . ·4 of its entire vertical height, *db*.

Wall of well-scabbled dry rubble . . ·5 of its entire vertical height, *db*.

With good masonry, however, we may take the height *ds* instead of *db*, and then the above proportions of *ds* will give a sufficient thickness at the ground-line, *os*.

When the backing is somewhat consolidated in horizontal layers, each of these thicknesses may be reduced, but no rule can be given for this.

The offset, *oe*, in front of the wall is not included in these thicknesses.

When, however, the backing is a pure clean sand or gravel, we should use only the full dimensions, inasmuch as the tremor caused by passing trains would neutralise any supposed advantage from ramming materials so devoid of cohesion. Such sand may be rammed with much advantage for the purpose of compacting it in foundations; but a different principle is involved in that case. When it is done, even with cohesive earths, with a view of saving masonry in retaining-walls, it is probable that the expense will generally be found quite equal to that of the masonry saved.

The base, *ab*, in fig. 45, is four-tenths of the height, *bd*. In the foregoing thicknesses at base, the back, *db*, of the wall is supposed to be vertical; and the face, *ca*, either vertical or battered (sloped or inclined backward) to an extent not exceeding about 1½ inch to a foot; which limit it is rarely advisable to exceed in practice, owing to the bad effect of rain, &c., upon the mortar when the batter is great. The base of a vertical wall need not, in fact, be as thick as one with a battered face; but when the batter does not exceed 1·5 inch to a foot the difference is very small.

A mixture of sand or earth with a large proportion of round boulders, paving pebbles, &c., will weigh considerably more than the materials ordinarily used for backing, and will exert a greater pressure against the wall, the thickness of which should be increased, say, about one-eighth to one-sixth part, when such backing has to be used.

The wall will be stronger if all the courses of masonry be laid with an inclination inwards, as at *oeb*; especially if of dry masonry, or if time cannot be allowed (as it always should be when practicable) for the mortar to set properly before the backing is deposited behind it. In retaining-walls, as in the abutments of important arches, the engineer should place as little dependence as possible upon mortar, but should rely more upon the position of the joints for stability.

An objection to inclining the joints in dry (without mortar) walls is that rain-water, falling on the battered face is thereby carried inward to the earth backing, which thus becomes soft, and settles. This may be in a great measure obviated by laying the outer or face courses horizontal; or by using mortar for a depth of only about a foot from the face. The top of the wall should be protected by a coping, *cd*, which had better project a few inches in front. After the masonry has been built up to the surface of the ground, the foundation pit should be filled up; and it is well to consolidated the filling by ramming, especially in front of the wall.

The back, *db*, of the wall should be left rough. In brickwork it would be well to let every third or fourth course project an inch or two. This increases the friction of the earth against the back, and thus causes the resultant of the forces acting behind the wall to become more nearly vertical, and to fall farther within the base, giving increased stability. It also conduces to strength not to make each course of uniform height throughout the thickness of the wall, but to have some of the stones (especially near the back) sufficiently high to reach up through two or three courses. By this means the whole masonry becomes more effectually interlocked or bonded together as one mass, and therefore less liable to bulge. Very thick walls may consist of a facing of masonry and a backing of concrete.

Where deep freezing occurs the back of the wall should be sloped forwards for 3 or 4 feet below its top, as at *co*, fig. 46, which should be quite smooth, so as to lessen the hold of the frost and prevent displacement.

FIG. 46.

A slight bulging in a new wall does not necessarily prove it to be actually unsafe. It is generally due to the newness of the mortar and to the greater pressure exerted by the fresh backing, and will often cease to increase after a few months. It need not excite apprehension if it does not exceed ¼ inch for each foot in thickness at *a*.

After a wall, *abco*, fig. 47, with a vertical back has been proportioned by Trautwine's rule, it may be converted into one with an offsetted back, as *aino*. This will present greater resistance to overturning, and yet contain no more material. Thus, through the centre, *t*, of the back, draw any line, *in* ; from *n* draw *ns* vertical ; divide *is* into any even number of equal parts (in the figure

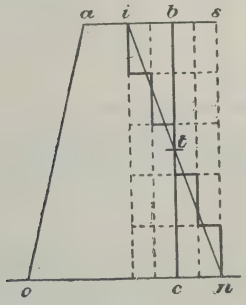


FIG. 47.

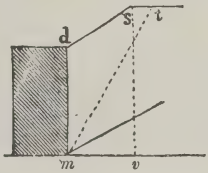


FIG. 48.

there are 4), and divide *sn* into one more equal part (in the figure there are 5). From the points of division draw horizontal and vertical lines for forming the offsets, as in the figure.

When, as in fig. 48, the backing is higher than the wall, and slopes away from its inner edge, *d*, at the natural slope, *ds*, of 1½ to 1, the following thicknesses at base will at least be found sufficient for vertical walls with sand. They are deduced from experiments.

In Table I., the first column contains the vertical height, *sv*, of the earth, as compared with the vertical height of the wall, which latter is assumed to be 1 ; so that the table begins with backing of the same height as the wall, as in fig. 156. Without increasing the base these vertical walls may be changed to others with battered faces to any extent not exceeding 1½ inch to a foot, or 1 in 8, without sensibly affecting their stability.

TABLE I.

Total Height of the Earth, compared with the Height of the Wall above Ground.	Wall of Cut Stone in Mortar.	Good Mortar Rubble or Brick.	Wall of Good Dry Rubble.	Total Height of the Earth, compared with the Height of the Wall above Ground.	Wall of Cut Stone in Mortar.	Good Mortar Rubble or Brick.	Wall of Good Dry Rubble.
	Thickness at Base, in Parts of the Height.				Thickness at Base, in Parts of the Height.		
1	·35	·40	·50	2	·58	·63	·73
1·1	·42	·47	·57	2·5	·60	·65	·75
1·2	·46	·51	·61	3	·62	·67	·77
1·3	·49	·54	·64	4	·63	·68	·78
1·4	·51	·56	·66	6	·64	·69	·79
1·5	·52	·57	·67	9	·65	·70	·80
1·6	·54	·59	·69	14	·66	·71	·81
1·7	·55	·60	·70	25			
1·8	·56	·61	·71	or more	·68	·73	·83

When the slope *nr*, fig. 49, of 1½ to 1 starts from the outer edge, *n*, of the wall, greater thickness is required. Poncelet gives the following for this case, for dry sand (Table II.).

TABLE II.

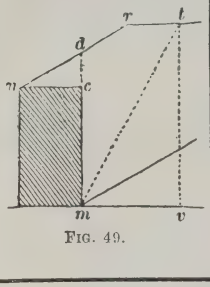


FIG. 49.

Total Depth of Earth, compared with Height of Wall.	Wall of Cut Stone in Mortar.	Wall of Brick-work.	Total Depth of Earth, compared with Height of Wall.	Wall of Cut Stone in Mortar.	Wall of Brick-work.
1	·35	·452	2·4	·762	1·02
1·1	·393	·498	3	·811	1·11
1·2	·439	·548	4	·852	1·18
1·3	·485	·604	6	·883	1·25
1·4	·532	·665	11	·909	1·28
1·5	·579	·726	21	·922	1·31
1·6	·617	·778	31	·926	1·32
1·7	·645	·824	Infinite	·934	1·34
1·8	·668	·847			
1·9	·690	·903			
2	·707	·930			

When the earth reaches above the top of the wall, as in figs. 48 and 49, the wall is surcharged, and the earth that is above the top is called the surcharge. When the surcharge is carefully deposited above the wall, so as to slope back at a steeper angle than 1½ to 1, as, say, at 1 to 1, theory does not require the wall to be as thick.

WHARF WALLS

are an instance where the thickness should be increased, notwithstanding that the pressure of the water in front helps to sustain them. The earth behind such walls is not only liable to be very heavily loaded when vessels are discharging, but is apt to become saturated with water, especially below low-water level, and thus to exert a very great pressure against the walls. Moreover, the water gets under the wall, and by its upward pressure virtually reduces its weight, and consequently its stability. The same cause, of course, diminishes the friction of the wall upon its base. Such walls are, therefore, very liable to slide if the foundation is smooth and horizontal, and have done so even when the foundation had a considerable inclination backward.

TABLE III

CONTENTS IN CUBIC YARDS FOR EACH FOOT IN LENGTH OF RETAINING-WALLS WITH A THICKNESS AT BASE EQUAL TO ·4 OF THE VERTICAL HEIGHT, IF THE BACK IS VERTICAL.

Ht. Ft.	Cub. Yds.	Ht. Ft.	Cub. Yds.	Ht. Ft.	Cub. Yds.	Ht. Ft.	Cub. Yds.	Ht. Ft.	Cub. Yds.	Ht. Ft.	Cub. Yds.
1	·013	10½	1·38	20	5·00	29½	10·9	48	28·8	74	68·5
1½	·028	11	1·51	20½	5·25	30	11·3	49	30·0	76	72·2
2	·050	11½	1·65	21	5·51	31	12·0	50	31·3	78	76·1
2½	·078	12	1·80	21½	5·78	32	12·8	51	32·5	80	80·0
3	·113	12½	1·95	22	6·05	33	13·6	52	33·8	82	84·1
3½	·153	13	2·11	22½	6·33	34	14·5	53	35·1	84	88·4
4	·200	13½	2·28	23	6·61	35	15·3	54	36·5	86	92·5
4½	·253	14	2·45	23½	6·90	36	16·2	55	37·8	88	96·8
5	·313	14½	2·63	24	7·20	37	17·1	56	39·2	90	101·3
5½	·378	15	2·81	24½	7·50	38	18·1	57	40·6	92	105·8
6	·450	15½	3·00	25	7·81	39	19·0	58	42·1	94	110·5
6½	·528	16	3·20	25½	8·13	40	20·0	59	43·5	96	115·2
7	·613	16½	3·40	26	8·45	41	21·0	60	45·0	98	120·1
7½	·703	17	3·61	26½	8·78	42	22·1	62	48·1	100	125·0
8	·800	17½	3·83	27	9·12	43	23·1	64	51·2	102	130·1
8½	·903	18	4·05	27½	9·45	44	24·2	66	54·5	104	135·2
9	1·01	18½	4·28	28	9·80	45	25·3	68	57·8	106	140·5
9½	1·13	19	4·51	28½	10·2	46	26·5	70	61·3		
10	1·25	19½	4·75	29	10·5	47	27·6	72	64·8		

If the back is stepped according to the rule on p. 631 the proportionate thickness at base will, of course, be increased. Face batter, 1½ inches to a foot, or one-eighth of the height. Back either vertical, or stepped according to Fig. 47. The strength is very nearly equal to that of a vertical wall with a base of ·4 its height. Experience has proved that such walls, when com-

posed of well-scabbled mortar rubble, are safe under all ordinary circumstances for earth level with the top.

A retaining-wall is usually in greater danger for a few months after its completion than after time has been allowed for the mortar to harden perfectly and for the backing to settle. When there are suspicions of the safety of a new wall, it would be well to place strong temporary shores against it, at about one-third to one-half of its height above ground. In some cases permanent buttresses of masonry may be built for the purpose. They should be well bonded into the wall.

The pressure of the earth backing will be much reduced if the first few feet of its height be made up in thin horizontal layers, to be consolidated by being used by the masons instead of scaffolding, as shown at *h*, fig. 45, p. 518. Frequently this can be done without inconvenience, and at very trifling cost.

CHIMNEYS.

Stability of Chimney Shafts.

The limit of stability to be observed for chimney shafts is that the wind must never bring tension on any joint. For this to be the case the 'weight moment,' that is, the weight of the length of shaft above any joint, multiplied by the diameter (or 'width' in the case of a square shaft) of the shaft at that joint, must not be less than the 'pressure moment,' *i.e.* the total wind pressure against the length of shaft above the joint, multiplied by the height of the shaft above the joint.

Assuming the shaft to be of uniform section from top to base, then

if

d = average breadth of shaft ; h = height, W = weight of the height h, in lbs. ; c = co-efficient of wind pressure per sq. ft. (this co-efficient varies with the cross-section of the shaft, and for square shafts = 56, for octagon shafts = 35, for round shafts = 28, though in the case of exceptionally heavy gales it may exceed these figures),

$$W = ch^2.$$

If the shaft is wider at the base than at other points of its length, then if d = average breadth and b = breadth of base,

$$W = c\frac{d\,h^2}{b}.$$

Example.—A square chimney shaft had an average breadth, d, of 8 ft., its base, b, being 10 ft. The height, h, of the shaft was 100 ft. What should the weight of the shaft be in order that the chimney may be stable ?

$$W = \frac{56 \times 8 \times 100 \times 100}{10} = 448{,}000 \text{ lbs.}$$

As brickwork weighs from 100 to 130 lbs. per cub. ft., such a chimney must average 13 inches thick to be safe. A round stack would weigh half as much, or have less base.

Dimensions of Chimneys, General Rules, and Formulæ.

Outside diameter at ground should not be less than $\frac{1}{10}$th the height, unless it is supported by some other structure.

Batter varies from 1 in 60 to 1 in 10 ; 1 in 24 is very common. Or the batter should be from $\frac{1}{10}$th to $\frac{1}{8}$ inch to foot on each side.

Thickness of brickwork : One brick (8 or 9 in.) for 25 ft. from the top, increasing $\frac{1}{2}$ brick (4 to 4½ in.) for each 25 ft. from the top downwards. If the inside diameter exceeds 5 ft., the top length should be 1½ brick, and if under 3 ft. it may be ½ brick for 10 ft.

Generally a much less height than 100 ft. cannot be recommended for boiler chimneys, as the lower grades of fuel cannot be burned as they should be with a shorter chimney.

Tall Chimneys should always stand alone ; if connected with the rest of the buildings, the increased settlement due to their weight on so small a foundation area causes rupture in the masonry. The distance from furnace to shaft should not exceed $\frac{2}{3}$ height of shaft. They should be built and allowed to settle before the connecting flue is made.

Circular form is best; with the same quantity of material it covers a great area, is therefore more stable, and the effect of wind upon it is much less.

In any case the flue should be circular; it can hardly be too large, as it can always be reduced by dampers. It should be lined with a detached skin of firebrick for a certain distance up, increasing in proportion to the heat of the vapours carried off, and separated from the main shaft by an air-space.

Chimneys to receive Vapours of a very high temperature are built altogether of fire-bricks.

Caps tie head of chimney together, but projecting caps catch the wind, and cause oscillation. A dangerous chimney has been saved by removing the cap.

The *Scaffolding* used for building a chimney should be so arranged that it does not prevent the chimney from settling.

The *Intensity of Draught* required varies with the kind and condition of the fuel and the thickness of the fires. Wood requires the least, and fine coal or slack the most. To burn anthracite slack to advantage a draught of 1½ inch of water is necessary, which can be obtained by a well-proportioned chimney 175 feet high.

Design of Tall Chimneys.

45 feet is an ordinary height to serve two steam boilers, but in some towns, as Manchester and Leeds, 90 ft. is the minimum allowed. They are sometimes proportioned for height, according to the coal burnt per week of 56 hours, thus:—

4	tons	per week	= 75 ft. high.	50	tons	per week	= 150 ft. high.
13	,,	,,	= 100 ,,	100	,,	,,	= 180 ,,
26	,,	,,	= 120 ,,	150	,,	,,	= 200 ,,

Another rule is to make the height of the chimney three times length of boiler plus twice distance of furthest boiler to chimney; this allows 1 ft. of height for every foot the gases travel round the boiler, and 2 ft. of height for every foot of external flue.

A round chimney should not exceed twenty-five times its internal diameter in height.

Prussian Government Rules.—In April 1902 the Prussian Government issued regulations for the construction of chimney-stacks. Square stacks to be designed for a wind pressure of 25½ lbs. per sq. ft., with the centre of pressure at the centre of gravity of diametral vertical plane. Octagonal stacks, 71 per cent. of the above pressure, and circular stacks, 67 per cent. (*Engineering Record.*)

Thickness of Brickwork.—Up to 150 ft. high or 5 ft. inside diameter the top length is generally one brick (9 ins.) thick; above that height or diameter the top length should be one brick and a half thick, and the thickness should be increased by a 4½-in. set-off at every 20 ft. below the top. The outlet at top and the throat or internal diameter at each set-off should be of the calculated size to give the necessary area plus allowance for friction.

Batter.—If the diameter of the throat is kept uniform and a 4½-in. set-off occurs at every 20 ft., the intermediate portions being of uniform thickness, a batter of 1 in 53·33 will be given, but the London County Council require a minimum batter of 1 in 48 = 2½ ins. in 10 ft., and besides that it will be requisite in most cases to provide a firebrick lining for at least the bottom 15 ft. or 20 ft., having a thickness of 4½ ins. and a clear space from the outer brickwork of 1½ in. to 2 ins., necessitating a still greater batter.

Chimney-Cap.—The chimney-cap or cornice, by the London County Council rules, must not project more than the thickness of the brickwork upon which it rests. It may be of cast-iron casings bolted together and filled with brickwork; or of granite or other good weathering stone the separate blocks being cramped together with galvanised-iron cramps, or with an iron hoop sunk into the top and protected with cement; or of brickwork in cement.

Diameter at Base—The London County Council require that the width of a shaft at the base if square on plan must be at least one-tenth, and if circular on plan, at least one-twelfth of the total height.

Firebrick Lining.—The firebrick lining must be entirely self-supporting and have a clear space behind, to allow for expansion and contraction independently of the main structure. It is therefore necessary to cover the top of the space by an over-sailing course of bricks built out from the inside of the main shaft, protecting the space, but allowing a clearance above the lining of ½ in. for every 5 ft. of its height, to permit of its expansion when heated. The average height of this lining is one-fifth height of shaft + 10 ft. A common height is 20 ft. and thickness 4½ ins.; when of greater height it is necessary to make the lower part 9 ins. thick. No air bricks should be inserted in the outer wall, as is sometimes done with the idea of cooling the air space, any such

openings being very detrimental to the draught. The only possible inlet for air to the chimney shaft should be through the fire-bars or over the surface of the incandescent fuel.

The smoke inspectors of the London County Council have lately recommended openings into the chimney-shafts near the bottom to reduce the smoke nuisance. The real effect is to dilute the smoke with air before it reaches the chimney-top so that it is not so black when expelled, but it does not reduce the actual amount of carbon emitted except so far as it tends to spoil the draught and so to reduce the fuel consumption and boiler efficiency.

Foundation.—It is important to note that a tall chimney should stand on an independent foundation in order that the settlement, or compression of the soil, may be uniform. When a chimney is built close alongside a boiler-house, or within the four walls of a warehouse, it is sometimes placed on an extension of the other foundations and bonded in with other work, but this invariably leads to unequal settlement and subsequent cracks in the brickwork.

(*Professor Henry Adams, M.Inst.C.E.*)

ARCHES.

Setting-out Arches.

Semi-elliptical Arch (fig. 50).—To set out a *semi-elliptical arch*, draw a line, AB, equal to the span or transverse axis of the ellipse. At the centre of AB draw the semi-conjugate, CD, at right angles, equal to the rise. Then, from the vertex, C, with radius AD or DB equal to half the span, describe an arc intersecting AB in E and F.

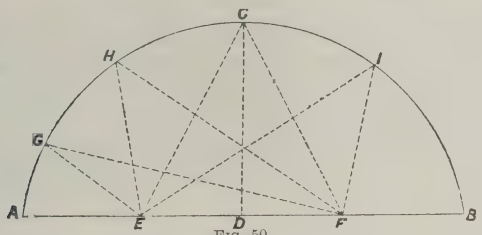


FIG. 50.

The points E, F, will be the foci of the ellipse. If two nails or pegs be fixed in the foci, and a line attached to them equal in length to AB, then the curve traced by a nail, keeping this line stretched, will be the ellipse required; the lines EGF, EHF, ECF, EIF, &c., being all equal to span AB, and to each other.

Many-centred Circular Arch.—Curves formed of arcs of circles of unequal radii, and similar in appearance to the ellipse, are sometimes adopted for the arches of bridges; with the same rise and span, they may be constructed to give a greater waterway, and in stone bridges they have been preferred by practical stonecutters, but in brick bridges they have no advantage in simplicity over elliptical arches. They may be described with three or a greater odd number of centres. The number of centres will depend on the relation between the span and rise; when the latter is one-third, or a greater fractional part of the former, three centres may be used, but if the rise is less than one-third of the span, then five, or a greater odd number, must be taken. In practice it will be found troublesome to describe arcs from a large number of centres, nor, indeed, will occasion be found for using curves of this description. The following is a method of describing a curve composed of three arcs each of 60°.

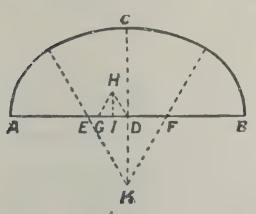


FIG. 51.

Let AB (fig. 51) represent the span, and CD the rise. Take DG = AD − DC, and on it describe an equilateral triangle, DGH; let fall the perpendicular HI, and take

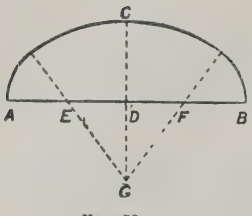


FIG. 52.

IE = HI. Lay off DF = DE. On EF describe the equilateral triangle EFK; then E, F, and K will be the centres required.

For an oval arch, rise one-third, bisect the half-spans AD and DB (fig. 52) in the points E and F, and produce versed sine CD to G, making DG = DC; then E, F, and G will be the three centres with which the curve may be described. In setting out arches, the practical difficulty arises from the elasticity of string. Instead of a string, soft wire about a tenth of an inch in diameter should be used. When the radius does not exceed 12 or 15 feet, a slip of wood may be used, with a nail at each end.

To find the Depth of Keystone for first-class Cut-stone Arches, whether Circular or Elliptic.

$$\text{Depth of key in feet} = \frac{\sqrt{\text{Rad.} + \text{half span}}}{4} + \cdot 2 \text{ foot.}$$

For second-class work this depth may be increased about one-eighth part; or for brick or fair rubble, about one-third.

TABLE.

Depths of Keystones for Arches

Of first-class cut stone. For second-class add about one-eighth part. For brick or fair rubble add about one-third part.

Span.	Rise, in Parts of the Span.						
	$\frac{1}{2}$	$\frac{1}{3}$	$\frac{1}{4}$	$\frac{1}{5}$	$\frac{1}{6}$	$\frac{1}{8}$	$\frac{1}{10}$
Feet.	Key. Feet.	Key. Feet.	Key. Feet.	Key. Feet.	Key. Feet.	Key. Feet.	Key. Feet.
2	·55	·56	·58	·60	·61	·64	·68
4	·70	·72	·74	·76	·79	·83	·88
6	·81	·83	·86	·89	·92	·97	1·03
8	·91	·93	·96	1·00	1·03	1·09	1·16
10	·99	1·01	1·04	1·07	1·11	1·18	1·26
15	1·17	1·19	1·22	1·26	1·30	1·40	1·50
20	1·32	1·35	1·38	1·43	1·48	1·59	1·70
25	1·45	1·48	1·53	1·58	1·64	1·76	1·88
30	1·57	1·60	1·65	1·71	1·78	1·91	2·04
35	1·68	1·70	1·76	1·83	1·90	2·04	2·19
40	1·78	1·81	1·88	1·95	2·03	2·18	2·33
50	1·97	2·00	2·08	2·16	2·25	2·41	2·58
60	2·14	2·18	2·26	2·35	2·44	2·62	2·80
80	2·44	2·49	2·58	2·68	2·78	2·98	3·18
100	2·70	2·75	2·86	2·97	3·09	3·32	3·55
120	2·94	2·99	3·10	3·22	3·35	3·61	3·88
140	3·16	3·21	3·33	3·46	3·60	3·87	4·15
160	3·36	3·44	3·58	3·72	3·87	4·17	
180	3·56	3·63	3·75	3·90	4·06	4·38	
200	3·74	3·81	3·95	4·12	4·29		
220	3·91	4·00	4·13	4·30	4·48		
240	4·07	4·15	4·30	4·48			
260	4·23	4·31	4·47	4·66			
280	4·38	4·46	4·63				
300	4·53	4·62	4·80				

In large arches it is advisable to increase the depth of the archstones toward the springs; but when the span is as small as about 60 to 80 or 100 feet, this is not at all necessary if the stone is good, although the arch will be stronger if it is done. In practice this increase, even in the largest spans, does not exceed from $\frac{1}{7}$ to $\frac{1}{5}$ the depth of the key, although theory would require much more in arches of great rise.

Amount of Pressure sustained by Archstones.

In bridges of the same width of roadway, if all the other parts bore to each other the same proportion as the spans, the total pressure would increase as the squares of the spans, while the pressure per square foot would increase as the spans. But in practice the depth of the archstones increases much less rapidly than the span; while the thickness of the roadway material, and the extraneous load per square foot, remain the same for all spans. Hence the total pressures, at key and at spring, increase less rapidly than the squares of the spans, but more rapidly than the simple spans; as do also the pressures per square foot. Thus, in two bridges of the same width, but with spans of 100 and 200 feet, with depths of archstones taken from the table on page 524, and uniform from key to spring, supposed to be filled up solid with masonry of 160 lbs. per cubic foot to a level of about 15 inches above the crown (including the stone paving of the roadway), with an extraneous load of 100 lbs. per square foot, the pressures will be approximately as follows:

Rise.	Span 100 Feet.				Span 200 Feet.			
	At Key.		At Spring.		At Key.		At Spring.	
	For 1 foot in width of its entire depth.	Per square foot.	For 1 foot in width of its entire depth.	Per square foot.	For 1 foot in width of its entire depth.	Per square foot.	For 1 foot in width of its entire depth.	Per square foot.
	Tons.	Tons.	Tons.	Tons.	Tons.	Tons.	Tons.	Tons.
$\frac{1}{6}$	42¼	13¾	58	18¾	126	29¼	179	42
$\frac{2}{6}$	36¾	12¼	57	19	112	27¼	181	44
$\frac{3}{6}$	31¼	11	57½	20	97	24½	188	47½
$\frac{4}{6}$	25	9	61½	22½	80¼	21	207	54¼
$\frac{5}{6}$	18	6¾	67½	25	57½	15½	230	61½

It will be seen that with the same span the pressure at the key becomes less, while that at the spring becomes greater, as the rise increases. Also, that when the archstones are of uniform depth, the pressure at either spring of a semicircular arch is about 4 times as great as at the key; whereas when the rise is but one-sixth of the span the pressure at spring averages but about one-third greater than at the key. These proportions vary somewhat in different spans.

The greater pressure per square foot at the springs may be reduced by increasing the depth of the archstones towards the springs. This, however, is not necessary in moderate spans, inasmuch as good stone will be safe even under this greater pressure.

Thickness of Arches.

Centre of pressure at every joint of arch and abutment should fall within centre third.

Thickness of arch at crown $= \sqrt{r \times B}$, where r is the longest radius of curvature at intrados; B, as deduced from practice, is as follows:—

	Feet.
Arch above ground standing solitary between abutments . . .	B = 0·12
Arch forming one of series of arches with piers between . . .	= 0·17
Underground archway in hard materials, such as rock or conglomerate	= 0·12
Underground archway in gravel or firm earth	= 0·27
Underground archway in wet clay or quicksand	= 0·48

Centering for Arches.

Centering is made of such stuff as is available. Fig. 53 shows what would be suitable for a 20 ft. arch of brick 2 ft. 3 ins. thick. The king post and struts might be of 2-in. stuff. The lagging is of 1½-in. plank, with centres 3 ft. apart; if 4 ft. to 5 ft. apart it should be of 2-in. plank. The struts are often placed at right angles to the arch, as shown in dotted lines.

In centering for *concrete arches* the lagging should be of narrow battens 3 ins. to 4½ ins., laid close, the joints being run in with whiting and plaster of Paris. The following dimensions have been followed:—25 ft. 6 ins. span; centres were 2 ft. apart, of 9-in. × 1½-in. stuff; lagging 4½ ins. × 1½ in., a 9-in. × 3-in. plank cutting into four for lagging, into two for ribs. There were two lines of intermediate supports. For flat or prismoidal roof of concrete 2-in. or 3-in. sheeting was used, on centres of 3-in. stuff, 4 ft. to 6 ft. apart.

The centres rest by the ends of their chords upon wooden *striking* or *lowering wedges*, for striking or lowering the centre after the completion of the arch. These consist of pairs of wedge-shaped blocks, as W, W, fig. 54, of hard wood from 1 ft. to 2 ft. long, about half as wide, and a quarter or more as thick (sufficient to lower the centre from, say, 2 ins. to 6 ins. or more, according to span and other circumstances).

It is of the utmost importance, especially in large arches, that the *centres should be lowered very slowly*, otherwise the momentum acquired by so heavy a body as an arch descending suddenly, even but 2 ins. or 3 ins., might possibly affect its shape, or even its safety. For this reason the wedges should not have a taper steeper than about 1 in 6 or 8 for arches of less than about 50 ft. span, or than 1 in 8 or 10 for larger spans. Vertical lines at equal distances apart should be drawn on the long sides of the wedges, as a guide for lowering them all to the same extent at a time; and this should not exceed in all about half an inch a day in intervals of about an eighth of an inch for 50 ft. spans, or about ·1 to ·25 of an inch per day in all for spans over 100 ft. Slowness is especially to be recommended in *brick arches*, not only because their greater number of

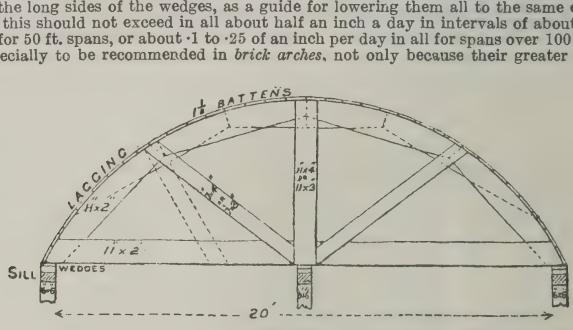


FIG. 53.

joints exposes them to greater derangement of shape, but because even good brick has much less than the average crushing-strength of good granite, limestone, or sandstone, and therefore is far more liable than they to crack, or even to crush, when the strains are thrown almost entirely upon their edges.

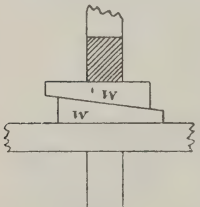


FIG. 54.

Easing and Striking Centres.—Much depends on temperature, mass of masonry or brickwork, composition of mortar, etc. With ordinary lime, centres should be eased while mortar is quite green, as soon as the arch is loaded to stability, and struck on an average after five to eight days. With small arches, one or two and a half bricks thick, 6 ft. to 8 ft. span, built in dry weather, the centres may be struck in three or four days. With slow-setting cement, if the arch be small and slight, so that it can be loaded to stability before the mortar of portion first commenced has set, the centres may be eased, as with lime, but they may be struck sooner, and the limit may be taken as the time at which the joints would become too hard set to admit of being raked out for pointing without seriously injuring the brickwork. If, in consequence of the cement being strong and quick-setting, or of the span or thickness of the arch being considerable, the arch cannot be completed before the portion first commenced has set, it is doubtful whether the centering should be eased at all, for fear of disturbing the setting of the cement. Centering for concrete should not be disturbed for at least fourteen days from the time of laying the concrete.

The main points to attend to are—to complete the arch to stability rapidly, and to prevent disturbance of the work while setting, by traffic over it, or vibrations from adjoining work, otherwise the setting power, of cement specially, is destroyed.

Piers.

Thickness of piers, generally from $\frac{1}{6}$ to $\frac{1}{8}$ span. Short piers have vertical sides; high piers may have side batter of $\frac{1}{12}$.

In viaducts and long bridges, 1 pier in every 5 or 6 may be thickened as an abutment pier, so that an accident to a part may not cause the destruction of the whole.

SECTION XIII

RAILWAYS

PART I

PERMANENT WAY

RAILS — BOLTS AND SPIKES — CHAIRS — SLEEPERS — ARRANGEMENT OF PERMANENT WAY — GAUGES — NARROW-GAUGE RAILWAYS—TUNNELS—RESISTANCES—CROSSINGS—CURVES . . . . . (pp. 529-569)

PART II

LOCOMOTIVES AND ROLLING STOCK

(Revised by H. Holcroft, M.I.Loco.E.)

LOCOMOTIVES—STEAM MOTOR CARS—CARRIAGES AND WAGONS—LIGHTING AND HEATING RAILWAY CARRIAGES—BRAKES . . . . . (pp. 571-643)

PART III

SIGNALS

PNEUMATIC SIGNALS—ELECTRIC SIGNALS (pp. 645-659)

PART IV

RAILWAY EXPENDITURE AND RECEIPTS—STANDING ORDERS WITH REFERENCE TO RAILWAY BILLS (pp. 661-665)

SECTION XIII

PART I

PERMANENT WAY.

RAILS — BOLTS AND SPIKES — CHAIRS — SLEEPERS — ARRANGEMENT OF PERMANENT WAY — GAUGES — NARROW-GAUGE RAILWAYS—TUNNELS—RESISTANCES —CROSSINGS—CURVES.

RAILWAYS.

RAILS.

There are two descriptions of rails in general use—viz. the ' Bull-headed Rail,' and the ' Flat Bottomed ' or ' Flanged Rail.'

Steel rails have superseded those of iron.

Dimensions of Rails Used on British Railways.

The standard rails of leading English railways are as follows : (1) London, Midland and Scottish Railway (Midland Section), 100 lbs. per yard, 45 ft. long ; (2) Great Western Railway, 97½ lbs., 30 ft. ; (3) London and North Eastern Railway (Great Northern Section), 96 lbs. ; (4) London and North Eastern Railway (Great Central Section), 96 lbs., 36 ft., and 86 lbs., 30 ft. long ; (5) London, Midland and Scottish Railway (North Western Section), 95 lbs., 60 ft. ; (6) London and North Eastern Railway (North Eastern Section), 90 lbs., 30 ft. ; (7) Southern Railway (Western Section), 90 lbs. ; (8) London, Midland and Scottish Railway (Lancashire and Yorkshire Section), 86 lbs. ; (9) London and North Eastern Railway (Great Eastern Section), 85 lbs. All of these are of the bull-head section, supported in heavy cast-iron chairs, to which they are secured by wooden wedges (or sometimes wedges of coiled steel). The chairs are fastened to the ties by round spikes and wooden spikes or trenails, through bolts being occasionally used.

More recently the London Midland and Scottish Railway has adopted a standard rail length of 60 ft., and the remaining British railways a standard length of 45 ft.

BRITISH STANDARD SPECIFICATION AND SECTIONS OF BULL-HEAD RAILWAY RAILS.*

(No. 9—Sept. 1928.) (*Abstract.*)

2. The steel for the rails shall be of the best quality made by the Open Hearth or Acid Bessemer process, as may be specified or required by the engineer (or the purchaser), and

* By permission of the British Engineering Standards Association.

shall show on analysis that in chemical composition it conforms to the limits given for such processes respectively in Tables 1 and 2.

The steel shall conform to the limits of chemical composition appearing in Tables 1 and 2 under the headings of ' Ordinary Carbon ' and ' Higher Carbon ' respectively, as may be specified or required by the engineer (or the purchaser).

TABLE 1.—STEEL MADE BY THE OPEN-HEARTH PROCESS.

Element.	Acid.		Basic.	
	Ordinary Carbon.	Higher Carbon.	Ordinary Carbon.	Higher Carbon.
	Per Cent.	Per Cent.	Per Cent.	Per Cent.
Carbon . .	0·45 to 0·55	0·50 to 0·60	0·45 to 0·60	0·55 to 0·65
Manganese . .	0·90 (max.)	0·80 (max.)	0·90 (max.)	0·80 (max.)
Silicon . .	0·15 (max.)	0·10 to 0·30	0·15 (max.)	0·10 to 0·30
Phosphorus .	0·07 (max.)	0·06 (max.)	0·05 (max.)	0·04 (max.)
Sulphur . .	0·07 (max.)	0·06 (max.)	0·06 (max.)	0·05 (max.)

TABLE 2.—STEEL MADE BY THE BESSEMER PROCESS.

Element.	Acid.	
	Ordinary Carbon.	Higher Carbon.
	Per Cent.	Per Cent.
Carbon . .	0·40 to 0·50	0·45 to 0·55
Manganese . .	0·70 to 1·00	0·90 (max.)
Silicon . .	0·15 (max.)	0·10 to 0·30
Phosphorus .	0·075 (max.)	0·07 (max.)
Sulphur . .	0·07 (max.)	0·06 (max.)

6. A falling weight test shall be made in the following manner:—A piece of rail 5 feet (1·52 m.) long, taken from one of the rails selected by the engineer or the inspector (or the purchaser), shall be placed in a horizontal position with the head uppermost upon the two steel bearers of the British Standard Falling Weight Testing Machine, the bearers being 3 feet 6 inches (1·07 m.) apart at their centres, and having their upper surfaces curved to a radius of 3 inches (76·20 mm.).

The test shall comprise two blows delivered midway between the bearers from a falling iron weight. The height of the drop for the various sections of rails shall be as tabulated below. The blows shall be sustained without fracture, and the permanent set measured on the specified distance between the centres of the bearers resulting from the two blows must not exceed the limits given in the table, p. 532.

Telegrams:
'SUMMERSON,
PHONE
DARLINGTON.'

Telephone Nos.:
2660
2669

THOMAS SUMMERSON & SONS LTD.

ENGINEERS AND
IRON AND STEEL FOUNDERS

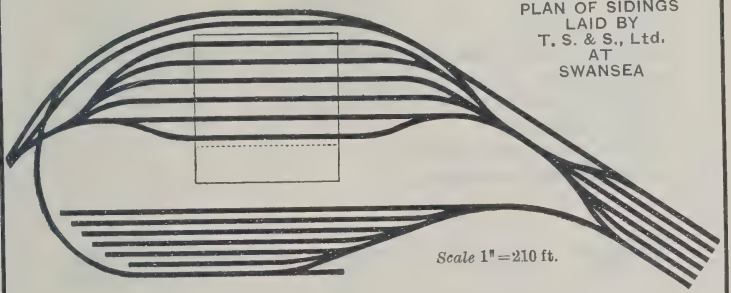
DARLINGTON

LONDON OFFICE — Established 1840 — 32 VICTORIA STREET, S.W.1

RAILWAY SIDINGS

SUPPLIED COMPLETE, INCLUDING PLATELAYING

Viewing of Sites, Preparation of Plans and Estimates of Cost



This Company has a unique practical experience in railroad work and was founded by the late Thomas Summerson, who, when engaged in operations connected with the original Stockton and Darlington Railway, designed and worked out the satisfactory application of the first set of switches and crossings made from rails. Upwards of ninety years of practical experience in all parts of the world has vested the technical staff of the Company with an intimate progressive knowledge of most suitable railroad practice, and this accumulated information is placed unreservedly at the service of all interested.

SEE NEXT PAGE

Telegrams:
'SUMMERSON,
PHONE
DARLINGTON.'

Telephone Nos:
2660
2669

THOMAS SUMMERSON & SONS LTD.

ENGINEERS AND
IRON AND STEEL FOUNDERS

DARLINGTON

LONDON OFFICE. Established 1840 32 VICTORIA STREET, S.W. 1.

RAIL-ROAD ACCESSORIES

SWITCHES AND CROSSINGS in all sections of rails, to all specifications.
LIGHT TRACK, Colliery and bar turns.
SWITCH LEVERS of all kinds. Rods, Cranks, Gauge Ties, etc.
FISH PLATES, plain and junction.
FISH BOLTS.
CHAIRS AND FASTENINGS of all kinds.
SLEEPERS, wood and steel.
BUFFER STOPS and wheel stops.
TURNTABLES.
PLATELAYERS' TOOLS.
RE-RAILING RAMPS.

Iron and Steel Castings.

This Company has a unique practical experience in railroad work and was founded by the late Thomas Summerson, who, when engaged in operations connected with the original Stockton and Darlington Railway, designed and worked out the satisfactory application of the first set of switches and crossings made from rails.

Upwards of ninety years of practical experience in all parts of the world has vested the technical staff of the Company with an intimate progressive knowledge of most suitable railroad practice, and this accumulated information is placed unreservedly at the service of all interested.

SEE PREVIOUS PAGE

BRITISH STANDARD BULL-HEAD RAILWAY RAILS.

B.S. Section No. 60.
60 lbs. per yard
(29·76 kgs. per metre).

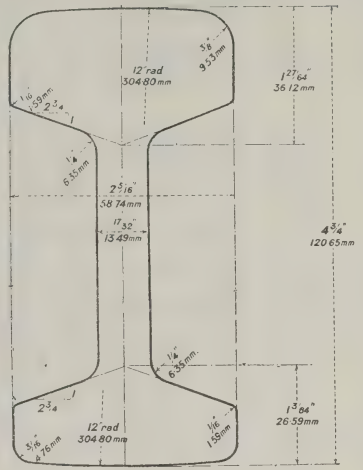


FIG. 1.

B.S. Section No. 80.
80 lbs. per yard
(39·68 kgs. per metre).

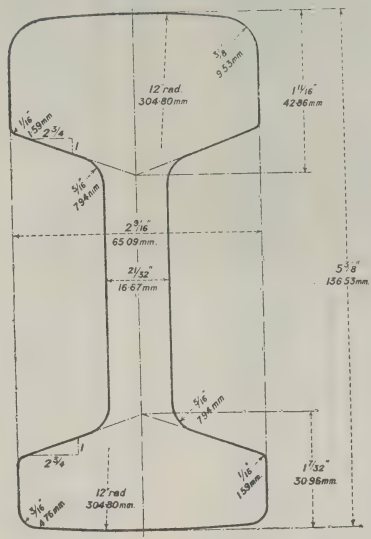


FIG. 2.

B.S. Section No. 100.
100 lbs. per yard (49·61 kgs. per metre).

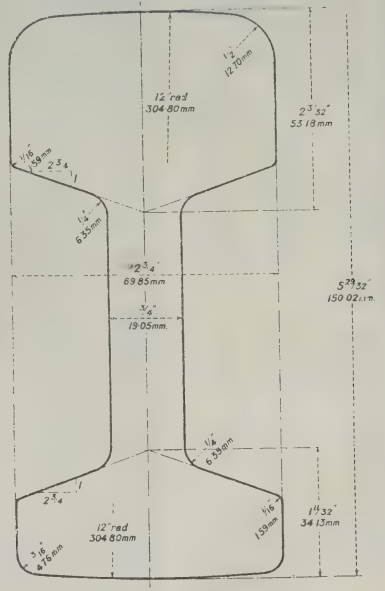


FIG. 3.

Falling Weight of 20 cwts. (2,240 lbs. = 1,016 kgs.)
Centres of bearers, 3 feet 6 inches (1·07 m.) apart.

No. of B.S. Section and Nominal Weight in lbs. per yard (kgs. per metre).	Height of Drop.		Maximum Permanent Set resulting from the two blows.		No. of B.S. Section and Nominal Weight in lbs. per yard (kgs. per metre).
	1st Blow.	2nd Blow.	Ordinary Carbon Rail.	Higher Carbon Rail.	
	ft. (m.)	ft. (m.)	in. (mm.)	in. (mm.)	
60 (29·76)	5 (1·52)	10 (3·05)	3·9 (99·06)	3·7 (93·98)	60 (29·76)
65 (32·24)	5 (1·52)	12 (3·66)	3·9 (99·06)	3·7 (93·98)	65 (32·24)
70 (34·72)	6 (1·83)	12 (3·66)	3·9 (99·06)	3·7 (93·98)	70 (34·72)
75 (37·20)	6 (1·83)	12 (3·66)	3·9 (99·06)	3·7 (93·98)	75 (37·20)
80 (39·68)	6 (1·83)	15 (4·57)	4·2 (106·68)	4·0 (101·60)	80 (39·68)
85R(42·16)	6 (1·83)	15 (4·57)	4·2 (106·68)	4·0 (101·60)	85R(42·16)
90R(44·64)	7 (2·13)	18 (5·49)	4·3 (109·22)	4·1 (104·14)	90R(44·64)
95R(47·13)	7 (2·13)	20 (6·10)	4·3 (109·22)	4·1 (104·14)	95R(47·13)
100 (49·61)	7 (2·13)	20 (6·10)	4·1 (104·14)	3·9 (99·06)	100 (49·61)

14. All ingots used in the manufacture of the rails shall be not less than 200 square inches (1,290·32 cm.²) in area at the larger end. The ingot shall be so fed into the cogging rolls that the bottom end arrives at the shears or hot saws first. From each end of the bloom and/or rail bar sufficient crop shall be cut to ensure that all unsound portions have been removed.

24. Dimensions and Weights of B.S.' Bull-Head Railway Rails.

No. of B.S. Section and Nominal Weight in lbs. per yard (kgs. per metre).	Height of Rail.	Width of Head.	Calculated Weight of Rail before Drilling. Lbs. per yard (kgs. per metre).	No. of B.S. Section and Nominal Weight in lbs. per yard (kgs. per metre).
	ins. (mm.)	ins. (mm.)		
60 (29·76)	$4\frac{3}{4}$ (120·65)	$2\frac{5}{16}$ (58·74)	59·79 (29·66)	60 (29·76)
65 (32·24)	$4\frac{7}{8}$ (123·83)	$2\frac{3}{8}$ (60·33)	64·58 (32·04)	65 (32·24)
70 (34·72)	5 (127·00)	$2\frac{7}{16}$ (61·91)	70·13 (34·79)	70 (34·72)
75 (37·20)	$5\frac{1}{8}$ (130·18)	$2\frac{1}{2}$ (63·50)	74·56 (36·99)	75 (37·20)
80 (39·68)	$5\frac{3}{8}$ (136·53)	$2\frac{9}{16}$ (65·09)	79·49 (39·43)	80 (39·68)
85R (42·16)	$5\frac{5}{16}$ (138·91)	$2\frac{11}{16}$ (68·26)	84·88 (42·11)	85R (42·16)
90R (44·64)	$5\frac{1}{2}$ (140·89)	$2\frac{3}{4}$ (69·85)	89·77 (44·53)	90R (44·64)
95R (47·13)	$5\frac{3}{4}$ (145·26)	$2\frac{3}{4}$ (69·85)	94·59 (46·92)	95R (47·13)
100 (49·61)	$5\frac{9}{16}$ (150·02)	$2\frac{3}{4}$ (69·85)	99·84 (49·53)	100 (49·61)

27. The standard length of the rails for straight line shall be 30 feet (9·14 m.), 36 feet (10·97 m.), 40 feet (12·19 m.), 45 feet (13·72 m.) and 60 feet (18·29 m.).

29. The rails shall be the specified length at a temperature of 62° Fahr. (16·67° C.). Any rail may be rejected which is more than three-sixteenths ($\frac{3}{16}$) of an inch (4·76 mm.) above or below the length specified at that temperature, whether for straight or curved line.

AREAS OF HEAD, WEB, AND FOOT OF 'B.S.' BULL-HEAD RAILWAY RAILS.

B.S. No. and Nominal Weight.		Head.			Web.			Foot.			Whole Section.			Calculated Weight.	
		Area.			Area.			Area.			Area.				
lbs. per yd.	kgs. per metre.	sq. in.	mm.²	Per Cent.	sq. in.	mm.²	Per Cent.	sq. in.	mm.²	Per Cent.	sq. in.	mm.²		lbs. per yd.	kgs. per metre
60	29·76	2·71	1749	46·4	1·29	830	21·80	1·87	1208	31·80	5·87	3786		59·79	29·66
65	32·24	2·95	1906	46·6	1·36	880	21·50	2·02	1303	31·90	6·34	4089		64·58	32·04
70	34·72	3·21	2068	46·7	1·45	938	20·70	2·22	1435	32·60	6·88	4441		70·13	34·79
75	37·20	3·43	2212	46·8	1·53	989	20·80	2·36	1521	32·40	7·32	4722		74·56	36·93
80	39·68	3·62	2334	46·4	1·73	1119	22·10	2·45	1582	31·50	7·80	5034		79·49	39·43
85R	42·16	4·07	2623	48·79	1·81	1165	21·67	2·46	1588	29·54	8·33	5375		84·88	42·11
90R	44·64	4·27	2752	48·42	1·97	1268	22·30	2·58	1665	29·28	8·81	5685		89·77	44·53
95R	47·13	4·48	2892	48·27	1·97	1268	21·17	2·84	1830	30·56	9·28	5990		94·59	46·92
100	49·61	4·91	3169	50·12	1·97	1268	20·05	2·92	1886	29·83	9·80	6322		99·84	49·53

31. The holes for fishbolts shall be drilled through the web from the solid at each end of the rail of the sizes and in the position shown in the British Standard Specification for Steel Fish-plates for Bull-head Railway Rails (Report No. 47—1919) or on a drawing to be supplied by the engineer (or the purchaser).

These holes shall be at right angles to the web of the rail, and clean cut, all burrs being carefully removed, and shall be checked by the manufacturer with suitable templates and gauges to be furnished by him and at his expense, and approved by the engineer (or the purchaser).

32. Should any of the holes vary from the correct size or position more than one-thirty-second ($\frac{1}{32}$) of an inch (0·79 mm.) the rail or rails in which such deviation occurs may be rejected.

FLAT-BOTTOMED OR FLANGED RAILS.

This form of rail is largely used in India, in the Colonies, and in America.

BRITISH STANDARD SPECIFICATION AND SECTIONS OF FLAT-BOTTOM RAILWAY RAILS.*

(No. 11—July 1926.) (*Abstract.*)

TABLE I (*Abbreviated*).—DIMENSIONS.

B.S. No. and Nominal Weight.	A	B	C	D	E	F	G	J	K
lb. per yd.	in.	in.	in.	in.	in.	in.	in.	in.	in.
25R	$2\frac{7}{8}$	$2\frac{3}{4}$	$1\frac{1}{2}$	$\frac{1}{4}$	$\frac{29}{32}$	$\frac{7}{16}$	$1\frac{13}{64}$	$\frac{5}{16}$	$\frac{3}{16}$
30R	$3\frac{1}{8}$	3	$1\frac{5}{8}$	$\frac{19}{64}$	1	$\frac{15}{32}$	$1\frac{19}{64}$	$\frac{5}{16}$	$\frac{3}{16}$
35R	$3\frac{3}{8}$	$3\frac{1}{4}$	$1\frac{7}{8}$	$\frac{21}{64}$	$1\frac{3}{32}$	$\frac{1}{2}$	$1\frac{25}{64}$	$\frac{5}{16}$	$\frac{3}{16}$
40R	$3\frac{5}{8}$	$3\frac{1}{2}$	$1\frac{7}{8}$	$\frac{23}{64}$	$1\frac{9}{64}$	$\frac{17}{32}$	$1\frac{33}{64}$	$\frac{5}{16}$	$\frac{3}{16}$
45R	$3\frac{7}{8}$	$3\frac{3}{4}$	$1\frac{3}{8}$	$\frac{3}{8}$	$1\frac{7}{32}$	$\frac{9}{16}$	$1\frac{39}{64}$	$\frac{5}{16}$	$\frac{3}{16}$
50R	$4\frac{1}{8}$	$3\frac{15}{16}$	$2\frac{1}{16}$	$\frac{25}{64}$	$1\frac{13}{64}$	$\frac{19}{32}$	$1\frac{23}{64}$	$\frac{5}{16}$	$\frac{3}{16}$
55R	$4\frac{5}{16}$	$4\frac{1}{8}$	$2\frac{3}{32}$	$\frac{27}{64}$	$1\frac{11}{32}$	$\frac{5}{8}$	$1\frac{21}{64}$	$\frac{5}{16}$	$\frac{3}{16}$
60R	$4\frac{1}{2}$	$4\frac{5}{16}$	$2\frac{1}{4}$	$\frac{7}{16}$	$1\frac{15}{32}$	$\frac{21}{32}$	$1\frac{7}{8}$	$\frac{5}{8}$	$\frac{3}{32}$
65R	$4\frac{11}{16}$	$4\frac{7}{16}$	$2\frac{5}{16}$	$\frac{15}{32}$	$1\frac{29}{64}$	$\frac{11}{16}$	$1\frac{31}{32}$	$\frac{7}{16}$	$\frac{1}{4}$
70R	$4\frac{7}{8}$	$4\frac{5}{8}$	$2\frac{3}{8}$	$\frac{1}{2}$	$1\frac{1}{2}$	$\frac{23}{32}$	$2\frac{3}{64}$	$\frac{7}{16}$	$\frac{1}{4}$
75R	$5\frac{1}{16}$	$4\frac{13}{16}$	$2\frac{7}{16}$	$\frac{33}{64}$	$1\frac{9}{16}$	$\frac{47}{64}$	$2\frac{1}{8}$	$\frac{7}{16}$	$\frac{1}{4}$
80R	$5\frac{1}{4}$	5	$2\frac{1}{2}$	$\frac{17}{32}$	$1\frac{39}{64}$	$\frac{49}{64}$	$2\frac{13}{64}$	$\frac{7}{16}$	$\frac{5}{16}$
85R	$5\frac{7}{16}$	$5\frac{3}{16}$	$2\frac{9}{16}$	$\frac{35}{64}$	$1\frac{43}{64}$	$\frac{25}{32}$	$2\frac{9}{32}$	$\frac{7}{16}$	$\frac{5}{16}$
90R	$5\frac{5}{8}$	$5\frac{3}{8}$	$2\frac{5}{8}$	$\frac{37}{64}$	$1\frac{23}{32}$	$\frac{13}{16}$	$2\frac{23}{64}$	$\frac{1}{2}$	$\frac{5}{16}$
95R	$5\frac{13}{16}$	$5\frac{9}{16}$	$2\frac{11}{16}$	$\frac{19}{32}$	$1\frac{25}{32}$	$\frac{27}{32}$	$2\frac{7}{16}$	$\frac{1}{2}$	$\frac{3}{8}$
100R	6	$5\frac{3}{4}$	$2\frac{3}{4}$	$\frac{39}{64}$	$1\frac{27}{32}$	$\frac{27}{32}$	$2\frac{1}{2}$	$\frac{1}{2}$	$\frac{3}{8}$

* By permission of the British Engineering Standards Association.

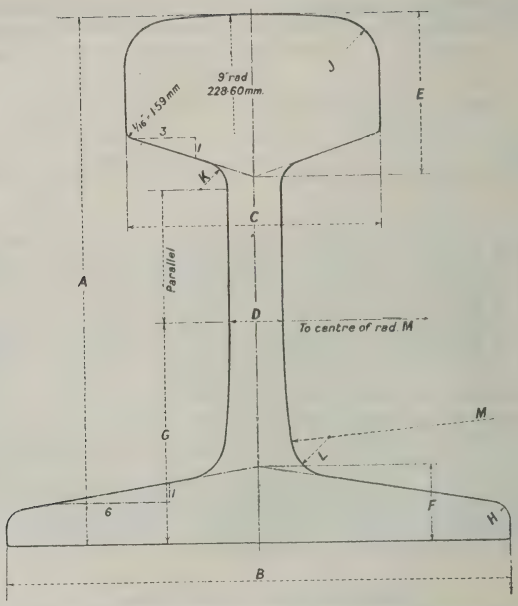


FIG. 4.—British Standard Flat-Bottom Railway Rails. Key to Table of Dimensions.

BRITISH STANDARD FLAT BOTTOM RAILWAY RAILS.

TABLE II (*Abbreviated*).—AREAS OF HEAD, WEB, AND FLANGE.

B.S. No. and Nominal Weight. lb. per yd.	Head.		Web.		Flange.		Whole Section.	B.S. No. and Nominal Weight. lb. per yd.
	Area. sq. in.	Per cent.	Area. sq. in.	Per cent.	Area. sq. in.	Per cent.	Area. sq. in.	
25R	1·129	46·04	0·438	17·82	0·886	36·14	2·453	25R
30R	1·355	45·96	0·563	19·10	1·030	34·94	2·948	30R
35R	1·598	46·49	0·656	19·09	1·183	34·42	3·437	35R
40R	1·771	45·14	0·805	20·52	1·347	34·34	3·923	40R
45R	1·996	45·21	0·898	20·33	1·522	34·46	4·416	45R
50R	2·242	45·66	0·978	19·92	1·690	34·42	4·910	50R
55R	2·420	44·76	1·120	20·71	1·867	34·53	5·407	55R
60R	2·644	44·90	1·193	20·25	2·053	34·85	5·890	60R
65R	2·804	44·00	1·342	21·05	2·227	34·95	6·373	65R
70R	2·978	43·30	1·470	21·37	2·430	35·33	6·878	70R
75R	3·193	43·45	1·588	21·61	2·568	34·94	7·349	75R
80R	3·375	42·95	1·698	21·62	2·783	35·43	7·856	80R
85R	3·598	43·09	1·825	21·85	2·928	35·06	8·351	85R
90R	3·781	42·83	1·886	21·37	3·161	35·80	8·828	90R
95R	4·026	43·15	1·992	21·35	3·313	35·50	9·331	95R
100R	4·274	43·56	2·066	21·07	3·471	35·37	9·811	100R

WEIGHT OF RAILS PER MILE.

Area.	Weight per Yard.	Weight per Mile.	Area.	Weight per Yard.	Weight per Mile.
Sq. Ins.	Lbs.	Tons. Cwts. Lbs.	Sq. Ins.	Lbs.	Tons. Cwts. Lbs.
5	50	78 11 48	7	70	110 0 0
5¼	52	81 14 32	7½	75	117 17 16
5½	55	86 8 64	8	80	125 14 32
5¾	56	88 0 0	9	90	141 8 64
6	60	94 5 80	10	100	157 3 84
6½	64	100 11 48			

No. of 15-ft. rails per mile . . .	704	No. of 18-ft. rails per mile . . .	587
„ 16-ft. „ „ . . .	660	„ 20-ft. „ „ . . .	528

Weight of rail per yard in lbs. multiplied by $\frac{11}{7}$ equals tons per mile of single track.
Approximately, the weight of a rail in kilograms per metre × 2 = weight in lbs. per yard.

ALLOWABLE RAIL WEIGHT.

If

L = maximum load on one driving wheel in tons; v = maximum speed in miles per hour,

Minimum Allowable Rail Weight in lbs. per yd. = $17 \sqrt[3]{(L + 0{\cdot}0001\,Lv^2)^2}$.

(*Sir Benjamin Baker.*)

Loss of Strength in Steel Rails through Use on Railways.

(*Abstract from Board of Trade Report*, 1900.)

(*a*) The preponderance of fractures near the ends of rails seems to show that the greater straining action due to discontinuity at the joint is a contributing cause of fracture, and this can be remedied by adopting rails of sufficient strength with webs of ample thickness and secure types of fastening, and by care that no looseness arises in service.

(*b*) The fact that worn rails are improved in strength and ductility by annealing proves that part of the deterioration of rails in service is of the nature of what is sometimes termed 'fatigue.'

(*c*) It appears certain that in some cases fractures of rails have been due to fissures formed during service. How far the minute transverse fissures, very often noticed in the running surface of old rails, give rise to these larger induced flaws requires further investigation. It is not likely that they usually spread into the substance of the rail, because they are common in old rails, and fractures would be more frequent than they are if that were the case. Also, the evidence as to the existence of visible flaws or defects in the fractured surface of rails is very conflicting. In some cases, undoubtedly, the combined effects of the weakening of a rail by wear and corrosion, possibly also increased straining action from defective packing of sleepers, and the presence of a flaw or fissure of not inconsiderable size have led to fracture of the rail. That such defects appear most commonly in the head of the rail is evidence to a certain extent that they are induced by the hammer hardening of the top surface.

(*d*) It is very desirable that the mechanical tests to which rails are subjected should be as far as possible standardised in connection with (1) the weight; (2) the section; and (3) the chemical composition of the rail.

As regards chemical composition the Committee does not think it desirable to insist upon too high a proportion of carbon, manganese, or silicon in the steel, having regard to the ordinary contingencies of manufacture, and the greater susceptibility of high carbon steel to thermal influences.

Mr. E. Windsor Richards (in conjunction with Mr. E. P. Martin, of Dowlais) considers, after careful consideration from the points of view of a manufacturer and user, that, exclusive of the iron, a steel rail should have the following range of composition:

	Carbon.	Silicon.	Sulphur.	Phosphorus.	Manganese.
Min.	·35	·05	·04	—	·75
Max.	·50	·10	·08	·08	1·00

The falling weight tests showed conclusively that double-headed rails worn in service on one head and afterwards turned down to rest on the chair so as to bring the unworn head into use, are dangerously brittle, owing to the worn head having developed numerous incipient small cracks, which really form 'nicks;' and such practice of using rails is recommended to be immediately discontinued.

THERMAL INFLUENCES.

Seasons at which Fractures most frequently occur.—Mr. Footner has given the following data from twelve years' experience on the London and North Western Railway: Fractures in spring, 29 per cent.; in summer, 16 per cent.; in autumn, 22 per cent.; in winter, 33 per cent. There is other evidence as to the frequency of fractures in winter, and no doubt when the permanent way is hardened by frost the effect of concussions on the rails is much more serious. There is no satisfactory evidence that the steel itself is sensibly weakened by any ordinary cold temperature.

According to all statistics more than 50 per cent. of fractures occur before the rails have had a service of fifteen years.

Mr. Footner states that of fractured rails

3 per cent. broke after ½ to 5 million carrying 5 to 50 tons. | 45 per cent. broke after 50 to 100 million carrying 100 to 200 tons.
35 „ „ „ „ „ | 17 „ „ „ „ „

So that 38 per cent. of the broken rails had carried less than 50 million tons.

According to Dormus, if f = breaking stress of rail steel in tons per square inch; l = gauge length in inches; Δ = ultimate elongation of length l in inches; then $100f^2\Delta \div l$ should not be less than 22,000; only the worst qualities give a lower value than 16,120.

Sir I. Lowthian Bell does not think that the Committee (as the result of the investigation) should recommend any change in the mode of management of the permanent way of the railways of the United Kingdom.

BREAKING LOAD OF STEEL RAILS.

At five feet bearings the breaking load of steel rails in pounds may be taken at 1,000 × weight of rail per yard in pounds.

Chemical analyses of average Bessemer rail steel, as made at present for heavy rails, run about as follows:—

	Per cent.		Per cent		Per cent
Carbon . .	·50	Sulphur . . .	·08	Manganese . .	1·00
Phosphorus . .	·10	Silicon . . .	·12	Iron . . .	98·20

(F. E. Abbott.)

MAXIMUM ALLOWABLE AXLE LOAD.

If

W = weight of rail in lbs. per yd. V = maximum booked velocity of trains in miles per hour, then,

$$\text{Maximum allowable axle-load in tons} = \frac{W^2}{200}\left(1 + \frac{V}{60}\right)$$

Rail-weight—lbs. per yd. . . .	40	50	60	75	85	90	100
Maximum booked velocity—miles per hour	15	22½	30	40	45	50	60
Maximum axle-load—tons . . .	6·4	9	12	16·7	18·3	22	25

(Remfry, 'Proc. Inst. Civ. Engrs.,' vol. ccxv. p. 338.)

Weight of rail in lbs. per yd. . .	30	35	40	45	50	55	60	65	70
Maximum axle loading in tons	4 to 6	5 to 7	6 to 8	7 to 9	8 to 10	9 to 11	11 to 13	13 to 15	15 to 17

(Ministry of Transport.)

Each 10 lbs. per linear yd. of rail on sleepers, not less than 14 per 30 ft. rail (2,464 per mile), should carry a safe load per wheel of 3,000 lbs. (1·34 tons).

(Baldwin Locomotive Works.)

The Government of India allows an axle-load of one-fifth of a ton for each pound per yard of rail-weight, but this rule is not now rigidly enforced.

DETECTION OF FATIGUE CRACKS IN AXLES.

The following method of testing axles for fatigue cracks is employed in some American railway shops.

The wheels are removed from an axle which is to be tested, and the surface of the axle near the fillets, where almost all fatigue cracks occur, is rubbed with oil and then wiped clean.

If cracks are present the oil works down into them, and is not removed when the surface is wiped clean. The surface is then painted over with a wash consisting of whiting and alcohol.

When this coating is dry the axle is rotated and struck smartly with a hammer. This hammering squeezes the oil out of the minute cracks and discolours the whiting, thus locating the cracks.

(University of Illinois Bulletin.)

Fish Plates.

BRITISH STANDARD SPECIFICATION AND SECTIONS FOR STEEL FISHPLATES FOR RAILWAY RAILS. (No. 47—1928.) (*Abstract.*)*

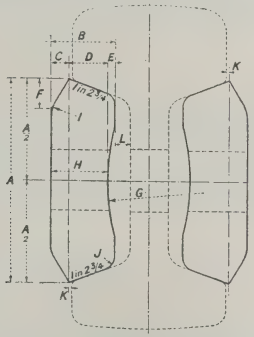


FIG. 5.

TABLE I.—DIMENSIONS OF SECTIONS OF FISHPLATES FOR BULL HEAD RAILS. (Fig. 5.)

No. of B.S. Section of Rail.	70	75	80	90R	100	No. of B.S. Section of Rail.	70	75	80	90R	100
	in.	in.	in.	in.	in.		in.	in.	in.	in.	in.
A	$3\frac{7}{32}$	$3\frac{7}{32}$	$3\frac{7}{16}$	$3\frac{1}{2}$	$3\frac{1}{2}$	G	$4\frac{1}{16}$	$4\frac{1}{16}$	$4\frac{1}{16}$	$4\frac{1}{16}$	$4\frac{1}{16}$
B	1	1	$1\frac{1}{16}$	$1\frac{1}{8}$	$1\frac{1}{8}$	H	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{15}{16}$	1	1
C	$\frac{9}{32}$	$\frac{9}{32}$	$\frac{5}{16}$	$\frac{21}{64}$	$\frac{21}{64}$	I	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$
D	$\frac{19}{32}$	$\frac{19}{32}$	$\frac{5}{8}$	$\frac{43}{64}$	$\frac{43}{64}$	J	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$
E	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	*K	$\frac{17}{64}$	$\frac{17}{64}$	$\frac{9}{32}$	$\frac{9}{32}$	$\frac{9}{32}$
F	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{17}{32}$	$\frac{17}{32}$	*L	$\frac{17}{64}$	$\frac{17}{64}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$

* K and L are given to the nearest $\frac{1}{64}$ inch.

WEAR PLATE FOR RAIL JOINTS.

A permanent way device which is being introduced on American railways to reduce the trouble with worn rail joints, is developed from a makeshift practice in vogue on both European and American railways. The arrangement, which has been introduced by the American Fork and Hoe Company, consists of a forged steel filler plate, fig. 6, which fits between the contact surfaces. This plate is 9 ins. to 12 ins. long, with a maximum thickness of about $\frac{1}{16}$ in. to $\frac{1}{8}$ in. for a length of 2 ins. or 3 ins. at the middle, and then tapering to the thickness of about No. 20 gauge sheet metal at the ends. It is $1\frac{1}{8}$ in. in width, and is bent to an angle shape, so that it is securely held in position. The length and thickness vary according to the design of rail joint and the amount of wear between rail and fish-plate. The device is applied easily by the track men, and is claimed to be an economical remedy for a rather frequent trouble.

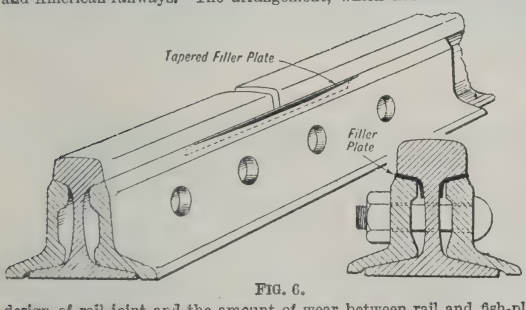


FIG. 6.

* By permission of the British Engineering Standards Association.

Cross-Overs.

Shortly after the Central London Railway was opened, practically all the cross-overs on it were changed to manganese steel. They have been in use for over ten years, and there has been no necessity for renewing them. With ordinary steel the life, with the traffic which the Central London carries, would be only from three to six months.

Check Rails.

Although practically all British railways state that they use a check rail on curves of 660 ft. or less radius in order to strengthen the track, the generally recognised purpose of the check rail is to safeguard rolling stock against derailment. The provision of a check rail on curves of 10 chains or less radius is compulsory under the regulations issued formerly by the Board of Trade and now administered by the Ministry of Transport. The Great Central section of the London and North Eastern Railway extends the use of a check rail to all curves under 20 chains radius, and, further, staggers the joints of the two running rails where the curve is of 30 chains radius or less.

The standard design of check chair now used for check rails allows a gap of 2 ins. only.

Specification for Bolts, Nuts, Screws, and Spikes.

London Midland and Scottish Railway (North Western Section.)

Specification for Steel Bolts and Nuts.

The steel from which the bolts and nuts are made is to have a tensile strength of not less than 36 tons per square inch, with an ultimate extension of not less than 20 per cent. upon 10-inch lengths. The heads are to be worked out of the solid, and not welded on, and are to be stamped with the letter 'S,' as approved by the engineer. The heads and nuts are to be at right angles with the centre line of the bolts, and central with the shank, which must be perfectly straight. They must be screwed to Whitworth's standard, and the threads must be clean and smooth.

Specification for Galvanised Steel Screws for Rail Chairs.

The steel from which the screws are made is to have a tensile strength of not less than 36 tons per square inch, with an ultimate extension of not less than 20 per cent. upon 10-inch lengths. The heads are to be worked out of the solid, and not welded on, and are to be at right angles with the centre line of the screws, and central with the shank, which must be perfectly straight. Each screw is to have a deposit of zinc of not less than 180 grs. smoothly and uniformly distributed all over.

Specification for Steel Spikes.

The steel from which the spikes are made is to have a tensile strength of not less than 36 tons per square inch, with an ultimate extension of not less than 20 per cent. upon 10-inch lengths. The heads are to be worked out of the solid, and not welded on, and are to be at right angles with the centre line of the spikes and central with the shank, which must be perfectly straight. The heads are to be stamped with the letter 'S,' as approved by the engineer.

Wooden Sleepers.

These are usually made of Baltic red timber, and measure 9 ft. long, 10 ins. wide, and 5 ins. deep; at the joints the width is 12 ins. Each load of 50 cubic feet should be impregnated with from 40 to 50 gallons of creosote, equal to from 2½ to 3 gallons per sleeper.

Plugging Worn Holes in Wooden Sleepers.

A plastic asphaltic composition is being used by the Western Australian Government Railways for filling worn spike holes in wooden sleepers. The composition is poured in hot, the spike is re-driven and in about 15 minutes it is so secure that a pull of 4,000 lbs. is required to draw a ⅝-in. spike. The composition is given as follows:—1 quart tar, 4 ozs. pitch, 1 lb. bitumen, ½ pt. kerosene, 2 lb. hardwood sawdust screened to remove material that will pass a 100-mesh screen.

Plugged sleepers have been used with marked success in large numbers by Chief Engineer Fredericia, of the Danish State Railway. A plain 1½-in. cylindrical creosoted plug of beech or birch is driven tight in a 1½-in. hole bored in the sleeper, and the spike is driven in a hole bored

in this plug. Worn-out sleepers plugged in this way have been found to give good service, as the rail seems to be held down with exceptional firmness, and deterioration due to the pounding of loose rails is prevented.

Length of Wooden Sleepers.

If

L = length in ft.; G = gauge in ft.

$$L = 1{\cdot}45, G + 0{\cdot}5$$

with a minimum of the gauge + 2 ft.

	Ft. Ins.	Ft. Ins.	Ft. Ins.	Ft. Ins.	Ft. Ins.	Ft. Ins.
Gauge . .	5 6	4 8½	3 6	*metre*	2 6	2 0
Sleeper length . .	8 6	7 3	5 7	5 3	4 6	4 0

(*Remfry*, '*Proc. Inst. Civ. Engrs.*,' *vol. cxv. p.* 339.)

Creosote Treatment of Sleepers.*

A special creosote treatment process of sleepers has been adopted by the Canadian Pacific Railway Company as follows:—

Prior to creosoting, incisions are made in the surface of the sleepers lengthways with the grain of the wood. The number, depth, and arrangement of these incisions are varied according to the kind of wood to be treated. The sleepers are then creosoted in the ordinary way.

Distribution of the preservative, both lengthwise with and across the grain, proceeds from each incision. The rate of distribution varies with the species of wood, moisture content, character of creosote oil, and details of treatment. The number and arrangement of incisions used

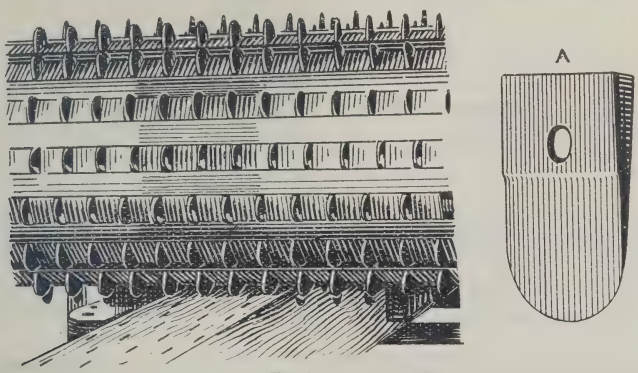


Fig. 7.

for any given kind of wood should be such that the penetration from each incision meets that from the adjoining ones in a reasonable time, thus forming a continuous treated zone of uniform thickness completely enclosing and protecting the inner and untreated portion of the wood. The thickness or depth of the treated zone is controlled by the depth of the incisions. The length of time of pressure necessary to secure a given depth of penetration in incised timber is much less than that required to obtain the same depth in the same timber not incised.

The sleepers are incised by passing them through a machine, fig. 7, provided with steel rollers equipped with removable incising cutters or members. The particular type of incising member adopted is shown by A. The cutters do not crush or break the fibres of the wood,

* W. Kynock, B.Sc.F., F.E., and J. A. Coderre, B.A., F.E. Forest Products Laboratories of Canada, Montreal. *The Engineer*, Feb. 9, 1923.

as some other forms tend to do, but merely press them apart slightly. This is so much the case that after treatment the incisions, narrow to begin with, close up and are hardly noticeable. Tests indicate that the reduction in strength due to incising is so small as to be quite negligible for practical purposes.

Incising presents the following advantages :—

(1) In the case of easily penetrable woods a very much better treatment than is usual without incisions, using less creosote oil per cub. ft. than is ordinarily specified.

(2) The successful creosote treatment of certain refractory woods.

(3) A shortening of the customary pressure period by at least one-half in the case of easily penetrable woods, and therefore—

(4) A considerably increased plant capacity with given equipment.

(5) Satisfactory penetration in every sleeper in a cylinder charge.

See also p. 361.

Steel Sleepers.

The Southern Railway has placed (1928) an order for 70,000 (5,000 tons) of steel sleepers of a special form for use with bull-headed rails, the chairs being an integral part of the sleepers, and the only separate parts being the two base-plates and the two keys for each sleeper. The supply ordered will enable about thirty-five miles of selected track to be equipped over a period of two years.

The sleepers are of the ' Sandberg ' type as shown by fig. 8.

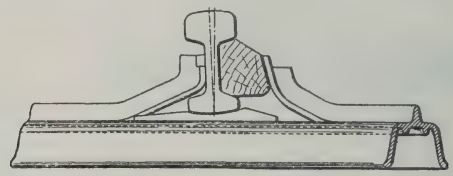


FIG. 8.—Sandberg Steel Sleeper.

The sleepers are 8 ft. long, special rolled section, with deep centre rib, pressed to shape, and with closed ends. Overall width when pressed, 11¼ ins.; depth of sleeper, 2½ ins.; height of rib, 1$\frac{1}{16}$ ins.

The key and rail jaws are formed by punching out the metal in the bed of the sleeper 3 ins. wide, and pressing to suitable shape. The centre rib adds considerably to the strength of the jaws and gives the appearance of a bent tee-iron stiffener.

A loose bearing plate, 8 ins. wide, forms the rail bed and helps to distribute the pressure over the bed of the sleeper plate where it is punched out for the jaws. A slot is provided, which engages the rail jaw, and prevents the bearing plate moving laterally.

Weight.—

Steel sleeper plate, 8 ft. long . . .	142 lbs.
Including two loose bearing plates . .	16 „
Total weight	158 lbs.

A special oak key shaped to suit the key jaw is used.

Fig. 9 shows a ' Harvey ' steel sleeper which is largely in use in India and the Crown Colonies. It is 8 ft. long, by 10¾ ins. overall width at rail seat, by 3½ ins. deep at rail seat, and is ' waisted ' at the centre in the usual manner. The ' waisting ' of the sleeper adds considerably to its strength and, in addition, tends to prevent the ballast working towards the middle and the sleeper becoming centre-bound.

Cast-iron chairs are securely held in correct position on the steel sleeper by means of metal clips attached to the sleeper by a patented process of spot-welding.

The chair has a spigot, as shown, cast on the underside, which engages a socket or cup-shaped depression in the bed of the sleeper.

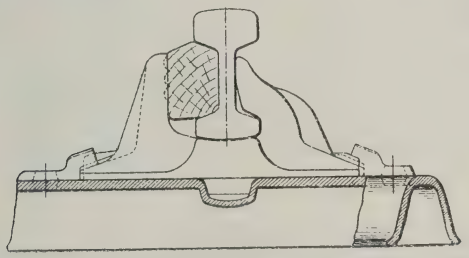


FIG. 9.—Harvey Steel Sleeper.

To attach the chair it is placed across the sleeper with the spigot engaging the socket, and turned into position, and firmly secured by the clips, which are arranged to grip the upper surface of the ends of the base of the chair.

Weight.—

Steel sleeper with clips	140 lbs.
Two cast-iron chairs	72 „
Total weight	212 lbs.

The rail is keyed in position with a standard oak key.

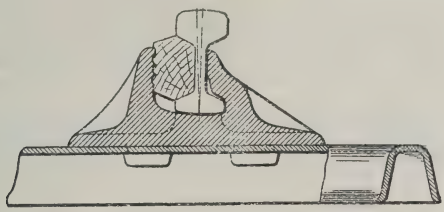


FIG. 10.—G.K.N. Composite Steel Sleeper.

This sleeper, fig. 10 (Guest, Keen & Nettlefolds, Ltd.), is of the rolled and pressed steel trough type, with closed ends, 8 ft. long by 11 ins. overall width at the rail seat by 3½ ins. deep at the rail seat; it is made parallel in width about the centre line of the rail, the centre of the sleeper being 'waisted,' thereby strengthening the tie and preventing the sleeper becoming centre-bound.

Modified standard cast-iron chairs are cast in correct position integrally with the sleeper, it being claimed by the makers that with this process the action of the cast metal contracting during cooling absolutely secures the chair plates to the steel sleeper plates, with an effective joint

Weight.—

Steel sleeper plate, 8 ft. long . . .	144 lbs.
Two cast-iron chairs . . .	90 „
Total weight	234 lbs.

The rail is keyed in position with a standard oak key.

Metal Keys.

A patent metal key (Mills' patent) which is largely used on one of the Indian railways is shown by fig. 11. This key, which is of steel, in one piece, is stated to be 'fool proof,' and requires

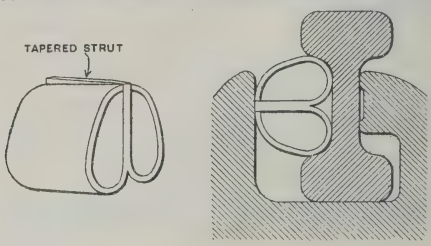


FIG 11

the same tools as are used for fitting wooden keys. The advantages claimed over the wooden key are: (1) the minimising of rail creep, (2) unaffected by climatic conditions, (3) longer life, (4) improved track alignment.

Ballast.

Cubic Yards per Mile of Ballast.

Depth of Ballast in Inches.	Side Slope of Ballast, 1 to 1. Width in Feet.					
	Single Track.			Double Track.		
	10	11	12	21	22	23
12	2,151	2,347	2,543	3,303	4,499	4,694
18	3,373	3,667	3,961	6,601	6,895	7,188
24	4,700	5,085	5,477	8,997	9,389	9,780
30	6,111	6,601	7,090	11,491	11,980	12,469

The depth of the ballast should be 2 feet below the top of the rails—that is, 1 foot 7 inches in thickness above the formation level. The ballast is usually laid in a double thickness, the upper layer consisting of stones not exceeding 3 cubic inches in bulk, and averaging about 2 cubic inches. The lower layer may be of stones varying in size from 3 to 30 cubic inches in bulk.

The width of the top ballast beyond the cross-sleepers should be about 18 inches, the total width in the case of a single line of rails being 12 feet; in the case of a double line, 23 feet; in the case of a triple line, 34 feet; and for a quadruple line, 45 feet. Slope of ballast at sides, 1 to 1.

Blast-furnace and steel slags are much used as ballast where supplies are readily available. Cinder ballast is now being superseded by stone or slag where a sufficient supply can be obtained at an economical cost; but where the railway is subjected to subsidences cinder ballast is used, otherwise successive liftings of the tracks would cause large quantities of the ballast gradually to disappear below the ordinary level of the ballasting. The use of slag ballast is generally avoided in tunnels, because in damp and confined situations it is considered to increase corrosion of the track; it is not used on the electrified lines of at least one administration, on account of certain chemical properties, and also because it is considered too retentive of moisture: this opinion is, however, not general, as it is ordinarily laid on other important electrified railways.

Gravel, when sufficiently cheap, makes excellent ballast (see p. 389).

Details of Permanent Way.

Clear space outside rail, 4 ft. Middle space between two sets of rails, 6 ft. Clear space outside rail, 4 ft. Least head for the top of ballast, and least width for archways and tunnels, 13 ft. 1½ in. for single lines; 24 ft. 3 ins. for double lines. Spaces required for slopes of ballast and benches beyond them on top of embankments, from 3 ft. 10½ ins. to 8 ft. 10½ ins. for single lines; and from 3 ft. 9 ins. to 8 ft. 9 ins. for double lines. Total breadth on top of embankments, from 17 ft. to 22 ft. for single lines; and from 28 ft. to 33 ft. for double lines.

Ministry of Transport Regulations for Permanent Way.

British Standard Specifications and Sections for all details of permanent-way to be adopted, unless authority to the contrary is given. On lines on which the normal traffic is heavy and worked at high speed, the weight of new rails should not be less than 85 lbs. per yard, and, in the case of a chaired road, the common chairs should not weigh less than 45 lbs. each. On lines where traffic conditions are less severe, chairs weighing not less than 40 lbs. each may be used. The minimum length of rail, as a rule, to be 30 ft. On lines normally used for light traffic and moderate speeds, lower weights of rails and chairs may be used.

Chairs, if used, must be secured to the sleepers, at least partially, by metal bolts, screws, or spikes. With flat-bottom rails, or bridge rails, the fastenings at joints and at one or more intermediate places to consist of fang or other through bolts, and such rails, on curves with radii of 15 chains or less, to be tied to gauge by iron or steel ties at suitable intervals.

Fixed diamond crossings must not be flatter than 1 in 8 except in special circumstances. Movable diamond crossings may be at any angle and are to be treated as worked facing points.

Lengthening Life of Track.

Perhaps the chief effort made towards lengthening the life of the track is annual lubrication of the fished rail joints, which, introduced some years ago, appears likely to become more generally adopted. The lubricants or preservatives—they serve as both—are tar or recovered machine oil. One railway is experimenting with the grease recovered from the gear-boxes of electric rolling stock; having more 'body,' it is anticipated that it will not dry up so soon as the 'thinner' substances. One Indian railway states that the fishplates are graphited and the nuts and bolts oiled once a year.

Rails laid in tunnels are in some cases painted with red lead, and in others are coated with tar. The painting or coating is usually done only when the rails are first laid; on one of the Irish railways, however, the tarring of rails and fishplates in tunnels and of fishplates exposed to the action of the sea is done periodically, and is said to have very materially prolonged their life. A similar practice is being tried on the Nigerian Railway in situations on the coast.

On the North Eastern section of the London and North Eastern Railway, when the bearing surfaces on the rails have become so worn as to cause ordinary fishplates to have little or no grip, it is found that the use of 'maintenance' fishplates, which are slightly greater in depth than the normal fishplates, materially reduces the wear and tear on the sleepers and fastenings at and close to the joint.

On the Great Northern section of the same railway sidecut rails are turned on all running lines. This practice is followed on several railways, and is being contemplated on others which previously discarded for main running lines rails the head of which is much worn on one side. The Canadian National Railways exchange the inner and outer rails on curves when wear has taken place; and another method, used in South Africa, is to exchange the rails in the curved and straight portion of the line.

On the District Railway, where the rails are particularly subject to uneven wear or 'battering,' used rails are taken up, machined to an even surface on the top, and replaced; this gives an increased life of about 100 per cent. while the cost of machining is only about 10s. a ton. The building up of used or damaged switches while in the track by oxy-acetylene flame is being tried in Canada.

Cost of Railway Construction.

In England, taking a fairly representative section of one of the main lines, outside the Metropolis, and constructed in recent years,* as an example, the percentages of the cost of about 40,000*l.* per mile, work out somewhat as follows:—

	Per Cent.
Land and compensation	10
Fencing	1½
Earthworks	24
Tunnels	12
Viaducts	8
Bridges for roads	9
Accommodation works	2
Culverts and drainage	5
Permanent way, ballast, including main line	11½
Sidings	3
Junctions and signals	1
Stations, including buildings (roadside only)	6½
Contingencies, including parliamentary, administration, legal and engineering expenses	6
Maintenance	½
	100

* Pre-war period.

LONDON MIDLAND AND SCOTTISH RAILWAY (NORTH WESTERN SECTION).
DETAILS OF PERMANENT WAY.
General Arrangement.

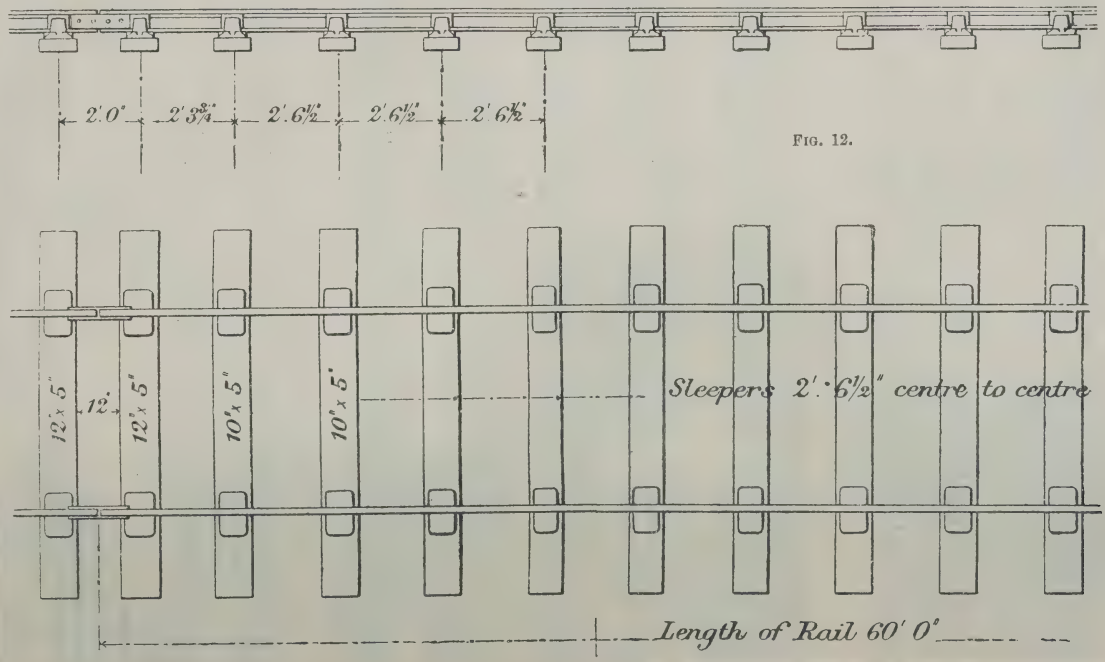
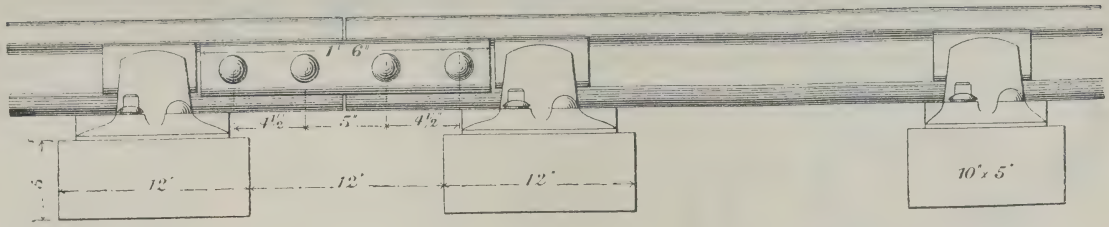


FIG. 12.

LONDON MIDLAND AND SCOTTISH RAILWAY (NORTH WESTERN SECTION).

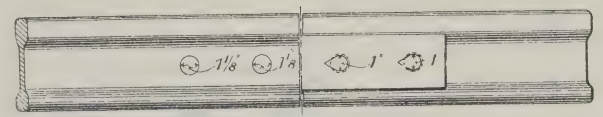
DETAILS OF PERMANENT WAY.

Rail Joints.



GENERAL ELEVATION OF JOINT

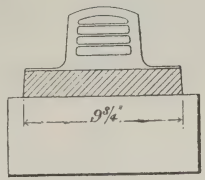
FIG. 13.



ELEVATION SHOWING HOLES IN RAIL

ELEVATION SHOWING HOLES IN FISHPLATE

FIG. 14.



SECTION OF JOINT CHAIR

WEIGHT 54 LBS.

FIG. 15.

L. M. & S. Railway (North Western Section).

DETAILS OF PERMANENT WAY.

Rail Setting.

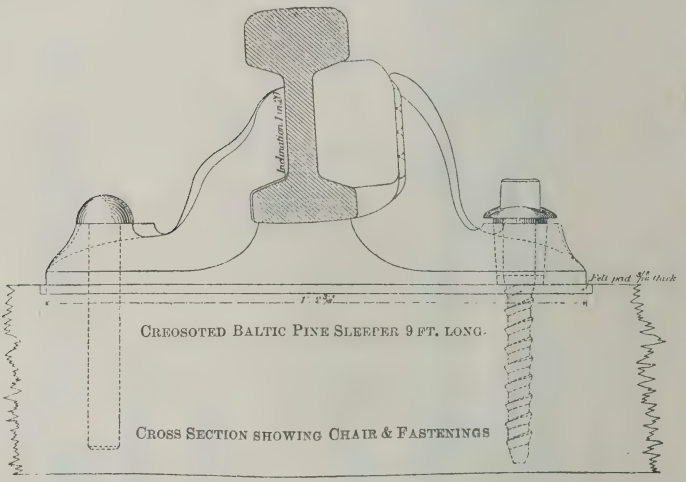
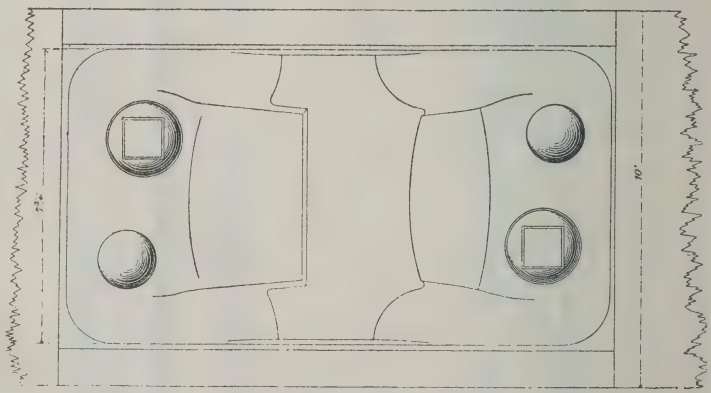


Fig. 16.



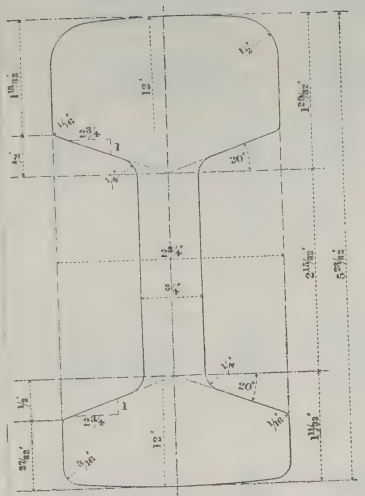
Plan of Intermediate Chair

(Rail and Key removed).

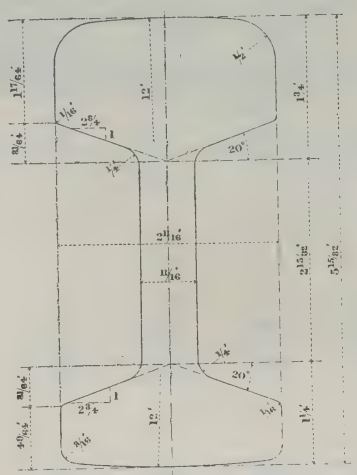
Weight 45 lbs.

Fig. 17.

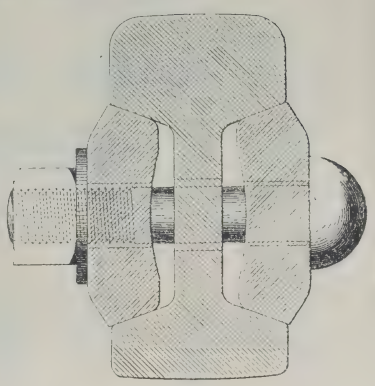
B.S. Rail Sections.



95 lbs. per yard. For Main Lines.



85 lbs. per yard. For Branch Lines



Section of Rail and Fish Plates.

Ferrule.

Bolts, Spikes, and Keys.

Bolt. Weight, 1 lb. 12 ozs.

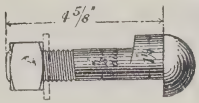
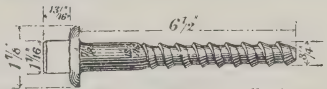
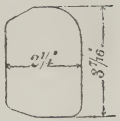


FIG. 18.



Galvanised Screw. Weight, 1 lb. 4 ozs.

Spike. Weight, 1 lb. 2½ ozs.



Oak Key.
6 ins. long

LONDON MIDLAND AND SCOTTISH RAILWAY (MIDLAND SECTION).

DETAILS OF PERMANENT WAY.

General Arrangement.

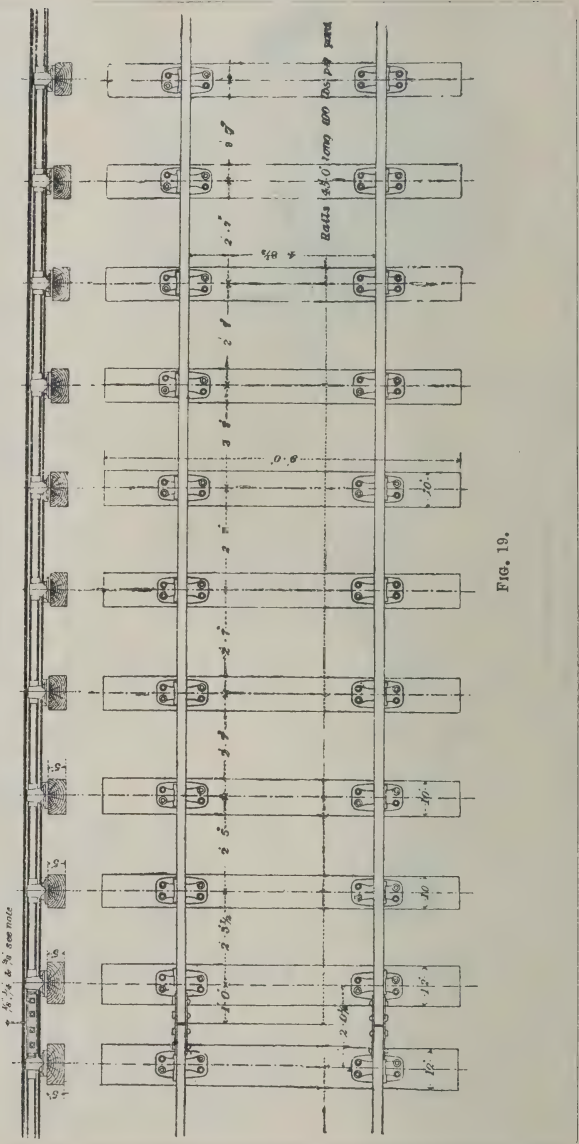


FIG. 19.

L. M. & S. Railway (Midland Section).

DETAILS OF PERMANENT WAY.

Rail Joints.

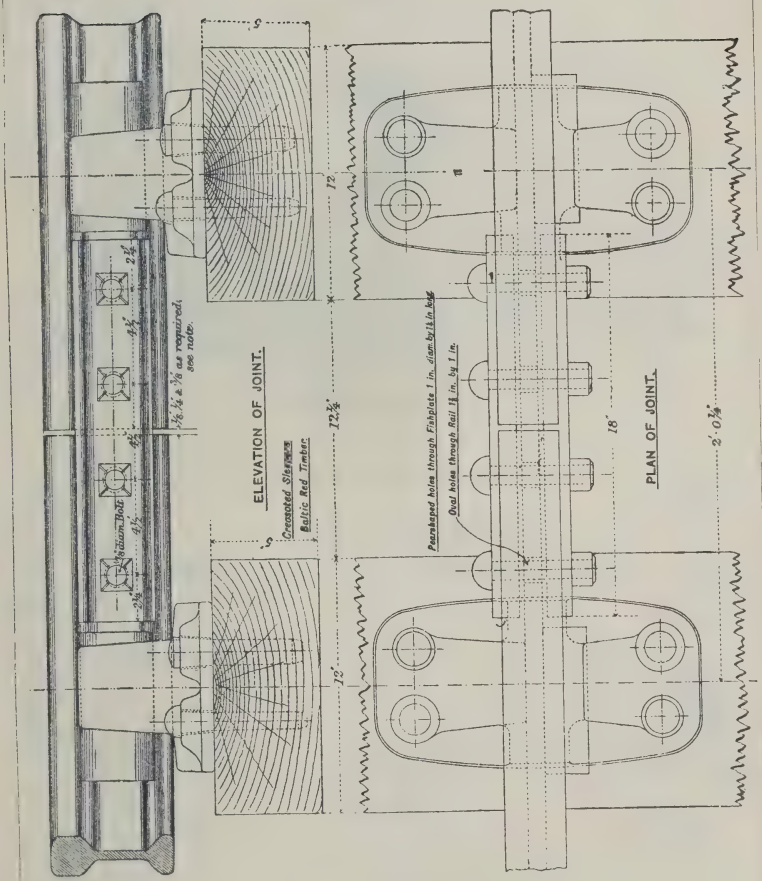
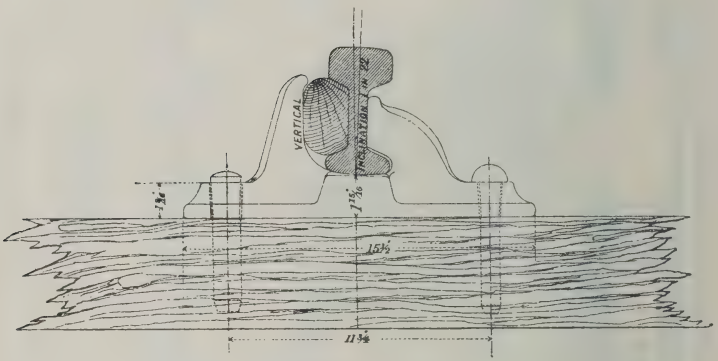


Fig. 20.

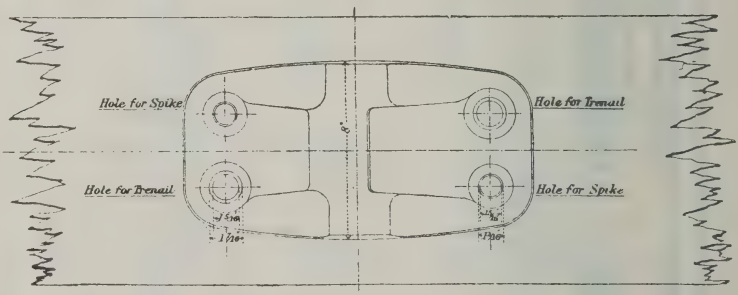
L. M. & S. Railway (Midland Section).

DETAILS OF PERMANENT WAY.

Rail Setting.



ELEVATION OF CHAIR.



PLAN OF CHAIR.

Fig. 21.

L. M. & S. Railway (Midland Section).

DETAILS OF PERMANENT WAY.

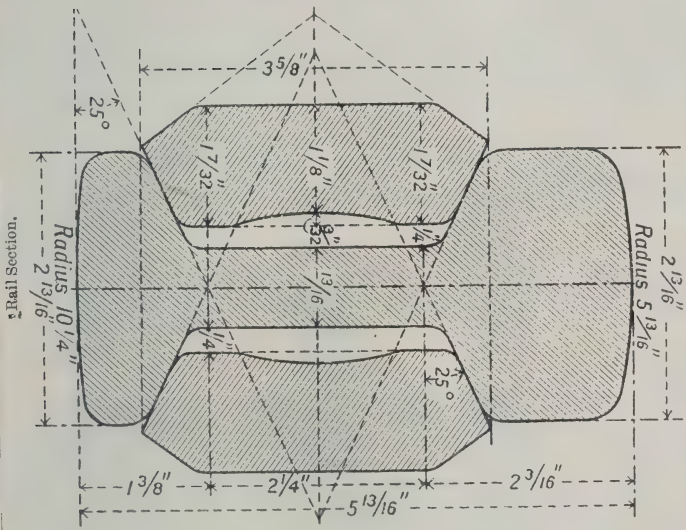
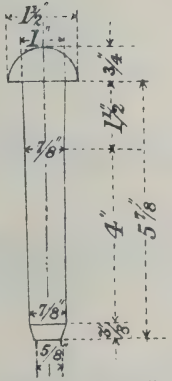
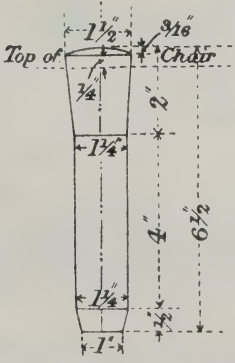


Fig. 22.

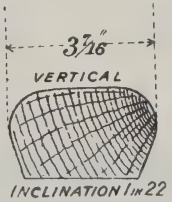
Spikes, Trenails, and Keys.



Wrought Iron Spike.



Solid Oak Trenail
(Before Driving).



Oak Key
7 ins. long.

Fig. 23.

From this it will be seen that, for country lines, the value of land does not form so important a factor in the total cost as is generally understood. The necessity for very easy gradients, and flat curves, in order to admit of high speeds, involves heavy earthworks, viaducts and tunnels, and expensive, well-ballasted permanent way, whilst stations and sidings have to be extensive to accommodate the traffic.

Experience has shown how to adapt the design of railways to their special surroundings. Their cost has varied from about 630,000*l.* per mile in London, and from an average of 52,378*l.* per mile in the United Kingdom, and of 8,127*l.* per mile in India, to a cost, in some parts of South Africa and South America, of 3,200*l.* to 3,500*l.* per mile, or, with equipment, of 4,000*l.* per mile—a rate at which, in new countries and under fairly normal conditions, railways can be provided, suitable for a speed of 30 miles per hour, and for heavy trains. (*Sir Douglas Fox.*)

Table showing Gauges of Various Railways

England, Scotland, and Wales	4 ft. 8½ ins. (1·435 metres)
Ireland	5 ft. 3 ins. (1·59 metres)
Canada	4 ft. 8½ ins.
New South Wales	4 ft. 8½ ins.
South Australia	5 ft. 3 ins.
Queensland	3 ft. 6 ins. (1·067 metres)
Victoria	5 ft. 3 ins.
Cape Colonies	3 ft. 6 ins.
New Zealand	3 ft. 6 ins.
British India	5 ft. 6 ins. (1·676 metres, & 3 ft 3⅜ ins. (1 metre)
Ceylon	5 ft. 6 ins.
Egypt	4 ft. 8½ ins. & 3 ft. 6 ins.
France	4 ft. 9 ins. (1·45 metres), 1·44 metres, & 3 ft. 3⅜ ins.
Belgium	4 ft. 9 ins.
Holland	4 ft. 9 ins.
Germany	4 ft. 8½ ins.
Austria	4 ft. 9 ins.
Switzerland	4 ft. 9 ins.
Italy	4 ft. 8½ ins. (1·445 metres)
Hungary	4 ft. 9 ins.
Spain	5 ft. 5⅝ ins. (1·67 metres)
Portugal	5 ft. 5⅝ ins. (1·67 metres)
Norway	4 ft. 8½ ins. & 3 ft. 6 ins.
Sweden	4 ft. 8½ ins.
Denmark	4 ft. 8½ ins.
Greece	1·44 metres
Poland	4 ft. 8½ ins.
Russia	5 ft. (1·524 metres)
Finland	5 ft.
Turkey	4 ft. 8½ ins.
Czechoslovakia	4 ft. 8½ ins.
Jugo-Slavia	4 ft. 8½ ins.
Lithuania	4 ft. 8½ ins.
Rumania	5 ft. & 4 ft. 8½ ins.
Sardinia	4 ft. 9 ins. & 3 ft. 1½ ins. (·95 metre)
United States	4 ft. 8½ ins.
Mexico	4 ft. 8½ ins. & 3 ft. (·91 metre)
Peru	4 ft. 8½ ins.
Brazil	3 ft. 3⅜ ins. & 5 ft. 3 ins.
Chili	5 ft. 6 ins. & 1 metre
Argentina	5 ft. 6 ins. & 1 metre
Uruguay	4 ft. 8½ ins.
Japan	5 ft. 6 ins.
China	4 ft. 8½ ins.

Widening of Gauge on Curves.

Under 8° . standard gauge (4 ft. 8½ ins.).
Above 8° . ¼ in. for each 2° or fraction thereof, up to a maximum of 4 ft. 9½ ins.

(*American Railway Engineers Association.*)

Quantity of Land required for a Line of Railway.

This depends upon the width as determined by the depth of cuttings and height of embankments. Assuming average depth and height of 3 feet, and slopes of 1½ to 1, 5 to 6 acres will be required per mile of single line (4 ft. 8½ ins. gauge), made as in England, exclusive of sidings or stations. For each foot added to or deducted from the average of 3 feet given above (same slopes), 1 yard in width, or ⅛ acre per mile, must be added or deducted.

Light or Narrow-Gauge Railways.

Narrow-gauge railways will be found very suitable in districts where a standard-gauge line could not be made to pay, on account of the greater cost of construction and working of the latter. Narrow-gauge railways occupy less land, and the earthworks are of a less expensive kind than for standard-gauge lines, as heavy embankments and cuttings can be avoided by adopting sharper curves than are admissible with the standard gauge.

For the permanent way light material can be used, the rolling stock can be constructed to give a more favourable proportion between the dead and the paying loads than is possible on

standard-gauge lines, particularly if the traffic is made up of a great number of small consignments. On narrow-gauge lines the average proportion of dead load to paying load may be taken at about 1 to 2·33.

Whether steel or wood sleepers should be used is principally a question of cost, but it should be borne in mind that steel sleepers have a much greater life than wood sleepers.

The gauge and weight of rail should be determined in every case after a careful inquiry into the earning powers of the line. For the rails a weight of 25 lbs. per yard would be about the minimum. The 2-ft. gauge has the great advantage that it can be worked best in combination with portable railways which are joined on to branches of permanent line and continued to the farmyards, fields, or factories. The 2-ft. gauge is especially suitable for private lines and for lines of public utility with limited goods and little passenger traffic. It can be worked at a speed of fifteen miles per hour with curves 100 to 150 feet radius and gradients 1 in 80, for public lines; while on private lines, curves down to 60 feet radius and gradients up to 1 in 25 are admissible, according to the speed required and the amount of traffic. For lines with extensive goods traffic or principally for passenger traffic, 2 ft. 6 in. or 3 ft. gauge would be preferable.

If the goods are of a nature that they will not stand handling, the narrow-gauge wagon bodies could be made of such dimensions that three or four would fit into one main-line truck. A very simple device is also in practical use, of putting the main-line trucks on specially constructed bogies, and thus transporting them over the narrow-gauge railway.

(*A. Koppel.*)

The use of locomotives is not profitable on gauges of less than 18 inches.

The speed on narrow gauges may be as follows:—

2 ft. 0 in. gauge, 15 miles per hour.	3 ft. 0 in. gauge, 25 miles per hour.
2 ft. 6 in. gauge, 20 miles per hour.	3 ft. 6 in. gauge, 30 miles per hour.

Cost of Light Railways.

A fact which has militated against light railway construction is that the cost in Great Britain is very little less than that of building standard railways, ranging from £12,000 per mile in difficult country to £3,333 under easy conditions, while the average return on capital has been only a little over 2 per cent. On the other hand, the Decauville system of light railway adopted in the Argentine can be laid for £1,240 per mile, and the Road-Rail system, as tested in Uganda, South Africa, and Morocco, for £1,250 per mile. Another cheap form of rail transport is the Rutway, which it is said can on ordinary ground, such as the plains of India, be laid down, in lengths of 20 miles or more, at a cost of only about £1,000 per mile, including the provision of rolling stock. In this case the wheels run without any flanges on flat metal surfaces at the bottom of two deep ruts in the lowest part of the permanent way, and are confined to these ruts by the vertical sides of the rail angles and by the stone ballast heaped up along either side of each rail.

See also Section XLV., 'Cost of Engineering Works.'

Formulæ for Calculating the Power of Electric Locomotives for Light Railways.

W = weight of locomotive in tons loaded for service; g = percentage grade (rise per 100); L = weight of load in tons.

$$W = \frac{12+g}{120-g} \times L.$$

T = tractive force in lbs.

$$T = 2{\cdot}24\,(g+12)\,(W+L).$$

H = effective horsepower at driving wheels; S = speed in miles per hour.

$$H = \frac{T \times S}{375}.$$

C = current in ampères; E = electromotive force in volts at the motor terminals; m = mechanical efficiency of motor at 80 per cent. of H, allowing 20 per cent. for friction and electrical leakage.

$$C = \frac{H \times 746}{E \times m}.$$

For example: If

$g = 18$ (*i.e.* 18 ft. per 100 ft.); L = 15 tons; S = 10 miles per hour; E = 500 volts

then,

$$W = \frac{12+18}{120-18} \times 15 = \frac{30}{102} \times 15 = 4{\cdot}4 \text{ tons.} \quad T = 2{\cdot}24 \times 30 \times 19{\cdot}4 = 1{,}304 \text{ lbs.}$$

$$H = \frac{1304 \times 10}{375} = 35 \text{ H.P.} \quad C = \frac{35 \times 746}{500 \times 0{\cdot}8} = \text{about 66 ampères.}$$

Hence the dynamo required would be of $66 \times 550^{*} = 36{,}000$ volt-ampères $= 36$ kilowatts.
Per kilowatt in the dynamo, there would be required, $1\frac{3}{4}$ H.P. in the steam engine,

or for 36 kilowatts, $36 \times 1\frac{3}{4} = 60$ H.P.

The strength of the dynamo must be calculated for the strongest current required.

For an installation intended to supply electricity both for lighting and power, a dynamo of 110 or 220 volts is the most suitable; if, however, the lamps are in circuit, a dynamo of 500 volts may be used. The advantage of the higher voltage is that the conductors may be made of a smaller section, while the loss of electromotive force remains the same.

The maximum loss of electromotive force can be taken at 20 to 25 per cent. of the voltage at the dynamo terminals.

By Ohm's Law,

where, $$E_1 = C \times R$$

E_1 = loss of electromotive force along conductor in volts; C = current in ampères; R = resistance of conductors in ohms.

If

d = diameter of copper wire in inches; W = weight of rails in lbs. per yard; l = length of track in yards, then,

$$R = l\left(\frac{1}{30000\,d^2} + \frac{1}{1100\,W}\right), \text{ and, } E_1 = C \times l\left(\frac{1}{30000\,d^2} + \frac{1}{1100\,W}\right).$$

(A. Koppel.)

Tunnels.

When deeply buried, a tunnel in soft earth is properly formed when its section is an ellipse whose major and minor axes have the ratio $c = \frac{\text{horizontal axis}}{\text{vertical axis}} = \sqrt{\frac{1 - \sin\phi}{1 + \sin\phi}}$, where ϕ is the angle of repose of the material. The horizontal axis must never be less than this ratio, but may be greater and will be still safe in firm earth. The whole ellipse is used in drains, but in tunnels the lower fourth or third is omitted, and a flat invert is put in.

Where tunnels are not deeply buried below the surface, the form of section is the 'Geostatic' arch, derived by transformation from the 'Hydrostatic' arch. To design a geostatic arch, given the depth of the crown below the surface $= x_o$, first design a hydrostatic arch and contract its horizontal ordinates in the ratio $c : 1$. This will give the figure required. As the greatest horizontal width will be a fixed quantity, the problem resolves itself into finding the depth below surface of the greatest hydrostatic breadth.

If x_o = depth of crown below surface; s = half span of geostatic arch; ϕ = angle of repose of earth; w = weight of a cubic foot of earth in lbs.; $c = \sqrt{\left(\frac{1 - \sin\phi}{1 + \sin\phi}\right)}$ = ratio of horizontal ordinates of two arches; y = half span of hydrostatic arch $= s \times \frac{1}{c}$; $s = cy$; $c = \frac{s}{y}$; $x_1 =$ depth of greatest horizontal diameter $= x_o + a$; a = rise of arch above greatest diameter; b = a function depending on the relations of x_o, a, and x, then

$$b = a\sqrt[3]{\frac{x_o + a}{x_o}}\ (1);\quad y = b - \frac{b^2}{30a}\ (2);\quad b = y + \frac{y^2}{30a}\ (3);\quad x_o = a\frac{a^2}{b^3 - a^3}\ (4).$$

By assuming a value for a in (1) the value of b may be found and placed in (2); if a does not agree with its value assumed, further trials can be made until a sufficiently approximate result is obtained.

If r_o = radius of hydrostatic arch at crown; r_1 = radius of hydrostatic arch at greatest horizontal diameter or depth, x_1; k_o = geostatic crown radius $= r_o\,c^2$; k_1 = geostatic radius at depth $x_1 = \frac{r^1}{c}$; then

$$r_o = a + \frac{a^2}{2x_o};\quad r_1 = a - \frac{a^2}{2(x_o + a)};\quad r_o = \frac{a^2}{1}\left(1 + \frac{b^3}{a^3}\right);\quad r_1 = \frac{a}{2}\left(1 + \frac{a^3}{b^3}\right).$$

* This is the electromotive force at the dynamo terminals. Generators for tractive work are usually constructed for 550 volts, and allow of a loss of 50 volts between the dynamo and the motor terminals.

Thickness of Walls = $\sqrt{\cdot 12R}$ up to $\sqrt{\cdot 48R}$, where R = greatest radius = $\frac{a^2}{s}$.

Pressure on Brickwork = $\frac{w[x_1(s+t) - 0{\cdot}8\ a.s]}{t}$ lbs. per square foot, where w = weight per cubic foot of earth.

Breadth of Footings.—If t_1 = breadth of footings, then t_1 must not be less than

$$\frac{W}{2}\left\{\frac{(1-\sin\phi)}{(1+\sin\phi)}\right\}^2, \text{ where W = half width of bricked tunnel.}$$

To Draw the Hydrostatic Arch (figs. 24, 25).—Make $a\,b = r_o$ and $c\,d = r_1$; from d strike an arc with radius $d\,e = f\,b - c\,d$; from a with radius $a\,f$ draw arc cutting this at e. Then d, e, and a are centres to strike half arch.

The radius of the hydrostatic arch at any point is inversely as its depth at that point below surface hence $r_o x_o = r_1 x_1 = r_n x_n$. Through mud the section will be practically the hydrostatic

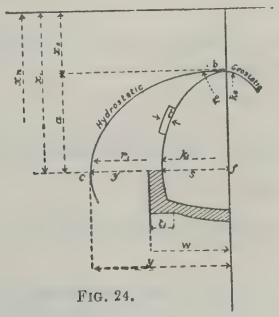


FIG. 24.

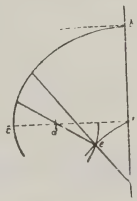


FIG. 25.

unless deeply buried, when the circular form would be correct; for in very soft material an invert resisting as an arch would be impracticable. An invert, however, could be used if strong enough to act as a beam against the upward pressure.

Superelevation of Outer Rail.

When a vehicle runs round a curve it tends either to overturn or to run off the line. The tendency to run off is counteracted by a high centre of gravity for the vehicle, because this throws a heavier pressure on the outer rail and keeps down the wheel flanges from mounting. Low centre of gravity produces detrusive effects upon the rail, and tends to force the flange over the rail.

Every speed has its proper degree of superelevation which brings in gravity to balance centrifugal force. The superelevation necessary for sharp curves and high velocities would cause slow trains to run off the rail at the *inside* of the curve.

In practice the maximum superelevation is limited to 7 or 8 ins., and the speed round the curve is regulated to fit that superelevation.

To determine the Superelevation of the Outer Rail of a Railway Curve.

Let v = speed of train in miles per hour; r = radius of curve in feet; G = gauge in feet.

$$\text{Superelevation in inches} = G \times \frac{v^2}{1{\cdot}25r} = \frac{v^2}{r} \times 3{\cdot}77 \text{ for 4 ft. 8½ ins. gauge.}$$

TABLE SHOWING SUPERELEVATION OF OUTER RAIL FOR A GAUGE OF 4 FT. 8½ INS.

Radius of Curve in Feet.	Velocity in Miles per Hour.				Radius of Curve in Feet.	Velocity in Miles per Hour.			
	10	20	30	40		10	20	30	40
	Superelevation of Outer Rail in Inches.					Superelevation of Outer Rail in Inches.			
500	·75	3·01	6·78	12·05	3,000	·13	·50	1·13	2·01
1,000	·38	1·51	3·39	6·03	4,000	·09	·38	·85	1·51
2,000	·19	·75	1·70	3·01	5,000	·07	·30	·68	1·21

Practical Rule for determining the Superelevation of the Outer Rail of any Curve.

(*Price.*)

Let v = speed of train in feet per second ; c = length of chord, in feet, whose versed sine equals the superelevation required ; G = gauge of railway in feet ; then

$$c = \tfrac{1}{4}\, v\, \sqrt{G}.$$

For 5 ft. 3 in. or 4 ft. 8½ in. gauge, and a speed of 40 miles per hour, c is approximately equal to 66 feet, or a Gunter's chain.

Stretch a 66-foot tape as a chord on the rail of the curve the superelevation of which it is required to find. Measure the distance of the rail from it at its centre point. This distance will equal the superelevation.

RADII OF CURVATURE AND VERSED SINES FOR RAILS.

Radius of Curvature in Feet.	Versed Sines for Rails, the Length in Feet being				Radius of Curvature in Feet.	Versed Sines for Rails, the Length in Feet being			
	15	16	18	20		15	16	18	20
	In.	In.	In.	Ins.		In.	In.	In.	In.
286	$1\frac{3}{16}$	$1\frac{5}{16}$	$1\frac{1}{16}$	$2\frac{1}{16}$	573	$\frac{9}{16}$	$\frac{11}{16}$	$\frac{7}{8}$	$1\frac{1}{16}$
301	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	2	636	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{15}{16}$
318	$1\frac{1}{16}$	$1\frac{3}{16}$	$1\frac{7}{16}$	$1\frac{7}{8}$	716	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{11}{16}$	$\frac{13}{16}$
337	1	$1\frac{1}{8}$	$1\frac{7}{16}$	$1\frac{3}{4}$	818	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{3}{4}$
358	$\frac{15}{16}$	$1\frac{3}{16}$	$1\frac{5}{8}$	$1\frac{11}{16}$	955	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$
382	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{9}{16}$	1,146	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{7}{16}$	$\frac{1}{2}$
409	$\frac{13}{16}$	$\frac{15}{16}$	$1\frac{3}{16}$	$1\frac{7}{16}$	1,432	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{7}{16}$
441	$\frac{3}{4}$	$\frac{7}{8}$	$1\frac{1}{8}$	$1\frac{3}{8}$	1,910	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$
477	$\frac{11}{16}$	$\frac{15}{16}$	1	$1\frac{1}{4}$	2,865	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{3}{16}$
521	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{15}{16}$	$1\frac{1}{8}$	5,739	$\frac{1}{16}$	$\frac{1}{16}$	$\frac{1}{16}$	$\frac{1}{8}$

The above radii are those corresponding (commencing at the top) to angles of deflection from 20 to 1 deg.

Upon sharp curves it is customary to widen the gauge ½ in. for curves of a radius of 750 to 1,000 feet, ¾ in. for a radius of 500 feet, and 1 in. for curves of a less radius.

Angles and Leads of Crossings.

Let R and r = radii of curves in feet; G = gauge of railway in feet; L = lead of crossing in feet (*i.e.* the distance from the nose of the crossing to the nose of the switch); θ = angle of crossing.

Curve Curving off a Straight.

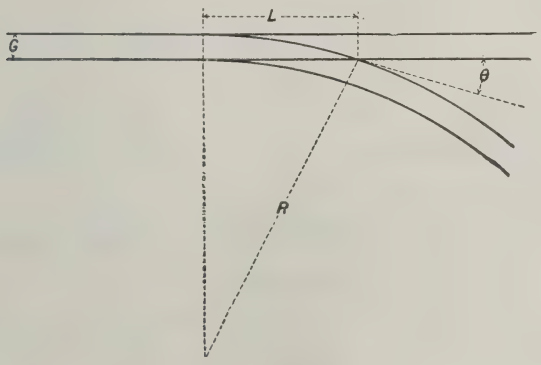


FIG. 26.

$$L = \sqrt{G(2R - G)} = \sqrt{2RG} \text{ (approximately)}; \quad \sin\theta = \frac{L}{R}$$

It is in some instances, where trains have to run fast over a crossing, thought necessary to adopt a curve with a larger radius than usual. In this case it may be considered advisable, in order to prevent an elbow occurring at the heel of the switch, to adopt a longer switch than usual. The length of the switch can be calculated from the following formula when the curve is off a straight.

$$\text{Length of switch} = \sqrt{2R(c + t - e)},$$

where c = the clearance between the rails and the heel of the switch (in feet), this is usually about 2 inches; t = thickness of head of rail; e = the projection of the nose of the switch, if any.

Alternative Data for a Curve Curving off a Straight.

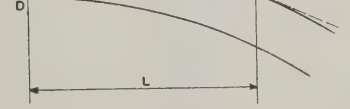


FIG. 27.

Angle BCD = crossing angle; Angle ACD = ½ angle BCD

Lead = L = G × cotan angle ACD.

Straight Track with Straight Switch and Curved Crossing.

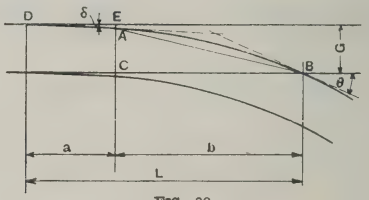


FIG. 28.

Angle ABC $= \frac{1}{2}(\delta + \theta)$; AC = EC − EA = G (guage) − heel opening.

b = AC × cotan angle ABC; a = DE = DA × cos δ = approximate switch-length.

Lead = L = $a + b$.

Straight Track with Straight Switch and Straight Crossing.

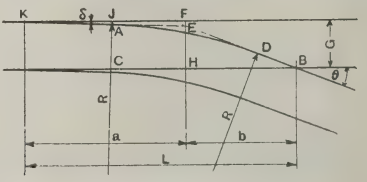


FIG. 29.

$$\sin\delta = \frac{AJ}{AK} = \frac{EF}{AK + AB} \qquad \sin\theta = \frac{EH}{EB} = \frac{G - EF}{DB + ED}$$

If

T = tangent length = AE = ED; a = (KA + T) cos δ; b = (DB + T) cos θ;

$$R = T \operatorname{cotan} \frac{\theta - \delta}{2}$$

Lead L = $a - b$.

Two Curves Curving in Opposite Directions.

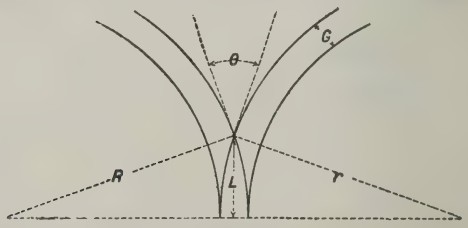


FIG. 30.

$$L = \sqrt{\frac{2RrG}{R + r}} \text{ (approximately)}; \qquad \sin\theta = \frac{2\sqrt{2Rr(R + r)G}}{2Rr + (R + r)G}.$$

Alternative Data for Two Curves Curving in Opposite Directions.

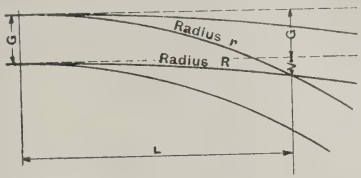


FIG. 31.

$$L^2 = \left\{2r - (G - V)\right\}(G - V) = (2R - V)V; \qquad V = \frac{2r - G}{R - (r - G)} \times \frac{G}{2}$$

Two Curves Curving in the Same Direction.

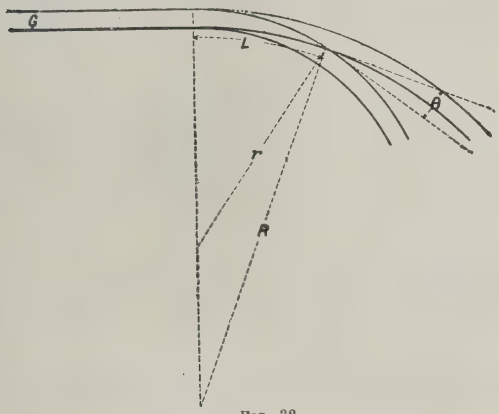


FIG. 32.

$$L = \sqrt{\frac{2Rr G}{R - r}}; \qquad \sin\theta = \frac{2\sqrt{2Rr(R - r)G}}{2Rr + (R - r)G}$$

Alternative Data for Two Curves Curving in the Same Direction.

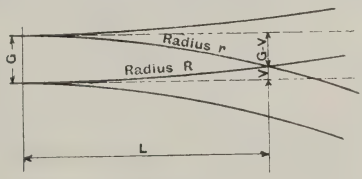


FIG. 33.

$$L^2 = \left\{2r - (G + V)\right\}(G + V) = (2R - V)V; \qquad V = \frac{2r - G}{R - (r - G)} \times \frac{G}{2}$$

Diamond Crossing with both Tracks Straight.

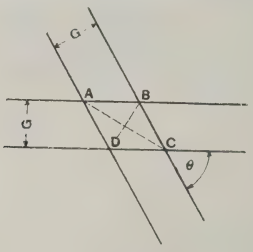


FIG. 34.

$AC = G \times \text{cosec } \frac{1}{2}\theta$ $BD = G \times \text{cosec angle } BDC$; Angle $BDC = \frac{1}{2}(180° - \theta)$.

Diamond Crossing with one Straight and one Curved Track.

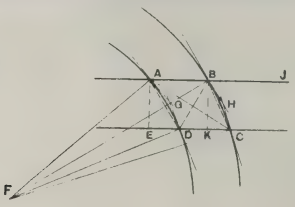


FIG. 35.

Angle $ADE = \frac{1}{2}$(angle BAG + angle EDG); Angle DCF = 90° − angle HCD;
$ED = AE \times \text{cotan angle } ADE$; $EC = ED + DC$ $AC^2 = AE^2 + EC^2$;
$BC = BK + \text{cosec } BCD$

Diamond Crossing with both Tracks Curved.

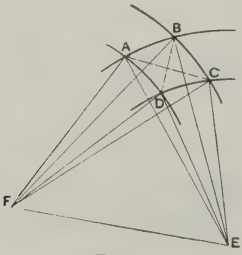


FIG. 36.

$$\cos \text{angle } FAE = \frac{AF^2 + AE^2 - EF^2}{2AF \,.\, AE}; \quad \text{Angle } AEC = \text{angle } FEC - \text{angle } AEF;$$

$$\cos \text{angle } AEC = \frac{AE^2 + EC^2 - AC^2}{2AE \,.\, EC}; \quad \text{Angle } BED = \text{angle } BEF - \text{angle } DEF;$$

$$\cos \text{angle } BED = \frac{BE^2 + DE^2 - BD^2}{2BE \,.\, DE}$$

TABLES OF LEADS AND ANGLES OF CROSSINGS CURVING OFF A STRAIGHT LINE.

I. GAUGE, 5 FEET 6 INCHES.

Radius of Curve (R) in Feet.	Lead (L) in Feet.	Inclination of Crossing, $\frac{1}{\theta}$.	Radius of Curve (R) in Feet.	Lead (L) in Feet.	Inclination of Crossing, $\frac{1}{\theta}$.
400	66	1 in 6·0	1000	105	1 in 9·5
450	70	„ 6·4	1050	107	„ 9·8
500	74	„ 6·7	1100	110	„ 10·0
550	78	„ 7·0	1150	112	„ 10·2
600	81	„ 7·4	1200	115	„ 10·4
650	84	„ 7·7	1250	117	„ 10·6
700	88	„ 7·9	1300	120	„ 10·8
750	91	„ 8·2	1350	122	„ 11·0
800	94	„ 8·5	1400	124	„ 11·2
850	97	„ 8·7	1450	126	„ 11·5
900	99	„ 9·0	1500	128	„ 11·7
950	102	„ 9·3	1550	131	„ 11·8

II. GAUGE, 3 FEET.

Radius of Curve (R) in Feet.	Lead (L) in Feet.	Inclination of Crossing, $\frac{1}{\theta}$.	Radius of Curve (R) in Feet.	Lead (L) in Feet.	Inclination of Crossing, $\frac{1}{\theta}$.
400	49	1 in 8·1	1000	78	1 in 12·8
450	52	„ 8·6	1050	79	„ 13·2
500	55	„ 9·0	1100	81	„ 13·5
550	58	„ 9·4	1150	83	„ 13·8
600	60	„ 10·0	1200	85	„ 14·1
650	63	„ 10·3	1250	87	„ 14·3
700	65	„ 10·7	1300	88	„ 14·7
750	67	„ 11·1	1350	90	„ 15·0
800	69	„ 11·5	1400	92	„ 15·2
850	71	„ 11·9	1450	93	„ 15·5
900	74	„ 12·1	1500	95	„ 15·7
950	76	„ 12·5	1550	96	„ 16·1

III. GAUGE, 3 FEET $3\frac{3}{8}$ INCHES (1 METRE).

Radius of Curve (R) in Feet.	Lead (L) in Feet.	Inclination of Crossing, $\frac{1}{\theta}$.	Radius of Curve (R) in Feet.	Lead (L) in Feet.	Inclination of Crossing, $\frac{1}{\theta}$.
400	51	1 in 7·8	1000	81	1 in 12·3
450	54	„ 8·3	1050	83	„ 12·6
500	57	„ 8·7	1100	85	„ 12·9
550	60	„ 9·1	1150	87	„ 13·2
600	63	„ 9·5	1200	89	„ 13·4
650	65	„ 10·0	1250	91	„ 13·7
700	68	„ 10·3	1300	92	„ 14·1
750	70	„ 10·7	1350	94	„ 14·2
800	73	„ 10·9	1400	96	„ 14·5
850	75	„ 11·3	1450	98	„ 14·8
900	77	„ 11·6	1500	99	„ 15·1
950	79	„ 12·0	1550	101	„ 15·3

IV. Gauge, 4 Feet 8½ Inches.

Radius of Curve (R) in Feet.	Lead (L) in Feet.	Inclination of Crossing, $\frac{1}{\theta}$.	Radius of Curve (R) in Feet.	Lead (L) in Feet.	Inclination of Crossing, $\frac{1}{\theta}$.
400	61	1 in 6·5	1000	97	1 in 10·3
450	65	„ 6·9	1050	99	„ 10·6
500	69	„ 7·2	1100	102	„ 10·7
550	72	„ 7·6	1150	104	„ 11·0
600	75	„ 8·0	1200	106	„ 11·3
650	78	„ 8·3	1250	108	„ 11·5
700	81	„ 8·6	1300	111	„ 11·7
750	84	„ 8·9	1350	113	„ 11·9
800	87	„ 9·1	1400	115	„ 12·1
850	89	„ 9·5	1450	117	„ 12·3
900	92	„ 9·7	1500	119	„ 12·6
950	95	„ 10·0	1550	121	„ 12·8

V. Gauge, 5 Feet 3 Inches.

Radius of Curve (R) in Feet.	Lead (L) in Feet.	Inclination of Crossing, $\frac{1}{\theta}$.	Radius of Curve (R) in Feet.	Lead (L) in Feet.	Inclination of Crossing, $\frac{1}{\theta}$.
400	65	1 in 6·1	1000	102	1 in 9·8
450	69	„ 6·5	1050	105	„ 10·0
500	72	„ 6·9	1100	107	„ 10·2
550	76	„ 7·2	1150	110	„ 10·4
600	79	„ 7·5	1200	112	„ 10·7
650	83	„ 7·8	1250	115	„ 10·8
700	86	„ 8·1	1300	117	„ 11·1
750	89	„ 8·4	1350	119	„ 11·3
800	92	„ 8·6	1400	121	„ 11·5
850	94	„ 9·0	1450	123	„ 11·7
900	97	„ 9·2	1500	125	„ 12·0
950	100	„ 9·5	1550	128	„ 12·1

Approximate Rule for Leads of Crossings.

The lead of any crossing is equal to twice the gauge multiplied by the angle of the crossing.

If curves are carefully plotted to an enlarged scale of, say, 10 or 20, or often even 33 feet to an inch, the leads and angles of the crossings can be scaled off with sufficient accuracy for all practical purposes.

The gauge should be generally about 1 inch tighter at the nose of the crossing and points than anywhere else on the lead, in order to prevent the wheels having any lateral play.

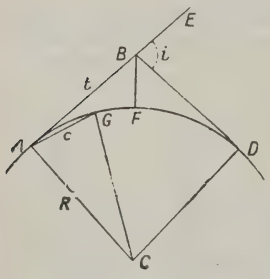


Fig. 37.

Ranging out Curves with a Theodolite.

R = radius of curve = AC. i = angle of inclination of intersection = EBD. t = tangent AB or BD = R tan $\frac{i}{2}$.

Length of curve = AFD = ·0002909Ri.

Tangential (or 'setting off') angle (angle between c and t) in minutes for any chord c

$$= 1718{\cdot}873 \frac{c}{R}.$$

R and c being expressed in feet and in minutes.

The angle ACG is called the 'deflection' angle, and is twice the tangential angle, *i.e.* the tangential angle is one-half the deflection angle.

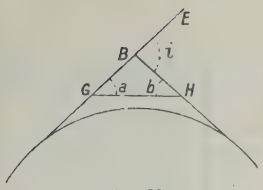


FIG. 38.

To find the Angle of Intersection and Point of Intersection where the latter is inaccessible.

If B is the inaccessible point, then—

Range the line GH.

$$i = a + b,$$

$$GB = \frac{GH \sin b}{\sin i}, \quad HB = \frac{GH \sin a}{\sin i}.$$

TABLE OF TANGENTIAL ANGLES FOR CHORDS OF 1 FOOT AND 100 FEET.

Radius of Curve.	Tangential Angle.		Radius of Curve.	Tangential Angle.		Radius of Curve.	Tangential Angle.	
	Chord 1 ft.	Chord 100 ft.		Chord 1 ft.	Chord 100 ft.		Chord 1 ft.	Chord 100 ft.
Ft.	Min.	Deg. Min.	Ft.	Min.	Deg. Min.	Miles.	Min.	Deg. Min.
500	3·438	5 43·77	3,000	0·573	0 57·30	$\frac{1}{8}$	2·604	4 20·44
600	2·865	4 46·48	3,500	0·491	0 49·11	$\frac{1}{4}$	1·302	2 10·22
700	2·456	4 5·55	4,000	0·430	0 42·97	$\frac{1}{2}$	0·651	1 5·11
800	2·149	3 34·86	4,500	0·381	0 38·20	$\frac{3}{4}$	0·434	0 43·41
900	1·910	3 10·99	5,000	0·344	0 34·38	1	0·326	0 32·55
1,000	1·719	2 51·89	5,500	0·313	0 31·25	$1\frac{1}{4}$	0·260	0 26·04
1,100	1·563	2 36·26	6,000	0·286	0 28·65	$1\frac{1}{2}$	0·217	0 21·70
1,200	1·432	2 23·24	7,000	0·246	0 24·56	$1\frac{3}{4}$	0·187	0 18·70
1,500	1·146	1 54·59	8,000	0·215	0 21·49	2	0·163	0 16·28
2,000	0·859	1 25·94	9,000	0·191	0 19·10	$2\frac{1}{2}$	0·130	0 13·02
2,500	0·688	1 8·76	10,000	0·172	0 17·19	3	0·109	0 10·85

A curve in America is expressed as an $n°$ curve (deflection angle), and is one of such a radius that a chord of 100 ft. subtends an angle of $n°$ at the centre.

Setting out Curves by Offsets.

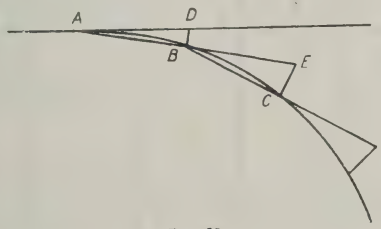


FIG. 39.

First offset BD from the tangent line.

$$DB = \frac{AD^2}{2 \text{ radius}}.$$

Offset from any chord AB produced

$$E = EC = \frac{AE \times EB}{2 \text{ radius}}.$$

Or where AB = BE, then

$$EC = \frac{AB^2}{\text{radius}}.$$

TABLE OF RADII OF CURVES.

(Chords=66 feet.)

V	R	V	R	V	R	V	R	V	R	V	R	V	R
0	Straight	1	99	2	49½ 49·500	3	33	4	24⅜ 24·750	5	19¼ 19·800	6	16½ 16·500
1/16	1,584 1584·01	1/16	93 3/17 93·176	1/16	48	1/16	32 1/49 32·326	1/16	24 2/55 24·369	1/16	19 5/9 19·5	1/16	16 3/17 16·330
1/8	792 792·005	1/8	88	1/8	46 1/17 46·588	1/8	31 1/25 31·680	1/8	24	1/8	19 1/41 19·317	1/8	16 5/16 16·163
3/16	528	3/16	83 7/19 83·368	3/16	45 9/25 45·257	3/16	31 1/17 31·059	3/16	23 3/17 23·641	3/16	19 7/33 19·084	3/16	16
1/4	396	1/4	79⅕ 79·200	1/4	44	1/4	30 6/13 30·461	1/4	23 1/17 23·294	1/4	18⅞ 18·857	1/4	15 3/25 15·840
5/16	316 4/5 316·800	5/16	75 3/7 75·428	5/16	42 1/47 42·811	5/16	29 4/33 29·886	5/16	22 2/33 22·956	5/16	18 5/44 18·635	5/16	15 5/101 15·683
3/8	264	3/8	72	3/8	41 1/19 41·684	3/8	29⅓ 29·333	3/8	22 2/25 22·628	3/8	18 1/43 18·418	3/8	15 9/17 15·529
7/16	226 2/3 226 285	7/16	68 2/3 68·869	7/16	40 5/13 40·615	7/16	28⅛ 28·800	7/16	22 3/11 22·310	7/16	18 6/9 18·207	7/16	15 3/103 15·378
1/2	198	1/2	66	1/2	39 3/5 39·600	1/2	28 2/7 28·286	1/2	22	1/2	18	1/2	15 3/13 15·230
9/16	176	9/16	63 9/25 63·360	9/16	38 2/41 38·634	9/16	27 1/19 27·789	9/16	21 1/13 21·700	9/16	17 1/19 17·798	9/16	15 3/176 15·086
5/8	158 2/3 158·400	5/8	60 1/13 60·923	5/8	37 1/7 37·714	5/8	27 3/99 27·310	5/8	21 1/5 21·405	5/8	17 1/5 17·600	5/8	14 4/13 14·943
11/16	144	11/16	58 2/3 58·666	11/16	36 3/43 36·837	11/16	26 1/99 26·847	11/16	21 3/25 21·120	11/16	17 37/91 17·407	11/16	14 8/107 14·804
3/4	132	3/4	56 4/7 56·571	3/4	36	3/4	26 2/5 26·400	3/4	20 1/8 20·842	3/4	17 2/3 17·217	3/4	14 2/3 14·666
13/16	121 1/3 121·846	13/16	54 1/9 54·621	13/16	35 1/7 35·200	13/16	25 5/9 25·967	13/16	20 4/7 20·571	13/16	17 3/31 17·032	13/16	14 5/109 14·532
7/8	113 1/7 113·143	7/8	52 4/5 52·800	7/8	34 10/23 34·434	7/8	25 1/7 25·548	7/8	20 4/13 20·307	7/8	17 1/47 17·851	7/8	14 2/5 14·400
15/16	105 3/8 105·600	15/16	51 3/31 51·097	15/16	33 3/47 33·702	15/16	25 1/7 25·143	15/16	20 1/79 20·051	15/16	16 4/85 16·674	15/16	14 1/97 14·270

V=versed sine in inches. R=radius of curve in chains.

NOTE.—If 33-foot chords are used for finding curves, the versed sine measured must be multiplied by 4 and then looked up in this table.

TABLE OF RADII OF CURVES (*continued*).

(Chords = 66 feet.)

V	R	V	R	V	R	V	R	V	R	V	R	V	R
7	$14\frac{1}{7}$ 14·143	8	$12\frac{3}{8}$ 12·375	9	11	10	$9\frac{9}{10}$ 9·900	11	9	12	$8\frac{1}{2}$ 8·250	13	$7\frac{8}{13}$ 7·615
$\frac{1}{16}$	$14\frac{2}{113}$ 14·017	$\frac{1}{16}$	$12\frac{1}{43}$ 12·280	$\frac{1}{16}$	$10\frac{334}{1445}$ 10·924	$\frac{1}{16}$	$9\frac{135}{1361}$ 9·838	$\frac{1}{16}$	$8\frac{56}{59}$ 8·949	$\frac{1}{16}$	$8\frac{40}{193}$ 8·207	$\frac{1}{16}$	$7\frac{1}{19}$ 7·579
$\frac{1}{8}$	$13\frac{17}{19}$ 13·894	$\frac{1}{8}$	$12\frac{1}{65}$ 12·181	$\frac{1}{8}$	$10\frac{23}{49}$ 10·850	$\frac{1}{8}$	$9\frac{7}{9}$ 9·7	$\frac{1}{8}$	$8\frac{80}{89}$ 8·899	$\frac{1}{8}$	$8\frac{16}{97}$ 8·165	$\frac{1}{8}$	$7\frac{13}{25}$ 7·543
$\frac{3}{16}$	$13\frac{89}{115}$ 13·774	$\frac{3}{16}$	$12\frac{12}{181}$ 12·091	$\frac{3}{16}$	$10\frac{33}{49}$ 10·775	$\frac{3}{16}$	$9\frac{117}{953}$ 9·718	$\frac{3}{16}$	$8\frac{152}{179}$ 8·849	$\frac{3}{16}$	$8\frac{8}{95}$ 8·123	$\frac{3}{16}$	$7\frac{197}{411}$ 7·507
$\frac{1}{4}$	$13\frac{19}{29}$ 13·655	$\frac{1}{4}$	12	$\frac{1}{4}$	$10\frac{26}{37}$ 10·702	$\frac{1}{4}$	$9\frac{27}{41}$ 9·658	$\frac{1}{4}$	$8\frac{4}{5}$ 8·800	$\frac{1}{4}$	$8\frac{4}{49}$ 8·081	$\frac{1}{4}$	$7\frac{24}{43}$ 7·471
$\frac{5}{16}$	$13\frac{7}{13}$ 13·538	$\frac{5}{16}$	$11\frac{121}{133}$ 11·910	$\frac{5}{16}$	$10\frac{94}{149}$ 10·630	$\frac{5}{16}$	$9\frac{3}{5}$ 9·600	$\frac{5}{16}$	$8\frac{136}{181}$ 8·751	$\frac{5}{16}$	$8\frac{8}{197}$ 8·041	$\frac{5}{16}$	$7\frac{31}{71}$ 7·436
$\frac{3}{8}$	$13\frac{28}{53}$ 13·423	$\frac{3}{8}$	$11\frac{55}{65}$ 11·820	$\frac{3}{8}$	$10\frac{14}{25}$ 10·560	$\frac{3}{8}$	$9\frac{15}{33}$ 9·542	$\frac{3}{8}$	$8\frac{61}{89}$ 8·703	$\frac{3}{8}$	8	$\frac{3}{8}$	$7\frac{43}{107}$ 7·402
$\frac{7}{16}$	$13\frac{37}{119}$ 13·311	$\frac{7}{16}$	$11\frac{11}{15}$ 11·73	$\frac{7}{16}$	$10\frac{74}{151}$ 10·490	$\frac{7}{16}$	$9\frac{51}{107}$ 9·485	$\frac{7}{16}$	$8\frac{20}{61}$ 8·655	$\frac{7}{16}$	$7\frac{193}{197}$ 7·960	$\frac{7}{16}$	$7\frac{76}{211}$ 7·367
$\frac{1}{2}$	$13\frac{1}{5}$ 13·200	$\frac{1}{2}$	$11\frac{11}{17}$ 11·647	$\frac{1}{2}$	$10\frac{8}{19}$ 10·421	$\frac{1}{2}$	$9\frac{3}{7}$ 9·428	$\frac{1}{2}$	$8\frac{11}{23}$ 8·608	$\frac{1}{2}$	$7\frac{23}{24}$ 7·920	$\frac{1}{2}$	$7\frac{1}{3}$ 7·333
$\frac{9}{16}$	$13\frac{1}{11}$ 13·09	$\frac{9}{16}$	$11\frac{77}{137}$ 11·562	$\frac{9}{16}$	$10\frac{17}{37}$ 10·353	$\frac{9}{16}$	$9\frac{63}{169}$ 9·372	$\frac{9}{16}$	$8\frac{104}{185}$ 8·562	$\frac{9}{16}$	$7\frac{59}{67}$ 7·880	$\frac{9}{16}$	$7\frac{95}{217}$ 7·300
$\frac{5}{8}$	$12\frac{80}{81}$ 12·983	$\frac{5}{8}$	$11\frac{11}{23}$ 11·478	$\frac{5}{8}$	$10\frac{2}{7}$ 10·285	$\frac{5}{8}$	$9\frac{27}{55}$ 9·317	$\frac{5}{8}$	$8\frac{18}{31}$ 8·516	$\frac{5}{8}$	$7\frac{85}{101}$ 7·841	$\frac{5}{8}$	$7\frac{29}{109}$ 7·266
$\frac{11}{16}$	$12\frac{38}{41}$ 12·878	$\frac{11}{16}$	$11\frac{55}{139}$ 11·395	$\frac{11}{16}$	$10\frac{34}{155}$ 10·219	$\frac{11}{16}$	$9\frac{5}{19}$ 9·263	$\frac{11}{16}$	$8\frac{8}{17}$ 8·470	$\frac{11}{16}$	$7\frac{263}{263}$ 7·803	$\frac{11}{16}$	$7\frac{1}{7}$ 7·233
$\frac{3}{4}$	$12\frac{24}{31}$ 12·774	$\frac{3}{4}$	$11\frac{11}{35}$ 11·314	$\frac{3}{4}$	$10\frac{2}{13}$ 10·153	$\frac{3}{4}$	$9\frac{9}{43}$ 9·209	$\frac{3}{4}$	$8\frac{20}{47}$ 8·425	$\frac{3}{4}$	$7\frac{13}{17}$ 7·764	$\frac{3}{4}$	$7\frac{1}{5}$ 7·200
$\frac{13}{16}$	$12\frac{84}{132}$ 12·672	$\frac{13}{16}$	$11\frac{11}{47}$ 11·234	$\frac{13}{16}$	$10\frac{14}{137}$ 10·089	$\frac{13}{16}$	$9\frac{27}{173}$ 9·156	$\frac{13}{16}$	$8\frac{8}{21}$ 8·381	$\frac{13}{16}$	$7\frac{148}{203}$ 7·727	$\frac{13}{16}$	$7\frac{37}{211}$ 7·167
$\frac{7}{8}$	$12\frac{4}{7}$ 12·571	$\frac{7}{8}$	$11\frac{11}{71}$ 11·155	$\frac{7}{8}$	$10\frac{2}{79}$ 10·025	$\frac{7}{8}$	$9\frac{3}{29}$ 9·103	$\frac{7}{8}$	$8\frac{32}{95}$ 8·337	$\frac{7}{8}$	$7\frac{77}{103}$ 7·689	$\frac{7}{8}$	$7\frac{4}{57}$ 7·135
$\frac{15}{16}$	$12\frac{60}{121}$ 12·472	$\frac{15}{16}$	$11\frac{1}{13}$ 11·077	$\frac{15}{16}$	$9\frac{51}{83}$ 9·962	$\frac{15}{16}$	$9\frac{9}{173}$ 9·051	$\frac{15}{16}$	$8\frac{56}{191}$ 8·293	$\frac{15}{16}$	$7\frac{15}{23}$ 7·652	$\frac{15}{16}$	$7\frac{233}{2253}$ 7·103

V = versed sine in inches. R = radius of curve in chains.

NOTE—If 33-foot chords are used for finding curves, the versed sine measured must be multiplied by 4 and then looked up in this table.

TABLE OF RADII OF CURVES (*continued*).

(Chords=66 feet.)

V	R	V	R	V	R	V	R	V	R	V	R	V	R
14	$7\frac{1}{14}$ 7·071	15	$6\frac{3}{5}$ 6·600	16	$6\frac{3}{16}$ 6·187	17	$5\frac{14}{17}$ 5·823	18	$5\frac{1}{9}$ 5·500	19	$5\frac{4}{19}$ 5·210	20	$4\frac{19}{20}$ 4·950
$\frac{1}{16}$	$7\frac{1}{28}$ 7·040	$\frac{1}{16}$	$6\frac{138}{341}$ 6·572	$\frac{1}{16}$	$6\frac{42}{257}$ 6·163	$\frac{1}{16}$	$5\frac{73}{91}$ 5·802	$\frac{1}{16}$	$5\frac{139}{286}$ 5·481	$\frac{1}{16}$	$5\frac{59}{305}$ 5·193		
$\frac{1}{8}$	$7\frac{1}{113}$ 7·008	$\frac{1}{8}$	$6\frac{6}{11}$ 6·54	$\frac{1}{8}$	$6\frac{6}{43}$ 6·140	$\frac{1}{8}$	$5\frac{107}{137}$ 5·781	$\frac{1}{8}$	$5\frac{67}{145}$ 5·462	$\frac{1}{8}$	$5\frac{3}{17}$ 5·176		
$\frac{3}{16}$	$6\frac{222}{227}$ 6·978	$\frac{3}{16}$	$6\frac{14}{27}$ 6·518	$\frac{3}{16}$	$6\frac{30}{259}$ 6·116	$\frac{3}{16}$	$5\frac{19}{25}$ 5·760	$\frac{3}{16}$	$5\frac{43}{97}$ 5·443	$\frac{3}{16}$	$5\frac{49}{307}$ 5·159		
$\frac{1}{4}$	$6\frac{18}{19}$ 6·947	$\frac{1}{4}$	$6\frac{30}{61}$ 6·491	$\frac{1}{4}$	$6\frac{6}{65}$ 6·092	$\frac{1}{4}$	$5\frac{17}{23}$ 5·739	$\frac{1}{4}$	$5\frac{31}{73}$ 5·425	$\frac{1}{4}$	$5\frac{1}{7}$ 5·143		
$\frac{5}{16}$	$6\frac{210}{239}$ 6·917	$\frac{5}{16}$	$6\frac{114}{145}$ 6·465	$\frac{5}{16}$	$6\frac{2}{29}$ 6·069	$\frac{5}{16}$	$5\frac{199}{277}$ 5·718	$\frac{5}{16}$	$5\frac{119}{203}$ 5·406	$\frac{5}{16}$	$5\frac{13}{103}$ 5·126		
$\frac{3}{8}$	$6\frac{102}{113}$ 6·887	$\frac{3}{8}$	$6\frac{18}{41}$ 6·439	$\frac{3}{8}$	$6\frac{6}{131}$ 6·045	$\frac{3}{8}$	$5\frac{97}{136}$ 5·697	$\frac{3}{8}$	$5\frac{19}{35}$ 5·387	$\frac{3}{8}$	$5\frac{17}{155}$ 5·109		
$\frac{7}{16}$	$6\frac{6}{7}$ 6·857	$\frac{7}{16}$	$6\frac{102}{247}$ 6·413	$\frac{7}{16}$	$6\frac{6}{263}$ 6·023	$\frac{7}{16}$	$5\frac{21}{31}$ 5·677	$\frac{7}{16}$	$5\frac{109}{286}$ 5·370	$\frac{7}{16}$	$5\frac{39}{311}$ 5·093		
$\frac{1}{2}$	$6\frac{24}{29}$ 6·827	$\frac{1}{2}$	$6\frac{12}{31}$ 6·387	$\frac{1}{2}$	6	$\frac{1}{2}$	$5\frac{23}{35}$ 5·657	$\frac{1}{2}$	$5\frac{13}{37}$ 5·351	$\frac{1}{2}$	$5\frac{1}{13}$ 5·077		
$\frac{9}{16}$	$6\frac{186}{283}$ 6·798	$\frac{9}{16}$	$6\frac{30}{43}$ 6·361	$\frac{9}{16}$	$5\frac{252}{268}$ 5·977	$\frac{9}{16}$	$5\frac{179}{251}$ 5·637	$\frac{9}{16}$	$5\frac{1}{3}$ 5·333	$\frac{9}{16}$	$5\frac{19}{313}$ 5·061		
$\frac{5}{8}$	$6\frac{10}{13}$ 6·769	$\frac{5}{8}$	$6\frac{42}{125}$ 6·336	$\frac{5}{8}$	$5\frac{127}{133}$ 5·955	$\frac{5}{8}$	$5\frac{29}{47}$ 5·617	$\frac{5}{8}$	$5\frac{47}{149}$ 5·315	$\frac{5}{8}$	$5\frac{7}{137}$ 5·044		
$\frac{11}{16}$	$6\frac{178}{265}$ 6·740	$\frac{11}{16}$	$6\frac{78}{251}$ 6·311	$\frac{11}{16}$	$5\frac{83}{89}$ 5·932	$\frac{11}{16}$	$5\frac{169}{283}$ 5·597	$\frac{11}{16}$	$5\frac{89}{299}$ 5·297	$\frac{11}{16}$	$5\frac{1}{35}$ 5·028		
$\frac{3}{4}$	$6\frac{12}{19}$ 6·711	$\frac{3}{4}$	$6\frac{2}{7}$ 6·285	$\frac{3}{4}$	$5\frac{61}{67}$ 5·910	$\frac{3}{4}$	$5\frac{41}{71}$ 5·577	$\frac{3}{4}$	$5\frac{7}{25}$ 5·280	$\frac{3}{4}$	$5\frac{1}{19}$ 5·012		
$\frac{13}{16}$	$6\frac{54}{79}$ 6·683	$\frac{13}{16}$	$6\frac{6}{23}$ 6·261	$\frac{13}{16}$	$5\frac{230}{266}$ 5·888	$\frac{13}{16}$	$5\frac{53}{95}$ 5·558	$\frac{13}{16}$	$5\frac{78}{301}$ 5·262	$\frac{13}{16}$	$4\frac{314}{317}$ 4·997		
$\frac{7}{8}$	$6\frac{78}{119}$ 6·655	$\frac{7}{8}$	$6\frac{30}{127}$ 6·236	$\frac{7}{8}$	$5\frac{13}{15}$ 5·86	$\frac{7}{8}$	$5\frac{7}{13}$ 5·538	$\frac{7}{8}$	$5\frac{37}{151}$ 5·245	$\frac{7}{8}$	$4\frac{52}{53}$ 4·981		
$\frac{15}{16}$	$6\frac{150}{239}$ 6·628	$\frac{15}{16}$	$6\frac{18}{85}$ 6·212	$\frac{15}{16}$	$5\frac{220}{271}$ 5·845	$\frac{15}{16}$	$5\frac{149}{287}$ 5·519	$\frac{15}{16}$	$5\frac{23}{161}$ 5·227	$\frac{15}{16}$	$4\frac{28}{29}$ 4·965		

(*C. E. Murray.*)

V=versed sine in inches. R=radius of curve in chains.

NOTE.—If 33-foot chords are used for finding curves, the versed sine measured must be multiplied by 4 and then looked up in this table.

Transition Curves.*

The superelevation of the outer rail on the curve which is needed at its full height for the whole length of the curve renders it necessary to begin canting the rails some distance back along the straight, called the 'run off,' in order that it may attain its full amount at the tangent point. To obviate the sudden centrifugal and effect also the difficulties due to the 'run off,' *transition curves*, or *spirals*, have been introduced. A true transition curve should determine, by slowly increasing its curvature, that the centrifugal force is applied gradually, and thus prevent any sudden oscillation, and also so that the full superelevation is gained gradually throughout its length.

The superelevation at any particular point in the transition curve should be the correct one for the curvature at that point, on the assumption that it has been designed to suit a certain maximum speed.

The formula $L = \sqrt{R}$, where L is the length and R the radius of the curve, gives very efficient transition curves for circular curves of less than 56·8 chains radius, which corresponds to about the maximum velocity used on any of our railways—84 miles per hour. But for curves of larger radius than 56·8 chains, the formula $L = 0{\cdot}00072428 \frac{V^3}{R}$ should be used, where V is the velocity in miles per hour, and L and R are in chains.

The formula $L = \sqrt{R}$ for determining the length of transition to be used has much to commend it, though it is used very little on British railways as yet, the custom at present being to take a length of 50, 60, or sometimes 66 ft. for every inch of superelevation on the circular curve. It might seem to a permanent-way engineer that the length provided by this formula for transitions to sharp curves was much too small, for instead of increasing in length as the curvature becomes sharper, it diminishes, consequently making the rate of gain of superelevation much quicker than 1 in. in 60 ft. But as the curves are sharper, the maximum speed permitted on them is also less, and thus in surmounting the superelevation at a lower velocity no uncomfortable motion is felt by passengers, as the discomfort of the motion varies as the cube of the velocity, a speed of 50 miles per hour causing nearly 5 times as much oscillation as that due to one of 30 miles per hour.

Again, for speeds of 70 and 80 miles per hour and a radius of between 40 and 60 chains, it may seem that the lengths of transitions are much too long, but here the excessive speeds demand a very gentle gradient on which to gain the full cant if the sensation of having the coach heaved up at one side is to be sufficiently gradual. Thus at speeds of 70 to 80 miles per hour the gain of superelevation should amount to 1 in. in 80 ft. or more if the best running is to be obtained. Therefore a general rule of, say, 1 in. in 60 ft. for the cant on all transition curves is unscientific and not productive of the best results. All the principal railways have now taken to the introduction of transition curves, where possible, at some parts of their systems.

The feeling that circular curves of large radius provide sufficiently good running to render them immune from the transition innovation is one which should be dissipated. As long as large easy curves are used for only *moderately high speeds* the need for a transition curve is not great. But the tendency on some of our principal railways to insert curves only of large radius and without transition curves on their main lines is a mistake, as the approach to any of these curves by trains at *high speed* is necessarily accompanied by irregularities of motion which would not occur had it been made on a transition curve.

Speaking broadly, it may be said that it is only on curves of large radius that there is any great need for a transition approach, as these curves are generally associated with high speeds, whereas the sharper curves are excluded from lines intended for fast traffic. Also, it must be remembered that on a high speed track an irregularity so slight as to escape the eye of the platelayer entirely may cause a very serious jar in the running at that point.

It is advisable that marks of a permanent nature be fixed along the 6-foot way, from which measurements may be taken to the curve in order to refix it in its true position and to the correct elevation during the operation of reballasting or other renewal work. The foreman platelayer should be provided with these and a plan giving the lengths of the offsets. A type of mark which has been adopted with some success is a centre peg made of a fish-plate set in a block of concrete, the fish-plate having a saw cut in it to indicate the centre line of the railway. It is not possible to fix these at the level of the rails, as their projection above the surface of the 6-foot way would be a danger to platelayers, also if a material train was unloaded at such a point a rail dropped on the marks would destroy their accuracy. However, if fixed well down in the pitching and formation so that the top surface of the plate is flush with the top surface of the sleeper or at a definite number of inches, say, 7 below the surface of the rail, they can be always used for re-establishing the rail in its correct position.

* Abstract from paper by Prof. S. W. Perrott, B.A., M.Inst.C.E., read before the Liverpool Engineering Society, March 7, 1917.

Another method, which has been used on the former Great Southern and Western Railway in Ireland, is that of using short lengths of old flat-foot rails with the flange turned towards the line and placed so that the ordinary road gauge measured the distance from the outside of the running rail head to the flange of the rail used as a mark. These pegs were put in at intervals of 25 ft., either on the inside or outside of the curve (the line was single), as the ground was more suitable, and each was set so that its top was at the correct level of the corresponding rail at the point and the correct superelevation was marked on it with a punch.

The custom of making the length of the transition proportional to the number of inches of superelevation was adopted in America, probably for the purpose of obtaining easy hauling for freight trains travelling at a moderate speed. The Canadian Pacific Railway, as also some of the leading railways in this country, adopts 60 ft. for every inch of superelevation on the circular curve, but the former seldom uses more than 5 ins. of superelevation. Also, transition curves are only used for curves of superelevation of 3 ins. and upwards. The tendency in America is to keep down the superelevation as much as possible. This is for the benefit of freight trains, which are considered of primary importance but travel at lower speeds, as the higher superelevation means increased resistance to hauling, caused by the wheels binding against the lower rail. The American Railway Engineering Association has laid down the rule that, in all cases of 2 ins. of superelevation and upwards transition curves should be inserted.

The principal point of danger of a sharp curve is at its entry. If trains could be brought on to curves without any sudden irregularity of motion before settling down to the uniform motion on the curve, very much sharper curves could be adopted for high-speed trains than are used to-day. Indeed the results of experience and experiments seem to show that trains traverse curves of 5 chains radius at 35 miles per hour with perfect ease and without showing any tendency to leave the metals.

There is no hard and fast rule adopted in this country with regard to the number of points set out on the ground to fix each transition curve, the tendency being to fix 8 or even 10 points on a long transition curve and 6 on the shorter ones. A point at every half chain is generally sufficient, but some prefer to fix points about 1 chain apart at the tangent end and half that amount at the circular curve end. This gives good results, as the curvature is so much sharper at that end, and wants its position defining at nearer points.

On the Canadian Pacific Railway it is usual to set out the centre point and two-quarter points only in addition to the extremities, except in the case of very long transition curves. If these are set out by deflection angles from the beginning of the spiral they are in the ratio of 1 : 4 : 9 : 16 as they vary as the square of the length. Also if set out from the end of the spiral the ratio of the four angles is 11 : 20 : 27 : 32. So when any one of these angles is found (generally that to the end point) the others are easily calculated.

Setting out Spiral Points.

A convenient formula for setting out spiral points by deflection angles is the following :—

The deflection angle in minutes to any chord point is given by $\frac{N^2L}{10}$ where N is the number of chords between the point where the instrument is set and the required point, and L is the length of the chord in feet. This formula will not apply if the instrument is set up at the intermediate chord points.

The operation of calculating the deflection angle θ for any point in a transition curve can be easily and rapidly done in the field without the need of any offset tables, as $\theta_n = \frac{L}{6R}$, which can be determined immediately the length of the transition is decided upon. Consequently the operation of setting out the transition spiral by deflection angles from the spiral tangent point is as simple as setting out the ordinary circular curve by deflection angles.

Example of a Transition.

A transition spiral is required to connect a straight with a 25-chain curve.

Taking the length of the transition $= \sqrt{R}$, then L = 5 chains,

$$\therefore \theta_n \text{ for the end of spiral} = \frac{L}{6R} = \frac{5}{150} = \frac{1}{30} \text{ radians}$$

$$\therefore \theta_n = \frac{57^{\circ}3}{30} = 1^{\circ}\ 54'\ 36''.$$

If the points on the spiral are put in every half-chain, there will be ten deflection angles as follows:—

			°	′	″		°	′	″
θ_1	=	$(\frac{1}{10})^2$	(1	54	36)	=	0	1	8·76
θ_2	=	$(\frac{2}{10})^2$	(1	54	36)	=	0	4	35·00
θ_3	=	$(\frac{3}{10})^2$	(1	54	36)	=	0	10	18·84
θ_4	=	$(\frac{4}{10})^2$	(1	54	36)	=	0	18	20·16
θ_5	=	$(\frac{5}{10})^2$	(1	54	36)	=	0	28	39·00
θ_6	=	$(\frac{6}{10})^2$	(1	54	36)	=	0	41	15·36
θ_7	=	$(\frac{7}{10})^2$	(1	54	36)	=	0	56	9·24
θ_8	=	$(\frac{8}{10})^2$	(1	54	36)	=	1	13	20·64
θ_9	=	$(\frac{9}{10})^2$	(1	54	36)	=	1	32	49·56
θ_{10}	=	$(1)^2$	(1	54	36)	=	1	54	36·00

In setting out the same transition curve by deflection angles from its end, it is usual to calculate the deflection angles for the circular curve for the corresponding points, and deduct from each its deflection angle as given in the previous table.

The circular deflection angles are simply found as follows:—

$$\frac{1718'\cdot87}{R} = \frac{1718'\cdot87}{25} = 68'\cdot75,$$

which is the deflection angle in minutes for a whole chain;

$\therefore$ 34′ 22″·5 is the angle for a half-chain.

The following table gives the deflection angles to be used with theodolite at point 10, in fixing points 9, 8, 7, etc.

(1) Deflection angle to point.	(2) Circular deflection angle.			(3) Corresponding values of θ_1, etc.			Difference between (2) and (3) giving deflection angles to spiral.		
	°	′	″	°	′	″	°	′	″
9	0	34	22·5	0	1	8·76	0	33	13·74
8	1	8	45	0	4	35·00	1	4	10·00
7	1	43	7·5	0	10	18·84	1	32	48·86
6	2	17	30	0	18	20·16	1	59	9·84
5	2	51	52·5	0	28	39·00	2	23	13·50
4	3	26	15	0	41	15·36	2	44	59·64
3	4	00	37·5	0	56	9·24	3	4	28·26
2	4	35	00	1	13	20·64	3	21	39·36
1	5	9	22·5	1	32	49·56	3	36	32·94
0	5	43	45	1	54	36·00	3	49	9·00

Vertical Curves for Joining Gradients.

For important railways, rates of change of 0·1 per 100 ft. on summits, and 0·05 in sags, should not be exceeded. On minor railways 0·2 per 100 ft. on summits and 0·1 in sags might be used.

The length of curve should be about 600 ft. at summits, and 800–1,200 ft. in sags.

(*American Railway Engineering Association.*)

Platforms.

The minimum clear width of any platform throughout its length to be 6 ft. At important stations the width to be not less than 12 ft., except for short distances at either end in any case of difficulty. In the case of island platforms, the minimum width for an adequate distance on each side of the centre of its length to be 12 ft. The descent at the ends of platforms to be by ramps and not by steps.

Columns for the support of roofs, and other fixed works, not to be less than 6 ft. clear from the edge of platforms. A general clear headway of not less than 8 ft. to be provided over platforms. The height of platforms above rail level may vary according to traffic and other conditions; as a rule, it should be 3 ft., but in no case less than 2 ft. 9 ins. or more than 3 ft. at permanent stations, without special approval.

The edges of the platforms to overhang not less than 12 ins., and the recess so formed to be kept clear as far as possible of permanent obstruction. (*Ministry of Transport.*)

Stations.

It is desirable to avoid constructing a station on, or providing a siding in connection with, a line which is laid upon a gradient steeper than 1 in 260. (*Ministry of Transport.*)

SECTION XIII

PART II

LOCOMOTIVES AND ROLLING STOCK.

LOCOMOTIVES — STEAM MOTOR CARS — CARRIAGES AND WAGONS—BRAKES.

(Revised by H. Holcroft, M.I.Loco.E., Editor of Journal, Institution of Locomotive Engineers.)

LOCOMOTIVES.

Some Recent Additions to British Locomotives.

For their heaviest passenger traffic, three of the four large British railways use the 4–6–0 type of locomotive with wheels about 6 ft. 9 ins. diameter. In the case of the L.M. & S.R. it has three high-pressure cylinders, while those on the G.W.R. and S.R. have four high-pressure cylinders. In the G.W.R. engine, the inside and outside cranks are set at 180° to one another, but the Southern engine has these cranks arranged at 135° to one another. By this means eight separate impulses are given to the wheels per revolution in place of four, as with cranks at 180°.

The L. & N.E.R. adopts the 4–6–2 or 'Pacific' type for the heaviest work. This engine has three high-pressure cylinders.

For lighter work the L.M. & S.R. continues to build 4–4–0 type engines. These are compound engines with one high-pressure cylinder inside the frames and two low-pressure outside. The L. & N.E.R. is building this type, but it has three high-pressure cylinders. The S.R. is also constructing a number of 4–4–0 type locomotives having three cylinders.

The 2–6–0 type with two outside cylinders and wheels about 5 ft. 6 ins. is being built by the L.M. & S.R. for mixed traffic purposes, and the G.W.R. and S.R. have also extensively adopted the type. The L. & N.E.R. has been constructing this class with three high-pressure cylinders. The S.R. is also building this type with 6 ft. 0 ins. wheels for fast passenger work, some two-cylinder and others three-cylinder. The G.W.R. has introduced a two-cylinder 4–6–0 type engine with 6 ft. 0 ins. diameter wheels.

For the heaviest mixed traffic work the G.W.R. uses a 2–8–0 type with 5 ft. 8 ins. wheels, while the S.R. prefers a 4–6–0 type having 5 ft. 7 ins. diameter wheels.

The heaviest mineral traffic on the L. & N.E.R. is run by three-cylinder engines of the 2–8–2 or 2–8–0 types.

The 0–6–0 type goods engine is still being built by the L.M. & S.R., G.W.R. and L. & N.E.R., and a large number of these engines remain at work on the other lines. Many of the older engines of the type have been modernised by rebuilding and superheating.

Large passenger tanks of the 4–6–4 type have been built by the L.M. & S.R.; also others of the 2–6–4 type.

The 'Garratt' articulated type of locomotive, which is extensively used on colonial railways, has been adopted to a limited extent on the L.M. & S.R. for hauling heavy mineral traffic. An engine of this type, but having two sets of three cylinders, has been put into use on the L. & N.E.R. for banking heavy mineral trains up a steep gradient.

No radical departures from normal locomotive design have been made on British railways as yet, although several experimental locomotives driven by turbines and fitted with condensers have been tried, and new high-pressure designs are in hand. Steady progress in efficiency is nevertheless being made by a process of gradual evolution. The Kitson-Still locomotive—a combination of the Diesel oil engine and a steam engine—has also undergone trials, but has not been put to regular service.

Section of Railway.	Type.	Class.	Service.	Cylinders. No.	Cylinders. Position.	Cylinders. Size.	Wheels. Bogie.	Wheels. Coupled.	Wheels. Trailing.	Wheel Base Coupled.	Wheel Base Total.
LONDON MIDLAND & SCOTTISH RAILWAY.							ft. in.	ft. in.	ft. in.	ft. in.	ft. in.
L.& N.W.	4-6-0	—	Hvy. pass.	2	In.	19 ×26	3 9	6 3	—		
,,	4-6-0	Claughton	,,	4	In. & out.	15½×26	3 3	6 9	—	15 3	29 0
,,	4-4-0	—	Pass.	2	In.	20½×26	3 3	6 9	—	10 0	25 1
Midland	4-4-0	—	Pass.	1 2	In. H.P. Out. L.P.	19×26 21×26	3 6½	6 9½	—	9 6	24 3
,,	0-6-0	—	Goods	2	In.	20 ×26	—	5 3	—	16 6	16 6
,,	0-10-0	—	Banking	4	In. & out.	16½×28	—	4 7½	—	20 11	20 1
L. & Y.	4-6-0	—	Hvy. pass.	4	In. & out.	16½×26	3 0⅜	6 3	—	13 7	25 7
Caledonian	4-6-0	—	Hvy. pass.	3	In. & out.	18½×26	3 6	6 1	—	15 0	28 8
,,	4-6-0	—	,,	2	In.	21 ×26	3 6	6 6	—		
,,	4-4-0	—	Pass.	2	In.	20½×26	3 6	6 6	—		
,,	2-6-0	—	Goods	2	In.	19½×26	3 6	5 0	—		
L.M.S. .	2-6-0	13,000	Mxd. traf.	2	Out.	21 ×26	3 6½	5 6	—	16 6	25 6
,,	4-6-0	RoyalScot	Hvy. pass.	3	In. & out.	18 ×26	3 3½	6 9	—	15 4	27 6
,,	0-8-0	G [3]	Mineral	2	In.	19 × 26	—	4 8½	—	18 3	18 3
LONDON & NORTH EASTERN RAILWAY.											
G.N. .	2-6-0	K[3]	Mxd. traf.	3	In. & out.	18½×26	3 2	5 8	—	16 3	25 2
,, .	2-6-0	K[2]	,,	2	Out.	20 ×26	3 8	5 8	—	16 3	25 2
,, .	2-8-0	O[2]	Mineral	3	In. & out.	18½×26	3 2	4 8	—	18 6	27 2
,, .	2-8-2	—	,,	3	In. & out.	20 ×26	3 2	5 2	3 8	18 6	36 2
,, .	4-6-2	A[1]	Hvy. pass.	3	In. & out.	19 ×26	3 2	6 8	3 8	14 6	35 9
G.C. .	2-8-0	—	Mineral	2	Out.	21 ×26	3 6	4 8	—	17 1	25 5
,, .	4-6-0	—	Mxd. traf.	2	Out.	21 ×26	3 6	5 8	—	15 3	27 6
,, .	4-4-0	D[11]	Pass.	2	In.	20 ×26	3 6	6 9	—	10 0	25 3
,, .	4-6-0	—	Hvy. pass.	4	In. & out.	16 ×26	3 6	6 9	—	15 6	28 1
,, .	4-6-0	—	Mxd. traf.	4	In. & out.	16 ×26	3 6	5 8	—	14 5	28 3
G.E. .	4-6-0	B[12]	Hvy. pass.	2	In.	20 ×28	3 3	6 6	—	14 0	28 6
,, .	4-4-0	D[15]	Pass.	2	In.	19 ×26	3 9	7 0	—	9 0	23 6
,, .	0-6-0	—	Goods	2	In.	20 ×28	—	4 11	—	17 8	17 8
N.E. .	4-4-2	—	Hvy. pass.	3	In. & out.	16½×26	3 7¼	6 10	4 0	15 7	29 6
,, .	4-6-0	—	Mxd. traf.	3	In. & out.	18½×26	3 1	5 8	—	13 6	27 8
,, .	0-8-0	—	Mineral	3	In. & out.	18½×26	—	4 7½	—	18 6	18 6
,, .	4-6-2	A[2]	Hvy. pass.	3	In. & out.	19 ×26	3 1¼	6 8	3 9½	15 0	37 2
N.B. .	4-4-2	—	Hvy. pass.	2	Out.	21 ×28	3 6	6 9	4 3	7 3	27 9½
,, .	0-6-0	—	Goods	2	In.	19½×26	—	5 0	—	16 11	16 11
,, .	4-4-0	—	Pass.	2	In.	20 ×26	3 6	6 6	—	9 6	23 7
L.N.E.	4-4-0	Shire	Pass.	3	In. & out.	17 ×26	3 1½	6 8	—	10 0	24 1
,, .	4-6-0	Sandringham	Hvy. pass.	3	In. & out.	17½×26	3 2	6 8	—	16 3	27 9

LOCOMOTIVES.

Working Pressure.	Grate Area.	Heating Surface.			Weight of Engine.		Tender.		Engine and Tender.		Tractive Effort at 85 per cent. of Boiler Pressure.
		Tubes.	Fire-box.	Super-heater.	On Coupled Wheels.	Total.	Coal.	Water.	Total Weight.	Total Wheel Base.	
lbs. sq. in.	sq. ft.	sq. ft.	sq. ft.	sq. ft.	T. C.	T. C.	Tons.	Galls.	T. C.	′ ″	lbs.
185	25·0	1,908	133	—	46 15	65 15	6	3,000	102 15		19,680
200	30·0	1,088 + 462	183	365	59 8	79 1	7	3,000	119 16	54 0	27,070
175	22·4	1,005 + 380	162	302	38 5	59 17	6	3,000	99 2	47 2¾	20,070
200	28·4	1,170	147	291	39 4	61 14	5½	3,500	104 8	47 1½	22,649 (Simple)
200	28·4				38 10	58 5	7	3,500	104 3		20,330
180	31·5	1,560	158	445	73 13	73 13	4	2,050	106 5	46 3¾	43,312
180	27·0	1,717	176	430	59 6	79 1	6	3,000	119 1	50 3½	29,470
180	28·0	1,700 + 500	170	270	60 0	81 0	5½	4,500	129 0	54 0½	28,000
170	26·0	1,666	148	516		73 0	5	5,000	130 0		21,240
170	21·0	1,220	145	330		59 0	5	4,600	115 0		19,760
170	20·6	1,071	119	267		54 0	4½	3,000	92 0		23,800
180	27·5	1,361	160	307	55 3	66 0	5	3,500	108 4	48 11½	25,800
250	31·2	1,892	189	445	61 12	84 18	5½	3,500	127 12	52 9½	33,150
200	23·6	1,434	150	352	60 15	60 15	4	3,500	101 19	41 10¼	28,250
180	28·0	1,900	182	407	60 0	71 14	6½	3,500	114 16	48 0½	30,000
180	24·0	1,629	152	305	53 18	64 8	6½	3,500	107 10	47 7½	23,400
180	27·5	1,340 + 528	163	430	67 7	75 16	6½	3,500	118 18	52 2¼	36,468
180	41·2	1,880 + 835	215	525	71 10	100 0	7	4,700	151 8	59 8	38,500 / 47,000 with Booster
220	41·2	1,399 + 1,123	215	706	66 0	96 5	8	5,000	152 11	60 10⅜	32,913
180	26·0	1,050 + 591	174	308	67 18	75 4	6	4,000	123 10	51 2½	30,813
180	26·0	1,050 + 591	174	308	53 18	72 18	6	4,000	121 4	50 8½	25,376
180	26·0	1,502	157	304	39 16	61 3	6	4,000	109 9	48 8½	19,385
180	26·0	1,881	163	343	57 12	79 2	6	4,000	127 8	52 9⅜	25,140
180	26·0	1,881	163	343	58 10	79 10	6	4,000	127 16	52 9½	29,950
180	26·5	1,123 + 366	143	286	43 8	62 19	4	3,700	102 4	48 3	21,970
180	21·6	1,561	118	—	34 12	52 4	5	3,450	91 9	43 8	17,100
160	21·6	874 + 283	118	155	47 7	47 7	5	3,500	85 13	38 7	25,820
175	27·0	1,298	185	392	39 14	79 4	5½	4,125	125 16	53 4	19,300
180	27·0	1,400	166	392	58 14	77 14	5½	4,125	124 6	52 6	30,032
180	27·0	1,663	166	392	71 12	71 12	5½	4,125	115 14	44 3⅜	36,910
200	41·0	1,572 + 694	300	810	60 0	101 0	5½	4,125	148 2	62 2¾	29,918
180	28·5	1,136 + 483	185	203	40 0	76 14	7	4,240	122 2	53 2	23,320
180	19·8	783 + 371	149	176	54 14	54 14	7	3,500	95 13	40 0	25,200
165	21·3	871 + 295	140	266	37 16	57 5	7	4,235	103 6	46 8½	18,700
180	26·0	872 + 354	171	246	42 0	66 0	7½	4,200	118 13	48 5¼	21,556
200	27·5	1,048 + 460	168	344	53 15	76 13	4	3,700	116 1	48 4	25,380

BRITISH TENDER

Section of Railway.	Type.	Class.	Service.	Cylinders.			Wheels.			Wheel Base.	
				No.	Position.	Size.	Bogie.	Coupled.	Trailing.	Coupled.	Total.
GREAT WESTERN RAILWAY.							ft. in.	ft. in.	ft. in.	ft. in.	ft. in.
G.W. .	4–6–0	King George V	Hvy. pass.	4	In. & out.	16½×28	3 0	6 6	—	16 3	29 5
,, .	4–6–0	Castle	,,	4	In. & out.	16 ×26	3 2	6 8½	—	14 9	27 3
,, .	4–6–0	Star	,,	4	Do.	15 ×26	3 2	6 8½	—	14 9	27 3
,, .	4–6–0	Saint	,,	2	Out.	18½×30	3 2	6 8½	—	14 9	27 1
,, .	4–6–0	Hall	Hvy. pass.	2	Out.	18½×30	3 0	6 0	—	14 9	27 1
,, .	2–8–0	—	Hvy. goods	2	Out.	18 ×30	3 2	4 7½	—	16 10	25 7
,, .	2–8–0	4,700	Mxd. traf.	2	Out.	19 ×30	3 2	5 8	—	20 0	29 3
,, .	2–6–0	4,331	Mxd. traf.	2	Out.	18½×30	3 2	5 8	—	14 9	23 6
SOUTHERN RAILWAY.											
Western	4–6–0	"Lord Nelson"	Hvy. pass.	4	In. & out.	16½×26	3 1	6 7	—	15 0	29 6
,,	4–6–0	"King Arthur"	,,	2	Out.	20½×28	3 7	6 7	—	14 6	27 6
,,	4–6–0	S^{15}	Mxd. traf.	2	Out.	20½×28	3 7	5 7	—	13 9	26 7½
,,	4–6–0	T^{14}	Pass.	4	In. & out.	15 ×26	3 7	6 7	—	14 4	27 8
,,	0–6–0	700	Goods	2	In.	19 ×26	—	5 1	—	16 6	16 6
Eastern	2–6–0	N	Mxd. traf.	2	Out.	19 ×28	3 1	5 6	—	15 6	24 4
,,	4–4–0	E_1	Pass.	2	In.	19 ×26	3 6	6 6	—	9 6	23 6½
,,	4–4–0	L	,,	2	In.	19½×26	3 7	6 8	—	10 0	24 3½
Central	4–4–2	H_2	,,	2	Out.	21 ×26	3 6	6 7½	3 6	6 10	26 4
,,	2–6–0	K	Mxd. traf.	2	Out.	21 ×26	3 6	5 6	—	15 6	23 9
,,	4–4–0	B^1/x	Pass.	2	In.	20 ×26	—	6 9	—	10 0	24 6

BRITISH TANK

Section of Railway.	Type.	Class.	Service.	Cylinders.			Wheels.			Wheel Base.	
				No.	Position.	Size.	Bogie.	Coupled.	Trailing.	Coupled.	Total.
LONDON MIDLAND & SCOTTISH RAILWAY.							ft. in.	ft. in.	ft. in.	ft. in.	ft. in.
L.&N.W.	4–6–2	—	Pass.	2	In.	20 ×26	3 3	5 8½	3 3	14 0	33 9
Furness	4–6–4	—	Pass.	2	In.	19½×26	3 2	5 8	3 2	13 3	40 9
L.M.S.	2–6–0	Garratt	Mineral	4	Out.	18½×26	3 3½	5 3	—	16 6	79 0
	0–6–2										
L.M.S.	2–6–4	—	Pass.	2	Out.	19 ×26	3 3½	5 9	3 3½	16 6	38 6
LONDON & NORTH EASTERN RAILWAY.											
G.N.	2–8–0	2,395 (Garratt)	Banking	6	In. & out	18½×26	2 8	4 8	2 8	17 10½	79 1
	0–8–2										
,, .	0–6–2	N2	Suburban	2	In.	19 ×26	—	5 8	3 8	16 3	23 9
N.E. .	4–6–2	—	Goods	3	In. & out.	16½×26	3 1½	4 7½	3 9½	14 6	34 3

LOCOMOTIVES—(*continued*).

Working Pressure.	Grate Area.	Heating Surface.			Weight of Engine.		Tender.		Engine and Tender.		Tractive Effort at 85 per cent. of Boiler Pressure.
		Tubes.	Fire-box.	Super-heater.	On Coupled Wheels.	Total.	Coal.	Water.	Total Weight.	Total Wheel Base.	
lbs. sq. in.	sq. ft.	sq. ft.	sq. ft.	sq. ft.	T. C.	T. C.	Tons.	Galls.	T. C.	′ ″	lbs.
250	34·3				67 10	89 0	6	4,000	135 14	57 5½	40,300
		2,514 (Total H.S.)									
225	30·3	1,885	164	263	58 17	79 17	6	4,000	126 11	54 6½	31,625
225	27·0	1,686	155	286	55 8	75 12		3,500	115 12		27,800
225	27·0	1,686	155	286	54 16	72 0		3,500	112 0		24,395
225	27·0	1,686	155	286	56 10	75 0		3,500	115 0	53 4½	27,275
225	27·0	1,686	155	286	61 18	68 6		3,500	108 6		33,490
225	30·3	2,062	169	323	73 8	82 0		3,500	122 0	56 9½	28,764
200	20·5	1,350	129	216	52 12	62 0		3,500	102 0		25,670
220	33·0	1,282 513	194	376	61 19	83 10	5	5,000	140 4	60 9	33,490
200	30·0	1,252 464	162	337	60 0	80 19	5	5,000	138 10	58 0	25,320
200	28·0	1,252 464	162	337	60 0	80 14	5	5,000	137 2	57 1½	29,860
175	31·5	873 407	158	269	53 3	75 15	5	5,800	136 3	55 7	22,000
180	20·3	523 266	117	167	46 14	46 14	—	3,500	83 8	39 1	23,500
200	25·0	1,391	135	203	50 18	59 8	5	3,500	98 13	47 9½	25,942
180	24·0	827 322	127	228	33 10	52 10	4½	3,450	91 10	45 2⅜	18,411
180	22·5	914 338	155	235	38 0	58 10	5	3,500	99 0	46 2¾	18,907
170	30·9	1,348 547	136	460	37 10	68 10	4	3,500	107 10	50 3	20,931
170	24·8	1,155	139	279	55 4	63 10	4	3,940	105 10	47 11	25,075
180	24·8	1,155	139	279	37 18	58 1	4	3,600	97 6	46 11	19,695

LOCOMOTIVES.

Working Pressure.	Grate Area.	Heating Surface.			Weight of Engine.		Tanks.		Tractive Effort at 85 per cent. of Boiler Pressure.
		Tubes.	Fire-box.	Super-heater.	On Coupled Wheels.	Total.	Coal.	Water.	
lbs. sq. in.	sq. ft.	sq. ft.	sq. ft.	sq. ft.	T. C.	T. C.	Tons.	Galls.	lbs.
175	23·9	638 309	138	248	44 0	77 0	3	1,700	22,580
170	26·0	1,850	153	—	54 18	93 5	4	2,200	21,006
190	44·5	1,954	183	500	116 10	148 15	7	4,500	45,620
200	25·0	1,083	137	266	51 8½	86 5	3½	2,000	23,125
180	56·4	2,757	237		143 18	178 0	7	5,000	72,940
170	19·0	880	118	207	52 5	70 6	4	2,000	19,950
180	23·0	1,508	140	—		87 8	5	2,300	29,420

BRITISH TANK

Section of Railway.	Type.	Class.	Service.	Cylinders.			Wheels.			Wheel Base.	
				No.	Position.	Size.	Bogie.	Coupled.	Trailing.	Coupled.	Total.
						in. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.
GREAT WESTERN RAILWAY.											
G.W.	2-8-0	—	Mineral	2	Out.	18½ × 30	3 2	4 7½	—		
,,	2-6-2	—	Goods	2	Out.	18½ × 30	3 2	5 8	3 8	14 9	31 9
,,	4-4-2	—	Pass.	2	Out.	18 × 30	3 2	6 8½	3 8	8 6	32 0
,,	0-6-2	5600	Mineral	2	In.	18 × 26	—	4 7½	3 8	15 3	21 9
SOUTHERN RAILWAY.											
Western	4-6-2	H[14]	Goods	2	Out.	21 × 28	3 7	5 7	3 7	15 0	36 6
,,	4-8-0	G[16]	Shunting	2	Out.	22 × 28	3 7	4 7½	—	18 0	32 0
Eastern	0-4-4	H	Suburban	2	In.	18 × 26	—	5 6	3 7	7 6	21 10
,,	0-6-4	J	Suburban	2	In.	19½ × 26	—	5 6	3 7	16 6	27 9
,,	0-8-0	Z	Shunting	3	In. & out.	16 × 28	—	4 8	—	17 6	17 6
Central	4-6-2	J	Pass.	2	Out.	21 × 26	3 6	6 7½	4 0	14 6	35 3
,,	4-6-4	L	Hvy. Pass.	2	Out.	22 × 28	3 6	6 9	3 6	14 9	40 10

AMERICAN TENDER

Railroad.	Type.	Class.	Service.	Cylinders.			Wheels.			Wheel Base.	
				No.	Position.	Size.	Bogie.	Coupled.	Trailing.	Coupled.	Total.
						in. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.
Pennsylvania	2-10-0	115	Freight	2	Out.	30½ × 32	2 9	5 2	—		
,,	2-8-2	—	,,	2	Out.	27 × 30	2 9	5 2	2 9		
,,	4-6-2	—	Pass.	2	Out.	27 × 28	3 0	6 8	3 0		
Lehigh Valley	4-8-2	5,000	Freight	3	In. & out.	25 × 28		5 9		18 0	41 2
New York Central	4-6-4	5,200	Pass.	2	Out.	25 × 28	3 0	6 7	3 0 & 4 3	14 0	40 4
Southn. Pacific	4-6-2	—	Pass.	2	Out.	25 × 30	3 0	6 1½	4 3½	13 0	35 6
,,	2-10-2	—	Freight	2	Out.	29½ × 32	2 9	5 3½	4 3½	22 10	42 4
Virginia	2-10-0 0-10-2	800	Freight	4	Out.	Comp'd. 30 × 32 / 48 × 32		4 8	—		
Northern Pacific	4-8-4	2,600	Pass.	2	Out.	28 × 30	3 0	6 3	3 0 & 3 9½	20 3	47 2
Union Pacific	4-12-2	9,000	Freight	3	Out. / In.	27 × 32 / 27 × 31	2 6	5 7	3 9	30 8	52 4
Delaware & Hudson	2-8-0	John B. Jervis	Freight	2	Out.	Comp'd 22½ × 30 / 38½ × 30	3 0	4 9	—	18 0	29 0

• Maximum cut-off limited to 50 per cent.

LOCOMOTIVES—(*continued*).

Working Pressure.	Grate Area.	Heating Surface.			Weight of Engine.		Tanks.		Tractive Effort at 85 per cent. of Boiler Pressure.
		Tubes.	Fire-box.	Super-heater.	On Coupled Wheels.	Total.	Coal.	Water.	
lbs. sq. in.	sq. ft.	sq. ft	sq. ft.	sq. ft.	T. C.	T. C.	Tons.	Galls.	lbs.
200	20·5	1,350	129	216	70 0	78 0		1,800	31,450
200	20·5	1,350	129	216	56 4	78 16		2,000	25,670
195	20·3	1,009	122		37 0	75 0		2,000	20,000
200	20·3	1,145	122	82	56 0	69 7	3½	900	25,800
180	27·0	910 357	139	231	59 0	96 8	3½	2,000	28,200
180	27·0	910 357	139	231	72 18	95 2	3½	2,000	34,000
160	16·6	1,002	102	—	33 12	54 8	2½	1,200	17,357
160	17·6	593 294	112	234	51 18	70 14	3½	2,000	20,378
180	18·4	1,173	106	—	71 12	71 12	3	1,500	29,844
170	25·0	997 464	125	357	54 10	89 0	3	2,000	20,931
170	26·6	1,535	152	383	56 10	98 5	3½	2,700	24,225

LOCOMOTIVES.

Working Pressure.	Grate Area.	Heating Surface.			Weight of Engine.		Tender.		Engine and Tender.		Tractive Effort at 85 per cent. of Boiler Pressure.
		Tubes.	Fire-box.	Super-heater.	On Coupled Wheels.	Total.	Coal.	Water.	Total Weight.	Total Wheel Base.	
lbs. sq. in.	sq. ft.	sq. ft.	sq. ft.	sq. ft.	T. C.	T. C.	Tons.	U.S.A. Galls.	T. C.		lbs.
250	70·0	2,731 1,313	290	1,418	152 14	166 0	17½	9,000	247 4		90,000*
205	70·0				117 0	151 10					57,850
205	70·0				89 7	140 0					41,845
200	84·3	2,832 1,505	392	1,249	110 0	164 14			254 8		64,700
225	81·5	4,203	288	1,965	81 5	153 2	18	10,000	246 9	76 1½	42,400
							Oil. Galls.				
200	70·4	3,069	282	836	80 8	133 0	4,000	12,000	230 6	75 9½	43,660
200	82·5	4,722	399	1,208	132 14	172 6	4,000	12,000	271 8	82 7½	75,150
							Tons				
215	108·2	5,592 2,511	532	2,120	275 10	305 8	12	13,000	401 2		176,600 147,200
210	115	4,115	485	1,992	116 1	190 4	24	15,000	333 2	90 0	75,500
220	108·2	4,459 803	529	2,560	158 10	221 0	21	15,000	350 0		96,650
400	82	788 1,116	1217	700	131 14	150 5	20	16,000	285 5	74 11½	85,000 (Simple)

T.E. taken at 75 per cent. of boiler pressure.

Locomotive Types.

The classification of locomotives is by wheel arrangement. Abbreviations are used the Anglo-American system denoting in their order leading carrying wheels, coupled wheels, and trailing carrying wheels by number of wheels to each group. Where there are no carrying wheels at either end a cipher is added. The letter T denotes a tank engine. Articulated engines are denoted by the wheel arrangement of their separate parts. The Continental system merely states number of axles coupled and total axles of engine, and gives the proportion of adhesive weight without specifying the actual arrangement of wheels. In addition to identity by wheel grouping, the various types receive code names in America. Example:

Diagram.	Code Name.	Anglo-American Notation.	Continental Notation.
	Atlantic	4–4–2	2–5
	Mogul	2–6–0	3–4

In France it is usual to classify a locomotive by groups of axles in their order, instead of by wheels: thus a 4–4–2 engine is known there as type 2–2–1; and a 2–6–0 engine is expressed as type 1–3–0.

Another system, in use in Germany and Switzerland, is similar to the French, but groups of coupled axles are represented by letters—A for one driving axle, B for two, C for three, etc.: so that a 4–4–2 engine becomes type 2–B–1. A 2–6–0 engine is denoted by 1–C only, a cipher not being necessary to indicate absence of carrying wheels with this sytem.

Types of Locomotives most generally used on British Railways.

Tender Engines.

Type.	Principal Use.	Type.	Principal Use.
0–6–0	Ordinary goods traffic	0–4–2	Branch passenger traffic
0–8–0	Mineral traffic	4–4–0	Fast passenger trains
2–8–0	Heavy goods traffic	4–4–2	Main line express
2–6–0	Mixed traffic	4–6–0	Heavy express
2–4–0	Branch passenger traffic	4–6–2	Exceptionally heavy express

Tank Engines.

Type.	Principal Use.	Type.	Principal Use.
0–4–0	Private yard shunting	0–4–2 2–4–0 4–4–0	Light passenger trains
0–6–0	Ordinary shunting		
0–8–0 0–8–2 0–8–4 2–8–4 4–8–0 2–8–0	Heavy shunting, banking and mineral trains	0–4–4 0–6–4 2–4–2	Suburban passenger trains
		2–6–4 4–4–2 4–4–4 4–6–2 4–6–4	Main line passenger trains
2–6–2 4–6–0 0–6–2	Main line goods trains		

The particular uses of the various types of tank engines are not so well defined as with tender engines. Several of the engines specified for goods traffic are often used on passenger trains, and *vice-versa*. Generally speaking, the total adhesion type (i.e. engines without carrying wheels) are mostly used for shunting and local work. Carrying wheels are added as the water and coal capacity are enlarged to permit of greater radius of action. The tank engines with two or more pairs of carrying wheels are capable of travelling considerable distances without replenishing fuel and water.

Articulated Locomotives.

On railways having a limited axle loading and tracks abounding in sharp curves and often of narrow gauge, a powerful locomotive can be provided by a division of the driving wheels into two independent groups, one or both being separated from the main framing of the engine and attached only by flexible connections. In the Fairlie and Kitson-Meyer designs the boiler, tanks, and coal-bunker are carried on a main frame supported by two steam-driven bogies, the centres of articulation being near the centre of the wheel base of each bogie. The 'Mallet' type, first used in France, has been developed to an enormous size on American railways as a tender engine. High-pressure cylinders drive the axles on the main frame which carries the boiler, while a steam bogie driven by low-pressure cylinders supports the smoke-box end of the boiler. The 'Garratt' type is now extensively used on British colonial railways, in South America, and elsewhere. The boiler alone is carried on the main frame, the water-tanks and coal-bunker being supported by two steam-driven bogies. By this arrangement the centres of articulation are brought nearer together, the pivots being towards the inner ends of the bogies. Very powerful locomotives of this type have been constructed for the 3 ft. 6 ins. gauge in South Africa and New Zealand, both for passenger and freight services.

The leading particulars of a recent 'Garratt' engine for the South African Railways are as follows :—Type 4–8–2 + 2–8–4, class GL, freight service, cylinders 4, size 22 ins. by 26 ins., bogie wheels 2 ft. 4½ ins. diameter, coupled wheels 4 ft. 0 ins., trailing wheels 2 ft. 9 ins. Coupled wheel base 13 ft. 3 ins., total wheel base of each unit 27 ft. 8 ins., total wheel base 83 ft. 7 ins. Centres of articulation 41 ft. 6 ins., working pressure 200 lbs. per sq. in. Grate area 74·5 sq. ft. Heating surface of tubes 3,036 sq. ft., of firebox 340 sq. ft., superheater surface 809 sq. ft. Adhesive weight 147 tons 8 cwt. Total weight 214 tons 2 cwt. Coal 12 tons, water in tanks 7,000 galls. Tractive effort at 75 per cent. of boiler pressure 78,650 lbs. The track on which these engines run is of 3 ft. 6 ins. gauge laid with 80-lb. rails, for which the maximum axle load is 18½ tons. The maximum gradient is 1 in 30, and the curves 300 ft. radius.

General Design of Locomotives.

The principal limitations governing locomotive design are :—

(1) *Loading gauge*, giving extreme dimensions as to heights and widths.
(2) *Rail gauge*, affecting stability, etc.
(3) *The track*, which limits the maximum axle load.
(4) *Bridges*, the strength of which restricts the weight per ft. run over total wheel base.
(5) *Curves*, which limit the length of engine and determine rigid wheel base.

Other factors to be taken into consideration are :—

(1) *Loads*, as to weight and type of rolling stock.
(2) *Gradients*, both ruling and maximum.
(3) *Resistance of curves.*
(4) *Speed.*
(5) *Prevailing atmospheric conditions.*

These are inter-related, load becoming a maximum for both directions of travel when there are no gradients or curves and the speed is low. An increase in speed or the presence of gradients, curves, and side winds reduces loads.

Some further points to be looked into when designing new engines are :—

(1) *Length and strength of turn-tables* and clearances round them.
(2) *Engine sheds*, as to clearances, length of pits, etc.
(3) *Coal stages*, height of platform, method of coaling, etc.
(4) *Position of water columns* at stations and in yards in relation to signals and points.
(5) *Distance between available coal and water supplies.*
(6) *Capacity of workshops*, in regard to load on cranes, length of pits and traversers, necessity for special tools, etc.

The bulk of the work on British railways is done by tender engines of the 0–6–0 type for goods and 4–4–0 type for passenger trains. Many of these engines are of fairly recent construction. Some of the older engines have been brought up to date by being rebuilt with larger boilers fitted with superheaters and with new cylinders provided with piston valves. The goods engines usually have cylinders 19 ins. × 26 ins. and wheels 5 ft. 0 ins. diameter ; the passenger engines 20 ins. × 26 ins. cylinders and 6 ft. 6 ins. wheels. Boiler pressure is about 180 lbs. per sq. in., The single-wheel engines once so numerous are now almost extinct on British railways, as are also compound engines, except on the Midland section of the L.M. & S.R.

For heavy passenger work the 4–4–2 and 4–6–0 types are resorted to. The former has no more adhesion than the 4–4–0 type, but increased boiler power permits heavy trains to be hauled at high speeds. The 4–6–0 type can also take heavy trains at high speed or, alternatively, exceptionally heavy trains at moderate speed on account of its greater adhesive weight. The 4–6–2 type hauls exceptionally heavy trains at high speeds. For mineral traffic and long goods

trains an eight-wheel coupled engine is used. The 2–6–0 type with wheels about 5 ft. 6 ins. is being used to an increased extent as a mixed traffic engine, and a similar engine of the 2–8–0 type has been introduced for heavier trains of this description. They are especially suitable for fast goods, excursion trains, etc. In the large passenger and goods engines outside cylinders predominate. The advantages claimed for the outside position are shorter wheel base, long connecting rods, no crank axle, direct drive through rods on all wheels, horizontal cylinders. The distance between centres of cylinders is about 6 ft. 8 ins. and cannot be reduced very appreciably on account of the necessity for finding sufficient bearing surface for coupling and connecting rods. Thus, the diameter of cylinders is limited by the remaining space within the load gauge, which varies on the different railways. Since the load gauge is at its narrowest near rail level, advantage

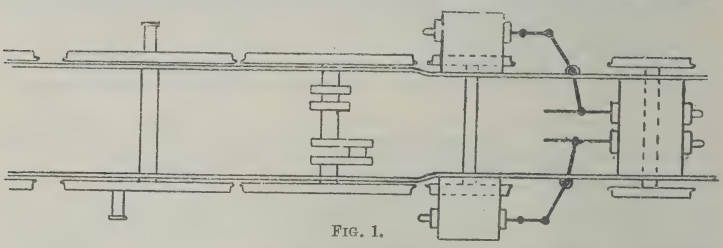


FIG. 1.

can sometimes be gained by inclining outside cylinders so that they are brought up into a wider part of the gauge.

The demand for larger cylinder capacity can be met by increased diameter or stroke of cylinders, or both. Another way is to multiply the number of cylinders, hence three- and four-cylinder high-pressure engines. Multi-cylinders are also being adopted for mechanical reasons, as they give better balancing and turning movement, less disturbance in running, subdivision of driving forces, increased bearing areas and other advantages. Multi-cylinder engines are necessarily more costly and complicated than two-cylinder. It is therefore desirable to reduce their working parts to a minimum. This can be done by providing two sets of valve gear only and operating the additional valve or valves through rocking levers attached to the valve gears. The arrangement adopted on the Great Western Railway four-cylinder engines is shown in fig. 1. Two sets of Walschaert gear drive the inside valves directly and the outside valves by means of rocking levers. The levers are not straight, the outer arms being set backwards, resulting in an improvement of the valve events. The London Midland and Scottish engines have Walschaert valve gear outside the frames and work the inside valves through straight rocking levers. All cylinders drive on one axle.

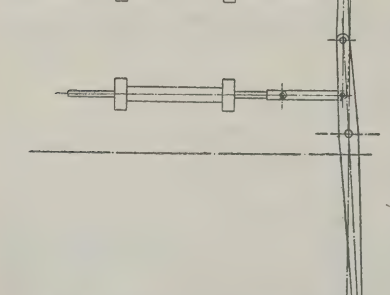


FIG. 2.

Three-cylinder engines can be arranged in a similar manner. Fig. 2 shows in plan the arrangement as used on the Great Northern section of the L. & N.E.R. for three-cylinder engines of several types. A long lever at the front of the engine is pivoted to a cross stretcher on the frame, the ratio of the arms being as 2 : 1. The longer end is attached to the tail spindle of the right-hand valve. To the shorter end another lever with equal arms is pivoted at its centre. The outer end of this is attached to the spindle of the left-hand valve, and the inner end to the spindle of the inside valve, all valves being in line and set at the same inclination; but in order that the crosshead may clear the leading coupled axle, the inside cylinder is inclined 1 in 8. The valves of the outside cylinders are, of course, directly operated by two sets of Walschaert valve gear. A variation of the same principle is in use in Germany and Spain, in which one rocking shaft having equal arms is mounted on the shorter arm of another rocking shaft having arms in ratio 2 : 1. Modifications of the arrangement are in use on the L.M. & S.R. (Caledonian section) and Southern Railway (Eastern section).

Four-cylinder high-pressure locomotives are in use on the Continent. In some of these there is a single piston valve for each pair of cylinders, one of which has crossed ports. In others a double-headed piston valve is used. British practice favours separate valves for each cylinder.

Four cylinders are more used in fast passenger engines. Since the forces and masses in each pair of motions are opposed, the balancing is good and the disturbances to smooth running reduced. The mechanical efficiency is high, but otherwise there is little advantage over two cylinders. Owing to the more regular turning movement, three cylinders are applied to engines in which the most is to be made of the available adhesion, such as the ' Atlantic ' and ' Pacific ' types and heavy shunting tanks. In the latter instance there is the further advantage of six exhaust beats per revolution in place of four, with less disturbance to the fire when working slowly.

Tank engines are used for fast passenger trains on the Tilbury section of the L.M. & S.R., Metropolitan Railway (Aylesbury extension), and to some extent on the Southern Railway.

Boiler pressures tend to increase ; 200 lbs. per sq. in. is now common, and recent engines carry 250 lbs. pressure.

' Poppet ' or double-beat lift valves, which have been used to a limited extent on the Continent for some years, are undergoing trials under British railway conditions. On the L. & N.E.R. several passenger and goods engines have had the cylinders with slide valves removed and new cylinders with ' Lentz ' valves substituted, the existing link motion valve gear being retained to operate the rocking cam shafts. Six of a new batch of engines are being fitted with Walschaert valve gear, and two with the valves operated by rotary sliding cams driven by gearing connected to the driving axle. A somewhat similar arrangement to the latter, the ' Beardmore-Caprotti ' valve gear, is in use on the L.M. & S.R. (see p. 596).

Some recent American locomotives are of enormous size. For the heaviest freight traffic Mallet articulated compounds are in use on the Virginian R.R. weighing, with tender, nearly 400 tons, a pair of cylinders being added to the tender to make use of its adhesive weight, the total tractive effort being about 80 tons. The latest engines on the New York Central and Northern Pacific have a trailing bogie below the firebox, being of the 4–6–4 and 4–8–4 types. On the Southern R.R. engines of the ' Mikado ' or 2–8–2 type are provided with steam tenders made from obsolete engines of the 2–8–0 type, in which the boiler is removed and a water tank substituted. The axle loads on the Pennsylvania R.R. reach 30 tons. In order to reduce dynamic loading of wheels the reciprocating parts are of alloy steel. With small wheels the revolving masses must likewise be reduced, since it is not always possible to make the balance weights in the wheel large enough. Outside valve gears such as the Walschaert, Baker-Pilliod, or Southern are general, and piston valves are of large diameter with inside admission. The two-cylinder engine with superheater has superseded all compounds except the Mallet type. Three-cylinder engines are now being extensively used. The valve gear for the middle cylinder is similar to that shown by fig. 2. Tank engines are not common, even switching engines having tenders.

A recent development is the ' booster.' This is a small steam engine geared to the carrying wheels, and which can be thrown in or out of action as required. It is used to assist the main engine when starting a heavy train from rest or in climbing grades. The device has also been applied to the tender wheels.

In the largest American locomotives a bogie is required under the firebox, in addition to a pony truck or bogie at the leading end, so that a large percentage of the weight of the engine is not available for adhesion. The addition of a booster partly overcomes this drawback, and enables the engine to start away with a heavier train than it could do in the normal manner.

Another feature peculiar to American practice is the ' limited cut-off ' locomotive, in which the maximum cut-off is fixed at about 50 per cent. in full gear. Under the operating conditions prevailing engines are at times worked ' all-out ' at the full gear cut-off of 80 to 90 per cent. of the stroke for long periods, which is most wasteful of steam. By enlarging the cylinder diameter and reducing the cut-off, and consequently the mean effective pressure, an equally powerful but more economical engine can be constructed. Positive starting is ensured by means of notched ports in the valve liners which allow a small supply of steam to fill the cylinders should the engine be on a ' dead centre.'

New locomotives for passenger service are mainly of the 4–6–2 and 4–8–2 types. For freight traffic the 2–8–2 type is mostly favoured, but the 2–10–2, 2–10–0 and 2–8–0 types are also being built to some extent. Three cylinder engines of the 4–10–2 and 4–12–2 types have been introduced for mixed traffic purposes on heavily graded roads.

An experimental engine, the ' Horatio Allen,' has been put into service on the Delaware & Hudson R.R. This is a cross compound of the 2–8–0 type, and is notable for having a water-tube firebox. The boiler pressure is no less than 350 lbs. per sq. in. It has been followed by another engine, the ' John B. Jervis,' having a pressure of 400 lbs. A more recent one has a pressure of 500 lbs. per sq. in. The Baldwin Locomotive Works has built a three-cylinder compound engine of the 4–10–2 type, having a water-tube firebox and pressure of 350 lbs. per sq. in.

In Russia a few locomotives driven by Diesel engines are in use. Some have electric transmission and others gear transmission. In the U.S.A. there are also a few Diesel-electric locos.

A high-pressure locomotive is undergoing trials in Switzerland. The boiler is of special water-tube construction, and the working pressure is 850 lbs. per sq. in. The engine is of the three-cylinder uniflow type, and is geared to a jack-shaft which transmits the drive to the wheels through outside connecting-rods. A German high-pressure engine has a boiler in three sections. A water-tube firebox generates steam at a very high pressure, and the heat from this is transferred to a second section of the boiler by the steam circulating through heating coils and condensing. Steam at a pressure of 850 lbs. per sq. in. generated in the second section passes to the high-pressure cylinder, and is exhausted to the steam-chests of the low-pressure cylinders. The third section of the boiler is tubular and generates steam at 200 lbs. per sq. in., which is used to supplement the steam to the low-pressure cylinders. This three-cylinder engine is otherwise of conventional design, except for the small diameter of the high-pressure cylinder. Experimental locomotives of this type are also being constructed in England (L.M.S.R.), France, U.S.A., and Canada. The L.N.E.R. has built an engine with a water-tube boiler having a pressure of 450 lbs. per sq. in. This is of the 4-6-2 type and is a four-cylinder compound with 10 ins. × 26 ins. high-pressure and 20 ins. × 26 ins. low-pressure cylinders. It weights 103 tons 12 cwt. without tender.

TRAIN RESISTANCE.

There are numerous train resistance formulæ, and they differ considerably. It is only safe to apply them to similar rolling stock to that from which the experimental results were obtained, as conditions vary. For instance, American formulæ are unsuitable for British or Continental rolling stock, and *vice versa*. In estimating the resistance of a train, it should be observed that the results can only be taken as approximate, due to various small factors which may influence them, such as condition of track, play between flange and rail; also empty rolling stock has a relatively higher resistance than when fully or partially loaded, etc.

The *starting* resistance of a train is very variable, and usually ranges from 12 to 18 lbs. per ton. It is dependent on how long the vehicles have been standing and on the temperature of the atmosphere, this variation being due to a deficiency in the lubrication and to the viscosity of the lubricant. In warm weather oil boxes offer greater resistance than grease boxes, as the oil is more quickly squeezed out of the bearing while standing, while the reverse occurs in cold weather when the grease is solidified and so causes a high starting resistance. Once movement of the vehicle takes place, efficient lubrication is restarted and the resistance falls to about 5 lbs. per ton at slow speed.

Resistance of Air to Train.

(1) The area of resistance presented by the first carriage is its front superficies; that of the following carriages, about 10 square feet, but depending upon the distance between them.

Q = resistance of air, in lbs. A = surface of resistance, in sq. ft. = about 70 ft. + 10(n − 1) for railway waggons, where n = number of waggons in train. (54 ft. according to M. Vuillemin, &c., 50 ft. Unwin.) v = velocity, in feet per second. e = 1·43 for a thin surface = 1·17 for a cube, = 1·07 for a train of 5 waggons, = 1·04 for a train of 25 waggons.

$$Q = \cdot 0011896 e A v^2 \text{ (Pambour).}$$

(2) If P = pressure on the front ends in pounds per square foot; V = velocity in miles per hour :—

$$P = \cdot 00265\ V^2.$$

Assuming the superficial area of the front end to be about 75 sq. ft., the following table shows the total pressure and the power required to overcome that pressure at various speeds :—

Speed in miles per hour.	Total pressure, lbs.	H.P.
20	79·5	4·24
40	318·0	33·92
60	715·5	115·28
80	1272·0	271·36
100	1987·5	530·0

It will be seen that at speeds below 40 miles an hour the effect of head end resistance is practically negligible; but at speeds of 60 miles an hour or more the power required to overcome head end resistance is by far the most important element in train resistance. As the number of cars in a train is increased, the relative effect of head end resistance is decreased.

An engineer of the firm which furnishes electric locomotives for the Simplon Tunnel says that at a speed of 50 miles an hour in the tunnel the air-resistance absorbs 400 horse-power, or 40 per cent. of the capacity of the locomotive, against 95 horse-power in the open air.

Resistance to Traction on a Level Railroad.

The most complete record of experiments on Train Resistance is probably that of Mr. J. A. F. Aspinall, given in a paper by him read before the Institution of Civil Engineers, November 26, 1901, and published in vol. 147, May 1902, of the Proceedings of the Institution. The conditions vary so much with the character of the train and other factors that no definite formulæ covering all cases and separating the various resistances making up the total can be given. The following are the principal formulæ given in the Proceedings referred to, abstracted from a list of 55, dating from that of Clark in 1855.

Formulæ for Train Resistance.

R = tractive resistance in lbs. per ton (2,240 lbs.).

V = velocity in miles per hour.

W = weight of engine and tender in tons (2,240 lbs.).

w = ,, carriages ,, ,,

A = area of front of train in square feet.

L = length of train in feet.

No.	Authority.	Value of R.	Conditions.	Reference.
1	Clark	$8+\frac{V^2}{171}$	Whole train	D. K. Clark, 'Railway Machinery,' 1855, p. 297.
2	,,	$6+\frac{V^2}{240}$	Carriages only	
3	,,	$12+\frac{V^2}{114}$	Whole train, Narrow gauge	Z. Colburn, 'Locomotive Engineering,' 1871, p. 291.
4	,,	$\left\{6+\frac{V^2}{240}+\left\{\left(\frac{2+V^2}{600}\right)\times\left(\frac{W+w}{W}\right)\right\}\right\}$	Engine and train	
9	Wellington	$4{\cdot}48+\frac{V^2}{116}$	20 loaded box cars (Up to 30 miles per hour)	Trans. Am. Soc. C.E., vol. viii. p. 52.
10	,,	$6{\cdot}72+\frac{V^2}{94{\cdot}7}$	40 empty box cars (Up to 30 miles per hour)	
11	,,	$\frac{V^2}{100{\cdot}8}$	20 loaded flat cars (Up to 30 miles per hour)	
12	,,	$\frac{V^2}{72{\cdot}3}$	40 empty flat cars (Up to 30 miles per hour)	
13	,,	$4{\cdot}48+\frac{V^2}{162}$	Engine and train	A. M. Wellington, 'Economic Theory of the Location of Railways,' 1887, p. 497.
14	,,	$4{\cdot}48+0{\cdot}0056\,V^2+0{\cdot}46\,\frac{V^2}{w}$	Carriages only	*Street Railway Journal*, 1899, p. 144.
22	Baldwin Co.	$3{\cdot}36+\frac{0{\cdot}56\,V}{3}$	—	*Engineering*, vol. lxviii. p. 1850.
23	,,	$1{\cdot}68+0{\cdot}224\,V$	47 to 77 miles per hour	*Railway Gazette*, 1897, p. 198.

FORMULÆ FOR TRAIN RESISTANCE—*continued*.

No.	Authority.	Value of R.	Conditions.	Reference.
24	Blood	$4{\cdot}48+0{\cdot}168V+0{\cdot}336\frac{V^{1\cdot8}}{W+w}$	—	*Street Railway Journal*, 1899, p. 145.
32	Clayton	$6{\cdot}5+0{\cdot}16V$	Oil-lubrication	—
33	,,	$6{\cdot}5+0{\cdot}005V^2$	Grease-lubrication	
35	Deeley	$3+\frac{V^2}{290}$	—	*The Engineer*, vol. lxxxv. p. 276.
36	Wolff	$3+\frac{V^2}{250}$	—	Students' Paper, Inst. C.E., 1898.
37	,,	$3\left(\frac{V+12}{V+3}\right)+\frac{V^2}{300}$	—	*Mechanical Engineer*, 1899, p. 256.
38	Eastern Railway of France	$0{\cdot}305V+5{\cdot}24-\frac{w}{89{\cdot}2}$	—	*Railway Engineer*, vol. ii. p. 344.
39	,,	$0{\cdot}305V+4{\cdot}1$	Train only $w=102{\cdot}6$ tons	
40	,,	$3{\cdot}96+0{\cdot}18V$	$w=185$ tons 7·5 to 20 miles per hour	*Street Railway Journal*, 1899, p. 143.
41	,,	$4{\cdot}032+0{\cdot}289V+0{\cdot}28\frac{V^2}{w}$	20 to 31 miles per hour	
42	,,	$4{\cdot}032+0{\cdot}289V+0{\cdot}194\frac{V^2}{w}$	31 to 40 miles per hour	
43	,,	$4{\cdot}032+0{\cdot}505V+0{\cdot}129\frac{V^2}{w}$	43 to 50 miles per hour	
48	Barbier	$\frac{3{\cdot}58+1{\cdot}658V}{\times\left(\frac{1{\cdot}609V+50}{1000}\right)}$	Four-wheeled coaches	*Le Génie Civil*, April, 1898, and Bulletin of the International Railway Congress, vol. xii. p. 1275.
49	,,	$\frac{3{\cdot}58+1{\cdot}644V}{\times\left(\frac{1{\cdot}609V+10}{1000}\right)}$	Bogie coaches	
50	,,	$\frac{8{\cdot}51+3{\cdot}24V}{\times\left(\frac{1{\cdot}609V+30}{1000}\right)}$	Locomotive and tender	
51	Aspinall	$2{\cdot}5+\frac{V^{\frac{5}{3}}}{58{\cdot}7}$	5 bogie-coaches and dynamometer-car	
52	,,	$2{\cdot}5+\frac{V^{\frac{5}{3}}}{65{\cdot}82}$	10 bogie-coaches and dynamometer-car	
53	,,	$2{\cdot}5+\frac{V^{\frac{5}{3}}}{73{\cdot}05}$	15 bogie-coaches and dynamometer-car	
54	,,	$2{\cdot}5+\frac{V^{\frac{5}{3}}}{80}$	20 bogie-coaches and dynamometer-car	
55	,,	$2{\cdot}5+\frac{V^{\frac{5}{3}}}{50{\cdot}8+0{\cdot}0278L}$	General formula, bogie-coaches, oil axle boxes	

Oil axle boxes

From tests carried out on the L. & N.E.R. with four-wheeled goods wagons of 6 tons tare and 10 tons capacity, and with bogie wagons of 17 tons tare and 50 tons capacity, the following results were obtained :—

Speed—m.p.h.	10	20	30	40	50
Resistance in lbs. per ton—10-ton wagons	4½	4½	5½	8	12½
Resistance in lbs. per ton—50-ton wagons	3½	3½	3¾	4¼	6¼

All wagons were fitted with oil axle boxes, and were fully loaded.

Resistance of Curves.

Resistance due to curvature of track varies inversely as the radius, and directly as the rail-gauge and length of rigid wheel base. It also depends to some extent on the state of the surfaces of wheels and rails. The practice of watering the rails or lubricating the flanges of the wheels lessens the resistance, whilst the occasional necessity of sanding the rails for the locomotive greatly increases it. On main lines the curve resistance is usually very small, and no allowance is made for it. In special cases, where the curves are sharper, such as in sidings, station yards, or at junctions, it is sometimes necessary to form some estimate of the resistance. The American Master Mechanics' Association gives the resistance per ton (of 2,240 lbs.) as 0·77 lbs. for each degree of curvature of track in the case of cars (with bogies), and double this amount—1·54 lbs. —for locomotives.

In British practice, curvature is expressed by the radius in feet or chains. As 1° of curvature corresponds to a radius of 5,730 feet, the radius of any curve in degrees is given in feet by $\frac{5,730}{x^\circ}$ or in chains by $\frac{87}{x^\circ}$.

Resistance in lbs. per ton $= \frac{4,412}{\text{radius in ft.}}$; or $\frac{67}{\text{radius in chains}}$ for cars.

" " $= \frac{8,824}{\text{radius in ft.}}$; or $\frac{134}{\text{radius in chains}}$ for locomotives.

To find the equivalent up-gradient, 1 in n, that offers same resistance as curve :—

$$n = \text{radius in ft.} \times 0{\cdot}5.$$
$$= \text{radius in chains} \times 33.$$
$$= \frac{2,871}{\text{curvature in degrees.}}$$

Resistance due to Gradients on Railways.

The force, F, required to move 1 ton up a slope of 1 in n, neglecting friction and resistance of air $= \frac{1 \text{ ton}}{n} = \frac{2,240 \text{ lbs.}}{n}$, or resistance in lbs. due to gravity on any incline $= \frac{2,240}{\text{rate of gradient}}$ per ton of train.

On a gradient of 1 in 300 the total resistance becomes about twice that on a level at a speed of 30 m.p.h., and therefore only half the utmost load can be carried, or the speed must be diminished. Gradients above 1 in 330 are considered first class ; 1 in 330 to 1 in 150, fair working gradients.

In American and Continental practice, gradient is given as rise per cent. or per thousand, so that grade of 1 in $n = \frac{100}{n}$ per cent., or $\frac{1000}{n}$ per thousand. Thus 1 in 300 = 0·33 per cent. or 3·3 per mille.

Rise in feet per mile $= \frac{5280}{n}$, or 17·6 ft. for 1 in 300.

Effect of Gradients on Speed of Passenger Trains.

The work done by the locomotive in a given time is proportional to total resistance × speed. For speeds over 25 m.p.h. on the level, the resistance of an average engine and train may be fairly represented in lbs. per ton by the simple formula $R = 5 + \cdot 003v^2$. The work done on the level is therefore proportional to $v(5 + \cdot 003v^2)$. On a gradient of 1 in n this becomes $v\left\{(5 + \cdot 003v^2) + \frac{2240}{n}\right\}$. If the engine is capable of maintaining a speed of 60 m.p.h. on the level, and the horse-power output is constant, the effect of the gradients on the speed is as follows :—

Inclination.		Level.	1 in 2,500	1 in 1,000	1 in 750	1 in 500	1 in 250	1 in 150	1 in 120	1 in 100	1 in 90	1 in 80	1 in 70
Speed	Miles per hour	60	59	57	55	53	47	40	36	33	31	28	26

TABLE SHOWING THE SPEED OF AN ENGINE WHEN THE TIME OF PERFORMING A QUARTER, HALF, OR ONE MILE IS GIVEN.

Speed per Hour.	Time of Performing			Speed per Hour.	Time of Performing			Speed per Hour.	Time of Performing		
	¼ Mile.	½ Mile.	1 Mile.		¼ Mile.	½ Mile.	1 Mile.		¼ Mile.	½ Mile.	1 Mile.
Miles.	M. S.	M. S.	M. S.	Miles.	M. S.	M. S.	M. S.	Miles.	M. S.	M. S.	M. S.
5	3 0	6 0	12 0	34	26	53	1 46	63	14	28	57
6	2 30	5 0	10 0	35	25	51	1 43	64	14	28	56
7	2 8	4 17	8 34	36	25	50	1 40	65	13	27	55
8	1 52	3 45	7 30	37	24	48	1 37	66	13	27	54
9	1 40	3 20	6 40	38	23	47	1 34	67	13	27	54
10	1 30	3 0	6 0	39	23	46	1 32	68	13	26	53
11	1 21	2 43	5 27	40	22	45	1 30	69	13	26	52
12	1 15	2 30	5 0	41	21	43	1 27	70	12	25	51
13	1 9	2 18	4 37	42	21	42	1 25	71	12	25	51
14	1 4	2 8	4 17	43	20	41	1 23	72	12	25	50
15	1 0	2 0	4 0	44	20	40	1 21	73	12	25	49
16	56	1 52	3 45	45	20	40	1 20	74	12	24	49
17	53	1 46	3 31	46	19	39	1 18	75	12	24	48
18	50	1 40	3 20	47	19	38	1 16	76	12	24	47
19	47	1 34	3 9	48	18	37	1 15	77	11	23	47
20	45	1 30	3 0	49	18	36	1 13	78	11	23	46
21	42	1 25	2 51	50	18	36	1 12	79	11	23	46
22	40	1 21	2 43	51	17	35	1 10	80	11	22	45
23	39	1 18	2 36	52	17	34	1 9	81	11	22	44
24	37	1 15	2 30	53	17	34	1 7	82	11	22	44
25	36	1 12	2 24	54	16	33	1 6	83	11	21	43
26	34	1 9	2 18	55	16	32	1 5	84	10	21	43
27	33	1 6	2 13	56	16	32	1 4	85	10	21	42
28	32	1 4	2 8	57	15	31	1 3	86	10	21	42
29	31	1 2	2 4	58	15	31	1 2	87	10	20	41
30	30	1 0	2 0	59	15	30	1 1	88	10	20	41
31	29	58	1 56	60	15	30	1 0	89	10	20	40
32	28	56	1 52	61	14	29	59	90	10	20	40
33	27	54	1 49	62	14	29	58				

Resistance of Locomotives.

The resistance of a locomotive is a very uncertain quantity, and there is no formula which can take account of all the variables. There are, however, several formulæ which give results approximating to the average resistances in ordinary working.

The resistance may be divided into three components:—

(1) Internal resistance of engine machinery, including driving wheels.
(2) Resistance of locomotive as a vehicle, including head-on air resistance.
(3) Resistance of tender as a vehicle.

According to Professor W. F. Goss, the internal resistance is fairly constant for any given cut-off for a wide range of speed, but that the later the cut-off, and consequently more even the turning moment, the less the friction.

As a vehicle the locomotive not only has the usual rolling resistance, journal friction, etc., but it has to encounter the atmospheric pressure on the surfaces perpendicular to the direction of travel, and it shelters to some extent the train behind.

The reaction of the steam pressure on the cylinder covers, applied alternatively to either side of the engine, causes it to move in a sinuous path and sets up further flange friction.

The tender has a resistance very similar to any other vehicle of the train but it is modified a little by the movement of the engine to which it is closely coupled.

Professor Goss gives as total resistance of engine and tender,

$$WR = 3{\cdot}8\ \frac{d^2 l}{D} + w(2 + \tfrac{1}{4}V) + 0{\cdot}11V^2,$$

where

d = diameter of piston in inches; l = stroke in inches; D = diameter of drivers in inches; V = speed in m.p.h.; w = total weight of engine and tender (W) less weight on drivers, in tons (of 2,000 lbs.).

This formula does not take into account the increase in friction with the number of pairs of coupled wheels.

Mr. Lawford H. Fry gives separate formulæ for engines with 4, 6, and 8 coupled wheels, which in lbs. per ton (2,240 lbs.) of engine and tender can be expressed as:—

$$R = 8{\cdot}5 + 0{\cdot}09V + 0{\cdot}004V^2 \text{ for 4 wheels coupled.}$$
$$= 10{\cdot}0 + 0{\cdot}12V + 0{\cdot}004V^2 \text{ for 6 wheels coupled.}$$
$$= 13{\cdot}4 + 0{\cdot}48V + 0{\cdot}004V^2 \text{ for 8 wheels coupled.}$$

The equations are derived from the mean of the results obtained by numerous European experimenters. (*The Engineer.*)

About 22 lbs. per ton at 50 m.p.h. is the usual resistance of a British 4-4-0 type engine; and at 60 m.p.h. some 4 h.p. per ton is expended in moving the engine and tender.

In order to see how much power the locomotive absorbed as compared with the train, a certain number of experiments were tried on the former Lancashire and Yorkshire Railway, and it was found that the ten-wheeled engine No. 1392 absorbed 34 per cent. of the total horse-power. Mr. W. M. Smith (the Institution of Mechanical Engineers) gave the result of his experiments as about 36 per cent. of the total horse-power; but the actual figure depends on the load behind the engine, the percentage being higher with light loads.

Train Acceleration.

The resistances to be overcome in moving a train are frictional and gravitational. Any draw-bar pull in excess of them is available for accelerating the train; or

$$F = T - W(R_1 + R_2 \pm R_3),$$

where

F = accelerating force in lbs.; T = tractive effort in lbs.; W = weight of train in tons; R_1, R_2 and R_3 = opposition to traction in lbs. per ton due to train resistance, curves, and gradients respectively. A negative result indicates retardation.

For the actual acceleration or retardation produced, allowing 10 per cent. for rotary acceleration of wheels and axles,

$$a = \frac{F \times 32}{1{\cdot}10W \times 2240} = \frac{F}{7iW} \text{ ft./sec.}^2$$

In American practice about 5 per cent. is allowed for rotary acceleration, as the car wheels are smaller in diameter and also bear a lesser ratio to the weight of car.

To produce a speed of V miles per hour in distance S feet,

$$F = \frac{82V^2W}{S} \text{ lbs.; or } F = \frac{82(V^2 - v^2)W}{S} \text{ lbs., } v \text{ being initial speed.}$$

To produce a speed of V miles per hour in t seconds,

$$F = \frac{112VW}{t} \text{ lbs.; or } F = \frac{112(V - v)W}{t} \text{ lbs.}$$

Power, Tractive Force, and Adhesion of Locomotives.

The pressure exerted on the piston is to the force exerted in the traction of the train inversely as the velocities of the piston and train, i.e., as twice the stroke of the piston to the circumference of the wheel.

Piston Speed.—Ordinary piston speed is about 900 feet per minute.

If d = diameter of piston, in inches; a = area of piston, in sq. inches; l = length of stroke, in inches; p = indicated pressure per square inch, in lbs.; D = diameter of driving-wheel, in inches; V = velocity, in miles per hour; n = number of revolutions per minute = $336\frac{V}{D}$; then

$$\text{Tractive force at rails} = \frac{d^2lp}{D} \text{ lbs.}$$

$$\text{Indicated horse-power} = \frac{d^2plV}{375D} = \frac{4anpl}{33{,}000} \times 1\tfrac{1}{3}.$$

The calculated tractive (or starting) effort

$$T = \frac{cd^2Pl}{D} \text{ lbs.}$$

where

P = boiler pressure, and *c* is a coefficient varying from 0·8 to 0·9, but usually taken at 0·85 both in British and American practice.

The above relates to the ordinary two-cylinder high-pressure engines. Multiply results by 1·5 for three-cylinder, and by 2 for four-cylinder locomotives. (For compound locomotives, see p. 592.)

The horse-power given out at the draw-bar is less than the indicated horse-power by the amount expended in moving the engine and tender. This is approximately 400 h.p. in modern locomotives running at 60 m.p.h., representing about 35 per cent. of the maximum indicated horse-power. (See Resistance of Locomotives, p. 586.)

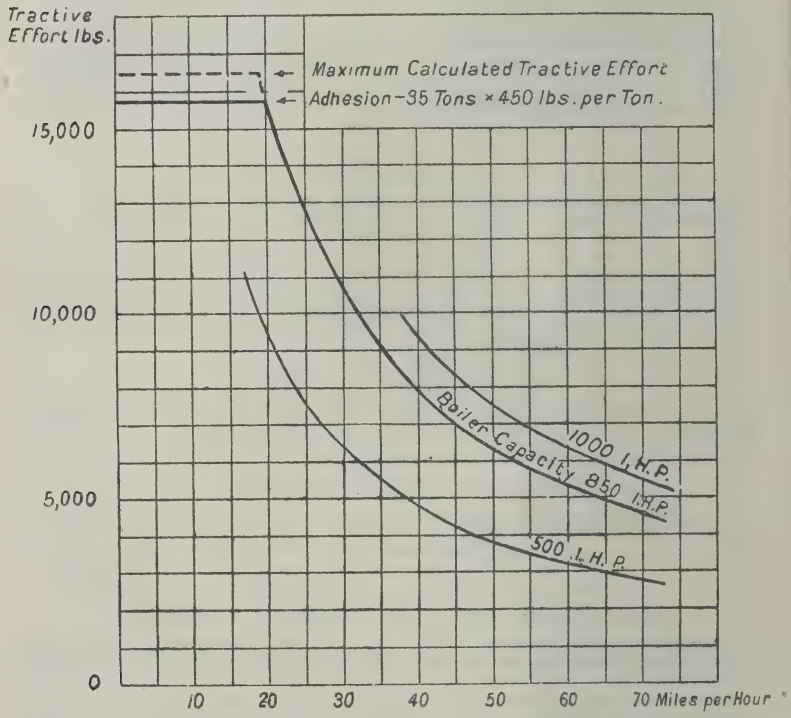


FIG. 3.

Draw-bar horse-power = 5·97 × pull (in tons) × speed (m.p.h.) = 0·0267 × pull (in lbs.) × speed (m.p.h.).

ADHESION.

The proportion of the weight on the driving wheels giving *adhesion* and available as tractive force varies considerably, from about 200 lbs. per ton in frosty and greasy weather to 600 lbs. in dry weather. It must, of course, exceed the tractive power, and is generally taken at about 350 lbs. per ton, this being quite feasible, as all locomotives are fitted with a sanding arrangement.

In average British practice, the factor of adhesion, that is, the ratio between adhesive weight and calculated tractive effort, is between 4 and 5 for four and eight-wheel coupled engines, and between 5 and 6 for six wheels coupled. In tank engines the factor varies with the amount of water and coal present. Multi-cylinder engines may have a lower factor on account of their more even turning moment.

The permissible axle load in relation to the maximum tractive effort determines the number of axles to be coupled, thus :—

$$\text{Max. T.E.} = \text{adhesion, } i.e. = \frac{\text{Adhesive weight.}}{\text{Factor of Adhesion.}}$$

If n = number of axles to be coupled, W = permissible weight on each axle, and K = factor of adhesion

$$\text{Max. T.E.} = \frac{n\text{W}}{\text{K}} \text{ or } n = \frac{\text{T.E.} \times \text{K}}{\text{W}}.$$

The relation between horse-power, tractive effort, and adhesion, is shown by the characteristic curve of a locomotive in fig. 3, where tractive effort is plotted against speed. Horse-power appears on the diagram as a hyperbolic curve, since for any given horse-power pull × speed = a constant. In the example given the maximum calculated tractive effort is 16,500 lbs. The actual pull is limited by adhesion to 15,750 lbs., adhesive weight being 35 tons and the adhesion 450 lbs. per ton. The boiler capacity is taken as 850 i.h.p. and is plotted as a hyperbolic curve. The intersection of the curve with the maximum available tractive effort gives the lowest speed at which the boiler is fully used. Until this point is reached horse-power is limited by cylinder capacity or adhesion, as the case may be. Beyond it, the horse-power is approximately constant and the pull falls off as the speed rises. A deduction must be made for engine and tender resistance to obtain actual draw-bar pull.

Relation of Power to Weight of Locomotives.

Ordinary locomotives are capable of developing from 9 to 12 horse-power for each ton of their weight (inclusive of tender).

Weight Restriction in Locomotives.

On English railways the heaviest load is usually 20 tons, but in the latest 3- and 4-cylinder engines a load of about 22 tons has been permitted.

The axle loads carried with the goods engines are about 15 tons in Germany and Italy, 17 tons in England, Belgium, and France, and 23 to 27 tons in the United States.

The following is a formula given by F. J. E. Spring for locomotive driving-wheels :—

Safe load = 200 × diameter of wheel in inches × width of rail head on which wheel bears in inches.

In addition to restriction of individual loads on wheels, the total weight of a locomotive is limited by the strength of permanent-way structures. In designing a new locomotive, or rebuilding an existing one, it is essential to study the resulting bending moment imposed on bridges. This varies with the total length of engine, wheel base and spacing of wheels, and the weights on individual pairs of wheels. A short engine with the wheels close together may stress a bridge more than a heavier engine in which the weight is spaced over a longer base. The locomotive engineer, however, is not concerned in the actual stresses produced, but his problem is to keep within the boundaries of a limiting bending moment curve. The permanent-way engineer sets out the strength of the weakest bridges of various spans, and from them deduces a curve showing the maximum equivalent uniformly distributed load that the different spans can sustain. The curve is either set out giving the total load on the spans, or the tons per foot run of span. In fig. 5 the upper curve shows a limit determined by the permanent-way engineer in lbs. per ft. run. No locomotive may run that sets up on any bridge a greater bending moment than this. If the excess is small, a slight modification of the wheel spacing or distribution of loads may rectify matters. As the locomotive constitutes a rolling load, any direct calculation is very tedious. The loads must be advanced across the span a foot or so at a time, and the bending moment ascertained for each step. It is obvious that maximum bending will occur under one of the loads, but it is not always certain which load it is. The maximum also changes from under one wheel to another at a certain point. The most satisfactory way is to first of all solve the problem graphically : if care is taken the results are very reliable. Once a solution is obtained the bending moment can be exactly calculated, and this acts as a check on the drawing and reading of scales. For the loading of the span it is assumed that there are several locomotives of the type under consideration, coupled together. As an example of the method, a 4–4–0 type engine is taken, the loads and spacing of wheels being shown on the line diagram at the foot of fig. 4. The first step is to set up a vertical line and mark it off to a convenient scale, say 1 in. = 10 ft., to represent the wheel loads to scale. Thus, *c–d* is the weight on the leading bogie wheels, *d–e* on the next pair, *e–f* that on the first pair of coupled wheels, and *f–g* for the trailing wheels. The loads on tender wheels are represented by *g–h*, *h–j*,

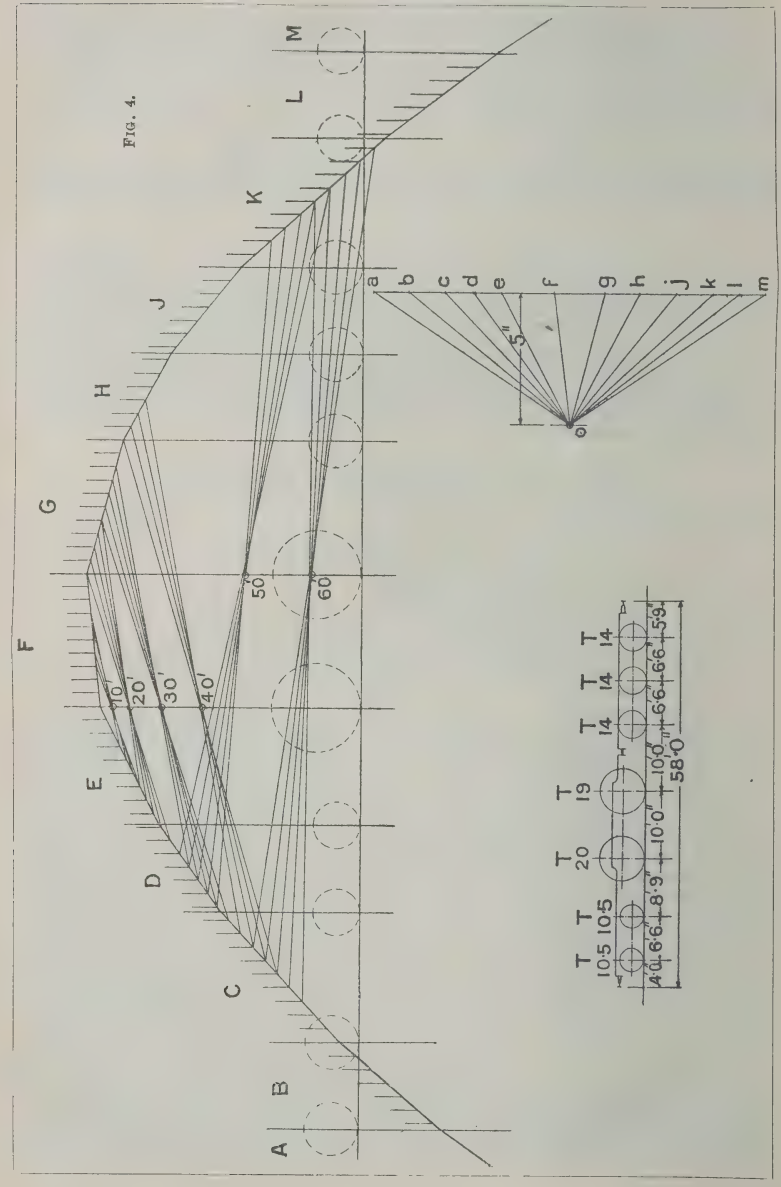

A
B
C
D
E
F
G
H
J
K
L
M
10'
20'
30'
40'
50'
60'
a
b
c
d
e
f
g
h
j
k
l
m
5"
T T T T T T T
10·5 10·5 20 19 14 14 14
4·0 6·6 8·9 10·0" 10·0" 6·6 6·6 5·9
58·0

Fig. 4.

j–k. The lengths *a–b*, *b–c*, are tender wheels of the preceding engine; while *k–l*, *l–m*, are for bogie wheels of the following engine. If the span is a long one more loads must be added in their proper order. Having done this a pole O is taken, approximately in line with the centre of gravity of the loads, and at a distance of, say, 5 ins. from the vertical line. The various points on the line are then joined to the pole. The next step is to draw a horizontal base line and mark on it the wheel base of the engine to a scale of, say, $\frac{1}{2}$ in. = 1 ft. Verticals are projected through the points, and the spaces are lettered A, B, C, etc., to correspond with the letters *a*, *b*, *c*, etc., on the vertical load line.

Commencing at any convenient place, a line is drawn through space A parallel to the polar line *o–a*. This is followed by line through space B parallel to *o–b*, and so on until complete. For convenience a number of vertical lines are spaced at intervals representing each foot of the span. In order to ascertain the bending moment for a 10-ft. span, the intersection of a vertical with the curve is joined to another ten spaces away and scaling 10 ft. when measured horizontally. The next and succeeding points are treated in the same way, and it will soon be evident where the greatest bending moment is enclosed. This is equivalent to moving the bridge span along under the engine. Having ascertained the greatest bending for a 10-ft. span, that for a 20-ft.

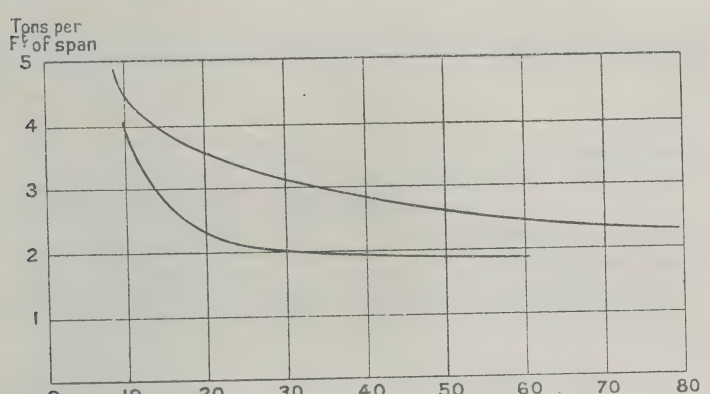


FIG. 5.

is obtained in the same way by joining verticals 20 ft. apart. The 30, 40, 50 ft., etc., spans are treated in like manner. It will be observed that the maximum B.M. moves from under the driving to the trailing wheels between 40- and 50-ft. spans. The maximum B.M. can be scaled by measuring vertically, and the results tabulated. If 1 in. = 10 tons of weight, and 1 in. = 2 ft. of span, and the polar distance is 5 ins., the scale of bending will be 1 in. = 10 × 2 × 5 = 100 ton-ft. The final step is to obtain the equivalent uniformly distributed load by equating each result to it—i.e., B.M. = $\frac{WL}{8}$, so that W = $\frac{8 \times \text{B.M.}}{L}$, where W = total distributed load on span. For load per ft. run of span, B.M. = $\frac{wl^2}{8}$, and $w = \frac{8 \times \text{B.M.}}{l^2}$.

The results obtained can be plotted on the bridge curve supplied by the permanent-way engineer (as the lower curve in fig. 5), and unless all the plotted points fall below the upper curve, the design of the engine will have to be modified in some way, either by reduction of weight, or possibly by an alteration to the wheel base, or the addition of another pair of wheels.

When the position of maximum bending and the loads producing it are known, calculation is a simple matter, and it acts as a check on the graphical working. The position of load system giving maximum bending is such that the centre of span bisects the distance between point of maximum bending (which occurs under one of the loads) and centre of gravity of loads on span.

Compound Locomotives.

The general claim for compound locomotives, by most authorities, is a saving of about 20 per cent. in steam. Expansive working, and consequently exhausting at a low temperature, is the main object.

Many of the larger locomotives recently constructed on the Continent are four-cylinder compounds, while the two-cylinder cross-compounds are returning to favour in the Argentine. A number of three-cylinder compounds have been built in England for the L.M. & S.R., and several experimental engines have been put into service in the U.S.A.

Ratios of the Areas of the High-pressure Cylinders to the Low-pressure.

Wordsell, Von Borries, and others, 1 to 2–2½; Webb, 1 to 2·3; Vauclain and America, 1 to 3. In modern compounds ratio is 1 to 2·4 on L.M.S. Railway (Midland Section), 1 to 2·5 Mallet compounds in U.S.A., and about 1 to 2·2 on Continent.

Tractive Force.

Von Borries, who recommends a ratio of 1 : 2·2 or 2·3, gives tractive power of two-cylinder compounds as,

$$\text{T. F.} = \frac{psd^2}{4\,D},$$

where,

p = boiler pressure; s = stroke in inches; D = diameter of wheel in inches; d = diameter of low-pressure piston.

For four-cylinder compounds, Baldwin gives

$$\text{T.F.} = \frac{2\,psc^2}{3D} + \frac{psd^2}{4D},$$

being diameter of high-pressure piston.

Another formula is given as

$$\text{T.F.} = \frac{0{\cdot}8\,psd^2}{(r+1)D} \text{ for two-cylinder compounds} = \frac{1{\cdot}6\,psd^2}{(r+1)D} \text{ for four-cylinder compounds,}$$

r being ratio of low to high-pressure piston area.

Cylinders of Simple and Compound Locomotives of Equivalent Power.

For proportioning the diameters of the cylinders of compound locomotives when designing them to generate a power equivalent to that of a given size of simple cylinder and of the same piston stroke, the average ratio of area of the simple cylinder and the high-pressure cylinder area of compound locomotives is 1 to 1·25 for the 2-cylinder and 1·10 for the 4-cylinder. In the case of the 4-cylinder engines the area is that of the sum of the two high-pressure cylinders.

(Railroad Gazette.)

Parts of Locomotives.

CYLINDERS.

The size of cylinders is governed by the maximum tractive effort required for starting the train, boiler pressure, diameter of driving wheels and adhesive weight available. $d^2l = \frac{\text{T.E.} \times \text{D}}{0{\cdot}85\text{P}}$

where d = diameter of piston in inches, l = length of stroke in inches, P = boiler pressure, D = diameter of driving wheels in inches, and T.E. = tractive effort in lbs.

Diameter of cylinder in inches = $\cdot75\ \sqrt{\text{I.H.P.}}$, where I.H.P. = maximum H.P. that engine will be required to develop continuously for any length of time.

Stroke and diameter of driving wheels should be so proportioned that the piston speed is about 900 ft. per minute, as this is found to be the most economical speed.

Thickness of barrel in inches = $\cdot02d + \cdot75$, where d = diameter in inches.

Thickness of metal of steam passages may be $\cdot7\ t$ and of the flanges $1{\cdot}25\ t$, where t = thickness of barrel in inches.

The studs for the cylinder cover should be calculated for a stress of 4,000 lbs. per square inch to allow a large margin of safety in case of priming.

For Flat Slide Valves.—Area of steam port $= d^2 + 15$, in the most modern practice. Length of ports should not be less than $\cdot 85\ d$. Area of exhaust port should be two and a half times that of the steam port.

Clearance between cylinder covers and piston is usually made from $\frac{1}{4}$ in. to $\frac{3}{8}$ in. The *clearance volume*, including the passages, varies from 6 to 8 per cent. of the cylinder volume. If the clearance is too small the compression line at high speeds rises above the steam chest pressure, and the engine does not run well.

Cylinders for Highly-superheated Steam.—Owing to the very high temperatures, it is necessary that the steam chest walls should be separate from the walls of the cylinders. There is a very great variation in the temperatures of superheated steam, on the one side, and of the exhaust steam on the other. If a common wall is used between the two the result is that the cylinders are liable to fracture. (*H. N. Gresley*, '*Proceedings, Inst. Loco. Engineers.*')

Pistons.—One railway company makes all pistons 0·0625 in. smaller in diameter than the bore of the cylinder. Another railway allows 0·002 in. per inch of diameter for clearance.

VALVES.

For Flat Slide Valves.—*Outside lap* usually $= 0 \cdot 8a$, where a is the width of the steam port. As a rule there is no inside lap, but in some cases a small inside clearance not exceeding $\frac{1}{16}$ in. is allowed. The *lead* is about $\frac{1}{4}$ in., and the *maximum travel* is about $2 \cdot 7a$. Valves of tank engines and others that make frequent stops should be of soft grey cast iron. For long-run engines, 5 parts copper, 1 part tin. (*Wolff.*)

Piston Valves.

Cylindrical or piston valves are extensively used in connection with superheated steam. A large proportion have internal admission of steam. Various packings are in use; some consist of segmental collapsible rings pressed against the liners by steam pressure; others have spring rings, of which there are several kinds. The advantages over flat slide valves are reduced friction, larger port opening, and wear is confined to the easily renewable packing rings.

The diameter of piston valves is about half that of the cylinders, so that circumferential length of ports is approximately $1 \cdot 5d$, where d equals diameter of cylinders.

The lap of the valves on the admission side is usually $\frac{3}{4}$ in. or 1 in., but this has been increased in recent years to $1\frac{1}{2}$ ins. or $1\frac{5}{8}$ ins., and the travel in full gear working has increased correspondingly from about $4\frac{1}{2}$ ins. to $6\frac{1}{2}$ ins. The advantages gained are sharper cut-off and freer exhaust, leading to greater mean effective pressure at high speed and greater economy in steam consumption.

Piston valves are of two kinds, inside admission and outside admission. In the former, steam is admitted between the heads, and exhaust takes place on the outside. The advantages are reduced area for radiation of heat and packing of spindle is only subjected to exhaust steam pressure. Outside admission valves are arranged like flat slide valves, with admission of steam on outside and exhaust inside.

The cast-iron valve liners are turned 0·003 in. to 0·004 in. larger than the steam chest, and are drawn into position by means of screw tackle.

MOTION WORK, ETC.

Allowances used in Fitting the Parts.

		In.			In.
Gland and neck rings	+ piston rod	0·03125	Slipper flanges	+ width of bar	0·03125
Eccentric straps	+ sheaves	0·015	Axle-box flanges	+ horn	0·03125
Side play of do.		0·03125	Brake gear holes	+ pin dia.	0·0156
Slide blocks	− bar opening	0·006	Valve gear pins	− holes	0·004

(*F. Thompson*, '*Journal of Inst. of Loco. Engineers.*')

Coupling and Connecting Rods.

In the case of outside cylinder engines the throw of coupling rods is usually the same as that of the connecting rods, namely, 12 ins. to 15 ins., but in a few cases the crank pins are turned eccentrically on their outer bearing, so that the throw of the coupling rod may be shortened a little. For inside cylinder engines the throws are from 9 ins. to 11 ins. Small throws require rods to be stiff laterally, in order to resist crippling loads, while large throws need a deep rod on account of the bending stresses set up by centrifugal force.

Where a fire-box comes between two coupled axles, some railways allow a little for expansion by heat of the frames by shortening the centres between axles by $\frac{1}{32}$ to $\frac{3}{64}$ in. When steam is raised the frames lengthen, so that the distance between centres of the axles is equal to the length of the coupling rods.

Most coupling rods have plain bushed ends, as have the solid-ended connecting rods of outside cylinder engines on the Great Western and other railways. The gun-metal bushes are turned

slightly larger than the hole by an allowance of 0·002 inch per inch of diameter. This requires a tonnage of 12 tons to press a 7-inch bush into place. A running clearance on the crank pin of 0·002 inch per inch of diameter is specified in one case, $\frac{1}{64}$ inch being a common figure. Another railway presses the bushes in with a load of 20 to 25 tons, and allows $\frac{1}{32}$ in. to $\frac{3}{64}$ in. clearance on the pin.

Valve Gear.

With inside cylinders Stephenson valve gear (see Section XXX, Part III) is general. In special cases, where eccentrics must be dispensed with, Joy's valve gear is used. Outside cylinder engines, with the valves inside the frames, also have the Stephenson gear; but when the valves are outside, the Walschaert valve gear is now preferred.

The outside cylinder engines on the Great Western Railway have 10-in. piston valves placed above the cylinders worked by Stephenson gear from the inside of the frames. The maximum valve travel is $6\frac{1}{4}$ ins., the lap being $1\frac{5}{8}$ ins., and the port opening $1\frac{1}{2}$ ins., although the steam ports are $1\frac{3}{4}$ ins. wide. The eccentrics are set to give $\frac{1}{8}$ in. lead at 25 per cent. cut off, resulting in the valves being about $\frac{1}{8}$ in. blind in full gear. This arrangement gives a sharp cut off, prolonged expansion and a free exhaust, resulting in large mean effective pressure at high speeds. See also p. 581.

Walschaert Valve Gear.

The Walschaert valve gear, figs. 6 and 7, is mainly used on outside cylinder engines. The travel of the valve is derived from two movements, the crosshead of the engine and a return

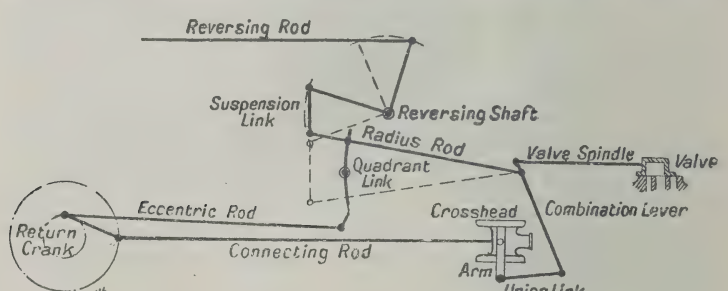


Fig. 6.

crank attached to the main crank pin, or, in the case of inside cylinder engines, from an eccentric on the crank shaft. The first movement is a reproduction of the travel of the crosshead to a reduced scale, the reduction being effected by a combination lever connected at its lower end to the crosshead through a link. In order that the angular movement of the lever shall not be too great by any restriction of length an arm is rigidly attached to the crosshead so that the lever may be suitably lengthened. The combination lever is also attached to the valve spindle and to the radius rod which forms its fulcrum. With an outside admission valve the valve spindle is attached above the radius rod and with inside admission of steam the positions are reversed (see figs. 6 and 7). The function of the combination lever is to give the mid-gear travel, equal to twice 'lap + lead,' and the lever is so proportioned that the stroke is reduced to this amount. The lead is constant for any length of travel.

The second movement, derived from the return crank, is transmitted to a quadrant link rocking on fixed bearings. The maximum angle through which this link swings should be kept within 45°. The radius rod is attached to the link through a die block and communicates its motion to the valve by way of the combination lever. This second movement is at a phase of 90° with the first, and its magnitude can be varied and its direction reversed by altering the position of the radius rod on the quadrant link. In mid-gear the second movement is nil, only the first movement being communicated to the valve. The position of the radius rod in the link is regulated by the reversing shaft, the attachment being either a vibrating suspension link (see fig. 6) or a slider consisting of a die block working in a long slot, as in fig. 7. In the latter case it is more convenient to use the bottom of the link for fore-gear running, while in the former either top or bottom can be used, since it is only necessary to alter the position of the return crank, which

is nominally 90° ahead or behind the main crank. For convenience of design the quadrant link is usually raised so that the eccentric rod is inclined, and this angle of inclination should be

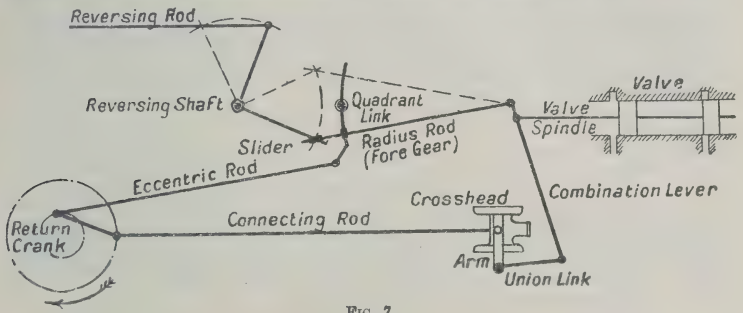


FIG. 7.

taken into account when setting the return crank, being added to or subtracted from the main angle of 90°. The setting of the return crank is as follows :—

Valve.	Fore Gear.	Return Crank.
Outside admission	At top of link	90° behind main crank
" "	At bottom of link	90° in advance of main crank
Inside "	At top "	90° in advance of "
" "	At bottom "	90° behind "

LENTZ POPPET VALVE GEAR.

Each cylinder is fitted with 4 poppet valves. These are disposed in pairs, one for admission and one for exhaust, at each end of the cylinder casting. A central annular chamber is provided in the cylinder casting, through which a camshaft runs at right angles to the valve spindles, which camshaft has mounted on it 2 cams, one of which operates the steam valves and one the exhaust. The cams may come into direct contact with the valve spindles, which in such cases are fitted with a roller and pin, the former bearing upon the cam profile. Alternatively intermediate levers are fitted which are arranged on a suitable fulcrum support.

The fulcrum support is a special feature of the design in question ; it carries the fulcrum pins and may be disposed above the cams or below them in the cam space, as may be most convenient. This fulcrum bar fits in the end of the cam space and also in the cam chamber cover, being securely held in order to prevent it from any possible movement. This arrangement permits of the withdrawal of the fulcrum bar, together with intermediate levers for inspection purposes, and also greatly facilitates the withdrawal of the cams and camshafts for a like purpose.

The intermediate levers are fulcrummed at one end ; at the opposite end they bear against the valve spindles, and in the centre a special roller and pin is fixed, the roller bearing against the cams. The advantage that the intermediate lever system has over what is known as the direct drive lies in the fact that better port openings are secured, as for any given cam profile the valve lift is doubled.

The valves are securely held on the spindles by an improved form of combined nut and inner spring cap, and the spindles, which are hardened and ground, work in cast-iron bushes pressed into the cylinder casting.

Each valve and spindle is fitted with a control spring and a control spring adjusting device fixed in the valve covers. The function of the springs is to keep the valves following the cams when the locomotive is running with steam shut off.

The poppet valve arrangement described is operated by one or other of the usual type of valve motion common in locomotive practice, and to this end the valve connecting-rod joins the upper part of the valve motion with a rocker arm securely keyed to the outer end of the camshafts. With the motion in full gear the camshaft has imparted to it a maximum angular movement corresponding to the greatest valve opening or travel. As the engine is linked up, the amount of motion imparted to the rocker arm becomes less, and likewise the valve opening, and in this manner the cut-off is made earlier or later, as may be desired.

With poppet valves more direct steam passages are obtainable than is possible with piston valves, and this is especially so for the exhaust. This together with the improved port openings

possible with poppet valves, as against piston valves for any given degree of cut-off, are amongst the important advantages offered by a poppet valve gear as described as against piston valve gear.

One of the chief principles underlying the design of poppet valves as arranged in the above manner is a slow opening followed by a quick movement of the valve, a rapid closing followed by a slow movement at the actual time of closure, and also the fact that up to the actual seating of the valve on its face the valve spindle either directly or indirectly (through the intermediate system) is in contact with the cam profile. It is in this way that absolutely noiseless operation is assured, there being no drop action whatsoever.

Steam-tightness of the valve is at all times assured through the special design of the inner and outer valve face.

The Lentz valves have also been constructed to work with a rotary cam drive instead of the usual valve gear.

Caprotti Valve Gear.

This is somewhat similar to the Lentz, but the valves are vertical instead of working in a horizontal direction. The valves are lifted by cams on a rotating shaft, which derives its motion from bevel wheels on one of the main axles. Variation in cut-off and reverse is a rranged by altering the angular advance of the admission cams on their shafts by an ingenious mechanism. The exhaust cams are operatated independently and give a fixed point of exhaust.

See also *The Engineer*, Sept. 2, 1927, and June 29, 1928.

CRANK AXLES.

If W = weight of engine in tons, when in working order ; D = diameter of driving wheels in inches ; then

$$d = K\sqrt[3]{WD}.$$

For the best modern practice, K = ·522. (*Wolff.*)

Dimensions taken from the crank axles of a number of modern British locomotives, with cylinders ranging from 17 to 21 ins. diameter, give the following rule :—

$$d = 0{\cdot}375C + 1,$$

d being the diameter of main bearings and C the diameter of cylinders.

The crank-pin journals are usually the same diameter as the main bearings, the barrel or centre portion about $\frac{1}{4}$ in. less and the wheel seat 1 in. more. Thickness of webs varies from 4 to $4\frac{5}{8}$ ins.

Built-up crank axles are common. The advantages over a forged axle are cheapness, more effective bearing surfaces due to absence of radii, fractured parts may be replaced without scrapping the whole axle. By extending the webs a balanced axle is obtained. Special steels may be used.

The parts are assembled by shrinkage fits, the webs being heated in a gas furnace and shrunk on to the shaft and pins in special fixtures to set the throws in their correct positions. Holes are drilled between web and shaft, into which screwed dowel pins are fitted as keys.

The shrinkage allowance, or the amount by which the hole is smaller than the pin, depends on the tensile strength of the crank webs. For steel to B.S.S. 8c this is 0·025 in. in Horwich practice for axles about 8 ins. diameter.

BEARING SURFACES.

Angle included at centre of journal by arc of contact is usually for coupled wheels, 160° to 170°, for bogie, carrying, or trailing wheels, 60° to 80°. The maximum bearing pressures are approximately 250 lbs. per sq. in. of projected area for coupled wheels, 350 lbs. for bogie, and 450 lbs. for tenders. Bearing brasses are commonly of gun-metal to Admiralty specification and the white-metal lining of Babbitt. In some cases alloys having a lead base are used. Maximum crown wear of brasses or axle-boxes is $\frac{1}{2}$ in.

Lubrication of bearings is by top feed from crown of box or from auxiliary oil boxes, and under pad of wool on spring frame. In case of bogie and tender wheels lubrication is often by underpad only.

The lubricating oil is a medium mineral oil mixed with about 10 per cent. of rape oil. An average oil consumption is 6 to 7 ozs. per bearing per 100 miles.

For crank-pin journals and coupling-rod pins, the pressure may be as much as 1,500 lbs. per sq. in., the load on connecting rod being taken as full boiler pressure on piston area.

For crosshead pins 4,000 lbs. per sq. in. is allowable on account of the small angular movement. Crosshead slippers should be about 60 lbs. per sq. in., a low pressure being advisable on account of the exposure of slide bars to sand and grit.

WHEELS AND TYRES.

Cast-steel wheels are general. The method of securing tyres by means of set screws through the rim of the wheel is the most common. The alternative is to depend on the shrinkage of the tyre to grip the wheel, a retaining ring being inserted to lock the tyre in place laterally. This is a cheap and satisfactory fastening, but there is a tendency for tyres to become loose as they wear thin.

The shrinkage allowance for tyres is usually between 1/750th and 1/1000th of the diameter of wheel centre.

In G.W.R. practice the total shrinkage is given as $\frac{1{\cdot}0625\ D + 10}{1{,}000}$, D being the diameter of wheel in inches.

Driving wheels have tyres with flanges about $\frac{1}{4}$ in. thinner than the other wheels. This eases the engine on curves and protects the crank axle to some extent. Blind or flangeless tyres are uncommon in England, although often resorted to in America.

Of modern types the 4–4–0 engine is the most economical in tyre wear. The flanges of the leading bogie wheels which guide the engine wear much faster than the second pair. Bogie wheels wear less than radial truck wheels, as a bogie sets itself tangentially to any curve. In engines of the 0–6–0 and 0–8–0 types mileage is determined by the flange wear of the leading coupled wheels.

Tyres with treads worn hollow are detrimental to points and crossings. A flange worn sharp is liable to cause derailment. Tyres are turned down to $1\frac{5}{8}$ ins. on tread and allowed to wear to $1\frac{1}{4}$ ins. before they are scrapped, except in case of large driving wheels, where $1\frac{3}{4}$ ins. or 2 ins. is the limit.

Wheels are forced on axles by hydraulic pressure. A minimum of 10 tons per inch diameter of wheel seat should be allowed when rape oil is used as a lubricant, 100 tons being a usual amount for large axles; or, if D is the diameter of wheel seat in inches, the load = 10D tons.

The wheel is bored slightly less in diameter than the axle, the difference expressed as a fraction of an inch being from $\frac{2D + 3{\cdot}5}{1{,}000}$ to $\frac{4D + 2}{1{,}000}$, depending on rigidity of the boss, quality of steel, etc.

In some cases the wheel and wheel seat are slightly tapered, the difference in diameter in the length of the wheel seat being $\frac{1}{32}$ in. The G.W.R. practice is to bore the hub parallel to and smaller than the axle by an allowance varying from 0·0015 to 0·002 per inch of diameter. Tonnage for pressing on the wheels by hydraulic press is from 8 to 12 tons per inch of diameter, the intensity of the grip depending to a large extent on the thickness of the hub. The best results are obtained when this is not less than half the diameter of the axle in thickness.

If the tyre is on the wheel, it takes 25 per cent. more pressure to force the wheel on the axle for the same allowance.

Keys should be tapered $\frac{3}{32}$ in. per foot and fitted to within $\frac{3}{4}$ in. before being driven home.

Crank pins should be forced in with a minimum pressure of 13 tons for each inch of diameter, and the ends of the pins riveted over inside the wheels.

BRITISH STANDARD CONTOURS FOR LOCOMOTIVE TYRES.*

(For British Railways 4 ft. 8½ in. Gauge.)

(No. 276—1927.) (*Abstract.*)

Contour A recommended for the leading and trailing wheels of bogies, the wheels of pony trucks and radial axles, or any pair of wheels which constitute the leading or trailing wheels of a locomotive, whether tender or tank engine.

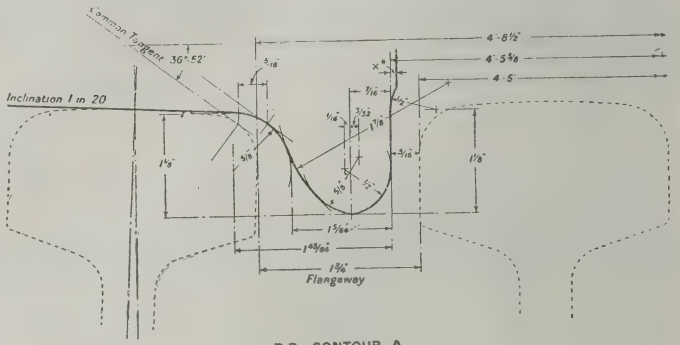
Contour A or G recommended for the leading wheels of a set of coupled wheels, when led by a bogie, pony truck, radial axle, or any pair of carrying wheels.

Contour A or G recommended for the trailing wheels of a set of coupled wheels, when followed by a bogie, pony truck, radial axle, or any pair of carrying wheels.

Contour G or E recommended for coupled wheels, other than the leading and trailing coupled wheels of groups 6, 8, and 10 coupled.

* By permission of the British Engineering Standards Association.

BRITISH STANDARD CONTOURS FOR LOCOMOTIVE TYRES FOR BRITISH RAILWAYS.


Common Tangent 36°-52'
Inclination 1 in 20
4'-8½"
4'-5⅜"
4'-5"
1⅛"
1⅝"
1⅝"
1⅝

Flexible Wheel Base.

Main-line engines should be capable of traversing a curve of 5 chains radius, and sufficient side-play is given to bogies and radial wheels for this. The theoretical amount required is not attained in practice, as engine frames and track spring a little on sharp curves.

There is a small amount of side-play allowed between flanges of wheel and rail. With new tyres on new rails, this is about $\frac{1}{16}$ in. each side, or a total of $\frac{1}{8}$ in. The flanges of driving wheels are thinned down to give about $\frac{3}{4}$ in. to $\frac{7}{8}$ in. total. Additional side-play is sometimes given by allowing a little clearance between axle-boxes and guides. It is very exceptional to give any side-play between journal and bearings.

Baldry's rule for the radius of pony trucks is

$$R = \tfrac{1}{2}\left(B - \frac{A^2}{B}\right),$$

where B is horizontal distance from truck wheels to centre of rigid wheel base and A is half rigid wheel base.

Bogies, in addition to turning on the pin, are allowed a certain amount of transverse movement. This is controlled by springs, suspension links, or inclined slides. Some bogies have curved slides giving radial instead of transverse movement. Fixed centre bogies are now but little used.

Counterbalancing Locomotive Driving Wheels.

The revolving and about 60 per cent. (this figure varies considerably) of the reciprocating parts should be counterbalanced. The tendency at the present time is to reduce this figure, and in recent three- and four-cylinder engines it has been as low as 40 per cent. The revolving parts develop centrifugal force, the horizontal component of which disturbs the engine longitudinally, to which must be added that due to the momentum of the reciprocating parts: also, these two forces produce a sinuous motion by acting first upon one side of the engine and then on the other. The vertical component of the centrifugal force generates a rolling motion by acting alternately upon each side of the engine. A peculiar effect is also produced by the obliquity of the connecting-rod to the slide bars, a turning upon a horizontal axis, together with an up-and-down movement at the crank axle. The horizontal forces are the most injurious, although some American engineers consider the vertical to be so; but English practice is to take a medium course between excessive horizontal and vertical disturbing influences. It has been clearly demonstrated that high centres of gravity are advantageous.

When of outside type, balance the balance weight by means of the moving parts on one side and a small weight, at the same angle from the balance weight, in the far wheel.

For inside cylinder engines place a balance weight in each wheel inversely proportional to the distance from the cylinder and diametrically opposite the crank. Thus there are two counterweights in each wheel, but these are combined to form a single balance weight, usually of crescent shape.

American practice is to allow 1/400th part of the engine weight to remain unbalanced in the reciprocating parts on each side. The remainder is then divided amongst the coupled wheels; or,

$$r = \left(R - \frac{W}{400}\right)/N,$$

where,

r = reciprocating weight in each wheel at crank pin radius; R = total weight of reciprocating parts; W = weight of engine in working order; and N the number of coupled axles.

Cast steel wheels have the balance weights cast solid with them; occasionally the weights have pockets cored out for the reception of lead. Another practice is to cast the wheels without weights and to rivet steel plates to the spokes. Molten lead is then run into the pockets formed between the spokes.

In order to obtain uniformity of results, some railways make use of a wheel-balancing machine. The pair of wheels to be tested is mounted on bearings supported by springs set radially round them. Weights are attached to the crank pins to represent the revolving and a proportion of the reciprocating parts. The wheels are then spun round at high speed and the counterbalance adjusted until steady running is obtained.

BOILERS.

Locomotive boilers should have a factor of safety of 5: considerations of weight make a larger factor undesirable. The copper fire-box stays should not be stressed greatly in excess of 4,000 lbs. per square inch, as copper loses much of its tensile strength at the steam temperature, while steel is not greatly affected (see pp. 171, 173).

Coned boiler barrels, although more costly, are adopted by some railways in order to obtain the most powerful boiler for a given weight, by the reduction in diameter at the front end, while allowing ample space by reason of the larger diameter in the region of the fire-box tube plate, where steam generation is at a maximum.

Some railways prefer the 'Belpaire' type of fire-box, while others adopt the round top with direct crown stays. In British practice the long narrow fire-box is almost universal. The maximum length practicable is about 10 ft. 6 ins., giving a grate area of some 30 sq. ft. For larger grate areas the wide fire-box is adopted in American practice, permitting of 70 sq. ft. or more, but it necessitates the addition of a small pair of carrying wheels.

Grate Area and Heating Surface.

Grate area in square feet = $\frac{\text{average I.H.P.}}{30 \text{ to } 40}$. Total heating surface should be not less than 60 × grate area, and at least 10 per cent. of this should be in fire-box.

The ratios giving the best results are as follows:—Grate to total heating surface, 1 : 55 to 65; grate to fire-box heating surface, 1 : 5 to 5·5; and fire-box heating surface to total heating surface, 1 : 11 to 12.

Bituminous coals require space in the fire-box for the proper combustion of their gases before entering tubes. The fire-box volume should be 6 or 7 cubic feet for every square foot of grate. Under these conditions 15 lbs. of air are needed for each pound of coal burnt. The air spaces through fire-bars should not be less than 33 per cent. of grate area, and the damper opening not less than 15 per cent. Air spaces of 50 per cent. and damper opening 20 per cent. are not too large. In coupled engines, where an axle comes close under the grate the slope of pan over it should be not less than 45°; and the restricted space for air between pan and fire-bars must have an area of at least 15 per cent. of the part of the grate which it supplies.

Fire-bars are of wrought iron or cast iron, and usually $\frac{3}{4}$ in. wide at top. Gaps of $\frac{3}{8}$ in. to $\frac{1}{2}$ in. between these give an air space of 30 per cent. to 40 per cent. of the grate area.

Boiler Tubes.

Diameter and Length of Tubes.—The diameter of tubes varies with the length of barrel. Short barrels require 1$\frac{1}{2}$ in. to 1$\frac{5}{8}$ in. outside diameter tubes; those over 10 ft. long, 1$\frac{3}{4}$ in. to 1$\frac{7}{8}$ in. tubes. Long barrels, 13 ft. and upwards, require tubes 2 ins. to 2$\frac{1}{4}$ ins. diameter.

Thickness of locomotive boiler tubes (Caledonian Railway), 5-in. superheater flue tubes, No. 8 W.G.: 1$\frac{1}{4}$-in. tubes, No. 11 W.G.

British Standard.—For 1$\frac{5}{8}$ in. outside diameter, 13, 12 or 11 s.w.g.; for 1$\frac{3}{4}$ in., 1$\frac{7}{8}$ in., and 2 ins., 12, 11 or 10 s.w.g.; for 2$\frac{1}{4}$ ins., 11 or 10 s.w.g.

Water space between tubes should not be less than $\frac{5}{8}$ in. for 1$\frac{1}{2}$ in. diameter tubes, and $\frac{3}{4}$ in. for 2$\frac{1}{4}$ in. tubes at the fire-box end, but may be pitched $\frac{1}{16}$ in. closer at the smoke-box end.

Life of Boiler Tubes.

The life of copper and steel tubes in locomotive boilers, taken over a period of eight years, is as follows:—

	Miles.
Copper 1st period (new)	110,000
„ 2nd period (drawn)	80,000
„ 3rd period (pieced)	50,000
Total	240,000

Subsequently 30 to 40 per cent. of those pieced are treated so again.

	Miles.
Steel 1st period (new)	70,000 to 80,000
„ 2nd period (pieced)	30,000 to 40,000

(*G. Hughes.*)

Calculation of the Heating Surface of Boilers for Locomotives.

The Association of Railway Locomotive Engineers' method of calculating the heating surfaces of locomotives is as follows:—

Fire-box.—The outside (wetted surface) of inner fire-box plates to be taken to top of foundation ring. Areas of tube-holes and sectional area to wetted side of fire-hole ring, or flanged plate, as the case may be, to be deducted; section of fire-hole being taken close up to the inner fire-box plate.

No deductions to be made for copper stays or roof stays. If the boiler has fire-box water tubes the surface to be taken on the inside (wet side) of tube.

Tubes.—The outside surface of tubes (wet side) to be taken, calculated on uniform diameters for both ordinary and superheater tubes; the length of tube being taken between tube-plates.

Elements.—The inside surface (steam side) of elements to be taken, the effective length of tube being from and to smoke-box end of large flue tube.

Smoke-box tube-plate.—No allowance to be made.

Equivalent heating surface of superheaters.—In order to compare the steaming capacities of superheater and non-superheater boilers, an equivalent heating surface is sometimes given for the superheater boiler, as owing to the lesser evaporative surface of the tubes, it appears less powerful. If 1·5 times the superheater surface is added to the evaporative heating surface, an equivalent heating surface is formed, which may be compared with that of a non-superheater boiler. Thus, a non-superheater boiler has 2,455 sq. ft. of heating surface, but when converted to superheater has 2,031 sq. ft. of evaporative heating surface and 460 sq. ft. of superheating surface. The addition of 1·5 times the superheating surface to the evaporative gives a total of 2,721 sq. ft., and indicates that the boiler is 11 per cent. more powerful when superheated.

Fire-Boxes.

Many attempts have been made to substitute steel for copper plates in fire-boxes of British locomotives, but the results have not been very successful. Steel fire-boxes are universal in

America, where conditions favour their adoption. Greater success had attended the introduction of steel side stays for copper fire-boxes. Owing to the higher tensile strength the diameter over threads is about ⅝ in. for a pitch of 3½ ins. The stay projects through the fire-box casing plate about ¼ in., but the projection through the copper plate is furnished with a hexagon nut of reduced thickness. This forms a renewable head for the stay and protects it from injury by the fire.

With the very high pressures now being used in America, the flat plates have given place to fire-box walls formed of steel tubes set closely together and expanded into drums, top and bottom, connected to the boiler barrel. A casing round the outside completes the fire-box.

In locomotive practice a pitch of about 4 ins. is almost invariably adopted for the screwed stud stays in the fire-box. Theoretically this is much closer than is necessary, but the practice is good, as it provides a substantial margin of wear in the copper plates.

Life of Fire-boxes.

The life of fire-boxes depends more upon the tons hauled than the actual miles run. Copper fire-boxes on the Lancashire and Yorkshire section of the L.M. & S.R. have a life of from 10 to 16½ years. The older engines with 140 lbs. pressure are found to run their fire-boxes up to as much as 500,000 miles: an average figure for 70 engines being 310,000 before replacement by new ones. In standard engines, such as the radial tank and 6-wheeled coupled goods classes, working at 160 lbs. pressure, the average life of a copper fire-box is eleven years, with average mileages of 272,000 and 235,000 respectively. These figures are based upon records of 120 locomotives. Individual copper tube-plates in the fire-boxes above named require renewing during the life of the box. In the radial tank class, records of 53 tube-plates show an average life of seven and three-quarter years with 205,000 miles before renewing, whilst the 6-wheeled goods average 135,000 miles for 144 tube-plates. The remaining plates, viz., door and wrapper plates, practically do not require replacing during the life of the fire-box, but certain repairs and patches are fitted if necessary.

Smoke-boxes.

In superheater engines the blast pipe is set some 30 ins. forward of the tube-plate, to allow ample room for the superheater header, and the orifice is about 7 ins. below the centre line of the boiler, but this figure varies considerably. The blast pipe should not be too low or there is risk of cinders being drawn into it. The diameter of the orifice should be as large as possible consistent with good steaming. For cylinders 16 ins. to 22 ins. diameter, the orifice varies from 4½ ins. to 5⅝ ins., 5 ins. being usual with a 19-in. cylinder, but a good deal depends on class of coal, work performed, and other conditions.

Chimneys are made as high as load gauge permits, and are extended downwards into the smoke-box. The main consideration as to dimensions is that a cone of steam diverging from the blast pipe orifice with a total angle of 1 in 6, should just pass clear through the top of the chimney, leaving a small annulus for the gases. The chimney should be slightly tapered and terminate in a bell, so that gases may be gradually entrained and increased in velocity.

In American practice a diaphragm is fitted across the smoke-box above the top row of tubes and carried down just below the blast-pipe orifice, terminating a little in front of it. By this means the draught through the tubes is equalised and the emission of sparks prevented. Some British engines are also fitted with this device.

The smoke-box vacuum obtained in modern superheater engines varies with the horse-power, and is about 5 ins. of water, with a maximum of 7 ins., in heavy passenger work, and about half these amounts for superheater goods engines.

Regulators.

The Hulburd 'Servo' Type Locomotive Regulator.

(*Hulburd Patents, Ltd.*, 28 *Park Road W.*, *Acton, London, W.* 3.)

In this arrangement a small auxiliary valve only is operated by the operating lever. The main valve, which is without mechanical connection with the operating lever, is *moved exclusively by the action of the steam* and in strict conformity with the spindle motion.

As the small resistance of the throttle may be overcome without effort, the regulator is made with a comparatively *long spindle stroke* contrary to short operating lever, the regulator is made with a comparatively *long spindle stroke* contrary to short operating regulators. This long stroke permits the slots in the lantern to be shaped so that other types of of steam which is admitted to the steam chests *increases gradually* during the greater part quantity spindle stroke. The pressure in the steam chests will therefore increase rapidly only towards the end of the stroke until perfect balance with the boiler pressure has been attained. This feature of the regulator, it is claimed, results in even the longest trains being started quietly and without loss of time, whilst jerks are absolutely eliminated: also it is claimed that 'Priming' is eliminated, even with a high water level.

Superheaters.

The superheater of a locomotive boiler performs two functions. In the first place, it acts as a steam drier by evaporating the moisture carried over with the steam to the cylinders. Secondly, the dried steam is heated some 200°–400° F. above its normal temperature at the working pressure, and a volumetric increase occurs. By superheating the steam, not only is the loss due to moisture avoided, but initial condensation in the cylinders is eliminated or greatly reduced. Losses due to leakage are also less. The expansion line of indicator diagrams taken from cylinders using

superheated steam show a more rapid fall of pressure than with saturated steam. On the other hand, the compression line is also lower, showing less back pressure, so that there is little difference in the mean effective pressure to that of cylinders using saturated steam at the same initial pressure and cut-off. The orifice of the blast pipe with a superheater engine needs to be about $\frac{1}{8}$ in. smaller as the weight of steam discharged in a given time is less than in saturated steam engines. The chief gain due to superheating the steam to a final temperature of about 650° F. results in a saving of from 15 to 25 per cent. in fuel and 20 to 30 per cent. in water. The steam consumption of saturated engines is approximately 27 lbs. and coal consumption 4 lbs. per i.h.p. The corresponding consumptions of engines superheated about 200° F. are reduced to 19 lbs. of steam and 3·0 lbs. of coal per i.h.p.

Boiler repairs are also less, due to smaller demand for steam and lower rate of combustion of fuel. In some cases the working pressure is reduced, larger cylinders being used on account of the increase in volume of steam due to superheating. Engines fitted with superheaters can run longer distances without replenishing water and fuel supply. This is particularly advantageous in the case of tank engines, where the supplies of both are limited. The superheater adds little or nothing to the starting power of an engine, but it enables a larger horse-power output to be maintained, and consequently the hauling power is increased when running. Engines which had become obsolete on account of the growth of train loads have been given a new lease of useful life by the addition of a superheater. In some cases it has been successfully applied to engines with flat unbalanced slide valves between the cylinders, the only alteration being the provision of new tube plates for the superheater flues. A moderate degree of superheat does not appear to affect bronze valves provided great care is taken with the lubrication. Either a mechanical force-feed or a sight-feed displacement lubricator can be used, but the lubricant must be of first-class quality. With piston valve engines, the chief trouble is the formation of carbon deposit. This clogs the packing rings, chokes the ports and coats pistons, covers valve heads and spindles, etc. When steam is shut off, the pumping action of the main pistons, when running, draws smoke-box gases dust, and ashes down the blast pipe, and solids adhere to the oily surfaces. Some decomposition of the oil also occurs. This action can be considerably lessened by (1) putting the reversing lever into full gear when steam is shut off; (2) by the provision of air valves to automatically admit air to the steam chests and destroy any vacuum created; (3) by automatic by-pass valves which put the opposite ends of the cylinder in communication when steam is shut off. Another practice is to admit a very small quantity of steam, either through the regulator or by means of a special valve. By filling the spaces with steam instead of air, decomposition of the oil is prevented.

Fire-tube Superheaters are principally employed. Several rows of large fire tubes are fitted in the boiler, and into these superheating tubes extend from a header or headers in the smoke-box towards the fire-box and back again. The construction of the header varies, as well as the arrangement of tubes, but the object in all designs is to direct steam through small tubes exposed to hot gases direct from the furnace and supplying an extended steam path, so that while travelling at a high velocity a considerable degree of heat is taken up by the steam.

The Swindon (used only on the Great Western Railway) and Vaughan-Horsey (previously fitted in Canada) designs are of this class, with their own constructional features, and both are designed for a moderate degree of superheat. The Schmidt design aims at a high superheat—from 250° to 530° F above the temperature of saturated steam. It consists of a suitable header or steam collector, to which are attached four-fold elements. This provides a long steam path, and in virtue of the high steam velocity it is able to take up the required heat.

To obtain superheated steam of from 625° to 675°, the area through flue tubes should be approximately 51 per cent. of the whole.

In conjunction with some fire-tube superheaters a damper mechanism operated by an automatic steam cylinder was fitted so that passage of furnace gases through the apparatus was prevented when steam was not being used. In some cases this was hand operated, so that the dampers could, if desired, be open during shunting, and like operations, to maintain the superheat. The provision of dampers or draught retarders is no longer considered essential.

Some railways fit anti-vacuum valves on the top of the smoke-box. When an engine is driftingwith steam shut off, air is drawn from the valves through the superheater elements and thus prevents overheating. At the same time the air is heated before coming into contact with the valves, and thus reduces the risk of the valve liners becoming loose by contraction from sudden cooling.

'M.L.S.' SUPERHEATERS.

(The Superheater Company, Limited, London and Manchester; 'Schmidt' and 'Robinson' Superheater Patents.)

The headers or collectors of this apparatus are all made of cast iron of a special semi-steel mixture, giving a tensile strength of not less than 28,000 lbs. per sq. in., and are situated in the smoke-box.

The elements, fig. 8, are of double loop type, giving a four-fold steam path; they are made from solid cold-drawn steel tubing, and the three return bends on each element are formed by the 'M.L.S.' Integral Machine Forged Return Bend Process, by which these bends are made integral with the tubing itself, and all electric and oxy-acetylene welding eliminated, thereby prolonging the life of the element and ensuring a clean smooth finish both internally and externally.

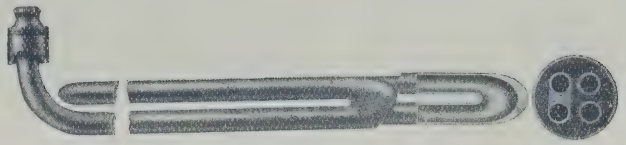


FIG. 8.—'M.L.S.' Ball oint Element.

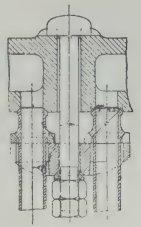


FIG. 9 —'M.L.S.' Ball Joint.

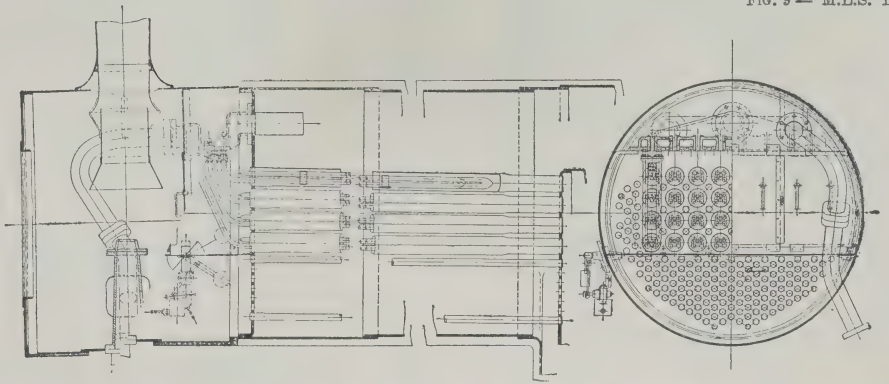


FIG. 10.—'M.L.S.' Through Bolt Header, Type 'A.M.,' with Ball Joint Attachment.

THE 'M.L.S.' SUPERHEATER.

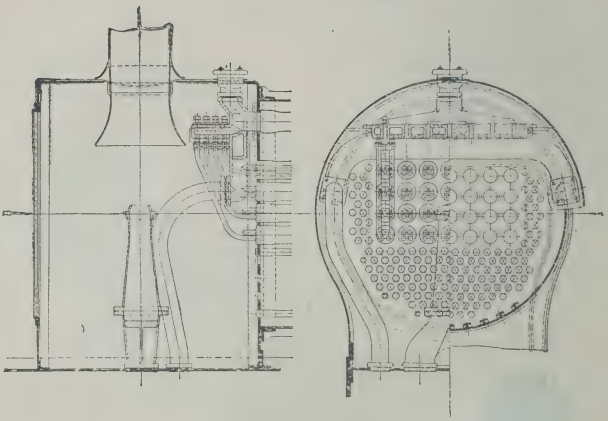


FIG. 11.—'Stirling' Through Bolt Header, Type 'AS.'

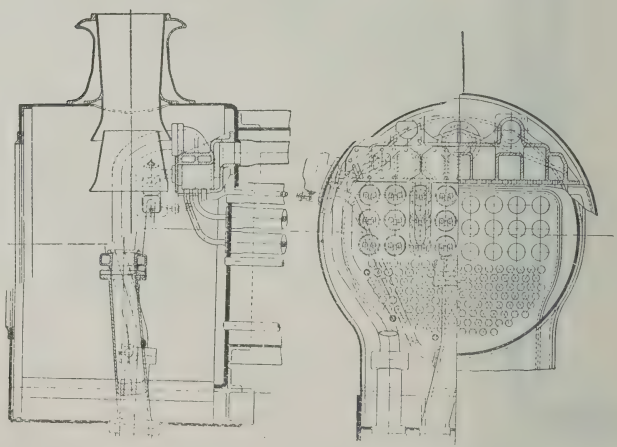


FIG. 12.—'Robinson' Expanded Joint Header, Type 'AR.'

Where elements are supplied having the ball joint attachment, fig. 9, the ball end is also made integral with the tube itself by upsetting the tube, thus avoiding any welding. It is afterwards turned and ground to gauge, having a diameter of $2\frac{1}{8}$ ins.

Where bolts are required for the joint, these are supplied in high tensile steel having a tensile strength of not less than 50 tons per sq. in. with a yield point of 35 tons per sq. in., so preventing the possibility of stretching when screwing up.

Other details, such as clamps and washers, are all made from drop forgings, which ensures sound material and maximum strength.

All elements are fitted with one or more steel bands and supports of robust construction, so arranged on the elements that the latter are properly positioned in the flue tube and at the same time reducing the obstruction to the flow of gases to a minimum.

Through Bolt Header, Type 'A.M.'

In this design of header, fig. 10, the casting is divided into transverse compartments or pockets communicating alternately with the saturated and superheated steam passages which are situated longitudinally at the back and front of the header respectively. Air passages which act as insulating spaces between these compartments are utilised for housing the element clamp bolts, the 'T' heads of which lie in a horizontal plane above the header, the nuts being below, or if desired the bolts may be used with their heads bearing upon the element clamp and the nuts on the top of the header. The ball joint attachment is almost universally fitted with this type of header, and is shown in fig. 10. As will be seen, any element can be easily withdrawn without removing the bolt.

'Stirling' Through Bolt Header, Type 'AS.'

This header, fig. 11, has similar insulating passages between the compartments or pockets. In this case, however, the saturated and superheated passages are arranged one above the other at the back of the header with communicating compartments or pockets at right angles. The slots or insulating passages, in addition to allowing for independent expansion of the compartment walls, are utilised for housing the clamp bolts with nuts either on top of the pockets or below the clamps as desired. It will be observed that to withdraw the element it is merely necessary to slacken the nuts, after which the bolt will slide out of the insulating space as the element is withdrawn, without having to remove the bolt from the clamp.

This type is very extensively fitted in India and the Colonies in conjunction with the ball joint attachment.

'Robinson' Expanded Joint Header, Type 'AR.'

This type of header, fig. 12, is designed to allow the superheater element inlet and outlet ends being expanded directly into the bottom wall, instead of being connected by bolts, nuts, flanges, etc. There are two longitudinal chambers, one for saturated steam at the back of the header, and one for superheated steam at the top, and these chambers open out alternatively into saturated and superheated steam compartments.

The latter are open at the front of the casting to permit of the introduction of tools for expanding and extracting the elements. Front cover plates, secured by studs and nuts, are provided for closing these openings.

The Weir System of Boiler Feeding and Feed-Water Heating.

In this system, fig. 13, the boiler feeding is carried out by means of a direct acting steam pump, B. The discharge side of the pump is connected through J to a surface heater, A, of special design. The heater is arranged in any convenient position on the locomotive, and uses part of the exhaust steam from the blast pipe, this steam passing to A through a pipe, D, which is teed off from the blast pipe. The heating steam impinges on the outside of the heater tubes. The orifice of the blast pipe is not altered in fitting the heater connection, so that the effective exhaust area is substantially increased with consequent reduction in back pressure on the engine.

The pump, B, has double-acting steam and water cylinders. The suction and delivery valves for both ends of the water cylinder are arranged within the casing in front of the pump. Steam to the cylinder passes through H, and is controlled by a Weir piston valve chest. No drain valves are provided on the steam cylinder, as the pump will start from cold, even with the cylinder full of water of condensation. Plugs are fitted, however, on the steam cylinder, the steam valve chest, the water valve chest, and the pump cylinder, and may be utilised when it is considered necessary to empty the system.

While the vertical type of pump is usually adopted special circumstances may arise in which the use of a horizontal type is more convenient; and to meet these cases the larger sizes of pumps are designed on the horizontal pattern.

In the design of the heater, strength and simplicity with ready facilities for inspection are special features. A rectangular shell of cast iron forms the steam space, with branches for the admission of the exhaust steam, and a connection for draining the condensed water. The tube plates are fixed to the ends of the shell by collar bolts, which also carry the end covers. The latter are provided with ribs which divert the feed water through the various passes of the heater. The tubes are of heavy solid drawn copper, and are so expanded into the tube plates that they form strong and reliable joints under all conditions.

THE WEIR SYSTEM OF BOILER FEEDING AND FEED-WATER HEATING.

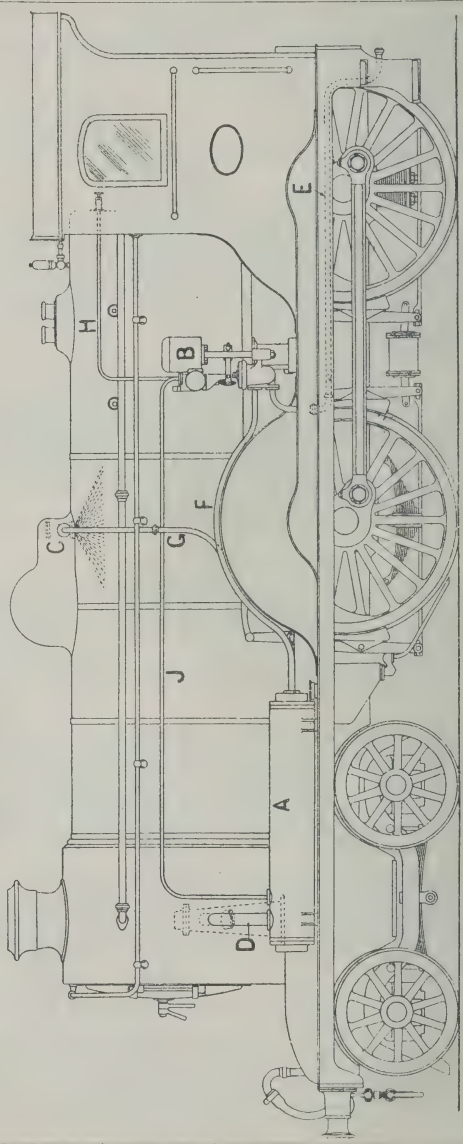


FIG. 13.

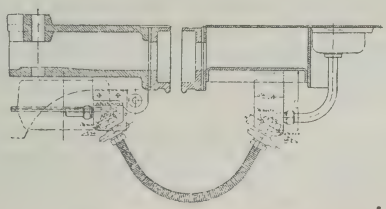


FIG. 14.—Feed-water Connection.

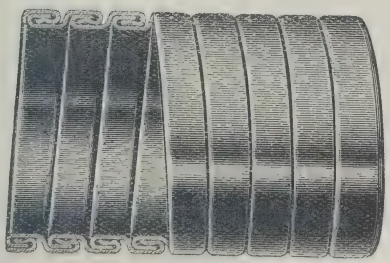


FIG. 15.—Section of Flexible Metallic Tubing
(United Flexible Metallic Tubing Co., Ltd.)

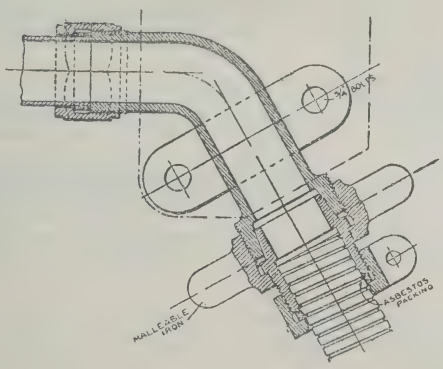


FIG. 16.—'Ball and Socket' Joint.

The feed water, after traversing the heater tubes, passes through G directly into the steam space, through a special spraying valve, C, situated on the top of the boiler. By this means the feed water reaches the water surface in the boiler at practically the steam temperature, and secures increased efficiency and decreased upkeep of the boiler, due to the uniformity of temperature maintained. Where chalky waters are in use, the heating of the feed water to the steam temperature has also a pronounced effect on the character of the deposit; this takes the form of a fine sludge, which may be readily washed out. Both pump and heater are of neat and compact design, the installation occupies very little space and is in no way obtrusive on the locomotive. The fitting of the gear is, moreover, simple, as it involves no serious alterations in design or structure.

It is claimed that by the system, with feed-water temperatures below 60° F. in the tender, and with any type of locomotive, saturated or superheated, a definite fuel economy of not less than 10 per cent. for the same locomotive duty is obtained, or for the same fuel consumption an increase of 10 per cent. in the evaporative power of the boiler; the feed pump gives a degree of feed flexibility and reliability which is not obtainable with injector feeding.

Hulburd's Seamless Soft Copper Jointings.

(*Hulburd Patents, Ltd.*, 28, *Park North Road, Acton, London, W.* 3.)

These copper jointings are made from pure copper, and are absolutely seamless. Unlike thin corrugated stamped metal jointings, they require no red lead or other filling, but are placed between the flanges just as they are, and make an absolutely metal to metal joint; under no conditions do they stick to the faces of the flanges; they are practically indestructible, as they may be used over and over again, and the joints made with them can be broken as often as desired; cases are known in which the same jointing has been used as many as eighteen times.

The jointings, which are made to any size and shape, will stand any known pressure, and are standard equipment on all the largest railways in the world in connection with cylinders, valve chests, boiler domes, safety valve seatings, manhole covers, cleaning doors, and superheater elements, etc., and all classes of pipe work.

Flexible Metallic Tubing.

On the locomotive, an essential fitting is the feed-water connection between the engine and the tender, fig. 14 (p. 607), this connection allowing any surplus steam from the boiler to be used to heat the feed water in the tender. Fig. 15 shows Flexible Metallic Tubing (*The United Flexible Metallic Tubing Co.*) which is specially suitable for the purpose, as rubber connections will not withstand the heat.

Flexible Metallic Tubing is also used for carriage heating, and being much stronger and lighter than rubber hose it will stand the full boiler pressure if the reducing valve fails, the pressure at the carriages being only about 40 lbs., whereas the boiler pressure is probably 200 lbs. The tubing when exposed to cold is in no way affected, even should any accumulation of water in the pipes be frozen solid.

With this Flexible Metallic Tubing there is no possibility of the drain valves of the couplings being stopped up by anything wearing or blowing off the interior of the tubes.

The 'Ball and Socket' joint, fig. 16 (designed by the locomotive engineer of the former Lancashire and Yorkshire Railway), when fitted to the tubing, gives great flexibility. The asbestos ring, upon which the 'ball' rests in a measure, allows the ball joint to be flexible when under steam pressure.

The tubing is manufactured in various lengths, and in sizes from ½ in. to 12 ins. internal diameter, and to resist pressures from 20 lbs. to 4,000 lbs. per sq. in.; it has been largely used in the place of rubber hose on account of its great strength, flexibility at all temperatures, nonliability to the attacks of vermin or insects, and that it remains indestructible by great heat, oils, or grease.

Flexible Metallic Tubing is specially suitable for tropical climates, as the sun's rays have no effect upon it, whether 'in stock' or in use.

Injectors.

Table showing Maximum Delivery of Combination Injectors.

(Gallons per hour working on dry steam, with feed water at 60° F., lifting 3 ft. and delivering against pressure equal to steam at back of nozzle.)

Size.	Boiler Pressure in lbs. per Square Inch.							Size of Pipes.			
	60	90	120	150	180	210	225	Steam.	Water.	Out-flow.	Deliv-ery.
								Ins.	Ins.	Ins.	Ins.
6	550	670	780	870	840	800	790	1	1¼	1¼	1¼
7	750	920	1,070	1,190	1,140	1,090	1,080	1	1⅜	1⅜	1⅜
8	980	1,200	1,400	1,550	1,490	1,420	1,410	1¼	1½	1½	1½
9	1,240	1,520	1,770	1,970	1,880	1,800	1,790	1¼	1¾	1¾	1¾
10	1,530	1,880	2,180	2,430	2.330	2,220	2,200	1½	1⅞	1⅞	1⅞
11	1,850	2,270	2,650	2,940	2,820	2,700	2,670	1½	1⅞	1⅞	1⅞
12	2,200	2,700	3,150	3,500	3,350	3,200	3,150	2	2	2¼	2¼

(*J. N. Gresham*, '*Jour. of Inst. of Loco-Engineers.*')

Metcalfe's Patent Exhaust Steam Injector.

The simplest method of using exhaust steam for feed-water heating is by an exhaust steam injector. A saving of 10 to 15 per cent. of coal, and 12 to 15 per cent. of water is claimed.

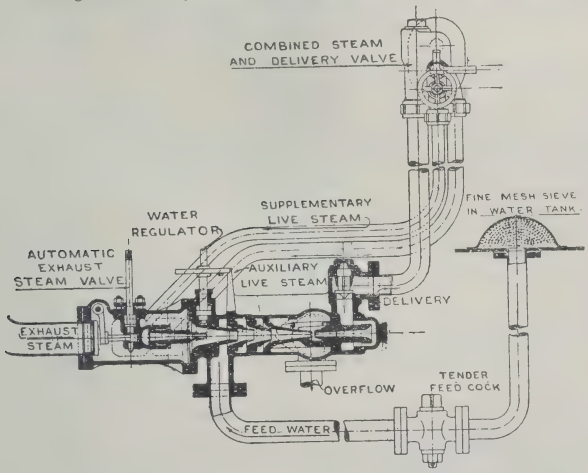


Fig. 17.—'Metcalfe' Patent Exhaust Injector (F. Type).

The 'F' type Metcalfe Patent Exhaust Injector is capable of working on exhaust steam alone against boiler pressures up to 150 lbs. per sq. in., and with the aid of a small jet of supplementary live steam it can feed against 300 lbs. pressure. The temperature of feed water delivered

to boiler varies from 200° to 220° F., depending on exhaust steam pressure. The amount of supplementary live steam is very small and does not exceed 2½ per cent. of evaporation of boiler. The water regulator, which slides the cone, can be adjusted so that the injector is kept continuously at work throughout the run, and will maintain a constant water level and steam pressure. By the introduction of very hot feed water the steaming power of the boiler is extended and the stresses in the plates, due to unequal contraction, are reduced. When the engine regulator is shut and the flow of exhaust steam ceases, the injector can be kept at work by the admission of a small quantity of live steam to take its place: this is termed auxiliary steam. The exhaust steam is conveyed from the cylinders to the injector by means of a large pipe. After leaving the cylinders the steam passes through a simple form of grease separator, consisting of a helical vane. Greasy water is discharged through an automatic drip valve.

In the latest 'H' type, the operation of the injector is entirely automatic upon the opening of the steam valve. The admission of steam opens the water valve and exhaust steam inlet, and these automatically close when the steam valve is shut. If the regulator is closed so that there is no exhaust from the cylinders, an auxiliary supply of live steam is automatically given in place of the exhaust when the injector is at work.

A size 10 injector will deliver a maximum of 2,300 gallons of water per hour, and this can be reduced by the water regulator to a minimum of 1,380 gallons per hour (working continuously).

Gresham's Patent Hot Water Combination Injector.

These injectors, fig. 18, work with either hot or cold water in the usual way, *i.e.* by opening the steam valve and water cock, and then regulating the water cock if necessary.

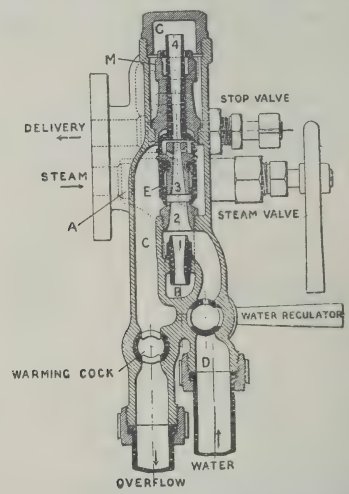


FIG. 18.—'Gresham' Patent Hot Water Combination Injector.

An internal steam pipe from the dome of the locomotive conveys steam to the injector steam valve A, which, being opened, admits steam to the steam nozzle 1 by the passage B. The steam issuing from the steam nozzle opens the two side flaps E; it also lifts the sliding delivery cone 4 making an aperture at M and passes out freely down the overflow pipe. In doing so it creates a partial vacuum in the water pipe D, and the water rises to the injector. The water then coming in contact with the steam travels with it through the lifting cone 2, and, in condensing, it creates a vacuum which closes the two side flaps E, thus preventing any air entering the cones. The velocity of the steam being now largely transferred to the water, the latter passes from the lifting cone 2 through the combining cone 3. After passing through this combining cone, the jet flows out of the aperture M and down the overflow pipe C until such time as it attains sufficient velocity to carry itself past this space and enter the delivery cone 4. When it reaches this

point the jet is at its maximum velocity, and is so great that it is sufficiently powerful to lift the back pressure valve (not shown) and so enter the boiler. At the same time the pressure thus created in the chamber G forces the sliding delivery cone down on to its seat and closes the aperture M, shutting in the water which otherwise would be boiled off when the injector is working with water at 130° F. When the water is below 100° F. the injector will work dry with the aperture M open, like an ordinary injector.

In order to heat the feed water, exhaust steam is conducted from the cylinders to the tender by a large pipe. This connects to a multi-tubular heater through which the feed water flows. The condensation of the steam gives up heat which is absorbed by the feed water, thereby raising its temperature.

BRITISH STANDARD SPECIFICATIONS FOR MATERIAL USED IN THE CONSTRUCTION OF RAILWAY ROLLING STOCK.

(Report No. 24—Parts I.–VI.—1921, 1925, 1928.) (*Abstracts.**)

Locomotive Crank Axles—B.S.S. 1—1928 (of Report No. 24, Part I.).

The crank axles shall be manufactured from the highest quality of steel made from the best selected material by the acid open-hearth process, and shall not show on analysis more than ·04 per cent. of sulphur or of phosphorus.

The metal shall show a tensile breaking strength of not less than 28 tons nor more than 33 tons (62,720 to 73,920 lbs.) per sq. in. (44·10 to 51·97 kg. per mm.^{2}), with an elongation of not less than 25 per cent. The yield point in either case shall be not less than 50 per cent. of the ultimate tensile strength.

A test-piece 9 ins. (228·60 mm.) long, 1¼ ins. (31·75 mm.) square, with $\frac{1}{16}$ in. (1·59 mm.) radius at the edges, machined from the crank axles, shall withstand being bent cold by direct pressure from a tool 2 ins. (50·80 mm.) wide, the end having a radius of 1 in. (25·40 mm.), until the sides of the test-piece are parallel. The ends of the test-piece are then to be brought together, without fracture, by direct pressure.

As an alternative to the cold bend test, the test-piece may be placed upon bearings having a clear span of 6 ins. (152·40 mm.), resting on a solid foundation, and shall withstand, without fracture, 20 blows from a weight of 1,120 lbs. (508 kg.), having a rounded end of 1½ in. (31·75 mm.) radius, falling 6 ins. (152·40 mm.). The test-piece shall be reversed after the first and every alternate blow. The fall shall then be increased to 12 ins. (304·80 mm.), and the test continued until fracture occurs.

All crank axles shall be oil hardened, and tempered.

Locomotive Straight Axles—B.S.S. 2—1928 (of Report No. 24, Part I.).

Quality of material same as for locomotive crank axles.

The axle shall be placed upon bearings resting on a block of metal of not less than 5 tons (11,200 lbs. = 5,080 kg.) weight supported on a rigid concrete or other solid foundation, and shall withstand, without fracture, five blows from a weight of 1 ton (2,240 lbs. = 1,016 kg.) falling 16 to 35 ft., according as the diameter is under 4 or over 6 ins., the distance apart of the bearings being from 3 to 5 ft., as shown by table.

The axle shall be turned after the first and third blow, and shall be broken after testing, both in the centre and at the journals.

The metal shall show a tensile breaking strength of not less than 35 nor more than 40 tons (78,400 to 89,600 lbs.) per sq. in. (55·12 to 63·00 kg. per mm.^{2}), with an elongation of not less than 25 per cent. with 35 tons and 20 per cent. with 40 tons; the sums of the intermediate tensile breaking strengths and corresponding elongations being not less than 60. The yield point shall be not less than 50 per cent. of the ultimate tensile strength.

A test-piece 9 ins. (228·60 mm.) long and 1½ ins. (31·75 mm.) sq., with $\frac{1}{16}$ in. (1·59 mm.) radius at the edges, shall withstand being bent cold by direct pressure from a tool 2 ins. (50·80 mm.) wide, the end having a radius of 1 in. (25·40 mm.), until the sides of the test-piece are parallel. The ends of the test-piece shall then be brought together, without fracture, by direct pressure.

N.B.—This test shall be taken only when a less number than 15 axles is ordered and the falling weight test has not been carried out.

All axles shall be either normalised or oil-treated.

Locomotive Straight Axles—B.S.S. 2A—1928 (of Report No. 24, Part I.).

The axles shall be manufactured from the highest quality of steel made from the best selected materials by the acid or basic open-hearth process. In the event of the basic open-hearth process being employed, the maximum percentage of sulphur or of phosphorus shall not exceed 0·05 per cent.

Tests and treatment as for B.S.S. 2.

* By permission of the British Engineering Standards Association.

Carriage and Wagon Axles—B.S.S. 3.—1928 (of Report No. 24, Part I.).

Same as for locomotive straight axles, B.S.S. 2, except that sulphur or phosphorus shall not exceed 0·05 per cent. and that range of falling weight test is from 15 to 30 ft. for axles under 3¾ ins. to over 6 ins.; distance between bearings, 3 to 4 ft.

Carriage and Wagon Axles—B.S.S. 3A (of Report No. 24, Part I.).

Same as for locomotive straight axles, B.S.S. 2A, except that range of falling weight test, etc., is as specified in B.S.S. 3.

Locomotive Tyres—B.S.S. 4—1928 (of Report No. 24, Part II.).

Quality of material same as for locomotive crank axles—B.S.S. 2.

The tyre shall be placed in a running position with the tread resting on a block of metal of not less than 5 tons weight supported on a rigid concrete or other solid foundation, and shall withstand, without fracture, blows from a falling weight of 1 ton. The weight shall be allowed to fall freely on to the tread from heights of 10 ft., 15 ft., 20 ft., and upwards, until the deflection of the tyre corresponds to that given in the following formula, in which d is the internal diameter of the tyre as rolled, in inches, and t is the thickness of the centre of the tread, as rolled, in inches, and s is the depth of the snip in inches.

Class C.—Tensile Breaking Strength 50 to 55 tons per square inch.	Class D.—Tensile Breaking Strength 56 to 62 tons per square inch	Class E.—Tensile Breaking Strength 63 to 69 tons per square inch.
$\frac{(d-3s)^2}{50\,t^2}$	$\frac{(d-3s)^2}{55\,t^2}$	$\frac{(d-3s)^2}{60\,t^2}$

A standard test-piece C taken from the position indicated shall show the tensile strengths and minimum elongations given in the table, the intermediate elongations being in proportion.

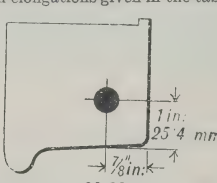


FIG. 19.

Class.	Tensile Strength. Tons per sq. in.	Minimum Elongation. Per cent.
C	50–55	13 to 11
D	56–62	10 to 8
E	63–69	10 to 8

When so specified in the inquiry, each tyre shall be allowed to drop freely in a running position, from the height specified below, upon a rail fastened to an iron block of not less than 2 tons in weight. Each tyre shall then be turned round through an angle of 90 degrees, and dropped a second time.

Internal Diameter of Tyre.	Up to 3½ ft.	Over 3½ ft. and up to 4½ ft.	Over 4½ ft. and up to 5½ ft.	Over 5½ ft. and up to 6½ ft.	Over 6½ ft.
	ft.	ft.	ft.	ft.	ft.
Height of fall . .	5	4	3½	3	2½

Carriage and Wagon Tyres—B.S.S. 5—1928 (of Report No. 24, Part II.).

The specification is the same as for locomotive tyres, with the addition of Class B tyres, tensile breaking strength 42 to 48 tons per sq. in. and minimum elongation 18 to 15 per cent.

Formula for deflection, Class B— $\frac{(d-3s)^2}{45\,t^2}$, and except that sulphur or phosphorus shall not exceed 0·05 per cent. in the case of steel for Class B and Class C tyres, and not more than 0·04 per cent. in the case of steel for Class D and Class E tyres. Height of fall for diameters over 3½ ft. is 4 ft.

Carriage and Wagon Tyres—B.S.S. 5A (of Report No. 24, Part I.).

Material as for axles B.S.S. 2A. Class E omitted and no drop test required.

LAMINATED SPRINGS—B.S.S. 6—1921 (of Report No. 24, Part III.).

The spring plates shall be manufactured from the highest quality of steel made from the best selected material by the acid open-hearth process. The steel must not show more than ·8 per cent. nor less than ·5 per cent. of carbon, nor more than ·05 per cent. of sulphur or of phosphorus. The steel for the top plates, which have to be welded at the eye, must not contain more than ·7 per cent. nor less than ·45 per cent. of carbon.

All finished springs shall be tested by being deflected by a quick-acting scrag before the buckle is put on, to an amount equal to $\frac{L^2}{900t}$, where L is the length of the top plate in inches measured along the arc as shown in fig. 20, and t is the thickness of the thickest plate in inches. They must then stand being deflected again three times in quick succession without showing any permanent set. The required deflections are shown in the table and must not be exceeded at any time during manufacture by more than 15 per cent.

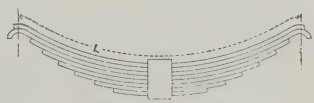


FIG. 20.

All plates shall be carefully hardened and tempered.

The buckles shall be made of good fibrous iron having a tensile breaking strength of from 20 to 24 tons per square inch, or of suitable steel having a tensile breaking strength of not less than 24 nor more than 30 tons per square inch. The elongation in each case shall be not less

L	Required Deflection under Scrag, in Inches.				
Ins.	$\frac{3}{8}$ in. plate.	$\frac{7}{16}$ in. plate	$\frac{1}{2}$ in. plate.	$\frac{9}{16}$ in. plate.	$\frac{5}{8}$ in. plate.
90	24	20⅝	18	16	14⅜
84	20⅞	17⅝	15⅝	14	12½
78	18	15½	13½	12	10⅞
72	15⅜	13⅛	11½	10¼	9¼
66	12⅞	11	9⅜	8⅛	7⅜
60	10⅝	9⅛	8	7⅛	6⅜
54	8⅜	7⅛	6½	5⅝	5⅛
48	6⅞	5⅞	5⅛	4½	4⅛
42	5⅛	4½	3⅞	3⅜	3⅛
36	3⅞	3⅛	2⅝	2⅜	2¼
30	2⅝	2¼	2	1¾	1⅝

than 20 per cent. (Standard Test Piece A), and the material must admit of bending, when cold, without showing crack or flaw, as follows:—

For iron, ¾-inch thick through an angle of 120 degrees.
" ⅝-inch " " 130 "
" ½-inch " " 140 "
For steel, all thicknesses, through an angle of 180 degrees.

In all cases the inside radius of the bend shall be not greater than 1½ times the thickness of the test-piece.

LAMINATED SPRINGS—B.S.S. 6A—1921 (of Report No. 24, Part III.).

Same as specification B.S.S. 6, except that spring plates shall be manufactured from steel made from selected material by the acid or basic open-hearth or the acid Bessemer process. In the event of the basic open-hearth process being employed, the maximum percentage of sulphur and phosphorus must not exceed ·05 per cent.

SPRING STEEL FOR LAMINATED SPRINGS—B.S.S. 6B—1921 (of Report No. 24, Part III.).

The spring plates and bars shall be manufactured from the highest quality of steel made from the best selected material by the acid open-hearth process, and must not show on analysis more than ·8 per cent. nor less than ·5 per cent of carbon, nor more than ·05 per cent. of sulphur or of phosphorus. The steel plates or bars to be used for top plates, which have to be welded at the eye, must not contain more than ·7 per cent. nor less than ·45 per cent. of carbon.

As a test, pieces of steel 30 ins. long shall be cambered to a radius of eighty times their thickness, as shown below, hardened and tempered, and after being pressed straight once, and the

camber carefully noted, must stand being pressed straight again six times in quick succession without any permanent set.

	Ins.	Ins.	Ins.	Ins.	Ins.
Thickness of plate . . .	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$
Required camber . . .	$3\frac{3}{4}$	$3\frac{1}{2}$	$2\frac{3}{4}$	$2\frac{1}{2}$	$2\frac{1}{4}$
Minimum camber after first blow and the next six consecutive blows	$3\frac{1}{2}$	$2\frac{7}{8}$	$2\frac{1}{2}$	$2\frac{1}{4}$	2

SPRING STEEL FOR LAMINATED SPRINGS—B.S.S. 6C—1921 (of Report No. 24, Part III.).

Material for spring plates and bars same as B.S.S. 6A. Tests same as B.S.S. 6B.

VOLUTE AND HELICAL SPRINGS—B.S.S. 7—1921 (of Report No. 24, Part III.).

The springs shall be manufactured from the highest quality of steel made from the best selected material by the acid open-hearth process. The steel must not show more than 1·3 per cent. nor less than ·8 per cent. of carbon, and not more than ·05 per cent. of sulphur or of phosphorus.

Each spring shall be tested by being driven home once by a quick-acting scrag, the height being carefully noted after release. The spring must then stand being driven home again five times in quick succession without showing any permanent set. Up to 5 per cent. of the springs shall also be tested under varying loads to determine the range and deflection per ton.

All springs shall be hardened in oil and tempered in a bath of molten lead.

VOLUTE AND HELICAL SPRINGS—B.S.S. 7A—1921 (of Report No. 24, Part III.).

The springs shall be manufactured from the highest quality of steel made from the best selected material by the acid open-hearth or the acid Bessemer process, and must not contain more than 1·3 per cent. nor less than ·8 per cent. of carbon.

Test and treatment same as B.S.S. 7.

STEEL FORGINGS FOR LOCOMOTIVES—B.S.S. 8—1930 (of Report No. 24, Part IV.).

The forgings shall be manufactured from steel made by the acid or basic open-hearth process. The tensile breaking strength and minimum elongation on Standard Test Piece C or D to be as follows :—

Class.	Description of Forging.	Process of Manufacture.	Maximum Manganese Content per cent.	Maximum Sulphur or Phosphorus Content per cent.	Tensile Breaking Strength in tons per sq. in.	Elongation Minimum per cent.
A	Special forgings which will be case-hardened	Acid or basic open-hearth or electric	0·70	0·05	24 to 28	29 to 25
B	Ordinary and boiler forgings	Do.	—	0·05	26 to 32	28 to 22
C	Special forgings without wearing surfaces, and bars	Acid open-hearth or electric	—	0.05	32 to 38	26 to 20
D	Special forgings with wearing surfaces, and bars	Do.	—	0·05	40 to 45	20 to 15

The sums of the tensile breaking strengths and corresponding elongations must be not less than 53 for Classes A, 54 for Class B, 58 for Class C, 60 for Class D. In the case of Classes C and D the yield point must be not less than 50 per cent. of the ultimate tensile strength.

A test-piece, 9 inches long and $1\frac{1}{2}$ inch square, or as near this size as the forging or forgings elected will permit, with $\frac{3}{16}$ inch radius at the edges, machined cold, must, without any reheating, withstand being bent cold by direct pressure from a tool $1\frac{1}{2}$ ins. wide in the case of Classes A, and B, $1\frac{3}{4}$ ins. wide in the case of Class C, and $2\frac{1}{2}$ ins. wide in the case of Class D,

the ends having a radius respectively of ¾ in., ⅞ in., and 1¼ ins., until the sides of the test-pieces are parallel. The ends of the test-pieces of all classes of forgings, except Class D, shall then be brought together without fracture by direct pressure.

STEEL BLOOMS FOR LOCOMOTIVE FORGINGS—B.S.S. 9—1930 (of Report No. 24, Part IV.).

Specification and tests for Classes A, B C, and D same as for B.S.S. 8.

The drawn-down section of bloom from which the test-pieces are cut may be normalised or otherwise heat treated before the test-pieces are cut therefrom.

STEEL CASTINGS—B.S.S. 10—1930 (of Report No. 24, Part IV.).

The castings shall be manufactured from steel produced by an approved process, and must not show on analysis more than ·06 per cent. of sulphur or of phosphorus.

All castings shall be thoroughly annealed by heating to a temperature not less than the normalising temperature and allowing to cool slowly from the maximum temperature in a practically uniform manner; or alternatively, normalised by heating in a similar manner and allowing to cool in air away from draughts.

The minimum tensile breaking strength and elongation on Standard Test-Piece C or D to be as follows :—

Description of Casting.	Minimum Tensile Breaking Strength per sq. in.	Minimum Elongation per cent.
Castings with wearing surfaces .	35 to 40 over 40	15 to 10 10
Other General Castings and Wheel Centres	26	20

In the case of castings other than those with wearing surfaces, a test-piece 9 ins. long, turned to 1 in. diameter, must withstand being bent cold through an angle of 120 degrees round a bar 2½ ins. diameter without fracture.

When wheel centres are tested to destruction, the test shall be made by dropping the wheel centre on to a block of metal of not less than 5 tons weight, supported upon a rigid concrete or other solid foundation. The wheel centre shall be raised and allowed to fall freely in a running position through distances of 5 ft. and 10 ft. on the end of a spoke (where such exists), and then turned through 90 degrees and again raised and allowed to fall freely in a running position through distances of 5 ft. and 10 ft. on the end of a spoke (where such exists). Any sign of failure under this test will render the wheel centres liable to rejection. The test shall then be continued, the fall being increased 5 ft. each time until fracture results or the wheel centre is doubled up.

COPPER PLATES FOR LOCOMOTIVE FIRE-BOXES—

B.S.S. 11—1925 (of Report No. 24, Part V.).

The plates shall contain not less than 99·20 per cent. of copper, and shall contain not less than 0·30 per cent. nor more than 0·50 per cent. of arsenic. Antimony shall not exceed 0·05 per cent. Bismuth shall not exceed 0·01 per cent.

Tensile breaking strength to be not less than 14 tons per square inch, with an elongation of not less than 35 per cent. on Standard Test-Piece A.

Pieces of the plate shall be tested both cold and at a red heat by the ends being bent through an angle of 180 degrees in opposite directions and doubled up close, without showing either crack or flaw on the outside of either bend.

ROLLED COPPER RODS FOR LOCOMOTIVE STAY BOLTS, RIVETS, ETC.—

B.S.S. 12—1925 (of Report No. 24, Part V.).

The rods shall contain not less than 99·20 per cent. of copper, and shall contain not less than 0·30 per cent. nor more than 0·50 per cent. of arsenic. Antimony shall not exceed 0·05 per cent. Bismuth shall not exceed 0·01 per cent.

Tensile strength to be not less than 14·50 tons per square inch, with an elongation of not less than 40 per cent. on Standard Test-Piece B. For rods up to 1¼ inch diameter, the test shall be made upon the unturned rod. For rods above 1¼ inch diameter, the rod may be turned down, when either Standard Test Pieces D or E shall be used, in which case the elongation shall be not less than 45 per cent.

The rod shall, both when cold and at a red heat, withstand, without machining, being doubled up close without cracking.

A piece of rod 1 inch long shall be placed on end and hammered or crushed down when cold to a thickness of ⅜ inch without showing either crack or flaw on the circumference of the resulting disc.

EXTRUDED COPPER RODS FOR LOCOMOTIVE STAY BOLTS, RIVETS, ETC.—

B.S.S. 12A—1925 (of Report No. 24, Part V.).

Tests the same as for B.S.S. 12.

Copper Tubes for Locomotive Boilers—

B.S.S. 13—1925 (of Report No. 24, Part V.).

The tubes shall contain not less than 99·20 per cent. of copper, and shall contain not less than 0·30 per cent. nor more than 0·50 per cent. of arsenic. Antimony shall not exceed 0·05 per cent. Bismuth shall not exceed 0·01 per cent.

Tensile tests shall be made on pieces of tube or strips cut from the tube. If the test is made on a piece of tube, the tensile strength shall be not less than 14·50 tons per square inch, with an elongation of not less than 50 per cent. on 2 inches. If the test is made on a strip, the tensile strength shall be not less than 14 tons per square inch, with an elongation of not less than 40 per cent. on a test piece having a gauge length of four times the square root of the area.

The test-piece shall stand bulging or drifting cold without showing either crack or flaw until diameter of bulged or drifted end measures not less than 25 per cent. more than the original diameter of the tube.

The test-piece shall stand flanging cold without showing either crack or flaw until the diameter of the flange measures not less than 40 per cent. more than the original diameter of the tube.

The test-piece shall stand the following test, both cold and at a red heat, without showing either crack or flaw. The test-piece shall be flattened down until the interior surfaces meet, and be doubled on itself, that is, bent through an angle of 180 degrees, the bend being at right angles to the direction of the length of the tube.

Each tube shall be tested by an internal hydraulic pressure of 750 lbs. per square inch, or by such internal pressure as may have been specified.

Brass Tubes for Locomotive Boilers—

B.S.S. 14—1925 (of Report No. 24, Part V.).

Brass for locomotive tubes may be of either 70/30 alloy or 2/1 alloy.

The tubes shall consist of an alloy of copper and zinc, and shall contain :—

70/30 Alloy.—Not less than 70 per cent. of metallic copper and not more than a total of 0·75 per cent. of materials other than copper and zinc.

2/1 Alloy.—Not less than 66·70 per cent. of metallic copper and not more than a total of 0·75 per cent. of materials other than copper and zinc.

The tubes shall be carefully annealed at both ends.

The test-piece shall stand bulging or drifting cold without showing either crack or flaw until the diameter of the bulged or drifted end measures not less than 25 per cent. more than the original diameter of the tube.

The test-piece shall stand flanging cold without showing crack or flaw until the diameter of the flange measures not less than 25 per cent. more than the original diameter of the tube.

The test-piece when cold shall stand the flattening and doubling-over test applied to copper tubes, B.S.S. 13. Tubes to be tested by an internal hydraulic pressure of 750 lbs. per square inch or such internal pressure as may have been specified.

Weldless Steel Tubes for Locomotive Boilers.

(B.S.S. No. 53—1927.)

The tubes shall be cold-drawn and weldless, and shall be manufactured from steel of the best quality made by the open-hearth process. They shall not show on analysis more than 0·035 per cent. of sulphur or more than 0·030 per cent. of phosphorus.

The tubes or strips cut from the tubes shall, without further annealing, show a tensile strength of not less than 20 tons per sq. in., nor more than 26 tons per sq. in., with an elongation of not less than 28 per cent. in 8 ins. for tubes, and not less than 22 per cent. in 8 ins. for strips cut from the

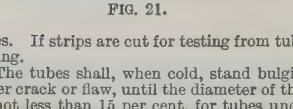


Fig. 21.

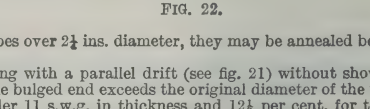


Fig. 22.

tubes. If strips are cut for testing from tubes over 2½ ins. diameter, they may be annealed before testing.

The tubes shall, when cold, stand bulging with a parallel drift (see fig. 21) without showing either crack or flaw, until the diameter of the bulged end exceeds the original diameter of the tube by not less than 15 per cent. for tubes under 11 s.w.g. in thickness and 12½ per cent. for tubes from 11 s.w.g. to 6 s.w.g. inclusive.

A piece of tube 2 ins. long shall be placed on end and shall, when cold, stand hammering down until it is reduced to 1⅓ in. high without showing either crack or flaw.

The tubes must, when cold, stand flattening without showing crack or flaw until the interior surfaces of the tubes meet, or, when the tubes are over 11 s.w.g. in thickness, until the interior surfaces are brought to a distance apart equal to the thickness of the wall of the tube (fig. 22).

Every tube shall be tested by an internal hydraulic pressure of at least 1,000 lbs. per sq. in.

Seamless Copper Pipes for Locomotives—

B.S.S. 15—1925 (of Report No. 24, Part V.).

The pipes shall contain not less than 99·20 per cent. of copper, and shall contain not less than 0·30 per cent. nor more than 0·50 per cent. of arsenic. Antimony shall not exceed 0·05 per cent. Bismuth shall not exceed 0·01 per cent.

Tensile tests shall be made on pieces of pipe or strips cut from the pipe. If the tensile test is made on a piece of pipe, the tensile strength shall be not less than 14·50 tons per square inch, with an elongation of not less than 50 per cent. on 2 inches.

If the tensile test is made on a strip cut from the pipe, the tensile strength shall be not less than 14 tons per square inch, with an elongation of not less than 40 per cent. on a test-piece having a gauge length of four times the square root of the area.

The test piece shall stand drifting cold without showing either crack or flaw until the diameter of the drifted end measures not less than 25 per cent. more than the original diameter of the pipe.

Flattening and doubling-over test the same as for copper tubes, B.S.S. 13.

Tubes to be tested by such internal hydraulic pressure as may have been specified.

Steel Plates, Angles, etc., and Rivets for Locomotive Boilers—

B.S.S. 16—1929 (of Report No. 24, Part VI.).

The plates, sections, and bars shall be manufactured from steel made from selected material by the acid or basic open-hearth processes, as may be specified. The steel must not show on analysis more than ·05 per cent. of sulphur or of phosphorus.

When plates are required for flanging purposes it shall be so stated at the time of inquiry.

Test-pieces to be cut crosswise from rolled material in case of plates, and lengthwise in case of sectional material. When material is annealed or otherwise treated before dispatch, the test-pieces shall be similarly and simultaneously treated with the material before testing. Any straightening of test-pieces shall be done cold.

Material.	Tensile Strength. Tons per sq. in.	Minimum Elongation. Per cent.		
		On Test-Piece A.	On Test-Piece B.*	On Test-Piece F.†
Plates	26–30	22	—	—
Sections and flats	26–32	20	—	—
Round and square bars (up to and including 1-in. diameter or thick)	26–32	—	20	—
Round and square bars (above 1-in. diameter or thick)	26–32	—	—	24
Rivet bars	24–28	—	27	32

Test-pieces for the cold-bend test shall be sheared crosswise from plates and lengthwise from sections and bars, and shall be not less than 1½ ins. wide, but for small material the whole section may be used. In all bend tests the sheared edges of the test-pieces may be removed by milling, planing, grinding, or other methods. The edges of the test-piece shall be draw filed to remove sharpness and roughness.

The test-pieces shall not be annealed or otherwise heat-treated unless the material from which it is cut is similarly annealed or heat treated, in which case the test pieces shall be similarly and simultaneously heat-treated before testing.

---

* Or in the case of square bars tested as rolled, a test-piece having a gauge length eight times the side.

† Or in the case of square bars tested as rolled, a test-piece having a gauge length four times the side.

The test-pieces shall withstand without fracture the bend test described in the following table. The bend tests may be made either by pressure or by blows.

Material.	Max. Internal Radius of Bend.	Description of Bend Test.
Plates . . .	The thickness of test-piece	Doubled over until sides are parallel
Sections and bars .	Do.	Do.
Rivet bars . .	—	Doubled over and flattened on itself

Tests for Manufactured Rivets.—The rivet shanks shall be bent cold, and hammered until the two parts of the shank touch, without fracture, on the outside of the bend. The rivet heads shall be flattened while hot without cracking at the edges until the diameter of the head is $2\frac{1}{2}$ times the diameter of the shank.

STEEL PLATES, ANGLES, ETC., AND RIVETS FOR LOCOMOTIVES—B.S.S. 17—1929 (of Report No. 24, Part VI.).

The plates, sections, and bars shall be manufactured from steel made from selected material by the acid or basic open-hearth process. The steel shall not show more than ·06 per cent. of sulphur or of phosphorus.

The tensile strength and ductility shall be determined from test-pieces cut lengthwise or crosswise from the rolled material in the case of plates, and lengthwise in the case of sections and bars. For all material, other than rivet bars, under $\frac{5}{16}$-in. thick, bend tests only are required.

TABLE OF TENSILE TESTS.

Material.	Dimensions.	Tensile Strength. Tons per sq. in.	Minimum Elongation. Per cent.		
			On Test-Piece A.	On Test-Piece B.*	On Test-Piece F.†
Plates—for other than cold-flanging or cold-pressing	$\frac{5}{16}$ in. thick and above	26–32	20	—	—
Plates—for cold-flanging or cold-pressing	Do.	24–28	24	—	—
Sections, flats, and bars—for other than welding or cold-working	$\frac{5}{16}$ in. thick and above	26–32	20	—	—
Do. for welding or cold-working	Do.	24–28	27	—	—
Round and square bars—for other than welding or cold-working	$\frac{5}{16}$ in. up to and including 1 in. in diam. or thickness	26–32	—	20	—
Do. for welding or cold-working	Do.	24–28	—	27	—
Do. for other than welding or cold-working	Above 1 in. in diam. or thickness	26–32	—	—	24
Do. for welding or cold-working	Do.	24–28	—	—	32
Rivet bars	—	24–28	—	27	32

* Or in the case of square bars tested as rolled, a test-piece having a gauge length of eight times the side.

† Or in the case of square bars tested as rolled, a test-piece having a gauge length of four times the side.

Cold-Bend Tests.—Test-pieces shall be sheared crosswise from plates and lengthwise from sections and bars, and shall be not less than 1½ ins. wide, but for small material the whole section may be used. In all bend-tests the sheared edges of the test-pieces may be removed by milling, planing, grinding, or other methods. The edges of the test-pieces shall be draw filed to remove sharpness and roughness.

The test-pieces shall not be annealed or otherwise heat-treated unless the material from which they are cut is similarly annealed or heat-treated, in which case the test-pieces shall be similarly and simultaneously heat treated before testing.

Material.	Dimensions.	Max. Internal Radius of Bend.	Description of Bend Test.
Plates—for other than cold-flanging or cold-pressing	All thicknesses	1½ times the thickness of test-piece	Doubled over until sides are parallel
Do. for cold-flanging or cold-pressing	$\frac{5}{16}$ in. thick and above	The thickness of test-piece	Do. Do.
Do. Do.	Below $\frac{5}{16}$ in. thick	—	Doubled over and flattened on itself
Sections, flats, and bars—for other than welding or cold-working	All thicknesses	1½ times the thickness of test-piece	Doubled over until sides are parallel
Do. for welding or cold-working	$\frac{5}{16}$ in. thick and above	The thickness of test-piece	Do. Do.
Do. Do.	Below $\frac{5}{16}$ in. thick	—	Doubled over and flattened on itself
Rivet Bars	All thicknesses	—	Do. Do.

For small sections these bend tests may be made from the flattened section.
Bend tests may be made by pressure or by blows.
Tests for Manufactured Rivets.—Same as for B.S.S. 16.

Steel Plates, Angles, etc., and Rivets for Locomotives—B.S.S. 17A—1929 (of Report No. 24, Part VI.).

Same as B.S.S. 17, but without analysis.

Steel Plates, Angles, etc., and Rivets for Carriages and Wagons—B.S.S. 18, B.S.S. 18A—1929 (of Report No. 24, Part VI.).

Specifications same as for locomotives, B.S.S. 17, B.S.S. 17A.

Wrought Iron.

(B.S.S. No. 51—1929.)

Specifications are given for wrought iron, Best-Yorkshire and Grades A, B, and C. The tensile tests, etc., vary with the sizes of the different sections rolled.

Other Materials used in Locomotive Construction.

Iron Castings.—The tensile test is the most trustworthy; 7 tons per sq. in. is the usual specification; 8 and 9 tons may be demanded in reason and expected. General foundry iron up to 11½ and 12½ tons per sq. in. and cylinder iron up to 13½ and 14 tons per sq. in. are not unknown, each having good fractures, and being good workable foundry metal. The usual commercial test is generally made transversely upon bars 3 ft. long, 2 ins. deep, and 1 in. thick, without being machined, 25 to 32 cwts. being the specified central breaking load. The test bars should be cast by means of an attachment to the casting, and the specified ultimate loads should not vary more than 1 per cent.

(For further details see Section XL., 'Metallurgy.')

For *Bronze, Brass, White Metals, and other Alloys*, see Section XL.

Use of Materials made to British Standard Specifications.

(Report No. 24—Parts I.–VI.)

No. 1.—Crank axles.

Nos. 2 *and* 2A.—Straight axles, locomotive and tender.

Nos. 3 *and* 3A.—Carriage and wagon axles.

No. 4.—Tyres for locomotive and tender. Class C is the most commonly specified.

Nos. 5 *and* 5A.—Carriage and wagon tyres. Class C for carriages and Class B for wagons are the most commonly specified.

Nos. 6 *and* 6A.—Laminated springs, purchased complete.

Nos. 6B *and* 6C.—Plates and bars for manufacturing laminated springs, spring wearing plates, flat springs.

Nos. 7 *and* 7A.—Manufactured volute and helical springs.

No. 8.—Steel forgings of Classes A to D (as below).

No. 9.—Steel blooms for making forgings. *Class A* is used for motion work and pins and other parts requiring to be case-hardened; spring hangers and links, brake work, draw bar, and safety links and pins, coupling screw and shackle, regulator rod, fire hole ring, foundation ring, and other parts requiring a weld. *Class B* for ordinary forgings, valve spindles, motion work not requiring case-hardening, reversing shaft and rod, reversing screw, buffer heads, regulator handle, roof slings, safety valve, and other seatings, wash-out doors and bridges, etc. *Class C*, connecting and coupling rods, crossheads, keys and cotters. *Class D*, piston rods, slide bars, crosshead cotters.

No. 10.—Steel castings. Wheel centres form one class; castings with wearing surfaces another, and this includes axle boxes and guides, bogie centres, bogie slide, brake shaft brackets, spring brackets, buffer plungers and guides. General castings without wearing surfaces include motion plate or slide bar brackets, smoke-box saddle, etc.

No. 11.—Copper fire-box plates.

Nos. 12 *and* 12A.—Copper stays and studs.

No. 13.—Copper boiler tubes.

No. 14.—Brass boiler tubes.

No. 15.—Copper pipes for feed water and steam.

No. 16.—Boiler barrel and fire-box shell plates, front tube plate, rivets, stays, nuts, angles, rods, etc., for boiler work. Expansion brackets, roof stays, palm stays.

Nos. 17 *and* 17A.—Locomotive and tender frame plates and cross stays, angles, bolts, rivets, plating, ash pan and doors, fire door, smoke-box and door, cab, tender tank plates, etc.

Nos. 18 *and* 18A.—Steel plates, angles, and other sectional material for carriage and wagon underframes.

Report No. 53 (1927).—Weldless steel tubes for boilers.

Contract Specification for Locomotive Construction.

The *Contract Specification* should provide for delivery and payment after running a guaranteed mileage (about 3,000 to 5,000). That all parts shall be constructed exactly to the drawings supplied, and no deviation made whatever without consulting those responsible for letting the contract, and that lack of information shall be no excuse for discrepancies in material or construction. The price quoted must be inclusive, and no responsibility shall devolve upon the purchaser for royalties and patent rights. The quotation should also include drawings (2 complete sets), photographs (12), lists of details, weights, etc., to be furnished with the (3rd or 5th) engine.

The specification should also stipulate that the workmanship should be accurate and well finished.

Liquid Fuel.

On the former Great Eastern Railway, express engines were fired with coal only on a consumption of about 50 lbs. per mile, or with coal and liquid fuel combined, on a consumption of 10 lbs. of coal and 30 lbs. oil fuel, including lighting up in each case.

The suitable oil fuels consist of petroleum residue, or Astatki, having a specific gravity of ·905, flash point 214° F.; distillates from coal gas tar with a specific gravity of 1·11, flash point 196° F.; coal gas tar with a specific gravity of 1·23, flash point 200° F.; or oil gas tar (refined) with a specific gravity of 1·05, flash point ·30° F.

The theoretical efficiency of oil fuel can be more nearly approached in practical working than that of coal, and with petroleum residue (the best oil fuel), 66·6 per cent. of its heat can be transferred into steam, whereas with steam coal 55·5 per cent. is about the average.

Engine No. 1862, a four-coupled bogie express, with a train weighing about 250 tons, exclusive of engine, ran from Liverpool Street to Cambridge on an expenditure of 120 gallons of oil fuel —oil gas tar—and a layer of coal and broken firebrick on the firebars. Steam was easily kept up during the trip, and no smoke was emitted from the chimney, thus showing that complete combustion of the fuel was effected (Holden's system).

Liquid fuel burning is seldom resorted to on British railways except at times of coal shortage resulting from prolonged strikes at the collieries. The conversion to oil burning is readily made.

When necessary to do so, the Midland section of the L.M. and S.R. burns oil in conjunction with a certain amount of coal; the burner is inserted through the fire-hole door, no alteration being made to the fire-box except by bricking up the portion of the tube plate under the brick arch (fig. 23). The oil issues from the top portion of the burner in a thin stream or ribbon, and is atomised by coming in contact with a flat jet of steam which issues immediately under it from an

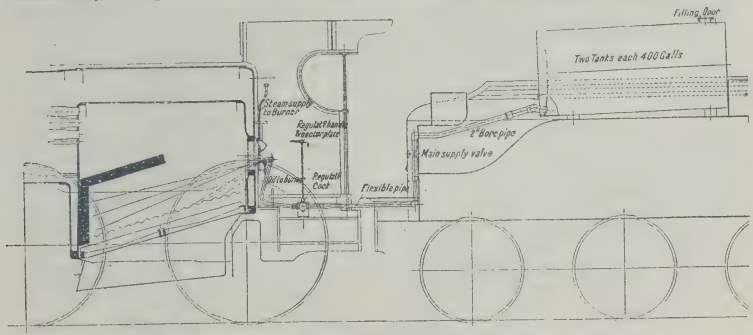


FIG. 23.—Oil-firing Arrangement, L.M. & S.R. (Midland Section).

orifice somewhat wider than that through which the oil flows. The details of the burner are shown in fig. 24.

The burner is inserted in the fire-hole, and is fed from two tanks placed on the tender, each of which holds from 400 to 500 gals. of oil. The flame is directed under the brick arch; the top portion of the fire-hole is closed by a special door which can be opened if it is necessary to use coal.

The fire-bars are covered with a layer of broken fire-brick, and if it is at any time necessary to use a little coal, it is put at the back of the fire-box. The fire can be started with wood, and as soon as the steam pressure reaches about 40 lbs. the burner can be used. In many cases no

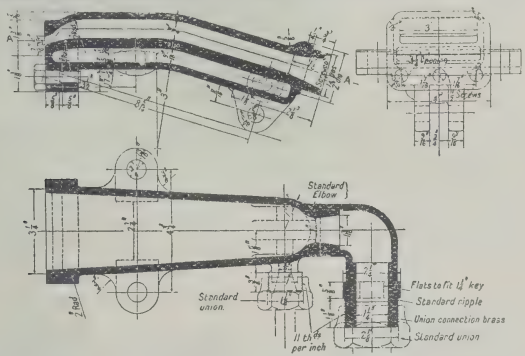


FIG. 24.—Oil Fuel Burner, L.M. & S.R. (Midland Section).

coal whatever is burnt for long periods, the engine steaming freely and the fire requiring very little attention. The best steaming results are, of course, obtained when there is a slight amount of smoke given off at the chimney. The locomotives fitted with this arrangement are able to deal easily with their accustomed loads. The advantage of the system is the ease with which it can be fitted up, and the simplicity in handling.

The company has one engine in which a similar burner is fitted in front of the ashpan, the fire-box being suitably bricked in on the sides and ends.

Oil is extensively burned in the South-Western States of America, Mexico, and the Argentine Republic, as coal has to be imported and oil is nearer at hand or is more cheaply transported.

See also Section XXXIV., 'FUELS.'

Mechanical Stokers.

Mechanical stokers are used by the Canadian National and the South African Railways. The former adopted them not for reasons of economy but because the locomotives had become too large for their full capacity to be maintained by hand firing. The consumption per unit of power developed is approximately the same with either hand or stoker firing. The same grade of fuel is used with mechanical stokers as is fired by hand. The proper use of these stokers to obtain best results calls for as much skill on the part of the fireman as hand firing, but the labour is reduced.

Mechanical stokers can be justified only on locomotives too large to be fired to full capacity by hand. The minimum economical size of engine for them are those of a rated tractive effort of 50,000 lbs., and then only when operating on long heavy grades. Engines having a rated tractive effort of 55,000 lbs. or over justify the use of mechanical stokers on any territory in freight service, as also engines of 50,000 lbs. tractive effort in passenger service.

Mr. Emerson states that the mechanical stoker for locomotives is a development of the last twenty years, in response to the necessity of securing greater capacity from single power units. Mechanical stoking permits more tonnage to be handled at a higher sustained speed, and gives greater average dependability than would be practicable with hand stoking. Its development has rendered it possible to design locomotives capable of burning coal continuously at a rate much in excess of the capacity of any fireman to supply it by hand. It is generally considered that a tractive capacity of 50,000 lbs., or a coal consumption of 4,000 lbs. per hour, will justify the use of a mechanical stoker. The present commercial production of mechanical stokers in the United States is confined to the overfeed or 'scatter' type, made by four companies. According to information supplied by those companies, 7,856 mechanical stokers were in service or on order on November 1, 1923, but their use has been considerably extended since then.

Pulverised Fuel.

Finely powdered coal can be burned in locomotive fire-boxes in a similar manner to oil fuel, by injection with air. The powder must, however, be perfectly dry for successful burning, but as it is slightly hygroscopic it absorbs moisture from the air and must therefore either be freshly made or kept perfectly dry in air-tight containers. The difficulty rather lies in the storage and distribution of the fuel than in the combustion.

Pulverised Fuel Locomotives.

An engine fitted for pulverised fuel firing by the A.E.G. system is shown in fig. 25.

The chief difference between this locomotive and that for ordinary firing is in the fitting of special burners and the alteration to the ashpan, which is lined with brick reaching up to a little above the foundation ring. Otherwise, the firebox is entirely free from brickwork (with the exception of the normal brick arch) which would keep the heat from this valuable heating surface.

The burners, consisting of long box-shaped tubes, are placed beneath the firebox, reaching from back to front on each side, and the fuel and air mixture is led to them from the tender. The fuel then burns in a number of short flames which issue laterally from the burner and meet those of the opposite burner in the centre of the firebox.

The fuel mixture is divided into a large number of narrow jets by means of deflecting strips of copper, suitably machined, in the burner boxes. The usual long pulverised fuel flame is thus replaced by a large number of short jets; each burner has a special long water-cooled projection, top and bottom, which are joined together at each end.

In addition to the main burners, there is an auxiliary burner at the back of the ashpan.

The pulverised fuel is kept in a cylindrical container on the tender, in the bottom of which two main and one smaller auxiliary fuel feeding screws are placed, one on each side. All these screws are driven by a small engine by means of a pinion engaging with a lay shaft, which in turn engages by means of gear wheels with the main screw driving shaft. These gear wheels can be engaged with the screw shaft by means of friction clutches. Each main feeding screw can be operated independently.

The fuel supply is regulated by altering the speed of the engine. The auxiliary feeding screw can also be operated independently.

The primary air, which is approximately 40 per cent. of the air required for combustion, is supplied by a turbo fan, and is forced through down pipes into the two mixing chambers placed at the delivery end of the feeding screws. The fuel and air are here mixed together intimately and then led to the burners, through pipes with flexible ball and socket connections between engine and tender.

The secondary air for complete combustion is obtained as usual by means of the suction caused by the blast-pipe exhaust. This air enters the ashpan through the front damper door, passes up an air passage formed between the front wall of the brickwork and the ashpan, leading up to an opening between the wall and the brick arch, and enters the firebox immediately under the brick arch.

The auxiliary screw also feeds the fuel into a mixing chamber.

The air for the auxiliary burner is provided by a small fan geared to the flywheel of the steam engine. The fuel and air mixture is driven through a hose-pipe connection to the auxiliary burner, which is placed beneath the foundation ring at the back end of the ashpan. The auxiliary burner is used for heating up, or when standing in a station, running down grades, etc.

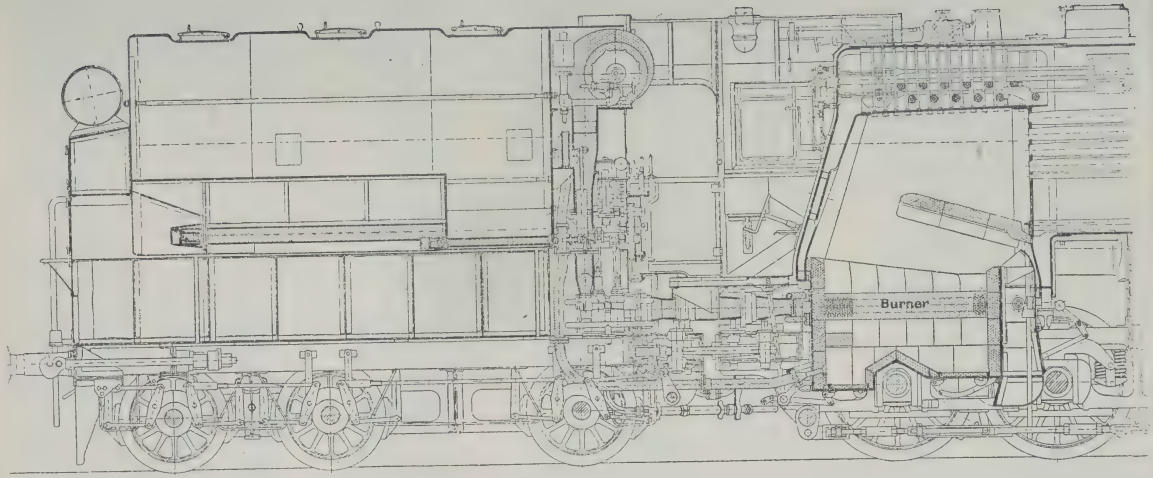


FIG. 25.—Arrangement of 'A.E.G.' Pulverised Coal Fired Locomotive showing lateral louvred Burners in Combustion Chamber or Fire-box, Main Screw Feeders, and Pilot Feeder and Burner.

The steam also drives a pump for the cooling water, which circulates through both burners and thus passes to a feed well, where it is used as feed water.

With the A.E.G. pulverised fuel system, fuel of low quality can be easily used. The efficiency of the boiler is increased, due to a much smaller quantity of air being required for complete combustion, thereby causing a considerable saving in fuel. The evaporative capacity of the boiler is not dependent on the strength or skill of the fireman.

A special advantage is that the steam requirements of the locomotive can immediately be suited, and that, further, the combustion is smokeless and sparkless.

The system is remarkable for its low fuel consumption and its high evaporative capacity over a long period. A fuel consumption of 1·24 kg./per h.p. hour with a fuel of 5,400 W.E/Kg. calorific value has been attained.

Another pulverised fuel system in use on German railways is that developed on behalf of a number of leading German firms by Messrs. Henschel and Sohn. The main points of difference with the A.E.G. system lie in the position and construction of the burners. These are fixed below the back plate of the fire-box and discharge the fuel towards the front. The end of each burner has a large square opening covered by a plate perforated with a considerable number of holes which form individual jets. No water-cooling of the burners is necessary, and the whole of the air of combustion is supplied through the burners.

See also Section XXXIV.

Coal Consumption.

The average rate of combustion in British express engines on heavy service is about 100 lbs. per sq. ft. of grate area per hour, and about 50 lbs. of coal per mile; the maximum rate may reach 160 lbs. per sq. ft. of grate area per hour.

According to the Railway Returns for 1925, the lbs. of coal consumed per engine-mile for the four British railway groups were: for passenger service, 40·00 to 53·11 lbs., for freight service, 54·82 to 70·06 lbs.

The evaporative power of locomotive boilers amounts to from 7 to 10 lbs. of water per pound of coal consumed. Longridge states that no rule can be fixed as to the best relative proportions of heating surface between grate area, fire-box, and tubes. It is an axiom that the capacity of a locomotive depends entirely upon its fire-box.

The consumption of fuel in a locomotive engine at rest, as when standing in a station or at a signal, is about one-fifth of that consumed when the engine is in motion. When standing at a shed with fire banked the consumption is very much less.

In a modern engine about 3½ cwt. of coal is actually consumed in raising steam in a boiler from cold water, and about 2½ cwt. from hot water. Some 10 cwt. is required for making up the fire before working a heavy train.

Water Consumption.

With a piston speed of 900 ft. per minute and a boiler pressure of 175 lbs. per sq. in., locomotives using saturated steam require about 24 lbs. per i.h.p. hour. The evaporation is approximately 8 lbs. of water per pound of coal, or 9½ lbs. from and at 212°F. The coal consumption is therefore 3 lbs. per i.h.p. hour. Locomotives using superheated steam require 25 per cent. less, or about 18 lbs. of steam per i.h.p. hour. The coal per i.h.p. is 12½ per cent. less, or roughly 2½ lbs. per i.h.p. hour, the evaporation being 7 lbs. of water per pound of coal.

Lubricating Oil Consumption.

The oil consumption of different railways varies considerably, but under favourable conditions 1 lb. of cylinder oil and 4 lbs. of engine oil are required per 100 miles. Superheater engines use rather more cylinder oil than non-superheater, and the quality of the oil is more expensive.

The oil consumption of the 4-6-0 Great Western engines averages 5 lbs. per 100 miles.

According to the Railway Returns for 1927, the consumption of lubricating oil per running-mile for the four British railway groups was: for passenger service, 6·40 to 7·26 pints; for freight service, 6·77 to 8·20 pints; for shunting service, 4·60 to 6·29 pints; for all services, 6·54 to 7·19 pints.

Sight feed lubricators on the hydrostatic principle, delivering oil to the steam pipes, are used on both superheater and non-superheater engines. It is more usual on superheater engines to employ a mechanical force feed lubricator driven by an attachment to some part of the engine motion. In this case oil is delivered direct to the piston valve heads and the middle of each cylinder bore.

The practice of feeding oil to the crowns of axle boxes by mechanical lubricators is also extending, more particularly in the case of heavy eight wheels coupled mineral engines.

For LUBRICATION AND LUBRICATORS, see p. 977.

Tenders.

The water capacity of tenders on British railways varies considerably. On the L.M. & S. Railway many tenders are under 3,000 galls., and the Great Western Railway uses 3,500 galls. for the largest engines, but these railways possess numerous water troughs, so that light six-wheeled tenders suffice. Large bogie tenders of some 5,000 galls. capacity are in use on the Southern Railway and on the Caledonian section of the L.M. & S. Railway.

The coal capacity does not vary so much, and lies between five and seven tons, depending on the size of the engine.

The connection between engine and tender consists of a simple eye-ended bar with one or both holes elongated to allow for the buffing which is taken up at the centre on a massive helical spring through which the bar passes. Safety links or loops are provided on either side, but only come into operation in the event of the bar breaking.

Several railways prefer side buffers as they help to damp out oscillations of the engine by the friction of the rubbing surface. The central arrangement is better for sharp curves.

In order that long non-stop runs may be made on the L. & N.E.R., some tenders have gangways through them which communicate with the train, and so enable a change of enginemen to be made on the journey.

Water Troughs.

Effective length is from 440 to 560 yards. Theoretically a quantity of about 1,000 gallons can be picked up, the mouth of the scoop being at rail level and $1\frac{1}{2}$ ins. below water level. In practice considerably more is obtained owing to the heaping up of the water in front of scoop, the maximum being obtained at 45 miles per hour with some 1,700 galls. At higher speeds there is a considerable loss by spray. The lowest speed at which water can be picked up is 15 miles per hour.

When running at 60 miles an hour over a water trough $\frac{1}{4}$ mile long, a locomotive tender will pick up about 1,000 gallons or 10,000 lbs. of water.

Examination of Locomotives at Running Sheds.

A failure on the road is a serious matter, and the best way of reducing breakdowns to a minimum is by periodical examinations. The following is a list of the usual examinations of the L. & N.E.R. (G.N. Section):—

Slide valves:—	
Piston valves	2 months
Richardson's	2 „
'D' valves, $\frac{7}{8}$ in. thickness	4 „
'D' „ $\frac{3}{4}$ and $1\frac{1}{16}$ in. „	3 „
'D' „ $\frac{5}{8}$ and $\frac{9}{16}$ ins. „	2 „
'D' „ $\frac{1}{2}$	1 month
Pistons	3 months
Tender tanks	1 month
Drip valves	1 „
Safety valves	6 weeks
Ball valves in water gauge frames (dependent upon quality of water in district)	2 months
Lead plugs	1 month
Large ends	3 months
Small ends	6 months
Vacuum regulating valves	1 month
Water scoops	1 „
Fire-boxes	1 week
Axle-box pads	3 months
Injectors	1 month
Main steam pipes on superheated engines	1 „
Tender wheels gauged	1 „
Leading wheels of 2–4–0 type gauged	1 „
Engine and tender tyres gauged	1 „
Mechanical lubricators to slide valves and pistons	2 months
Mechanical lubricators to axle-boxes	1 month

Cracked frames, flaws in connecting and coupling rods, or any part of motion, loose tyres, etc., should also be watched for. (*G. A. Musgrave*, '*Proceedings*, *Inst. Loco. Engrs.*')

Engine and Tender Tools and Equipment.

CARRIED IN TOOL BOXES.

Flat chisel.
Pin punches (large and small).
Chipping hammer.
Wrench for union nuts.
Complete set of spanners.
Adjustable wrench.
Tommy bar.
Large can for engine oil (6 to 8 quarts).
Large can for cylinder oil (6 to 8 quarts).
Small can for paraffin (about 1 quart).
Spring valve oil feeder, long spout.
Spring valve oil feeder, short spout.
Flare lamp.
Case of fog signals.
Red flag.
Hand sweeping brush.
Packing drawer } (if required).
Tallow pot }
Spare glasses for water gauge.
Spare glasses for sight feed lubricator (if fitted).
Spare packing rings for glasses.
Copper wire for trimmings.
Worsted for trimmings.
Spare cotter pins.
Spare nuts.

CARRIED ON TENDER.

Fire rake
Pricker
Dart
Tube scraper
Clinker shovel
} (length to suit firebox).
Slice (to suit depth of firebox at back).
Coal hammer.
Coal pick.
Pinch bar.
Bucket.
Fireman's shovel.
Screw-lifting and traversing jack (if specified).

OTHER EQUIPMENT.

Full set of head and tail lamps.
Full set of head boards or discs (when used).
Water gauge lamp.
Sight feed lubricator lamp (if required).

SPECIAL EQUIPMENT AT SHED FOR CLASS OF ENGINE.

Spanner for brake screw adjusting nuts.
Box spanner for fusible plugs.
Ashpan rake.

Brakes.

In addition to hand brakes the majority of engines and tenders are fitted with power brakes, usually the standard Westinghouse or Automatic Vacuum apparatus (page 640). Some of the large railways manufacture their own vacuum brake fittings. Goods engines not fitted with a continuous brake have a simple steam brake. Many engines are fitted with a steam brake working automatically with the vacuum by means of a valve of simple construction which can either be controlled by the vacuum in the train pipe or by hand. This is found to be a convenience, and the brake cylinders being of small diameter occupy but little space under the engine. The arrangement is extensively used and gives very satisfactory results.

Instead of a small ejector for maintaining the vacuum in the train pipe, several railways employ a vacuum pump about 5 ins. bore directly driven from one of the crossheads of the engine. As the pump is capable of creating a high vacuum, a relief valve is provided to limit the vacuity. On the Great Western Railway this is set for 25 ins., resulting in a running vacuum of 26 ins. of mercury. Other railways set the valve for the standard 20-inch vacuum, which results in 22 inches being maintained when running. Less trouble from pumps running hot is experienced with the higher vacuum, and the size of brake cylinders or leverage is reduced a little on account of the greater available atmospheric pressure, but no practical difficulties arise in the interchange of rolling stock with other railways.

Under average conditions as to load, leakage, and number of stops, the steam required by a vacuum ejector in maintaining a 20-in. vacuum results in the consumption of 42½ lbs. of coal per hour. The power absorbed by a vacuum pump coupled to the cross-head of the engine is equivalent to a consumption of 15 lbs. of coal per hour with a 20-in. vacuum, and 17½ lbs. per hour with a 25-in. vacuum.

(*J. N. Gresham, 'Journal of Inst. of Loco. Engineers.'*)

The percentage of brake power—that is, pressure on brake blocks in relation to load on wheels—should not exceed 75 per cent. for coupled wheels and 100 per cent. of weight when empty for tenders. If bogie wheels of engine are braked the percentage should not exceed 50 in order to minimise any chance of skidding.

To avoid frequent taking up of blocks brake leverage should not be too great. With steam the air brakes leverage is about 7 to 1; vacuum brakes are sometimes as much as 14 to 1 on account of difficulty of finding room for large cylinders. With this ratio a 7-inch piston travel allows ¼ inch clearance for blocks and ¼ inch wear.

Steam Rail Motor Cars.

A large number of rail motor cars were built between the years 1904 and 1909. The Great Western Railway still have many of these cars, some of which are 70 feet long over headstocks. The cylinders are 12 ins. by 16 ins., the wheels 4 ft. diameter, water capacity of tanks 450 galls. The boiler is vertical, 4 ft. 6 ins. diameter with cone top, and has 458 tubes 1⅛ in. diameter, total heating surface 659 sq. ft. and grate area 11½ sq. ft. The pressure is 160 lbs. per sq. in. and the tractive effort 6,912 lbs. The 70 ft. cars seat 61 passengers and the weight on motor bogie is 26·9 tons, and on the trailing bogie 16·25 tons. Walschaert valve gear is fitted.

The practice on the L.M.S.R. and S.R. is to use up obsolete locomotives and carriages by suitably converting them for motor services. A small tank engine is attached to one or more trailer cars to form a unit which can be driven from either end. When pushing the car, the locomotive is controlled by the driver from the opposite end of the train by means of levers and rods with universal joints connected to the regulator. A brake valve and whistle cord are provided and an electric bell signal enables the driver to communicate with the fireman on the footplate. In some cases control of the regulator is by means of compressed air or vacuum.

A recent development is the 'Sentinel-Cammel' design of car. This is of very light construction, and approximates more to tramway practice. The 'Sentinel' type of steam road wagon boiler and engine supplies the motive power. In one instance the coal consumption (inclusive of lighting up) is given as 6·27 lbs. of Welsh coal per mile, the water consumption 3·89 gallons per mile, and the total cost (inclusive of fuel, wages, repairs, depreciation, etc.) as 6·16 pence per mile. A similar type of car is the 'Clayton.' Both types are in use on branch lines of the L. & N.E.R., and numbers are at work on Colonial railways.

The 'Sentinel' and 'Clayton' power units have also been applied to small locomotives for light work on branch lines or for shunting purposes.

ROLLING STOCK.

(Revised by H. Holcroft, M.I.Loco.E.)

Recent Developments in Carriage and Wagon Stock

No very radical departures have been made in carriage and wagon stock on British railways, but in the United States steel is being use to a large extent for passenger and freight car bodies. On British railways galvanised steel plates of about $\frac{1}{16}$ in. thickness are now being used for panels of carriages, but anything approaching an all-steel construction for bodies is very much heavier than with wood. On account of the doors with the compartment system, it is not possible to make use of the side of the body as a girder, as in the open American cars with end doors. The L.M.S. Railway have put into service a number of steel coaches of the open type with central gangway and end doors. The under-frame, roof, and sides form a box girder, and no truss rods are needed.

The L. & N.E.R. and Southern Railway use the American type of automatic coupling and vestibule for some of their main-line trains. In order that these vehicles can be attached to ordinary stock, the coupler is hinged and can be dropped down, exposing a hook for screw coupling. Side buffers are provided, and these have a loose sleeve which can be removed to allow the buffers to be pushed back when the automatic coupling and vestibule with central buffers are in use.

Pullman cars of the American type are extensively used on the Southern Railway, and to a lesser extent on the L. & N.E.R., L.M. & S., and Metropolitan Railways. Mr. Gresley's twin coaches, as used on the L. & N.E.R., consist of a pair of bodies hinged together The outer ends rest on bogies in the usual way, but there is only one bogie for the inner ends, the bogie pin being at the hinge between the coaches. A flexible gangway connects the two bodies. This system has been extended to complete trains, so that a set train of five coaches has only six bogies.

The G.W.R. has also built some trains of articulated coaches.

The high-capacity bogie wagon makes but slow progress on British railways, owing to the lack of terminal facilities. The tendency is to build 4-wheel wagons of 12, 15, or 20 tons capacity. The fitting of a continuous brake to goods trains is limited to special trains for perishables, ballast, etc. Its more general application is retarded by the presence of a large number of private owners' wagons and the high capital costs involved.

The container system of transporting goods has been introduced on British railways. Large boxes fitted with doors are filled at the factories, transported by road and then placed on railway wagons by a crane.

Another new type of wagon is that for carrying milk. This consists of a large cylindrical tank, glass lined.

General utility vans have been introduced by the Southern Railway. These are 4-wheeled vehicles, with a wheel base of 21 ft., and they have end as well as side doors. By means of special fittings they are adaptable for a number of purposes besides passengers' luggage, such as milk cans, motor-cars, road vehicles, fruit baskets, theatrical scenery, aeroplane parts, etc.

See also 'All-metal Cars for British Railways,' Vol. ccviii. (April 1921), *Proceedings Inst. C.E.*

CARRI-

Section of Railway.	Type of Vehicle.	Length of Body.	Width of Body.	Height from Rail to Top of Roof.	Corridor or Non-Corridor.
LONDON, MIDLAND & SCOTTISH RAILWAY.					
L. & N.W.	Brake Third	65 6	9 0	12 5½	Corr.
,,	Compo.	65 6	9 0	12 5½	Corr.
,,	Brake Compo.	65 6	9 0	12 5½	Corr.
Caledonian	Third	68 0	9 0	—	Non-Corr.
,,	First	65 0	9 0	—	Non-Corr.
,,	Third	65 0	9 0	—	Corr.
L. & Y.	First	56 0	9 0	—	Centre Corr.
,,	Third	56 0	9 0	—	Centre Corr.
,,	Third	56 0	9 0	—	Side Corr.
L.M.S.	Third	57 0	9 0	12 7½	Centre Corr.
LONDON & NORTH-EASTERN RAILWAY.					
G.N.	First Dining	65 6	9 0	—	Corr.
,,	Compo. Dining	65 6	9 0	—	Corr.
,,	Sleeping	58 6	9 0	—	Corr.
,,	Third	58 6	8 6	—	Corr.
,,	Brake Compo.	58 6	8 6	—	Corr.
N.E.	Third	52 0	8 6	—	Non-Corr.
G.C.	Brake Compo.	56 0	8 6	—	Corr.
,,	Compo.	56 0	8 6	—	Corr.
,,	Third	60 0	8 10½	—	Corr.
,,	First	60 0	8 10½	—	Corr.
G.E.	First	54 0	9 0	—	Non-Corr.
,,	Third	54 0	9 0	—	Non-Corr.
,,	Brake Van	50 0	8 9	—	—
L.N.E.	Compo. Dining Triple Set	153 7 over vestibules	—	—	Corr.
GREAT WESTERN RAILWAY.					
G.W.	Brake Third	70 0	9 0	—	Corr.
,,	Third	70 8¼	9 0	—	Corr.
,,	Compo.	71 4½	9 0	—	Corr.
,,	Dining	71 4½	9 0	—	Corr.
SOUTHERN RAILWAY.					
Western	Brake Third	57 0	9 0	12 5	Corr.
,,	Third	57 0	9 0	12 5	Corr.
,,	Pantry Third	57 0	9 0	12 5	Corr.
,,	First	57 0	9 0	12 5	Corr.
Central	First	57 0	9 0	—	Non-Corr.
,,	Third	54 0	8 6	—	Non-Corr.
,,	Compo.	56 0	8 6	—	Non-Corr.
Eastern	First	62 0	8 0¾	12 2	Corr.
,,	Brake First	62 0	8 0¾	12 2	Corr.
,,	Second	62 0	8 0¾	12 2	Corr.
,,	Brake Second	62 0	8 0¾	12 2	Corr.

AGES.

Weight Empty.	Compartments. 1st Class.	Compartments. 3rd Class.	Seats. 1st Class.	Seats. 3rd Class.	Bogie.	Bogie Wheel Base.	Centre of Bogies.	Remarks.
T. C.						′ ″	′ ″	
39 10	—	—	—	—	6 wheel	12 6	43 0	West
42 0	—	—	—	—	,,	12 6	43 0	Coast
41 4	—	—	—	—	,,	12 6	43 0	Stock
—	—	11	—	132	,,	—	—	—
—	9	—	72	—	,,	—	—	—
—	—	9	—	72	,,	—	—	—
26 0	—	—	40	—	4 wheel	—	—	—
25 19	—	—	—	66	,,	—	—	—
26 3	—	8	—	64	,,	—	—	—
30 0	—	—	—	56	,,	9 0	40 6	All-steel
40 0	2	—	20	—	—	—	—	—
41 8	1	1	8	18	—	—	—	—
35 0	11	—	11	—	—	—	—	—
29 7	—	8	—	48	—	—	—	—
29 19	3	3	12	18	—	—	—	—
24 0	—	8	—	80	—	—	—	—
34 6	2	3	8	24	4 wheel	—	—	—
38 9	2	4	12	32	,,	—	—	—
36 3	—	8	—	64	,,	—	—	—
36 16	7	—	38	—	,,	—	—	—
29 4	8	—	80	—	6 wheel	—	—	Suburban
27 9	—	10	—	120	,,	—	—	—
22 8	—	—	—	—	4 wheel	—	—	—
81 8	—	—	36	42	4 wheel	8 6	47 0 42 1 47 0	Articulated: 3 Cars on 4 Bogies
—	—	4	—	32	4 wheel	9 0	53 6	—
—	—	10	—	80	,,	9 0	53 6	—
—	4	5	24	40	,,	9 0	53 6	—
—	—	—	18	32	,,	9 0	53 6	—
—	—	4	—	32	4 wheel	9 0	41 0	5-coach
—	—	8	—	64	,,	9 0	41 0	Pantry
—	—	7	—	56	,,	9 0	41 0	Sets
—	7	—	42	—	,,	9 0	41 0	6Lavatories
28 10	6	—	42	—	,,	—	—	2 ,,
24 0	—	8	—	78	,,	—	—	—
26 10	4	4	24	40	,,	—	—	—
31 14	6	—	24	—	,,	8 0	44 0	Continental Boat Train Stock
30 9	5	—	20	—	,,	8 0	44 0	
31 8	—	7 Second	—	42 Second	,,	8 0	44 0	
30 9	—	6 Second	—	36 Second	,,	8 0	44 0	

WAG-

Section of Railway.	Type of Wagon.	Carrying Capacity.	Length over Body.	Width over Body.
LONDON, MIDLAND & SCOTTISH RAILWAY.				
		Tons.	′ ″	′ ″
L. & N.W.	Open Goods	10	18 0	7 10
,,	,, ,,	20	21 0	7 9
,,	Covered ,,	10	18 0	8 0
Midland	Open ,,	10	16 0	7 10
,,	,, ,,	12	16 0	7 10
,,	,, ,,	20	21 6	7 11
,,	Covered ,,	10	17 6	7 8
L. & Y.	Open ,,	12	16 0	7 10
,,	,, ,,	20	21 6	8 0
,,	Covered ,,	10	20 0	8 0
,,	,, ,,	30	35 0	8 0
LONDON & NORTH EASTERN RAILWAY.				
G.N.	Open Goods	10	19 0	8 0½
,,	,, ,,	20	21 6	8 5
,,	Covered ,,	8	19 0	7 8
N.E.	Open ,,	10	15 0	8 0
,,	,, ,,	12	17 0	8 0
,,	,, ,,	20	20 0	8 0
,,	Covered ,,	10	17 0	8 0
,,	,, ,,	25	37 0	8 6
,,	Coal ,,	40	36 0	8 0
G.C.	Open ,,	10	19 0	8 0½
,,	,, ,,	20	21 6	7 9
,,	Covered ,,	10	19 0	7 8
G.E.	Open ,,	10	17 0	7 7
,,	,, ,,	20	21 6	8 2
,,	Covered ,,	10	19 0	7 9
,,	Guard's Brake	—	17 6	7 9½
,,	Cattle Wagon	10	19 3	7 11½
GREAT WESTERN RAILWAY.				
G.W.	Open Goods	10	16 0	8 0
,,	,, ,,	20	21 0	7 9
,,	Covered ,,	10	16 0	7 10
SOUTHERN RAILWAY.				
Western	Open Goods	10	16 0	7 11½
,,	,, ,,	12	18 0	7 11
,,	Covered ,,	10	18 0	8 0
Central	Open ,,	10	15 5	7 9
,,	,, ,,	12	16 6	7 11
,,	Covered ,,	8	18 4	7 8½
,,	Ballast	15	20 0	8 0
,,	Cattle Wagon	6	18 4	8 0
S.R.	Guard's Brake	—	13 10½	8 4 (over look-outs)

ONS.

Depth inside.	Tare Weight.	Wheel Base.	Centres of Journals.	Size of Journals.	Remarks.
′ ″	T. C. Q.	′ ″	′ ″	″ ″	
3 0	6 13 0	9 9	6 6	9×4	
4 7	8 0 2	12 0	6 6	10×5	
7 8⅜	7 10 1	9 9	6 6	9×4	
3 2	5 17 3	9 6	6 6	9×3¾	
4 1⅜	6 10 1	9 6	6 6	9×4¼	
5 1⅛	8 7 0	12 0	6 6	10×5	
7 9	7 12 3	10 0	6 6	9×3¾	
4 1¼	6 14 2	10 6	6 4	9×4¼	
5 11 5/16	9 16 0	12 0	6 6	10×5	
7 8	7 11 1	10 6	6 4	9×4¼	
—	14 17 1	—	—	—	Bogie Wagon
3 5	6 9 0	10 0	6 4½	8×3¾	
5 2⅝	8 9 0	12 0	6 6	10×5	
7 1	6 18 0	10 0	6 4½	8×3¾	
2 2½	5 10 0	8 6	6 5	8×4	
3 1½	6 7 0	9 6	6 5	9½×4¼	
6 10½	9 1 0	10 6	6 6	10×5	
7 9	7 2 0	9 6	6 5	8×4	
7 10	15 0 0	—	—	—	Bogie Wagon
6 0	16 5 0	—	—	—	Bogie Hopper Wagon
3 5	6 10 0	10 0	6 6	8×4	
4 10	7 17 0	12 0	6 6	10×5	
7 1	7 10 0	10 6	6 6	8×4	
4 0½	6 7 2	9 6	6 6	8×3¾	
4 7	9 1 1	12 0	6 7½	10×5	
7 0⅞	6 9 2	10 6	6 6	8×3¾	
—	20 0 0	—	—	—	
—	7 6 2	—	—	—	
3 3	6 0 0	9 0	6 6	8×3¾	
4 9¾	8 12 0	12 0	6 6	10×5	
7 7½	6 10 0	9 0	6 6	8×4	
2 8⅞	5 19 0	9 0	6 8	9×4	
4 8⅜	6 18 2	10 6	6 8	9×4	
7 7½	6 17 0	10 6	6 8	9×4	
2 10¾	5 18 1	9 3	6 3	8×4	
4 2⅜	6 17 1	9 0	6 6	9×4¼	
7 1	6 8 0	9 9	6 3	8×4	
—	7 0 0	—	—	—	
—	6 9 0	—	—	—	
—	26 0 0	16 0	—	—	

Axle Boxes.

These are usually of cast iron: for carriages oil boxes are universal, and a large number of new wagons are now being fitted with them instead of grease boxes. An oil reservoir in the bottom of the box conveys oil by capillary attraction, either by cotton-waste packing or by means of a worsted pad with tail trimmings supported on springs. The bearing or 'brass' is of gun-metal, or gun-metal lined with white metal. Some white metals have a tin base and contain about 60 per cent. tin and 25 per cent. of lead, with 10 per cent. antimony and a little copper. The lead base alloys have about 76 per cent. of lead, 15 per cent. of antimony, 6 per cent. tin, and a small percentage of copper.

The oil used is a mineral oil, and sometimes contains a trace of fatty oil.

A leather-faced dust shield prevents dirt or water creeping in at the back of the box.

Roller Bearings.

Experiments made on British railways seem to indicate that the friction of a plain bearing lubricated by an underpad is even less while running than a roller bearing, although the starting effort required with the latter is reduced, as also the oil consumption.

Owing to the higher capital cost there is, however, little or no inducement to adopt the roller bearing on British railways. On railways abroad, where the workmanship, maintenance, and materials are not so high-class, so that friction is greater, some advantage is gained by the adoption of roller bearings, and a small saving in fuel is found, sufficient with the other advantages realised to justify the adoption, this tendency being more marked in the Northern countries in Europe having cold winters. The bearings are also applied to trains de luxe in France and elsewhere on the Continent, such as the 'Blue Train' and the 'Golden Arrow.' Favourable conditions for the use of roller bearings are also found in North America, where some of the steel passenger coaches weigh as much as 70 to 80 tons each. Owing to the axle loading the journals are of large diameter, while the wheels are smaller than in England, so that the frictional resistance is high, and the risk of heating increased. The use of roller bearings reduces the friction, enables an easier and smoother start to be made, and obviates the risk of hot bearings, besides reducing oil consumption.

Wheels.

For carriages the wheel with wooden centre, once universal, has now been superseded to a large extent by a steel disc centre of slightly dished form, on which is shrunk a steel tyre having a lip on outside, and secured by retaining ring inside. Tyres are 5 ins. wide and are coned 1 in 20. In order to ensure smooth running, all carriage wheels are carefully balanced. Each pair of wheels is placed on a machine, the journals resting in bearings mounted on springs. When the wheels are revolved at high speed any irregularity in the running can be detected, and a correction made by fastening small plates to the inside of the wheel, near circumference, so that smooth running results.

Wood centres must be bonded by a copper strip between tyre and hub, so as to conduct electric current from rail to rail for operating signalling appliances. Tyre shrinkages are $\frac{3}{16}$ in. for wood centres, and about $\frac{5}{64}$ in. for steel centres.

Wagon wheels are made in a variety of ways. The bossed spoke wheel consists of spokes made from flat bar bent into triangular shape. These are placed together in a mould and a cast-iron boss is cast round them. The outer ends are then welded together to form a solid rim. Instead of casting a boss, it may be forged by placing two discs brought to welding heat on either side of the spokes, and forcing them together in a powerful press. Spoked wheels have also been made by forging spoke bars and building up a wheel. But this has been largely superseded by a cast-steel spoke wheel. A few wheels are being made with steel disc centres, and there are a few solid cast-iron wheels with chilled treads running on British railways. These are on wagons originally built for war service in France.

Journals of axles are sometimes case-hardened, but very good results can be obtained by the simple process of burnishing in the lathe. After the journal is machined, a tool having a small roller at the end is fixed in the slide rest and the journal is rolled all over with considerable pressure; this closes all the pores and leaves a hard, bright surface.

The maximum bearing pressure for journals varies from 500 to 700 lbs. per sq. in. of projected area.

A reduction in diameter by $\frac{1}{4}$ in. is allowed before axles are scrapped.

Springs.

Laminated bearing springs are usually $3\frac{1}{2}$ ins. wide for carriages, and 4 ins. for wagons, with plates $\frac{3}{8}$ to $\frac{1}{2}$ in. thick, approximately.

If L = distance in inches between centres of bearings measured along top plate; b = breadth of plates in inches; t = thickness of plates in sixteenths of an inch; n = number of plates; W = safe load in tons; D = deflection in sixteenths of an inch per ton of load,

$$n = \frac{15LW}{bt^3}; \quad W = \frac{bt^2n}{15L}; \quad D = \frac{2L^3}{bt^3n}.$$

If there are different thicknesses of plates in one spring, it is necessary to find the equivalent number of plates of uniform thickness from the square (or cubes in the case of deflection) of the thickness, thus :—

$$n = \frac{n_1 t_1^2 + n_2 t_2^2}{t_1^2}.$$

See also p. 613.

Buffer springs are largely made of rubber, and are ormed of rubber discs 2½ to 3 ins. thick placed between 6 or 8 metal discs.

For *Helical Springs* see p. 614.

Couplings.

Draw bars should be 1¾ ins. diameter ; they should be made from wrought iron, Grade A, or welding steel. Area of end of eye = area of two sides × 1·2.

Couplings should be made of wrought iron, Grade A, or welding steel. Standard diameter = 1½ in.

Screw coupiings are of wrought iron, or sometimes of steel (B.S.S. 9B).

Draw hooks should be of wrought iron, Grade A, or welding steel. If stamped out, the grain of the iron should be kept running with the hook, an important point sometimes overlooked in second-rate work.

FOR BRITISH STANDARD SPECIFICATIONS FOR MATERIALS USED IN THE CONSTRUCTION OF RAILWAY ROLLING STOCK, see pp. 611 to 620.

LIGHTING AND HEATING RAILWAY CARRIAGES.

STONE'S SYSTEM OF ELECTRIC LIGHTING FOR RAILWAY CARRIAGES.

In this system, fig. 26, the dynamo for producing the electric current is suspended from the under-frame of the carriage so that its pulley is at approximately the same level as the pulley on the axle, from which it is driven direct by means of a belt.

Speed of Pulley on Axle of Motor (representing Axle of Railway Carriage).	Equivalent Speed of Train in Miles per Hour (approximate).	Speed of Dynamo, Revolutions per Minute.	Equivalent Speed of Dynamo if no Slip were to take place.	Output of Dynamo. Amperes.
Revs. per Minute 100	12	500	500	0
150	18	750	750	6
175	21	870	875	16
195	23	900	975	18½
225	27	915	1,125	20
275	33	915	1,375	20
325	39	915	1,625	20
375	45	915	1,875	20
450	54	915	2,250	20
600	72	915	3,000	20

However the speed of the train may vary, after a certain predetermined limit has been reached, *the dynamo always runs at a uniform speed and consequently always produces practically the same amount of electricity.* This result is obtained by suspending the dynamo by one corner of its frame by means of an adjustable link, so that the driving belt draws the dynamo out of the diagonal position in which it would naturally hang, thus putting a definite tension on the belt just sufficient to absorb power equivalent to the amount of electricity required. When the pull on the belt (owing to increase in speed of the train) exceeds the weight on the belt, due to the one-sided suspension of the dynamo, the latter is automatically drawn towards the driving pulley on the axle, thus allowing the belt to slip while the armature continues to revolve at its normal speed. That the arrangement is an effectual one is shown by the above table.

The latest developments in the system include the introduction of the firm's 'Liliput' dynamo compactly built on modern lines and provided with ball bearings throughout, a special feature being that all the parts are very easily accessible for inspection or removal. The armature is of the drum wound type with former wound coils, the machine itself being shunt wound.

The centrifugally operated switchgear embodied in the older dynamos is omitted from the 'Liliput' machine, the switching being now effected by a separate automatic cut-in and cut-out switch operating on voltage ; and in the case of a double battery system, a 'Pegoud' battery change-over switch is included, so as to change over the position of the charging and regulating batteries every time the dynamo switches into circuit.

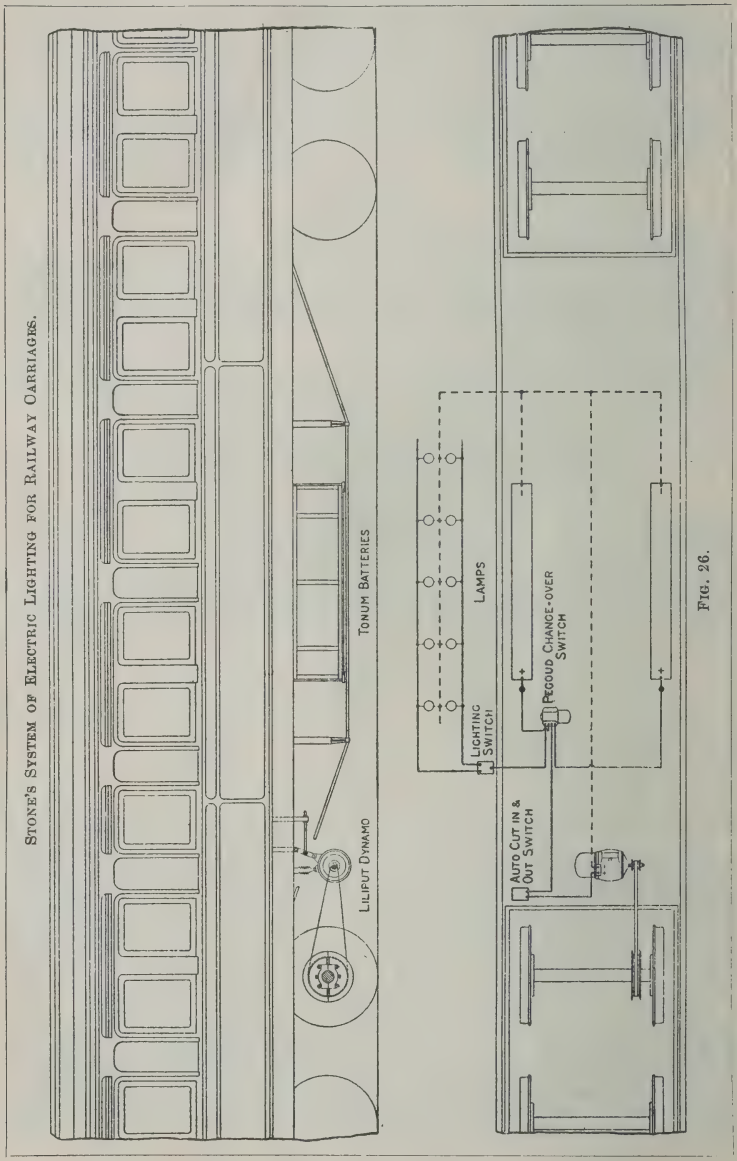

Stone's System of Electric Lighting for Railway Carriages.
Lilfut Dynamo
Tonum Batteries
Lighting Switch
Lamps
Pegoud Change-over Switch
Auto Cut in & Out Switch

Fig. 26.

The 'Liliput' dynamo (in which is embodied the pole-charging device) is made in two forms, either to run at a constant speed, as described above—the output being controlled by means of a slipping belt—or at a variable speed, that is, the speed of the dynamo following the speed of the train, the output remaining constant at all speeds, and as the lamps are switched 'on' or 'off' so the dynamo output increases or decreases according to the demand.

The accumulator being always in connection with the lamps causes the light to be absolutely steady, acting as a regulator to absorb any excess of current generated by the dynamo.

The working parts of the dynamo are enclosed in a movable but dust-proof casing.

A switch is provided to each carriage, so that the lights can be either fully on or entirely off, or where there are tunnels, and the light is required in the daytime, half the lights may be used. When the lamps are not alight, the current is being stored in the accumulator.

'Tonum' batteries are used with either special formation pasted or Plante positive plates. When a single battery is installed, individual cells have twice as many positive plates and are twice as heavy as the cells in the equivalent double battery; and as the latter, *i.e.*, the double battery system, gives the best regulation and is more reliable in working it is always recommended. One battery being directly connected to the lamps keeps the light absolutely steady while the other battery is being charged by excess current generated by the dynamo over and above that required for the lamps, which are supplied by the dynamo through the lamp resistance. On the train stopping, both batteries supply the lamps.

The 'Tonum' type of cell is designed to withstand the rough handling to which cells are unavoidably subjected in railway work, and embodies several valuable features.

The grids of the plates are of such design as to combine efficiency with mechanical strength. The sections are contained in a lead-lined wood box fitted with a lid which does not require any metal holding down bolts or fasteners, and the top of the box is specially formed and the lining so fitted as to prevent any acid from running down between the lining and the wood and gradually attacking the box.

The positive and negative plates are separately supported on the bottom block, eliminating leakage from plate to plate through lodgment of sediment on the surface of the supporting strips. Ample space is provided beneath the plates for any possible accumulation of sediment, and above the plates to allow of a generous supply of acid, so that the cells do not require frequent 'topping' to make good evaporation.

A float is fitted which indicates the level of the electrolyte in the cell, showing at a glance—without opening the lid of the cell—whether the cell requires water.

The electrical qualities of the cell are (1) retention of capacity throughout its useful life and (2) the maintenance of a high initial voltage for a considerable time after the commencement of the discharge.

The whole of the apparatus is of British manufacture and made in London, the system being largely adopted by the leading railway companies throughout the world.

Heating Railway Carriages by Steam.

Heating of trains is by steam from the locomotive boiler. The train piping consists of 1½ in. bore iron piping, and flexible rubber hose connections with universal couplings are provided between carriages. Automatic ball valves and thermostats are fitted for drainage of water. A pressure of 60 lbs. per square inch is the maximum, but 30 lbs. is quite an average pressure for ordinary working.

Several systems are in use, but they may be classed under two headings, high-pressure and atmospheric. In the high-pressure systems, long cylindrical heaters, 4 ins. to 5 ins. diameter, are placed under one seat of a compartment. These are in direct communication with the train pipe, and steam under pressure fills them. As condensation takes place the water is drained away through a thermostatic valve. There are several modifications of this system.

In the atmospheric systems the heaters have an open outlet at one end, and a jet of steam entering at the opposite end expands to atmospheric pressure and condenses on the walls of the heater, the water draining away at the outlet. In some systems regulation of steam is by means of a thermostat, which automatically controls the steam. In others a choke plug in the branch pipe to the heater restricts the supply of steam to the necessary amount.

With coal of such a quality that it produces 7½ times its own weight of steam in the locomotive boiler, the average consumption of coal for the three coldest months should not exceed about 6½ lbs. per axle per hour.

Westinghouse System.

In the Westinghouse system of heating, steam at atmospheric pressure only is used in the heaters, and the temperature of the heaters is thus maintained equal throughout the train.

Figs. 27, 28 show a typical Westinghouse heater. It consists of a tube A with external gills, which provides a large heating surface in comparatively small space, the heating surface being approximately 2¾ times that of a plain tube of the same external diameter.

The gilled tube A, constructed of steel tube with steel gills, is provided with covers B and J. Steam from the branch pipe enters the heater by the inlet D on the lower side of cover J. Tube E is made of a special alloy having a high ratio of expansion, and is attached at one end to socket F. At the other end of tube E is carried valve K, for which the edge of the steam inlet forms a seat at

L. On socket F is formed a projection C. Attached to the cover B, by two studs G and H, is a plate M. The projection C is kept in contact with plate M by spring N (which also serves to press the gland O against packing P). By adjustment of nuts Q, R, S, T, plate M is fixed so that one end

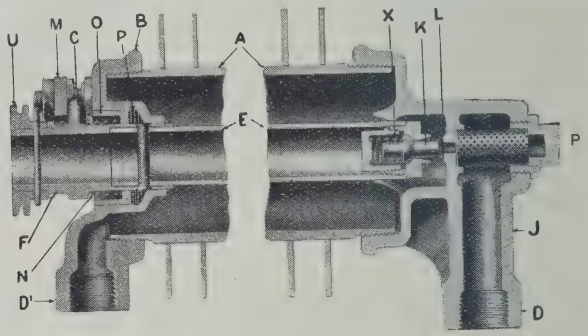


FIG. 27.

of it (*i.e.*, the left-hand end) is slightly further from the heater cover than the other end. In consequence, when socket F is rotated, the projection C slides along the plate M and expansion

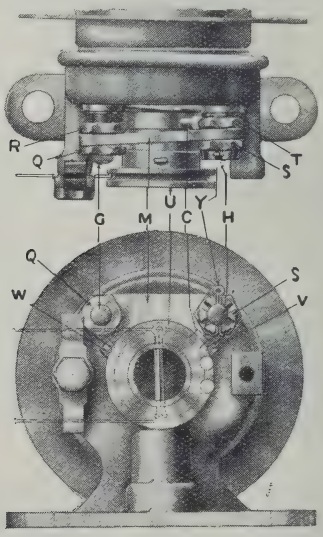


FIG. 28.

tube E moves further in or out of the heater, as the case may be, causing valve K to approach or recede from seat L. The total movement of tube E can be varied to suit requirements by adjusting the inclination of plate M.

Socket F is rotated through a pulley mounted on its outside end by means of Bowden wire, as shown in figs. 27, 28 ; or a crank for rod operation can be fitted if desired.

When the heater is in the ' On ' position, i.e., with projection O rotated to the left-hand side, tube E is drawn back and unseats valve K, allowing a free passage of steam which enters and completely fills the heater, thus rapidly heating up tube A and other parts ; the air and water of condensation escape from outlet D^1, from which a short piece of pipe is led through the floor of the carriage to the outside. As the heater becomes hot, so does tube E, which expands more than tube A for the same rise of temperature. This causes valve K to approach seat L and, when tube E reaches a predetermined temperature, valve K closes the steam inlet and prevents further ingress of steam until the heater cools slightly, when tube E contracts and again allows valve K to open and admit a further quantity of steam. The temperature of the heater is thus maintained practically constant.

If the pulley or crank is moved towards the ' Off ' position, valve K is brought nearer to seat L, so that the steam is cut off sooner, because tube E has to expand less to bring valve K on to its seat L.

If the pulley or crank is again moved towards the ' On ' position, the opposite effect takes place. The temperature of the heater is therefore regulated to a nicety by manipulating the crank or pulley.

When the pulley or crank is moved right over to the ' Off ' position, the valve K is brought on to the seat L, thus shutting off the steam ; and as the heater cools down and tube E contracts, the valve K is kept up against its seat L by the pressure of spring X.

It will be seen, therefore, that a variation of pressure in the main pipe does not affect the temperature of the heater, which is controlled entirely by the position of the pulley or crank. As the steam in the heater is never at more than atmospheric pressure, the highest possible temperature therein is 212° F.

A strainer is provided to prevent the passage of foreign matter into the heater. The cap P should be unscrewed and the strainer cleaned when necessary.

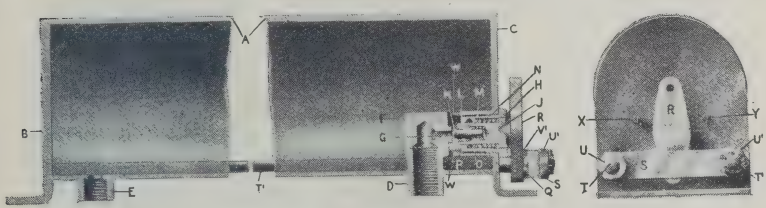


FIG. 29.

Another type of heater is illustrated in fig. 29. In this case the thermostatic control is obtained by expansion and contraction of the main heating tube against two external solid rods which are fixed to one end of the main tube and cause movement of the valve in relation with the seat of the valve mounted in the main tube.

The end covers B and C, the steam inlet connection D, and drain connection E are welded to tube A, and the valve casing F is welded to cover C. The valve seat G, and bush H, are screwed into casing F. In bush H slides a stem J, carrying valve K, which is held in position by a nipple L, and pressed towards the left by spring M. Another spring N is interposed between gland O and a shoulder in stem J, and serves to keep the gland tight against packing P, and also to keep projection Q on lever R in contact with plate S. Two rods, T and T^1, are fixed in cover B, and extend outside and parallel to tube A, through holes in plate S to which they are attached by nut U and castle nut U^1, and lock nuts V and V^1. Steam enters at D and passes into the heater at openings W. Water of condensation escapes to atmosphere through E.

When tube A is hot it expands, and, as the external rods T and T^1 remain cooler, the nuts U and U^1 press against plate S and cause stem J to slide towards the left, pressing valve K against its seat. As tube A cools it contracts, and seat G is withdrawn from valve K, allowing steam to enter again.

The temperature control operates in exactly the same way as in heaters described above—*i.e.* the plate S is adjusted out of parallel with the end cover so that rotation of lever R causes stem J to move in or out of bush H. This adjustment is set before the heater leaves the works, and further adjustment in service should be confined to moving the nut U and its lock nut V (behind plate S) very slightly.

If necessary at any time, the valve can be removed from the heater while the latter is in position by removing the 'On' and 'Off' stops X and Y, turning lever R clear of plate S and withdrawing stem J.

The Westinghouse system of heating is in use on the chief railways in the United Kingdom and on many railways in foreign countries, including some of the coldest in the world.

BRAKES.

Action of Brakes.

f = proportion which resistance produced by brakes bears to weight of train.

v = speed in feet per second.

V = speed in miles per hour.

Distance in feet in which the train is stopped on level $= \frac{v^2}{64{\cdot}4f} = \frac{V^2}{30f}$.

Brakes self-acting on all the wheels, $f = {\cdot}14$.

Ordinary brakes worked by hand in brake-vans, $f = {\cdot}031$ to ${\cdot}023$.

On gradient *ascending* 1 in n, resistance $= f + \frac{1}{n}$.

On gradient *descending* 1 in n, resistance $= f - \frac{1}{n}$.

Distance in feet for stopping $= \frac{f \times \text{distance on level}}{f \pm \frac{1}{n}}$

Power required by Brakes.

The following results have been obtained as regards consumption of energy with different kinds of air brakes :—1. On city lines, the consumption of energy for excentric compressors was 41·5 watt-hours per car kilom. (66·4 watt-hours per car mile); with geared compressors, 31·2 watt-hours per car kilom. (49·9 watt-hours per car mile); and with motor compressors, 16·16 watt-hours per car kilom. (25·86 watt-hours per car mile). 2. On suburban lines, the consumption for excentric compressors was 22·8 watt-hours per car kilom. (36·5 watt-hours per car mile); with geared compressors, 14·6 watt-hours per car kilom. (23·4 watt-hours per car mile); and with motor compressors, 6·32 watt-hours per car kilom. (10·11 watt-hours per car mile).

The Westinghouse Automatic Brake.

The Westinghouse automatic brake is continuous throughout the train and is operated by compressed air furnished by the compressor and stored in the main reservoir on the engine. This compressed air is fed by the driver's brake valve into the train pipe and, past the triple valves, into the auxiliary reservoir on each vehicle.

The brake is applied by reducing the pressure in the brake pipe, which causes the pistons of the triple valve to move and permit some of the compressed air stored in the auxiliary reservoirs to pass to the brake cylinders, the pistons of which are forced outwards, applying the brake blocks to the wheels.

The brake is released by restoring the air pressure in the train pipe, which causes the triple valves to close the communication between auxiliary reservoirs and brake cylinders and open a port from the brake cylinder to the atmosphere, through which the compressed air escapes from the cylinder. The spring in the cylinder can then push back the piston and withdraw the blocks from the wheels.

The brakes are usually applied by the driver, or in cases of emergency by the guard, but a break-away, or rupture of a hose coupling, or other accident causing an escape of air from the brake pipe, also immediately applies the brakes.

It is recommended that a pressure of 90 lbs. per sq. in. be maintained in the main reservoir, and 70 lbs. per sq. in. in the brake pipe.

Improved Triple Valve.

This valve has been designed to give a closer approach to simultaneous actioe of all the triple valves in a train than heretofore. As shown in fig. 30, the the valve is in the release position and the bulb is open to the atmosphere. When the slide valve 6 is moved in applying the brakes, the bulb 3 is closed to the atmosphere and opened to the brake pipe by the cavity p and port s shown in dotted lines. The local reductions of brake pipe pressure thereby produced by the forward triples in a train cause an earlier action of the rearward triple valves, which results in a more nearly simultaneous braking effect being produced throughout the train in every case of first setting the brakes than heretofore, with a smoother and more efficient brake service generally.

A further improvement, also tending to smooth and even action of all the brakes, is shown in the passage leading from port a in the seat of the slide valve 6 to the brake cylinder at B. It consists of a removable plug 13, perforated with holes w and x, which are always open, and an additional hole y carrying an automatic check valve 14, supported by a spring 15. It is so arranged that, when the brakes are first set, the automatic valve is opened against the spring resistance by the excess of air pressure above it, so that the hole x controls the rate of flow of air to the brake cylinder: as the air pressure on the under side of the valve increases, the spring closes the valve, when the smaller holes w control the rate of flow to the brake cylinder. The diameters of the regulating holes w and x are made proportional to the diameter of the brake cylinder with which the triple valve is to be used. By changing these perforating plugs, the same pattern of valve is thereby suited to cylinders of various sizes.

The bulb at the bottom of the triple valve is made in different sizes, proportioned in content to the volume of the brake pipe of the vehicle on which the valve is to be used.

A separate passage a^1 is provided in the triple valve body from brake cylinder to slide valve 6 for the air to escape from the brake cylinder on releasing the brakes. The rate of release is controlled as heretofore, by a nipple screwed into the outlet hole c in the triple valve body.

Driver's Equalising Brake Valve.

The principle of this brake valve, fig. 31, is not to allow the driver in ordinary applications of the brakes to discharge air directly from the brake pipe, but from a small reservoir connected with chamber T of the valve. The reduction of air pressure, thus effected in the reservoir, is then at once automatically and properly repeated in the brake pipe by means of a small equalising piston, 11, placed between the air in the chamber T, and that in the brake pipe E. The piston moves in accordance with the variations in pressure above and below it, and governs an exhaust valve U in such a manner that the air pressure in the brake pipe must always equalise with that in the small reservoir connected with the valve chamber T. Even if the exhaust of air from the brake valve reservoir is abruptly stopped by the driver, the discharge valve U is always gradually closed by the piston, when equilibrium of pressure is established throughout the train. The equalising device, therefore, will under all circumstances provide for a uniform reduction of the pressure in the main brake pipe, thus ensuring a uniform application of the brakes to all the vehicles in the train.

The compression of air is effected by a compressor (Westinghouse), a description of which is given in Section XXIII.

Pipe Couplings.

In the connecting-pipe *coupling*, which is shown by fig. 32, the two halves of couplings, 1, are exactly alike, and an air-tight joint is formed by means of the rubber packing-ring, 3, 1 in each, which rings face each other when the couplings are united. The air-pressure in the

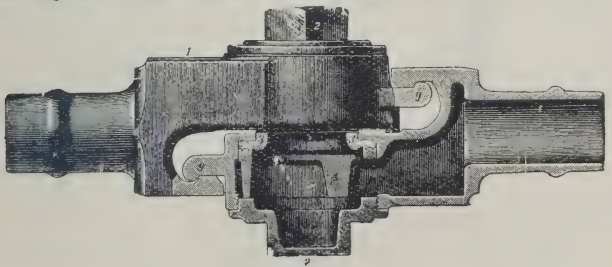


Fig. 32.

couplings tends to force these rings towards one another, so that the joint becomes tighter with increase of pressure. The tendency of the air-pressure in the couplings is also to force them apart in a direction at a right angle to the line of the india-rubber hose-pipes, and, consequently, the greater the pressure, the more firmly are the couplings held together by the projecting piece, g,

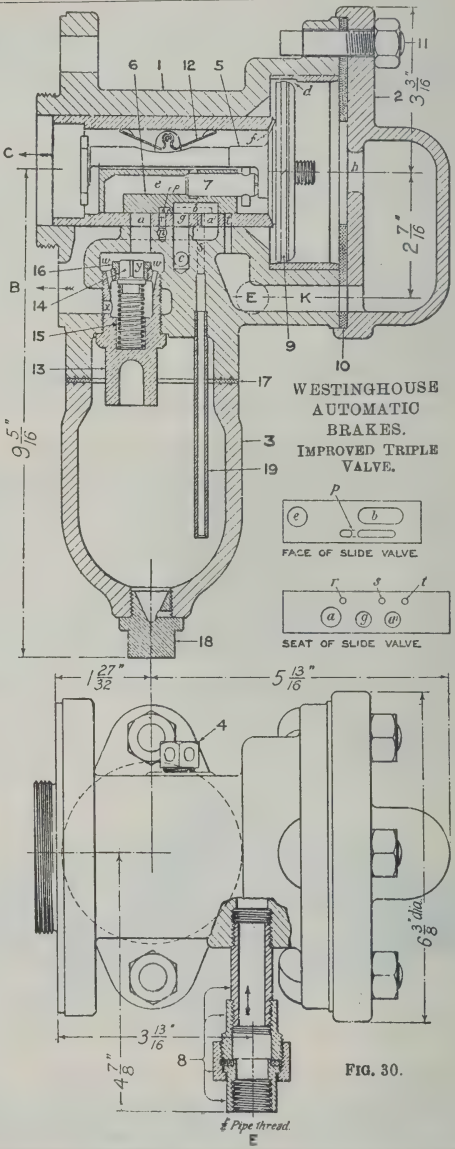

WESTINGHOUSE AUTOMATIC BRAKES.
IMPROVED TRIPLE VALVE.
FACE OF SLIDE VALVE
SEAT OF SLIDE VALVE
Pipe thread.

FIG. 30.

WESTINGHOUSE AUTOMATIC BRAKE
DRIVER'S EQUALISING BRAKE VALVE.

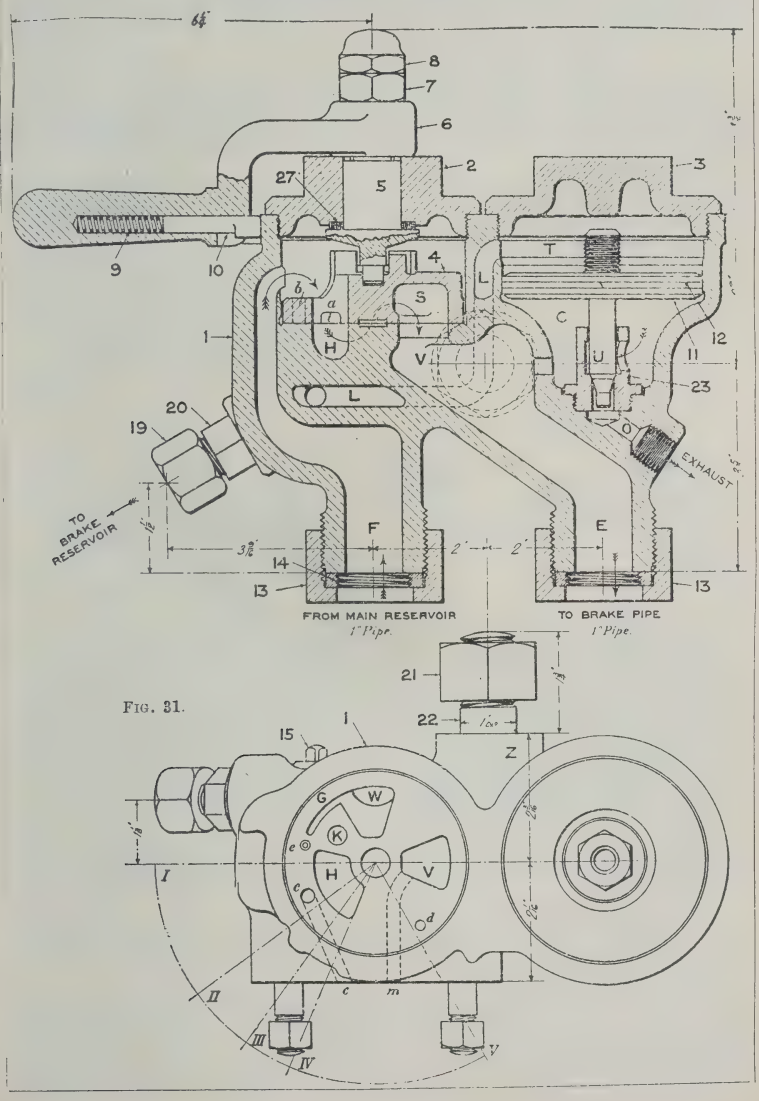


FIG. 31.

of each coupling, which fits in a corresponding groove of the other coupling. No damage is done if the couplings be drawn apart forcibly by the separation of the train, as the rubber rings, 3, 3, are forced into their respective couplings far enough to permit the projections to disengage from their grooves. These couplings are united by simply placing them face to face at right angles, the stop-pins being on the under side, and then turning the projection of the one into the groove of the other.

The Vacuum Automatic Brake.

The Vacuum Automatic Brake is applied by admitting air to the brake cylinders through a continuous brake pipe, and released by withdrawing the air. To charge the brake cylinders, air from above the piston is withdrawn through a non-return ball valve to below the piston and thence direct through the brake pipe to a high-speed rotary exhauster in the case of an electric locomotive, or an ejector if a steam locomotive.

A service brake application is made by the driver admitting air to the brake pipe through a valve on the locomotive, and atmospheric pressure entering the brake cylinder below the piston acts against the vacuum above the piston and applies the brake at a pressure proportionate to the reduction of vacuum in the brake pipe. Air is prevented from entering the evacuated space above the piston by the non-return ball valve.

An emergency brake application may be made with the guard's van valve at rear of train, or a slow brake application by the passenger emergency valve fitted on each carriage. The brake is automatically applied throughout the train if an accident occurs sufficient to part the couplings.

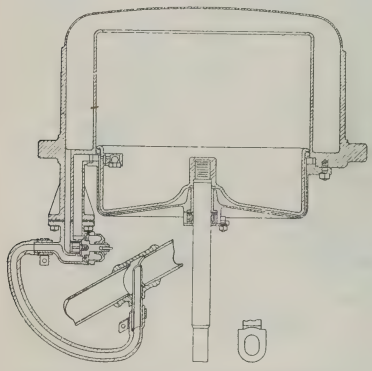


FIG. 33. "Prestall" Wrought Steel Vacuum Brake Cylinder with internal ball valve and external release valve. Piston in release position. Air above piston is exhausted through the internal ball valve. The brake pipe connection to cylinder (shown in section) is provided with a siphon to prevent condensed moisture from being drawn into cylinder.

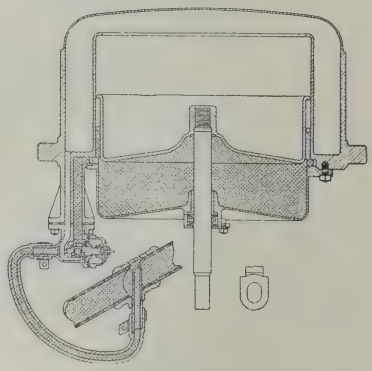


FIG. 34. "Prestall" Wrought Steel Vacuum Brake Cylinder with external ball valve. Piston in 'brake on' position. A brake application is made by admitting air to the shaded portion below piston. The non-return ball valve prevents air entering the space above piston.

Passenger trains are generally operated at 20-in. vacuum, but a few railways use 24-in. vacuum; and freight trains at 20-in. vacuum, or 16-in. vacuum on long trains where high vacuum is difficult to maintain.

Brake cylinders are either of the combined type with the vacuum chamber forming the outer casing of the cylinder (see figs. 33 and 34) or separate type with an independent vacuum chamber. In the former case, the cylinder is connected to the brake pipe by a single branch valve, and in the latter case a double branch is used, the second branch connecting the vacuum space above piston with an independent vacuum chamber.

The ball valve is housed either inside the piston as shown in fig. 33, or in the casing of a detachable valve as fig. 34. The former type of valve is less liable to leak through grit being drawn on to the valve seat, and it possesses a further advantage that when the piston moves upwards during a brake application the port in the wall of the piston (leading to the ball valve) passes to the vacuum side of rolling ring, and thus isolates the valve and cuts off a possible source of leakage that might render the brake inoperative. The poppett type release valve (see fig. 33) used on cylinders with an internal ball valve, effects a more rapid manual release of the brake than is

possible where the ball valve (see fig. 34) combines the functions of check valve and manual release valve as with the externally housed ball valve.

The tendency to reduce deadweight wherever possible, and the use of higher tensile materials in the construction of rolling stock has led to the introduction by the *Westinghouse Brake & Saxby Signal Co.*, of brake cylinders made entirely from pressed steel. These are 30 to 40 per cent. lighter than cast-iron cylinders and possess greater resistance to fracture or breakage. These cylinders, known as the "Prestall" wrought steel cylinders (figs. 33 and 34) have a detachable base for the easy renewal of the rubber rolling ring packing between cylinder and piston, the total detachable weight being 50 per cent. less than the cast-iron type in which it is necessary to remove the entire cylinder from casing in order to renew the rolling ring. The piston rod is guided by a gun-metal bush and rendered airtight by a rubber gland arranged so that atmospheric pressure acting on the outside diameter of the gland forces it with an even pessure around the piston rod. The rubber gland is easily renewed without dismantling the cylinder by simply unbolting the steel cover-plate forming the gland cap.

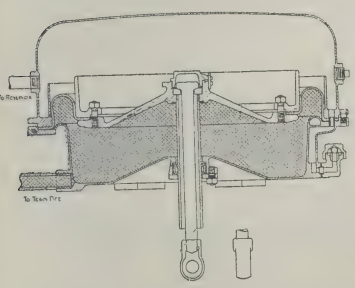


FIG. 35. "Prestall" Wrought Steel Diaphragm Cylinder. The rolling action of the diaphragm between the vertical flanges is shown with the Piston at 'brake on' position; the shaded portion represents atmospheric pressure.

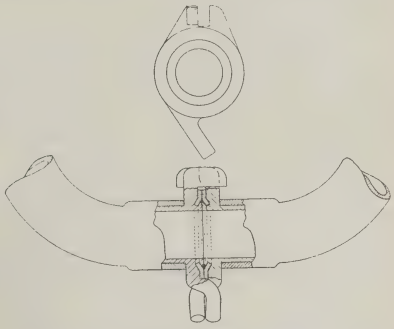


FIG. 36. Vacuum Hose Coupling. *Above*, End Elevation; *Below*, Sectional side Elevation.

In all cylinders with rolling ring packing a certain amount of leakage is liable to occur after the rolling ring has become worn, and to avoid this, certain railways use a cylinder with a reinforced rubber diaphragm, since the vacuum can be maintained above piston for a longer period with a diaphragm than is possible with the rolling ring packing. This is an important feature for railways with long gradients, since it is impossible to recreate the vacuum above piston without first releasing the brake. The "Prestall" wrought steel diaphragm cylinder, fig. 35, is provided with a vertical guide flange on the piston follower, and a corresponding flange in the dome of the cylinder, so that the diaphragm rolls between these two walls and the atmospheric pressure beneath the diaphragm is confined to a vertical path in the line of piston travel. This enables a longer and more powerful stroke than is possible when the diaphragm is not so guided.

The continuous brake pipe is generally 2-in. bore, but a few narrow-gauge railways use 1½-in. bore. The brake pipe is connected between vehicles by means of flexible internally armoured rubber hose, at the end of which is secured a metal coupling head with rubber jointing washer. These couplings when connected are held together by the horn at the bottom of coupling, which hooks on to a similar horn on the opposite coupling, whilst a lug and slot at the top of the coupling fits into a similar lug and slot on the opposite coupling, and keeps the bore of the two couplings in register. This coupling in the form approved by the Railway Clearing House is shown in fig. 36.

SECTION XIII

PART III

RAILWAY SIGNALS.

MECHANICAL SIGNALS.

THE WESTINGHOUSE BRAKE AND SAXBY SIGNAL CO., LTD., SIGNALLING SYSTEMS.

Double Wire Working.

In this system two wires are utilised as the operating transmission for the working of point locks, locking bars, derailers, signals, etc. The system, which is employed practically universally on the continent of Europe, has been developed with a view to making it suitable for operation with the British type of signal. A great advantage of the arrangement is the facility with which the operating wires for points, etc., can be run, compared with the installation of the ordinary rod. These operating wires can, if necessary, be placed at a sufficient height to avoid interference with the ordinary use of pathways and sidings. There is a further advantage in definitely pulling the signal to 'danger,' as compared with the gravity method, because it prevents any possibility of an arm remaining in the 'clear' position due to accidental causes, such as freezing of the signal arm or connections; also points and signals can be operated at a greater distance than is possible with the ordinary mechanical system. Points 500 yards from the cabin are well within the limits of working, and thus it is possible with the system to dispense with two interlocking frames where long passing loops or sidings are installed. Fig. 1 (p. 646) shows the type of lever generally used for double wire working. Fig. 2 (p.646) shows a simple form of point mechanism, and indicates how, with a breakage of one wire, protection is given against accidental movement of the points; the catches A and B prevent the driving pulley, to which the operating wires are connected, moving, and, consequently, shifting the points, unless an even tension is maintained in both operating wires; and, of course, functions in both positions of the points. Means are also provided, in case of a breakage of the return wire on signalling systems, for the immediate disconnection of the pulling wire, and the return to 'danger' of the semaphore arm, by gravity.

POWER SIGNALLING.

Electro-Pneumatic.

In the electro-pneumatic system the control and the indication of the operation of the points and signals is carried out by means of electric power, at, as a rule, a moderately low voltage, and at the expense of quite small current consumption, this electric power being required only for the operation of small valve magnets, whereas the actual heavy work of shifting points, signals, etc., is carried out by means of compressed air. This compressed air may be at any pressure, but, as a rule, a pressure of 60 lbs. per sq. inch is chosen. Electric control may be by direct current, or by alternating current.

Simple cylinders are used, to which air is admitted by the electro-magnetic valves before mentioned. The motion of the cylinder being in a straight line, it is unnecessary to provide

special gear for changing the rotary into linear motion, as is necessary with apparatus operated by rotating electric motors.

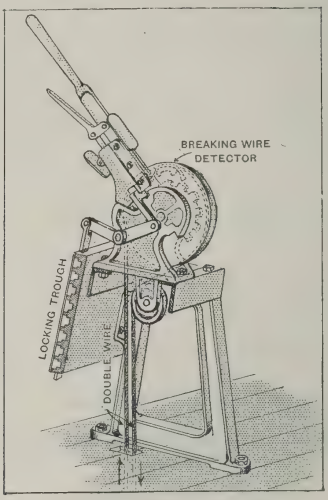


FIG. 1.

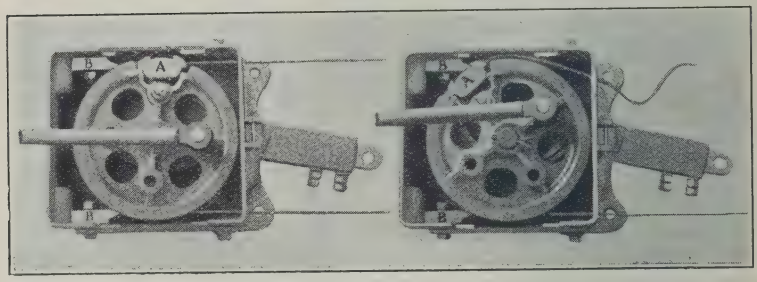


FIG. 2.

All-Electric System.

With the all-electric system, electric power is used both for operation and control. As a rule, for both direct-current and alternating-current systems, a pressure of 100 volts is used, this being applied directly to the motors for the signals or point machines. For operation on direct current, the motors are generally series wound, having two field windings, one of which is connected to the normal and the other to the reverse control wire; the remaining two wires of the field coil are connected to one armature brush, the other brush being connected to the return wire. Application of current to the normal control wire causes the motor to run in one direction, and in the opposite direction when current is applied to the reverse control wire.

Installations operating on this system have been in service at Baker Street (Metropolitan Railway), on the Berlin Underground Railways, and elsewhere, for many years.

With the alternating current system, split phase induction motors are used for the operation of the signals. The point motors are series repulsion type motors, and as the current taken by

these is considerably heavier than that required by direct-current motors exerting the same horse-power, it is generally found advisable to supply current to them from a local transformer, the control from the cabin comprising a polarised contactor and two wires which may be energised with current in one direction, so connecting the field and armature together as to cause the motor to run in one direction, whereas when the current in the control wires and contactor is reversed the field and armature connections to the motor are also reversed, causing it to run in an opposite direction. The current for operating the polarised controller is very small, and comparatively thin conductors can be used between the lever and the controller. This patented system of supplying current locally enables alternating current to be applied in a very simple manner to the heavy work of shifting points.

Point Indication and Detection.

The system now almost universally adopted by the Westinghouse Co. employs a point detector at the points which, with one set of contacts, simultaneously detects both switch tongues and facing point bolt. This point detector can be wired up as a double pole change over switch, to which are connected two wires, which are taken to the signal cabin and operate a relay of the polarised type, so that the relay will be energised in one direction when the points are normal and in the other when the points are reversed. When there is no current on the wires the relay is de-energised. The system can be used either for direct current or alternating current, although the tendency is to use alternating current in all cases.

The arrangement enables all detection of signals over points to be carried out by taking the signal control circuits over the contacts of the relays located in the signal cabins.

Light Signals.

During the past few years there has been a very large development in the use of colour-light signals for daylight work. With these signals the same indication is secured both by night and by day, and signals can be easily distinguished in bright sunshine at 1,000 yards or more. Although the Westinghouse Co. have a number of designs of day colour-light signals, the type which has been most widely installed is that in which systems of lenses, one for each colour, are provided, behind which there are fixed special electric lamps accurately placed at the focal centre of the lens combination, the illumination of the various lamps being controlled by relay or lever contacts. The signal has therefore no moving parts and is very easy of maintenance. The first installation in the British Isles was made on the Liverpool Overhead Railway in 1921, and since that date further installations have been made on the L. & N.E.R. between Marylebone and Wembley, the L.E.R. from Golders Green to Edgware, on the Met. R. between Harrow, Rickmansworth, and Watford, also a very extensive installation for some 200 miles of line in New Zealand. Colour light signals, including a large number of the four-aspect type, have also been installed throughout the extensive area of electrification on the Southern Railway lying between and including Charing Cross, Cannon Street, and London Bridge. In service the indications given by these signals have proved very satisfactory. The latest type of signal developed by the Westinghouse Co. provides a short-range indication in addition to the main beam lights, so as to make the signal indication readily visible to the driver standing directly below and slightly at the side of the signal. With day colour-light signals the alternating current system of control is always used, as it is necessary with such signals to concentrate the source of light in as small a space as possible, and this can only conveniently be done with low-voltage metallic filament lamps. With alternating current, power can be transmitted from a cabin or other source to the signal at 100 volts and transformed down at the signal itself to a low voltage suitable for the lamps.

A development of electric signalling is the operation and control of outlying points and signals. Many such installations are in service in locations where points have to be handled from a greater distance than is possible, or permissible, by mechanical levers and rodding. In some cases the power is supplied by either primary or secondary batteries located near the points, and controlled by wires from the cabin.

Another method is by the hand generator system developed by the *Westinghouse Brake & Saxby Signal Co.*, in which the current is provided by a generator operated by hand when movement of the points or signals is required. Control and interlocking in this case is provided by either a slide lever frame, a small power frame, or even by a standard mechanical locking frame. The distance from which signals and points are operated by this system is, in many cases, more than a mile.

Track Circuits.

Track circuits may be of either direct-current or alternating-current type. In the first, primary batteries are often used with small direct-current relays, but all large modern installations or track circuiting are now carried out on the alternating-current system, in which transformers are used to step down the main supply of voltage to that suitable for use on the track rails, say one to two volts, and an alternating current relay is used. These alternating-current relays are so designed as to make them immune from operation by direct current, and they are therefore suitable for operation on lines where direct-current propulsion is used for electric trains. Where the propulsion current returns to the power station through the running rails, alternating-current track circuits are the only ones which it is possible to use. With these, impedance bonds are necessary at the ends of each track circuit, which impedance bonds are so designed as to allow the direct-current propulsion to pass without hindrance, whilst preventing the alternating-current from passing from one section to the other.

Alternating-current track circuits were first installed in Great Britain on the C.L.R. and G.W.R. about 1910. Since then numerous alternating-current track circuits have been installed in all parts of the world.

WESTINGHOUSE ELECTRO-PNEUMATIC RAILWAY SIGNALLING SYSTEM.

This consists essentially of electrical circuits working in conjunction with a compressed air supply. The electrical system connects the signalman's levers at the cabin with the points and signals, whereby instantaneous response of the signals and points to the motions of the levers is secured, and also an instantaneous 'indication' of the movements of the points and signals is returned to the locking frame. The compressed air of about 60 lbs. per sq. in. pressure serves as the actual power moving the signals and points. The levers in the signal cabin are merely small electrical switches, and they involve no heavy labour on the part of the signalman for their operation. The locking frame, the levers of which are closer together and far less in number than those in the hand system, is provided with two kinds of interlocking—a mechanical system and an electrical system. The mechanical interlocking is practically identical with that used in hand-power signalling systems, excepting, of course, that the parts are very much lighter. The electrical locking consists of a series of electro-magnets connected in the indication circuits from the signals and points; the armatures of these magnets engage with projecting lugs or sectors fitted to the shafts of the levers. By means of this combination of electrical and mechanical locking, the many complicated problems of special locking required by modern methods of signalling are successfully solved. The electrical contacts of the lever shafts, by which the circuits between the battery and the electro-magnetic valves of the point and signal motors are made or broken, consist simply of phosphor-bronze strips, working over contact pieces mounted on an ebonite bushing surrounding the lever shaft. The electric current is usually drawn from a small battery of storage cells, the pressure being about 14 volts. Primary cells are often used, the current required being only about 0·05 ampere per signal or point motor. Small wires made up into cables connect the locking frames with the various point and signal motors.

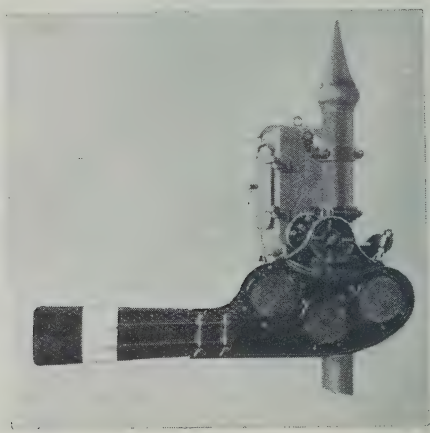


FIG. 3.

The actual work of pulling off a signal or throwing over a set of points is done by a small compressed air-motor fixed close by the signal or point to be operated. The definite, straight line motion of such motors is particularly suitable for the movement of signals and points. The valves controlling the air supply to these motors are opened by small electro-magnets. With this system selection of signals becomes simple, since it is easy to arrange the electrical contacts in such a way that the path of the electric current is made through the signal only which appertains to a route which is set by the movement of certain point levers, and when the signal lever is pulled over that signal only is operated by the electric current; whereas, if other point levers are operated or another route set, the path of the current being changed, another signal is operated when the same signal lever is pulled over.

The pressure of the compressed air used is, as before stated, about 60 lbs. per sq. in. Experience has proved this pressure to be the most suitable for the purpose having regard to the

size of the motors, and the general convenience of installing, operating, and maintaining the system. The quantity of air used does not amount, even at the busiest centres, to more than half a cubic foot of free air per minute per lever.

The electro-pneumatic signal (fig. 3) is fitted direct to the post, and is designed for either two- or three-position working. The air cylinder C (see fig. 4) operates the arm to the 'caution' or

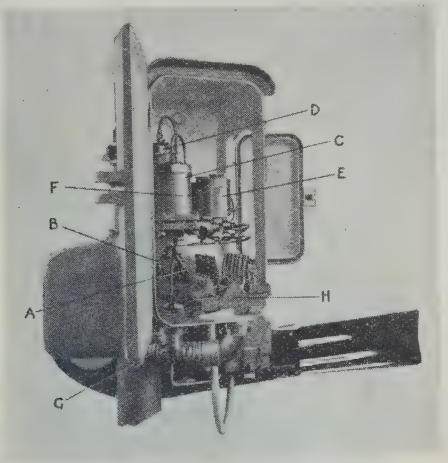


FIG. 4.

45° position, under the control of electro-pneumatic valve E, through crank A, which rotates the spindle on which is mounted the semaphore arm. This arm is operated to the 'clear' or 90° position by the air cylinder D, under control of electro-pneumatic valve F, through crank B. The apparatus is so designed that the 90° position cannot be reached by the semaphore arm unless the electro-pneumatic valve controlling the 45° position has first been energised. This arrangement greatly simplifies control wiring. Each signal has an internal circuit controller H, which provides contacts for controlling such external circuits as for indications, repeaters, etc. The return to the 'danger' position of the semaphore arm is by gravity, immediately the pressure is cut off, and the cylinders exhausted, by the electro-pneumatic valves. The exhaust of air is choked by an exhaust nipple which provides a cushioning effect, so that the arm will not drop too heavily.

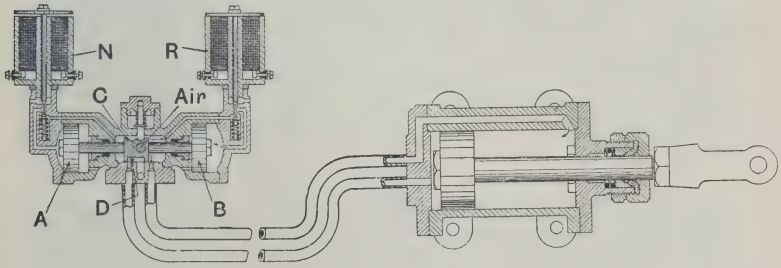


FIG. 5.

The point motor, fig. 5, consists of a double-acting cylinder working horizontally, and is usually installed outside the track, alongside the points. Compressed air is admitted to this cylinder by the point valve, which is usually mounted on a post near-by and connected to the cylinder by two

pipes. The point valve consists of a slide valve C moved by pistons A and B, these pistons being in turn controlled by electro-pneumatic valves N and R respectively, which admit air (or exhaust it) behind pistons A and B, according to which valve is energised. The chamber in which the slide valve is located is under constant air pressure, and when the slide valve is moved to right or left it uncovers one or other of two ports which communicate direct with the back or front ends of the cylinder. At the same time the covered port is bridged by a cavity on the face of the slide valve to a third port open to atmosphere.

The latest development is the point valve style C (fig. 6), which employs a third electro-pneumatic valve, the purpose of which is to lock the slide valve into position, and to cut off the air supply after each operation of the cylinder. This arrangement enables metal-packed pistons to be used in the point cylinder, and also economises in compressed air, as the cylinder is under air pressure only while the movement is taking place.

When the style 'C' point valve is used it can be arranged so that should the point tongues be accidentally or deliberately displaced, air is automatically applied to the point cylinder so that the tongues will be closed again.

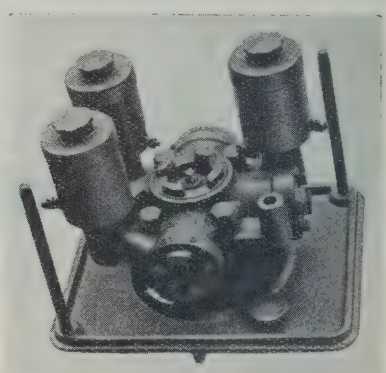


FIG. 6.

Westinghouse electro-pneumatic signals are also largely used in connection with automatic signalling, both on steam and electric traction roads. For this purpose the running rails are divided into suitable sections or blocks which are electrically insulated from one another: the presence of a train or any vehicle in a block keeping the signals controlling the entrance to that block at danger. The signals are not cleared to admit a second train until the first train has cleared the block by a certain distance, depending upon the braking distance required by a train.

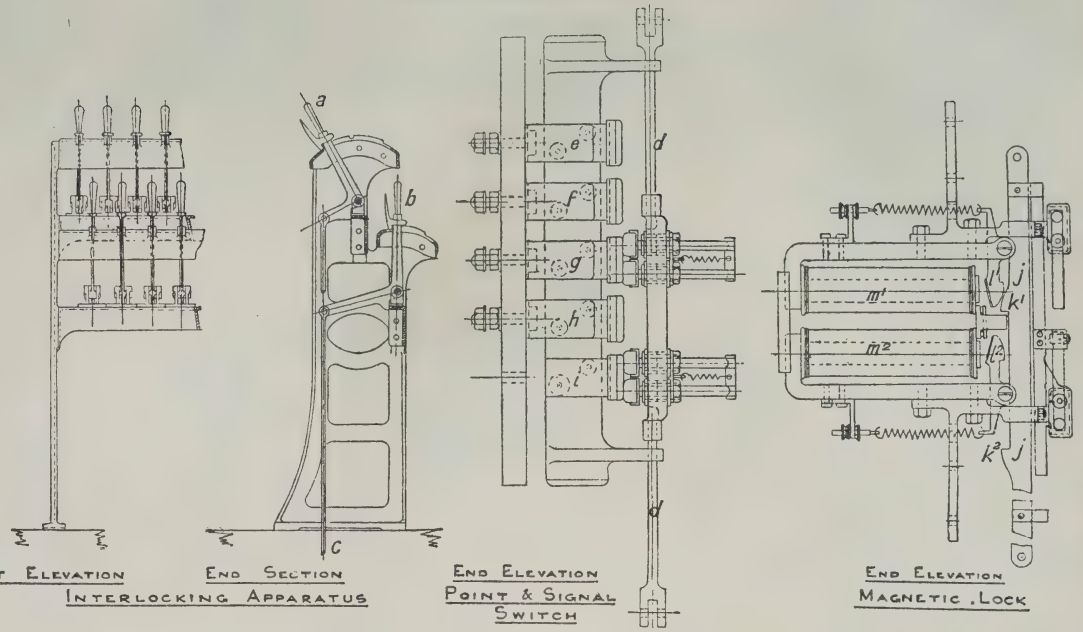
Such installations are largely used in America, there being nearly 10,000 signals of this type in operation. In Great Britain complete installations have been working for several years on the Metropolitan Railway, the Metropolitan District Railway, the London Electric Railways (consisting of the Baker Street and Waterloo; Great Northern, Piccadilly and Brompton; and Charing Cross, Euston and Hampstead Railways), the Central London and the East London Railways. These installations deal with a service of forty trains per hour, and have the additional protection of a train stop at every signal location which automatically applies the brakes on a train should it overrun a signal at danger.

The electro-pneumatic signalling apparatus, both for power-operation of points and signals and for automatic signalling, is manufactured and supplied by The Westinghouse Brake and Saxby Signal Co., Ltd.

'CREWE' ALL-ELECTRIC SYSTEM.

The Railway Signal Company, Ltd., are the manufacturers of a system, of which there are several installations on the London Midland and Scottish Railway (North Western Section)—hence its name—of operating points and signals by electricity.

Fig. 7 shows the locking-frame, and it will be seen that the levers are in two tiers, those, *a*, in the upper tier taking odd numbers and, generally, working points, and those, *b*, in the lower the even numbers and, generally, working signals. To each lever is connected a down-rod *c* to which, immediately under the signal-box floor, are coupled cranks. These actuate the interlocking tappets in the usual interlocking-boxes. At the lower end of the down-rod is connected the switch *d*, the carbon contacts of which join, when the lever is moved, switches *e f* and *g h*, and so cause power to be sent to the points or signal as the case may be.



FRONT ELEVATION — END SECTION
INTERLOCKING APPARATUS

END ELEVATION
POINT & SIGNAL SWITCH

END ELEVATION
MAGNETIC LOCK

FIG. 7.

As in other power-signalling systems, the full stroke of a point lever from normal to reverse and from reverse to normal, and the complete stroke of a signal lever from the 'off' to the 'on' position, cannot be made until an indication is received that the operation set up by the initial movement of the lever has been completed. This, in the 'Crewe' system, is achieved by the check lock j attached below the switch. On the check lock j are two lugs k^1 k^2 which engage pawls l^1 l^2, the armatures of which can be attracted by the magnets m^1 m^2. When a point lever is moved, say, from normal to reverse, the down rod is raised and the lug k^2 on the check lock will then abut against and be stopped by the pawl l^2, but as soon as the current set up by the switches above has done its work, the return indication current energises the magnet m^2, so that the pawl p^2 is attracted and frees lug k^2, consequently the lever can then be pulled the remainder of the stroke. When the points are being restored to normal the down-rod descends and lug k^1 is then stopped by pawl p^1 until magnet m^1 is energised. When the points are reversed the switch d has its contacts on switches e and g, and when normal on switches g and i.

Fig. 8 is a plan and elevation of the lay-out of a pair of facing points. The motor (see also fig. 9), which requires a current of 20 amperes and 100 volts, is fixed in a cast-iron box below the rail level. The motor a (fig. 9), by means of a worm, drives the worm wheel b, fixed loose on the same shaft as the cam wheel c. Movement is transmitted to the cross shaft by means

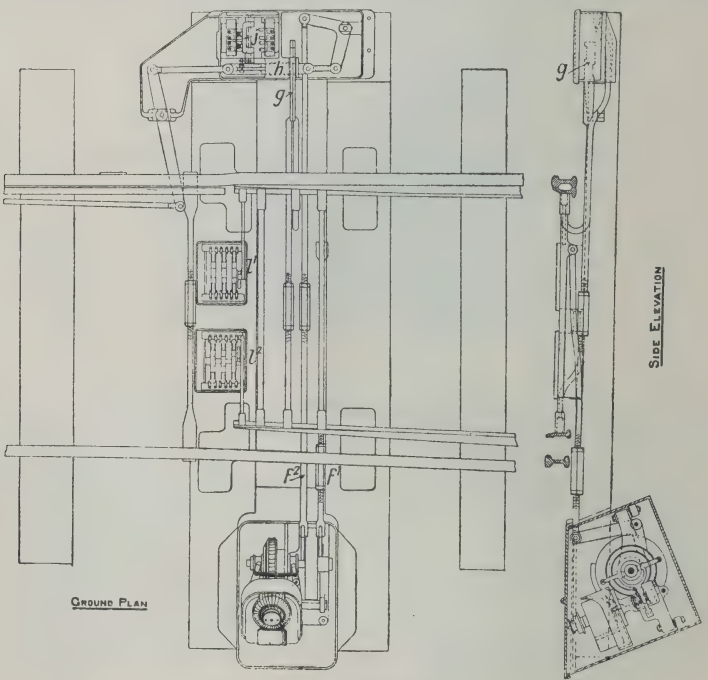


FIG. 8.

of one or other of the clutches d^1 d^2, which are right- and left-handed, so that if the worm wheel turns in one direction the right-handed clutch is engaged and if in the other direction the left-handed clutch. On either face of the cam wheel c is a cam-shaped slot of irregular contour in which works a roller attached to a crank, e. The crank on one side works the points by means of the rod f^1, and the other crank actuates the plunger and locking bar by means of rod f^2. Each point has an extension g, suitably slotted on the upper side for one position of the points and on the lower side for the other position of the points. Both extensions and, therefore, both switches of the points, must be accurate in their position or the plunger h will not pass through them. To the end of the plunger is connected a rod leading to the locking bar. On the upper side of the plunger

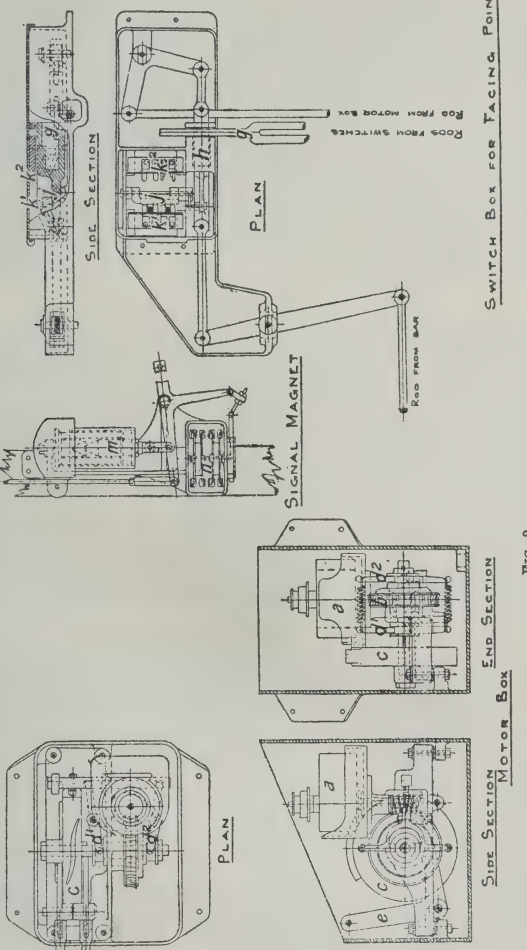

'Crewe' All-Electric System.
Rod to switches
Rod to bar
Plan
Side Section
Motor Box
End Section
Signal Magnet
Side Section
Plan
Rods from switches
Rod from motor box
Rod from bar
Switch Box for Facing Points

Fig. 9.

are guides that move over a short lever and place the contact maker j in or out of contact with the springs k^1 k^2. The latter are connected to the magnets m^1 m^2 of the return indication in fig. 7 and cause pawls l^1 l^2 to be attracted and allow for the down rod c to complete its upward or downward movement and for the switch d to make contact with e g—if an upward movement—or with g i if a downward movement. This cuts off the current from the motor. But as will now be readily understood, this cannot occur until the plunger is shot and the switches held, nor can the plunger be shot unless and until the switches are in their correct position. Further detection is given by means of the contacts l^1 l^2, which are in the circuits to the signals concerned. The switches for these contacts are rotated by means of short rods connected as near as practicable to the 'nose' of both point switches.

Signals in the 'Crewe' system, whether semaphore signals or the dwarf signals used in shunting operations, are changed from the 'danger' to the 'clear' position by means of a long-pull magnet m, fig. 9. A current of about 15 amperes and 100 volts is required for a fraction of a second to clear a signal, which current is then passed, by means of the switch n, through a resistance, and about 2 amperes is enough to hold the signal at 'clear.' When the lever in the signal-box is replaced partly to normal the current supply to the signal is completely cut off and the arm goes to danger by its own gravity. When the signal is fully 'on' the switch n sends the return indication and allows for the lever to complete its stroke. No return indication is necessary to show that a signal has gone from 'danger' to 'clear.' What is of importance is that it has gone from 'clear' to 'danger.'

SYKES ELECTRO-MECHANICAL SYSTEM.

About 2,000 levers have been installed in the electro-mechanical system of the W.R. Sykes Interlocking Signal Co., Ltd., wherein all signals are actuated by electricity, the circuits for which are completed by a small slide being drawn forward or pushed back, and all points are moved by the usual mechanical lever, elevations of which are given in fig. 10

The slide a carries contact pieces b^1 b^2 which, when the slide is drawn out, put springs c^1 c^2 into contact with the plate d. This completes the circuit to the signal governed by the lever by means of wires passing through the box e. When it is necessary to control the movement of a slide, e.g., from another signal-box, or by a fouling bar or track-circuit, a pin rests normally in the slide which can only be lifted when the magnet f is energised. The slides are interlocked as in a mechanical frame, but on a miniature scale. The slide itself acts as its own tappet, and the locks are in the locking-boxes g g.

The tappet of the mechanical lever is h, and a stud h^1 turns the shaft h^2, and so, by a stud h^3, actuates a co-acting tappet in the electrical part. By this the mechanical levers and electrical slides are interlocked.

By an ingeniously arranged escapement wheel the movement of the mechanical tappet is only 2 ins., the same as the electrical slide. A lock j in the tappet k holds point levers when required until they are freed by the lock being lifted out of the tappet by the magnet l being energised.

Sykes Railway Signalling Devices.

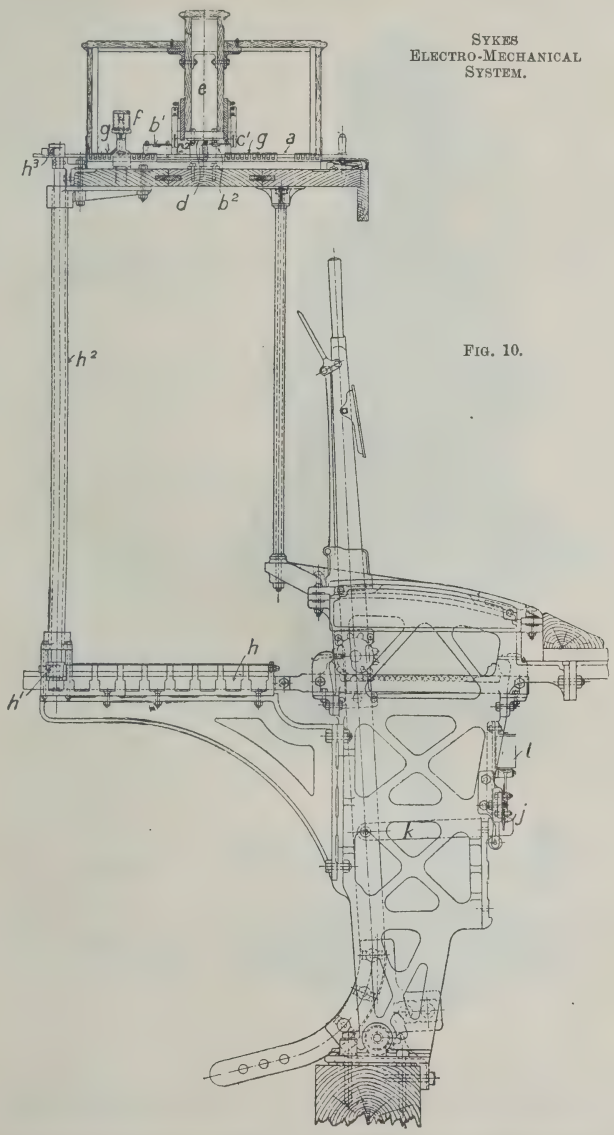
For block signalling, three wire and one wire instruments of various patterns are made, a special system, extensively used on the Southern and L.N.E. Railway's suburban lines being also supplied. In this latter arrangement the signal levers are directly interlocked with the block instruments, so that they must be worked in unison. The system is peculiarly adapted to a heavy suburban or mixed traffic, as it can be worked very rapidly with ease and safety. Modifications have been supplied to Colonial lines, and in recent years a new lock-and-block system called the Style 'L' has been devised, and is used in India, Argentina, Brazil, Cuba, Ceylon, Siam, and elsewhere. The apparatus has been designed with a view to eliminating the difficulties often experienced with native signalmen. An alarm signal gong is sounded whenever a movement has not been made at the right time, and continues till the omission is rectified, while the train itself automatically announces its departure ahead if the signalman fails to do this. In addition, the new system is free from the old cancelling key difficulty, it being impossible to reset the instruments if a train is in the section.

A lock-and-block system for single line working has been devised which enables long and short sections to be worked with great facility, without the complications found in staff and tablet systems. A 'switching-out' instrument can also be used with these systems, and has been adopted by several lines. Messrs. Sykes also make a special instrument for non-crossing block posts, where staff or tablet apparatus is in use.

The 'Intermediate Sidings Key' apparatus is an improvement on the 'Annett' Key, as it enables the key to be taken to either end of the section after use, whereas with the older apparatus it must return to the issuing point. The sidings key instrument may be looked on as a staff instrument for trains using the siding only.

Electrical point and bolt detectors, by means of relays, with interlocked armatures, enable the position of the points to be continuously repeated in the cabin, the electric locking circuit being local.

Electric lever locks are made for track circuit locking and other electrical controls, and also for block instruments. Rail contacts, operated by the deflection of the rail itself—the usual method—or constructed as depression bars worked directly by the wheels, are made.



SYKES
ELECTRO-MECHANICAL
SYSTEM.

FIG. 10.

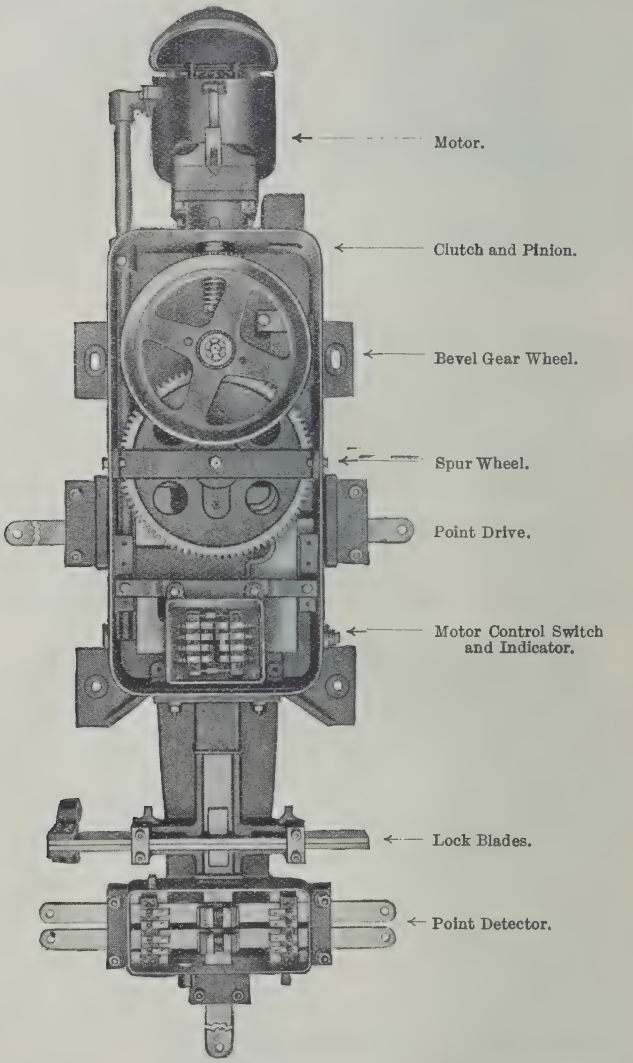


FIG. 11.—British Power Railway Signal Co.'s patented Point Operating Machine.

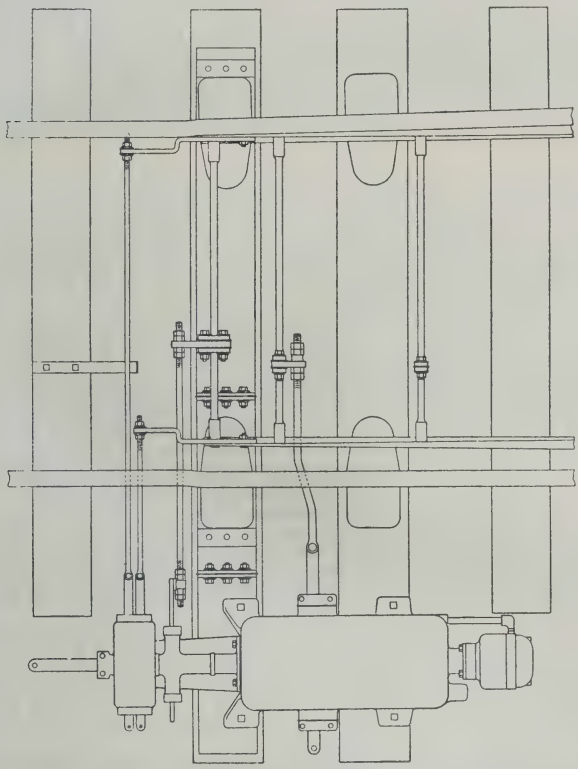


FIG. 12.—Low-Volt Quick-Acting B.P. Point Operating Mechanism, with Outside Point Lock and Detector Attachment.

For 'track circuiting' Messrs. Sykes devised some years ago a flange type joint, which is highly efficient and still in use.

What is termed 'intermediate signalling' is now coming into favour, to enable simple intermediate block posts to be abolished, and the signals controlled from the adjacent cabins. Track circuiting and, generally, lock-and-block control are used for this, and the plan enables considerable economies to be effected. Messrs. Sykes have laid down several installations, a recent one being near Manchester, on the Cheshire Lines.

Recently a new system of signals for single line tramways has been designed and put into service on the Mumbles Rly. The signals are operated from the 700-volt trolley circuit by special contactors, with which the trolley itself makes contact, and exhibit red and white transparent discs, illuminated from behind. The indications are exceedingly clear and have greatly facilitated the single line working on the line. In addition, a novel form of facing point lock has been applied to the points at the ends of the loops, which are spring points, normally bolted. The bolt is withdrawn just before a car passes through in the trailing direction by a depression bar actuated by the treads of the wheels, and returns to the locking position when the point tongue springs back. A shearable piece is incorporated in the bolt to protect the points should the bar become disconnected. A colour light signal, having red and green lights, faces towards the single line, and exhibits a green light when the points are closed in position and bolted. The working of the whole arrangement is automatic.

LOW-VOLTAGE POINT MOVEMENT.

(*The British Power Railway Signal Company, Ltd.*)

The Low-Voltage Point Mechanism, which is used to a considerable extent on single lines, makes it possible to control both points and signals from a distance, and affords a convenient and economical means of operating many outlying points where the cost of a pointsman or interlocking frame would be prohibitive.

The main feature of the arrangement is that the actual operating battery consists of a small set of primary cells or accumulators situated somewhere near the points. Current is supplied to the point motor through control relays, which operate with a fractional amount of current, and can therefore be governed by small line wires over long distances.

Indicators provided in the operator's cabin show at all times the position of the points, and any irregularity of working is immediately detected. Each mechanism box is supplied with a special metal crank handle, which, when fixed on to the shaft of the motor, can be used to mechanically wind over the points for maintenance purposes or in case of battery failure. When not in use, and resting in its normal place in the mechanism case, this handle forms part of the electrical return circuit common to both the operating control relays, and also the signalman's indicators, so that the latter set themselves to show something amiss immediately the handle is removed from its proper place. The control relays are also rendered inoperable.

If track circuit is provided, as in a complete scheme, this can be utilised to prevent a movement of the switches while vehicles are fouling any part of the crossover.

The controlling switch can be placed in any convenient position, and it is sometimes found useful to fix this in the telegraph or stationmaster's office, for instance, when there is only one pair of points for a siding, passing loop, or other purpose in the vicinity, but of course it can also be fixed to and operated by an ordinary lever interlocked with other signals and point levers if required.

Fig. 11 shows the British Power Railway Signal Co.'s patented point operating machine fitted with Point Lock and Point Detector outside attachment. This machine is standard for high (110 v.) or low voltage (20 to 30 v.) operation.

The main operating mechanism consists of a chain of gears comprising :—A bevel pinion fitted with safety clutch device and meshing with a bevel wheel keyed to a spur pinion, this in turn meshing with the main spur wheel. This spur wheel is fitted with roller devices for driving the power bars.

The motor is series wound, enclosed in a substantial weather-proof case. Provision is made for hand operation of the points by means of a crank handle, which requires an isolating switch contact arm to be swung out of the way before the handle can be placed in position.

Figs. 11 and 12 show one of a large number of variations or combinations of machine fixing and point connections.

ALL-ELECTRIC POWER SIGNALLING.

(*The British Power Railway Signal Company, Ltd.*)

Amongst signal engineers the all-electric system of point and signal operation has met with more favour generally in recent years than those systems, such as the electro-pneumatic, where two sources of power are required for operation. Each system has its own advantages and disadvantages, and the choice between the two systems depends on the circumstances at the particular station or yard where an installation is contemplated.

The system of all-electric signalling adopted by the British Power Railway Signal Company, Ltd., is based on the principles of dynamic indication in combination with an automatic stroke completion movement of the signal or point lever, and also a complete cross protection system. The following is the arrangement :

Locking Frame.

The lever slides are arranged in easily detachable units at 3 in. pitch, each slide having cam-shaped slots cut therein for the operation of the interlocking, which is arranged vertically at the front of the machine. A further cam slot is also cut in each lever for the purpose of obtaining the automatic lever movement by solenoid action acting on the diagonal portion of the cam slot for indication purposes.

An extension of the lever slide carries a contact maker which makes and breaks with ' normal ' and ' reverse ' contact springs for operation of the points and signals. A horizontal rod extension from the lever slide operates the combination circuit closer arranged vertically at the back of the frame for the control of the various circuits. The safety and indication selector relays are fixed conveniently at the back and below the lever slides, as also the polarised relays for cross protection purposes.

Point Mechanism.

The power point mechanism comprises a 110-v. motor in weatherproof casing and bolted solidly to the gear-case. The motor operates the train of spur gears and the main cam gear for unlocking, throwing the points and relocking. A friction clutch is provided to protect the mechanism from shock in case of obstruction. A pole-changing device is provided, all self-contained in its own casing within the main gear-case. The pole-changer is required for the changing of the motor connections at the end of each point operation ; this is effected by a cam slot on the lock bar connection, and in addition is also operable by solenoid magnets, in order that the points may be reset in their previous position in the event of any obstruction in the points. Connections from the lock bar may be arranged to work a combined facing point lock with point and lock detector, or the design of point mechanism is arranged for an outside lock or combination of out-side lock and point detector attachment bolted securely to the end of the mechanism casing to meet varying conditions and requirements. Energy for the dynamic indication stroke of the lever is obtained by the overrun of the point motor after the completion of the movement of point function, and in addition the generated indicated current provides an automatic snubbing action to the motor and all parts are brought to rest without shock. Two wires are used for operation, ' normal ' and ' reverse,' and these are used for the return indication current ; the ' normal ' operating wire is used for the ' reverse ' indication, and the ' reverse ' operating wire for the ' normal ' indication.

For detection, one ' normal ' and one ' reverse ' wire is used.

Signal Mechanism.

The signal mechanism comprises a 110-v. motor with a train of gears operating a circuit controller and main driving shaft. A spring attachment is operated by this driving shaft and is placed under tension when the signal lever is operated. When a train enters a track section govern-ing the signal the operating current is broken and the signal falls by gravity to the ' stop ' position. The spring tension comes into action, when the lever is returned to ' normal,' for the purpose of rotating the motor armature, which generates the dynamic indication current and automatically completes the indication stroke of the lever.

The design of mechanism is similar for running signals or dwarf signals ; for the latter an alternative signal of the solenoid type is sometimes used, in which case battery indication is provided.

D.C. OR A.C. LOW-PRESSURE ELECTRO-PNEUMATIC SIGNAL SYSTEM.

The British Power Railway Signal Co., Ltd., specialises in d.c. or a.c. electro-pneumatic systems for terminal or through stations, in which the principles of the automatic indication of the stroke completion of the lever, and cross protection, are retained, as in the All-Electric Signalling System. The points are operated by 8-in. diameter cylinders controlled by electro-pneumatic valves, the operating air pressure varying from 15 to 20 lbs. The signal operating cylinders are 4 in. diameter.

For gravity or sorting yards, the point operating cylinders are 7 in. diameter and are controlled by electro-pneumatic valves energised from a push-button machine placed in the control cabin.

Mechanically Operated Points.

The limit of distance from levers working points to be 350 yds., unless the points are power worked and occupation of the lines is electrically indicated in the signal-box, in which case the distance may be indefinitely extended. (*Ministry of Transport.*)

SECTION XIII

PART IV

RAILWAY EXPENDITURE AND RECEIPTS.*

Financial Statements.

The capital expenditure of the railways in Great Britain at 31st December, 1929, amounted to 1,222,851,206*l.*, as compared with 1,216,234,299*l.* at the end of 1928, an increase of 6,616,907*l.* Capital receipts at 31st December, 1929, amounted to 1,170,532,425*l.*, the amount received during the year 1929 being 2,098,395*l.* The excess of capital expenditure over capital receipts was thus increased from 47,800,269*l.* to 52,318,781*l.*

The gross receipts from railway working amounted to 195,409,523*l.*, compared with 194,005,049*l.* in 1928, a increase of 1,404,474*l.*

The amount appropriated for the payment of interest and dividends in 1929 was 48,355,050*l.*, as compared with 44,873,928*l.* in 1928, representing 4·13 per cent. upon the receipts from capital issued as compared with a corresponding figure of 3·84 per cent. in 1928.

The Reserves and Depreciation Funds of the companies, which in 1928 amounted to 80,873,477*l.*, were reduced to 79,876,621*l.* in 1929.

Way and Works.

Mileage of Lines.—The total length of road open for traffic at the 31st December, 1929, was 20,419 miles, a increase of 10 miles during the year. Expressed as single track, the total mileage of running lines was 37,169 miles (an increase of 26 miles) and of sidings 15,514 miles (an increase of 87 miles), representing a total increase of 113 miles during 1929.

Rolling Stock.

Locomotives.—The number of steam, electric, and petrol locomotives owned, excluding those employed solely on departmental services, at the 31st December, 1929, was 23,454, a decrease of 212 during the year.

Coaching Vehicles.—The number of electric and steam or petrol rail motor vehicles showed an increase of 344 during the year, and of carriages worked by steam power, a decrease of 1,062.

The total number of passenger-carrying vehicles owned was 51,243, a decrease of 718, and the seating capacity of these vehicles was 2,800,272, a decrease of 8,851.

The stock of Pullman cars in use increased from 204 to 229.

Merchandise, Mineral, and Railway Service Vehicles.—The number of merchandise and mineral vehicles owned at the 31st December, 1929, was 700,093, a decrease of 5,988 during the year. The total tonnage capacity excluding brake vans (13,527) was 7,690,405, a increase of 16,147 tons. The average capacity per wagon rose from 11·08 tons to 11·20 tons.

The service rolling stock vehicles decreased during the year from 49,520 to 47,304.

---

* 'Returns of the Capital, Traffic, Receipts, and Working Expenditure, etc., of the Railway Companies of Great Britain for the Year 1929.' (Ministry of Transport. H.M. Stationery Office, Adastral House, Kingsway, London, W.C. 2. Price 6*s.* 6*d.*)

Operation.

Engine-Mileage.—The total engine-miles run increased by 14,312,955, or 2·39 per cent. Coaching train-miles were higher by 2·20 per cent., freight train-miles increased by 3·28 per cent., coaching and freight shunting-miles increased by 2·7 per cent., and other miles (assisting light and departmental) increased by 1·26 per cent.

Fuel and Lubricating Oil Consumed.—For all services combined the consumption of coal increased from 53·55 lbs. to 54·02 lbs. per engine-mile, and the consumption of lubricating oil increased from 6·62 pints to 6·63 pints per 100 engine-miles. The total tonnage of coal consumed and pints of lubricating oil used by steam locomotives and steam rail motors on standard gauge railways were 13,413,308 tons and 36,947,035 pints respectively, as compared with 13,051,812 tons and 36,200,671 pints in 1928.

Wagon-Load.—The average wagon-load for all descriptions of freight increased from 5·54 tons to 5·64 tons, and the average train-load increased from 127·09 tons to 130·90 tons.

Traffic.

Passenger Traffic.—The total number of ordinary and workmen's journeys increased by 3·37 per cent.

The total number of passenger-journeys, including season-ticket holders on the basis of 600 journeys per annual ticket, increased by 2·30 per cent.

Parcels and Miscellaneous Traffic.—The gross receipts from parcels and miscellaneous passenger-train traffic (excluding parcels post and mails) on standard gauge railways were 14,316,990*l.*, an increase of 0·81 per cent. compared with 1928.

The average receipt per coaching train-mile from passenger-train traffic, including mails, parcels, etc., and other merchandise conveyed at passenger train rates during 1929 was 6*s.* 2*d.* as compared with 6*s.* 5*d.* in 1928.

Freight Train Traffic.—The total tonnage of goods and minerals conveyed (excluding companies free-hauled traffic) was 329,578,743 as compared with 306,128,804, tons in 1928.

The total tonnage and net ton-miles of freight traffic (including live stock and companies free-hauled traffic on standard gauge railways) increased by 7·02 and 6·31 per cent., respectively, compared with the year 1928.

On standard gauge railways the average lengths of haul were for general merchandise 91·15 miles, for coal, coke, and patent fuel 42·61 miles, and for other minerals 55·49 miles, in 1929, compared with 89·47 miles, 43·05 miles, and 54·36 miles respectively in 1928. For all descriptions of traffic the average haul was 53·69 miles, compared with 54·05 miles in 1928. The net tonnage miles per route-mile were 933,921, compared with 878,659, and the net ton-miles per engine-hour 446·65, compared with 438·16 in 1928.

The average receipts per freight train-mile from all freight were 14*s.* 10*d.* during 1929, as in 1928.

The average receipts per ton-mile on standard gauge railways were 2·269*d.* for general merchandise, 1·044*d.* for coal, coke, and patent fuel, and 1·105*d.* for other minerals in 1929, compared with 2·346*d.*, 1·059*d.*, and 1·123*d.* respectively in 1928.

Electrical Working of Railways.

The total length of road over which electrical trains were run increased from 560 miles in 1928, to 586 miles in 1929.

The car or wagon miles for passenger and freight services for 1929 compared with 1928 were as under :—

Service.	Car or Wagon Miles.	
	1928.	1929.
Passenger	251,219,260	273,848,467
Freight	3,201,992	4,278,389

RAILWAY STATISTICS OF THE UNITED STATES AND TWENTY OTHER COUNTRIES OF THE WORLD.

Year.	Country.	Miles of Line.	Capital, or Cost of Construction.	Receipts from Passenger Traffic.	Receipts from Freight Traffic.	Receipts from other Sources.	Total Receipts.
			£	£	£	£	£
1906	United Kingdom	23,063	1,253,424,374	48,585,823	56,875,967	8,718,213	114,180,005
1904	Russia in Europe	30,800	580,820,600	11,431,042	47,810,400	5,872,699	65,114,141
1906	German Empire	34,562	722,698,741	34,033,000	79,516,147	12,846,450	126,395,598
1904	France	24,755	662,396,000	26,482,802	31,057,550	928,035	58,468,388
1905	Austria	12,976	282,098,809	7,248,518	20,935,807	1,088,474	29,272,799
1905	Hungary	11,259	143,746,328	3,494,840	9,232,920	501,520	13,228,080
1903	Italy	10,022	222,851,200	5,190,309	7,690,384	623,998	13,904,689
1902	Spain	8,559	113,243,200	—	—	—	11,321,380
1904	Sweden	7,708	48,937,234	1,923,162	3,662,011	112,072	5,697,245
1906	Norway	1,575	11,795,544	395,882	523,925	18,491	938,295
1906	Denmark (State only)	1,164	11,249,700	924,993	952,448	146,688	2,024,129
1905	Belgium	2,844	95,849,477	3,483,859	7,052,261	209,143	10,745,265
1905	Holland	2,100	34,769,800	2,025,760	1,956,400	221,920	4,212,880
1905	Switzerland	2,644	55,190,014	2,820,362	3,481,983	310,381	6,612,727
1905	Portugal	1,425	32,477,056	802,839	1,064,575	84,787	1,952,501
1904	Roumania	1,974	30,115,975	750,932	1,492,786	78,627	2,322,365
1907	Canada (companies)	20,660	234,387,561	9,146,130	19,147,653	1,053,869	29,347,652
	Canada (Government)	1,792	16,548,987	412,477	832,757	56,603	1,301,837
1904	Argentine	12,065	113,579,198	2,584,375	8,450,887	1,138,774	12,074,036
1906	Japan	4,783	41,155,903	3,413,162	2,869,950	620,203	6,903,313
1907	New South Wales	3,452	43,538,024	1,448,998	2,847,047	290,915	4,586,980
	Total	220,183	5,751,323,729	166,499,269	307,453,850	34,921,867	520,607,982
1906	United States	222,340	2,612,855,861	102,006,516	328,077,331	35,069,186	465,153,033

Year.	Country.	Total Expenses.	Ratio Receipts to Expenses.	Passengers Carried.	Mean Journey.	Tons of Freight Carried.	Mean Haul.
		£			Miles.		Miles.
1906	United Kingdom	70,889,525	62·09	1,240,347,000	7·95	488,790,683	24·86
1904	Russia in Europe	42,663,090	65·40	123,208,000	68·58	170,039,234	152·49
1906	German Empire	81,637,773	64·55	1,209,224,072	14·21	479,226,874	62·48
1904	France	30,377,544	51·90	433,912,983	19·59	130,143,810	78·86
1905	Austria	19,592,404	66·91	189,931,211	23	133,764,959	65
1905	Hungary	8,122,560	61·40	86,485,000	25	51,948,000	83
1903	Italy	10,058,508	72·30	68,031,232	27	23,688,410	66
1903	Spain	5,558,400	49·10	39,600,000	34	—	—
1904	Sweden	3,773,554	66·22	39,354,777	16·47	27,624,444	44·64
1906	Norway	691,525	73·71	10,074,372	15·37	4,273,189	34·10
1906	Denmark (State only)	1,572,676	77·70	19,467,616	21	4,530,833	53
1905	Belgium	6,717,124	62·13	163,422,017	14·03	65,515,247	—
1905	Holland	3,527,440	83·74	38,810,000	18 66	13,936,000	56·97
1905	Switzerland	4,294,574	64·95	82,424,588	12	13,971,540	44
1905	Portugal	885,247	45·34	13,446,043	20	3,775,559	89
1904	Roumania	1,327,509	57·10	5,234,683	42	4,731,195	89
1907	Canada (companies)	20,749,734	70·70	32,137,319	64	63,866,135	183
	Canada (Government)	1,265,779	97·23	2,277,218	54	2,673,217	234·60
1904	Argentine	6,408,815	53·05	23,312,987	24	20,123,575	104
1906	Japan	3,215,517	46·58	113,675,403	22	21,530,064	63
1907	New South Wales	2,434,746	53·08	41,413,084	9·96	8,793,832	66·66
	Total	325,764,047	62·60	3,975,789,605	15	1,732,946,800	65
1906	United States	307,489,740	66·08	797,946,116	31·54	1,631,374,219	132·33

(Sloson Thompson.)

Census of Staff.—The total number of persons employed by the railway companies in Great Britain (including the Railway Clearing House, but excluding the Manchester Ship Canal Co.) during the week ending 10th March, 1928, was 677,148, compared with 683,077 during the week ended 30th April, 1927.

Canals.—The tonnage originating on railway-owned canals in Great Britain in 1928 was 1,863,152 tons, as compared with 1,987,799 tons in 1928. Traffic originating on canals, other than railway-owned, in 1928, amounted to 12,710,685 tons, as compared with 13,185,072 tons in 1927.

Ministry of Transport Regulations with Reference to Railways.

See 'Requirements for Passenger Lines and Recommendations for Goods Lines of the Ministry of Transport in regard to Railway Construction and Operation.' H.M. Stationery Office, Adastral House, Kingsway, London, W.C. 2. Price 3*s.* net.)

Cost of Renewal of Railway Tracks.

At the Railway Rates Tribunal (1925) it was stated that on the Great Western Railway it was the general practice to renew the track on the main and principal lines with new materials; the serviceable materials recovered were used on secondary and branch lines. The mileage of the G.W.R. system which must be renewed with new materials was 4,194. The cost worked out at 3,000*l.* per mile. In 1913 the total mileage maintained was 6,709, and this was increased in 1924 to 6,915.

STANDING ORDERS OF THE HOUSE OF COMMONS WITH REFERENCE TO RAILWAY BILLS.

The following are the principal Orders relating directly to railways.

Estimate.

37. The estimate for any works proposed to be authorised by any railway, tramway, tramroad, subway, canal, dock, or harbour Bill shall be in the following form, or as near thereto as circumstances may permit :—

ESTIMATE of the proposed (Railway).

Line, No. . Whether Single or Double.

Length of line . . . . . *Miles. Fgs. Chs.*

Earthworks :	*Cubic Yds.*	*Price per Yd.*	£ *s. d.*	£ *s. d.*
Cuttings—Rock				
Soft Soil				
Roads				
Total				
Embankments, including roads	*Cubic yds.*			
Bridges—Public roads	*Number*			
Accommodation bridges and works				
Viaducts				
Culverts and drains				
Metallings of roads and level crossings				
Gatekeepers' houses at level crossings				
Permanent way, including fencing :				
Miles. Fgs. Chs. at	*Cost per Mile.* £ *s. d.*			
Permanent way for sidings, and cost of junctions				
Stations				
Contingencies		per cent.		
Land and buildings : A. R. P.				
		TOTAL	£	

The same details for each branch, and general summary of total cost.

Plans.

42. In all cases where it is proposed to make, vary, extend, or enlarge any railway, the plan shall exhibit thereon the distances in miles and furlongs from one of the termini; and a memorandum of the radius of every curve not exceeding one mile in length shall be noted on the plan in furlongs and chains; and where tunnelling as a substitute for open cutting is intended, the same shall be marked by a dotted line on the plan, and no work shall be shown as tunnelling in the making of which it will be necessary to cut through or remove the surface soil.

43. If it be intended to divert, widen, or narrow any public carriage-road, navigable river, canal, or railway, the course of such diversion and the extent of such widening or narrowing shall be marked upon the plan; and if it be intended to divert any public foot-path, the course of such diversion shall be marked upon the plan.

44. When a railway is intended to form a junction with an existing or authorised line of railway, the course of such existing or authorised line of railway shall be shown on the deposited plan for a distance of 800 yards on either side of the proposed junction, on the same scale as the scale of the general plan.

Sections.

47. The section shall be drawn to the same horizontal scale as the plan, and to a vertical scale of not less than one inch to every one hundred feet, and shall show the surface of the ground marked on the plan, the intended level of the proposed work, the height of every embankment and the depth of every cutting, and a datum horizontal line, which shall be the same throughout the whole length of the work, or any branch thereof respectively, and shall be referred to some fixed point (stated in writing on the section), near some portion of such work, and in the case of a canal, cut, navigation, public carriage-road or railway, near either of the termini, the distance of such fixed point above or below an ordnance bench mark in the locality of the proposed works, and near one of the termini, and the height of such bench mark above ordnance datum shall also be stated.

48. In cases of Bills for improving the navigation of any river, there shall be a section which shall specify the levels of both banks of such river; and where any alteration is intended to be made therein, it shall describe the same by feet and inches, or decimal parts of a foot.

49. In every section of a railway the line of the railway marked thereon shall correspond with the upper surface of the rails.

50. Distances on the datum line shall be marked in miles and furlongs, to correspond with those on the plan; a vertical measure from the datum line to the line of the railway shall be marked in feet and inches, or decimal parts of a foot, at the commencement and termination of the railway, and at each change of the gradient or inclination thereof; and the proportion or rate of inclination between every two consecutive vertical measures shall also be marked.

51. Wherever the line of the railway is intended to cross any public carriage-road, navigable river, canal, or railway, the height of the railway over or depth under the surface thereof and the height and span of every arch of all bridges and viaducts by which the railway will be carried over the same, shall be marked in figures at every crossing thereof; and where the railway will be carried across any such public carriage-road or railway, on the level thereof, such crossing shall be so described on the section; and it shall also be stated if such level will be unaltered.

52. If any alteration be intended in the water level of any canal, or in the level or rate of inclination of any public carriage-road or railway which will be crossed by the line of railway, then the same shall be stated on the section, and each alteration shall be numbered; and cross-sections, in reference to the numbers, on a horizontal scale of not less than one inch to every three hundred and thirty feet, and on a vertical scale of not less than one inch to every forty feet, shall be added, which shall show the present surface of such road, canal, or railway, and the intended surface thereof when altered; and the greatest of the present and intended rates of inclination of the portion of such road or railway intended to be altered shall also be marked in figures thereon; and where any public carriage-road is crossed on the level a cross-section of such road shall also be added, and all such cross-sections shall extend for two hundred yards on each side of the centre line of the railway.

53. Wherever the extreme height of any embankment, or the extreme depth of any cutting, shall exceed five feet, the extreme height over or depth under the surface of the ground shall be marked in figures upon the section; and if any bridge or viaduct of more than three arches shall intervene in any embankment, or if any tunnel shall intervene in any cutting, the extreme height or depth shall be marked in figures on each of the parts into which such embankment or cutting shall be divided by such bridge, viaduct, or tunnel.

54. Where tunnelling as a substitute for open cutting, or a viaduct as a substitute for solid embankment, is intended, the same shall be marked on the section, and no work shall be shown as tunnelling in the making of which it will be necessary to cut through or remove the surface soil.

55. When a railway is intended to form a junction with an existing or authorised line of railway, the gradient of such existing or authorised line of railway shall be shown on the deposited section, and in connection therewith, and on the same scale as the general section, for a distance of 800 yards, on each side of the point of junction.

SECTION XIV

ELECTRIC TRACTION

ELECTRIC TRACTION ON LONG-DISTANCE RAILWAYS—ELECTRIC LOCOMOTIVES — LOCOMOTIVE BRAKES — SUBURBAN RAILWAYS — MOTOR-COACH CONTROL EQUIPMENT — TRAMWAYS — RAILS — TRACTIVE RESISTANCE — TRAMWAY MOTOR CONTROL — BRAKES — ELECTRIC EQUIPMENT OF TRAMWAYS — TROLLEYS — OVERHEAD CONSTRUCTION — POLES—TROLLEY WIRE—RAIL BONDING — CABLES—MINISTRY OF TRANSPORT REGULATIONS—SPECIFICATIONS — RAILLESS ELECTRIC TRACTION—BATTERY VEHICLES.

(Pp. 669–707.)

(Revised and Amplified by Prof. S. Parker Smith, D.Sc., Assoc.M.Inst.C.E., M.I.E.E.)

SECTION XIV

ELECTRIC TRACTION

ELECTRIC TRACTION ON LONG-DISTANCE RAILWAYS—ELECTRIC LOCOMOTIVES — LOCOMOTIVE BRAKES — SUBURBAN RAILWAYS — MOTOR-COACH CONTROL EQUIPMENT — TRAMWAYS — RAILS — TRACTIVE RESISTANCE — TRAMWAY MOTOR CONTROL — BRAKES — ELECTRIC EQUIPMENT OF TRAMWAYS — TROLLEYS — OVERHEAD CONSTRUCTION—POLES—TROLLEY WIRE — RAIL BONDING — CABLES — MINISTRY OF TRANSPORT REGULATIONS—SPECIFICATIONS—RAILLESS ELECTRIC TRACTION—BATTERY VEHICLES.

(Revised and Amplified by Prof. S. Parker Smith, D.Sc., Assoc.M.Inst.C.E., M.I.E.E.)

ELECTRIC TRACTION ON LONG-DISTANCE RAILWAYS.

There are at present four systems for electric traction on long-distance railways: (1) Direct-current, (2) Three-phase, (3) Single-phase, and (4) Split-phase—a combination of (2) and (3). The Three-phase system, although a complete success, has made considerable headway in but one country—Italy; its drawback is the double overhead conductor. The Direct-current, Single-phase, and Split-phase systems only are in active conflict to-day.

In most European countries the choice of system has been definitely decided by national commissions: these bodies, after weighing the facts, have set forth the details of the system most suited to their needs. Thus, in Great Britain the well-known report of the Electrification of Railways Advisory Committee has recommended the adoption of Direct-current at 1,500 volts. The same conclusion has been reached by the authorities in Holland, Belgium, Czecho-Slovakia, and France; whereas Switzerland, Germany, Austria, Sweden, and Norway have standardised the Single-phase system at low frequency. In the U.S.A. the electrification of the Chicago, Milwaukee, and St. Paul Railway, a large trans-continental route, on the Direct-current system at the high pressure of 3,000 volts, has had a considerable influence on some European decisions. The more recent adoption by the Virginia Railroad of the Split-phase arrangement for one of the heaviest freight services in the world has, however, again drawn attention to the capabilities of A.C. for electric train-haulage, while the Pennsylvania Railroad has adopted single-phase for a main line with heavy traffic.

The motors employed for Direct-current traction are universally of the Series type, yielding a torque which is greatest at starting, falling off as the speed rises. (See Section XXXV.) *Speed regulation*—a very important matter for locomotive work—is obtained by series-parallel grouping of the motors—the standard method employed in traction to vary the motor terminal voltage. Further speeds entail weakening of the motor field flux: a method, while economical, resulting in the reduction of the useful torque per ampere of armature current. Starting is effected by the insertion of resistance in the motor circuit. For passenger and express work the

losses in the rheostats may well be insignificant, but for goods and shunting operation, or where stops are frequent, these losses may become important.

For Single-phase traction in Europe, the Single-phase neutralised series motor is used almost universally; this machine is similar in principle and performance to its D.C. prototype, but has some inherent defects; these have now been overcome by sound design, although compromise and cost have figured largely in the solution, which involves the adoption of a low frequency ($\frac{50}{3} = 16\frac{2}{3}$ periods). The decisive advantage, however, is the feasibility of a high pressure on the contact line: 20,000 volts has been successfully tried, and 16,000 volts is standard practice. The stationary transformer is largely responsible for the success of Single-phase traction, both in its capacity of voltage transformation and of providing simple speed-regulation for the motors. No series resistance is required in the armature circuit, as the motor can be switched on to various tappings on the secondary side of the main transformer. This is of especial value for goods locomotives; while, in addition, starting can be economically graded to take up the slack in the wagon couplings.

In America the later developments in Single-phase train-haulage have been with the Split-phase arrangement. Here a higher frequency can be used on the lines with advantage. The high-voltage Single-phase energy taken from the trolley wire is transformed down on the locomotives and passed through a phase-converter, which supplies two- or three-phase energy to the induction motors used for traction. The use of these robust and efficient motors is a great advantage, and commutators are avoided. This is important in the U.S.A., where 25 cycles has become the standard frequency for A.C. traction. Polyphase induction motors are lighter than either D.C. or Single-phase machines, but speed regulation is difficult, complex pole-changing and cascading devices being required. Rheostatic starting is essential. The recuperation of energy by the motors when descending a grade is automatic, and this explains the adoption of the Split-phase system on lines dealing with heavy mineral trains on steep gradients; moreover, braking is positive; energy is returned to the line; tyres, brake blocks, and rails are subjected to less wear; and the high line-voltage makes the collection of large quantities of energy a simple matter. More recently the Single-phase-D.C. converter locomotive has been developed.

The straight Three-phase system possesses the disadvantage of requiring two overhead conductors, the track forming the third phase. The result is that the voltage between phases is limited, and the load is always more or less unbalanced. Even here, however, modern developments have improved the conditions, and the phase voltage has, experimentally, been trebled. Induction motors are used as for the Split-phase system, and three or four speeds are becoming common practice.

The systems employed for electric traction which have proved to be technically sound are shown in the following table, which traces the connection from locomotive to power-station:—

Equipment of			Supply.
Locomotive.	Track.	Substation.	
D.C. motor	Overhead wire Third rail	Rotary converter Motor generator Mercury rectifier and transformer	
Polyphase induction motor	Two overhead wires	Transformer	Three-phase at normal or low frequency.
Polyphase induction motor, with phase converter	Overhead wire	Transformer	
Single-phase series motor and transformer	Overhead wire	Motor generator and transformer Transformer	Single-phase at low frequency.

The choice of an electrification system in any specified case is first and foremost a matter of finance. In the main, the installation which is cheapest when taken from every standpoint is the system to be installed. An important secondary matter is interference with communication circuits. The following table will give some idea of the comparative advantages of the several systems:—

Item.	D.C.	Three-phase.	Single-phase.
Locomotives—			
Speed regulation	fair	worst	best
Starting economy	poor	poor	good
Motor transformer	none	optional	essential
Power factor	unity	good	fair
Efficiency	good	best	lowest
Recuperation of energy	fair	simple	fair
Current collection	worst	medium	best
Weight	heavy	light	heavy
Cost	medium	cheapest	dearest
Substations and Track Equipment—			
Contact line	good	complex	best
Interference (telephonic)	may be bad	may be bad	safeguards essential
Number and size of substations	largest	medium	smallest
Cost	greatest	small	lowest
Transmission lines—			
Cost	high	high	low
Power Stations—			
Generators	cheapest	cheapest	dearest
Overall cost	least	least	most
Complete Installation—			
Overall efficiency	47–61 per cent.	45 per cent.	50–68 per cent.
Overall cost	not much difference		

The following summary shows the chief developments in Long-distance and Main-line electric traction. Unless otherwise indicated the lines are now running electrically.*

Austria.—The decision on July 23, 1920, by the National Assembly to electrify 650 km. of route was influenced by the urgent need of reduction in the consumption of imported coal and utilisation of available water-power resources to assist in restoring economic independence. The system of electrification has been under consideration for many years, and the commission formed for that purpose has decided to adopt the Single-phase system at 15,000 volts, $16\frac{2}{3}$ periods. The chief lines running or under construction, together with some privately owned lines, are noted in the table below :—

Railway.	Section.	Route Length. Km.	System.	Remarks.
Austrian Federal Railways	Innsbruck–Lindau	235	Single-phase, 15,000 volts, $16\frac{2}{3}$ periods.	Mountain line.
	Salzburg–Worgl	192		Heavy traffic.
	Schwarzach–Villach*	117		Mountain line.
	Stainach–Attnang	107		—
Mittenwald	Innsbruck–Reutte	105	Single-phase, 15,000 volts, $16\frac{2}{3}$ periods.	Connects Tyrol–Bavaria.
Lower Austrian Railways	Vienna–Pressburg	68	Single-phase, 15,000 volts, $16\frac{2}{3}$ periods.	D.C. in Vienna and Pressburg.
	St. Polten–Mariazell	91	Single-phase, 6,600 volts, 25 periods.	Tourist traffic.

The present work on the Federal Railways was due for completion in 1925, after which a further group, totalling 1,135 km., was to have been converted: financial difficulties have delayed progress. Cheap coal may also retard development.

* An asterisk denotes lines at present under construction.

France.—Although the coal situation in France is not so critical as in some other parts of Europe, it is nevertheless serious. The question of main-line railway electrification has been much to the fore since 1918, when a commission investigated existing railways in Switzerland, Italy, England, and the U.S.A. The operation of the Chicago, Milwaukee, and St. Paul Railway, in the Western States of the Union, was closely studied, and a report was made in favour of the D.C. system, but at 1,500 volts. Three of the French railway companies, as detailed below, have embarked on large electrification schemes. Supply will be obtained solely from hydro-electric stations, except in the case of the Paris–Orleans line, for which a supply will also be taken from parallel-working steam stations near Paris.

Railway.	Section.	Route Length. Km.	System.	Remarks.
Paris–Orleans .	Paris–Vierzon .	916	D.C., 1,500 volts	Heavy traffic—partly completed.
	Vierzon–Brive* .			Mountain lines.
	St. Sulpice–Gannat*			
	Brive–Clermont*			
Paris–Lyons–Mediterranean .	Culoz–Modane* .	149	D.C., 1,500 volts	Mountain line.
	Nice local lines* .	174		Suburban.
Midi . .	Bordeaux–Toulouse and branches. Part now working.	1,782	D.C., 1,500 volts	Main line connecting with mountain branch lines, partly converted from one-phase system.

As it is still an open question whether the overhead wire or the third rail should be used, locomotives, etc., are being provided with two types of collecting gear.

Germany.—Electrification on a large scale had already begun in Germany previous to the war, and several experimental tracks had been equipped. The Leipzig–Magdeburg line, in the plains of the Elbe, the Silesian Mountain Railway, the Wiesenthal Railway, in Baden, and some of the South Bavarian lines were already electrified. After 1918 it was realised that the natural resources of the country had to be conserved. For districts lacking in water-power, large steam-stations capable of using inferior lignite coal were proposed. For Bavaria, an extensive scheme is being carried out, the electric power network being fed from hydro-electric stations at Walchensee and on the Middle Isar River: a separate supply network will feed the railways. The question of system was investigated both before and after the war, and in the report of December 1921 it was unanimously decided that the system best suited to the governing conditions was the Single-phase at 15,000 volts, $16\frac{2}{3}$ cycles, with a separate supply for the railways.

Railway.	Section.	Route Length. Km.	System.	Remarks.
German Federal Railways .	Silesian Mountain Rly.	259	Single-phase, 15,000 volts, $16\frac{2}{3}$ periods.	Mountain lines.
	Magdeburg–Leipzig .	177		Heavy traffic.
	Bavarian lines* .	672		Hydro-electric supply.
	Wiesenthal Railway† .	49		

Electrification is sufficiently far advanced in Germany to enable a decision to be reached as to rolling-stock standardisation, and the administration have standardised six locomotive types to suit all the conditions met with.

Great Britain.—Great Britain has comparatively little to show in the way of long-distance railway electrification, although there are numerous schemes and projects under consideration.

Standards for British Railway Electrification.—An attempt has been made to standardise the system, etc., for future developments, but a certain amount of latitude has been allowed, as the following will show:—

The Electrification of Railways Advisory Committee, appointed by the Minister of Transport in March 1920, issued its final report in September 1921, recommending that all new electrification

* An asterisk denotes lines at present under construction.
† 15 periods.

should be on the direct-current system, and that the standard pressure should be 1,500 volts, except in the case of railways already operating at 600 or 1,200 volts D.C., or with alternating current. The adoption of 750 or 3,000 volts should be permitted where approved by the Minister. Both overhead and rail conductor collection were recommended, provided that the form and design of the conductors and structures were in accordance with specified regulations. Current should be generated on the three-phase system, at the frequency in general use in the district concerned; 50 cycles could be used satisfactorily. Suggested regulations were given, with drawings, for the standardisation of contact rails.

The Railway Electrification Committee (1927) reviewed the 1921 recommendations in a report issued in 1928. It was found that extensions had been made to existing lines at 600 and 650 volts, making a total of 1,404 miles of track at low-voltage D.C., 77 miles of 1,500-volt D.C., and 19 miles of Single-phase, the Brighton section of the Southern Rly. having been converted to D.C. The Committee therefore recommended the adoption of D.C. working at 'maximum' standards of 750 and 1,500 volts, with third rail and overhead contact systems respectively, and with generation at 50 cycles, 3-phase. The Committee did not consider it necessary to provide for the inter-running of multiple-unit rolling stock between high- and low-voltage electrified sections. For locomotives, however, to enable inter-running to be effected, two maximum loading gauges were prepared. No. 1, with a maximum height of 12 ft. 8 ins. above rail level, will be suitable for inter-running on all British railways, except on certain short lines, such as light railways, tube lines, and goods branches, totalling 264 miles. Such locomotives will have a scope of practically the whole of the 20,000 route miles of railway. No. 2 takes advantage of the higher (13 ft. 2 ins.) loading gauges of the L.M. & S., L. & N.E., G.W., and Southern (central section main line) Railways, and locomotives constructed to this gauge will have a scope of rather more than 13,000 miles. A suitable overhead-contact current collector was suggested. To permit of inter-running electrically, locomotives should be equipped with 750-volt motors capable of being connected in series-pairs for operation on 1,500 volts, and both overhead and shoe collectors should be provided. The reservation was made that the electrical standardisation need not be effected by a railway company until one-third of its total locomotive stock has been replaced by electric engines. Details of third rail and overhead clearances were specified. The Committee considered it undesirable to set limits to the voltage drop and uninsulated return conductors, a decision with which representatives of the G.P.O. were in disagreement.

A further Government inquiry into main-line electrification was instituted in 1929.

Holland.—In consequence of the recent decision of the Dutch commission on the subject the future development of electric traction will be on the high-voltage D.C. system. The railways will be fed from a unified network of steam power-stations, and it is considered that for this purpose D.C. offers the most decided advantages. At the same time, the only long-distance line at present in operation is the Rotterdam–Scheveningen Railway, which employs single-phase energy at 10,000 volts, 25 cycles. The line intended for the first step in future D.C. electrification is the Rotterdam–Haarlem–Amsterdam railway.

Railway.	Section.	Route Length. Km.	System.	Remarks.
Dutch State	Rotterdam–Scheveningen	36	Single-phase, 1,000 volts, 25 periods.	Semi-suburban traffic.
	Rotterdam – Haarlem–Amsterdam* . .	—	D.C., 1,500 volts.	Main route.

Hungary.—The rearrangement of the international boundaries after the war lost to Hungary her complete oil and natural gas resources, two-thirds of her coal-fields, and nearly the whole of her water-power. The railways must, therefore, be electrified, in order to reduce imports, to conserve what coal and peat remain, and because the available fuel is only suitable for power-station combustion. While plans are ready for the eventual conversion of 1,385 km. of route, only a short 15-km. line has at present been equipped between Budapest and Dunakeszi–Alag. The system proposed is the split-phase arrangement with 15,000 volts, 50 periods on the track, in order to supply the line from phases of the industrial network, and tests under these conditions have proved satisfactory.

Italy.—The electrification of the Italian railways dates back to the beginning of the present century, and provides practically the earliest example of main-line electrification in Europe. This fact largely explains the adoption of the three-phase system at 3,000 volts, 15 periods, at

* An asterisk denotes lines at present under construction.

the outset, for at the time the high-voltage D.C. and Single-phase systems were in their infancy. Although it seems likely that Italy will adhere to the three-phase system, yet there are indications that the line voltage will be raised to 10,000 volts, and the frequency to that commonly met with in the surrounding industrial districts. The reasons for electrification here, as elsewhere, were economical, owing to lack of coal and the existence of abundance (in some districts) of water-power. The Valtellina line in Northern Italy was the first electrified, and other lines have been added from time to time, until now about 670 km. of route is in operation, while very extensive programmes exist for future developments. High-voltage D.C. traction is being tried in the south.

Railway.	Section.	Route Length. Km.	System.	Remarks.
Italian State Railways	Milan–Varese	73	D.C., 650 volts.	Third rail : date 1901.
	Valtellina lines	143		First 3-phase line.
	Giovi lines Savona–Sampierdarena Savona–Ceva Alessandria–Voghera Novi-Tortona–Ronco	241	Three-phase, 3,000–3,700 volts, 15–17 periods.	Among the most heavily worked lines in Italy. Steep gradients and long tunnels.
	Mt. Cenis line	113		Through Mt. Cenis Tunnel.
	Turin lines	173		
	Rome–Tivoli	40	Three-phase, 10,000 volts, 45 periods.	Industrial frequency.
	Benevento–Foggia	—	D.C., 3,000 volts.	Experimental.

Scandinavia.—In Sweden the question of main-line railway electrification has been discussed since 1899. In 1915 the first section of the Riksgransen Railway was put into operation. This line carries heavy mineral trains from the iron-ore mines of Kiruna and Gellivare to the coast at Narvik and Lulea. The line is run at 16,000 volts, single-phase, 15 cycles, under the most severe climatic conditions. Since 1918 the Swedish State Railways Electrification Commission have considered afresh the question of system, and the choice of Single-phase has been confirmed. The Norwegian authorities have come to a like decision. The Swedish Stockholm–Gothenburg line is in process of conversion to electric traction : this line will be fed from the three-phase industrial network through converter stations at 15,000 volts, $16\frac{2}{3}$ cycles. The Norwegian developments comprise the completion of the Riksgransen Railway to Narvik and the equipping of the Kristiania–Drammen Railway.

Railway.	Section.	Route Length. Km.	System.	Remarks.
Swedish State Railways	Narvik–Riksgransen–Lulea*	472	Single-phase, 16,000 volts, 15 periods.	Mineral line.
	Stockholm–Gothenburg	458	Single-phase, 15,000 volts, $16\frac{2}{3}$ periods.	Main line traffic.
Norwegian State Railways	Kristiania-Drammen	53	Single-phase, 15,000 volts, $16\frac{2}{3}$ periods.	Semi-suburban.

Switzerland.—The need for national independence by utilising water-power instead of importing coal has long been recognised. While a few local railways are operated with D.C., and two important lines with three-phase, the main bulk of the electrified railways use the Single-phase system, at $16\frac{2}{3}$ cycles, which, after a protracted study, was recommended by the Administration of Railways in 1912. The single-phase lines comprise the Federal Railways, the Loetschberg Railway, and the metre-gauge Rhaetian Railways. The latter are operated at

* The Narvik–Riksgransen part is in Norway.

11,000 volts, and the others at 15,000 volts. All the power is obtained from hydro-electric stations fed from rivers or lakes in such a way that the supply is maintained reasonably constant all the year round. All the Federal main lines have been electrified, and the Simplon line changed to single-phase.

Railway.	Section.	Route Length. Km.	System.	Remarks.
Loetschberg and Bernese Subsidised Railways	Berne–Thun–Brieg and branches	194	Single-phase, 15,000 volts, $16\frac{2}{3}$ periods.	International through route.
Rhaetian Railways	Chur–Landquart–Klosters–Filisur, etc.	277	Single-phase, 11,000 volts, $16\frac{2}{3}$ periods.	Metre-gauge.
Swiss Federal Railways	St. Gothard line, Rhone Valley, etc.	1645	Single-phase, 15,000 volts, $16\frac{2}{3}$ periods.	International routes.
	Simplon Tunnel	22		Continuation of Loetschberg and Rhone Valley lines.

United States of America.—Largely on account of its immense size and development, most of the important and extensive railway electrification projects are to be found in the United States.

Chicago, Milwaukee, and St. Paul Rly.—At present the C.M.S.P. Railway is the largest electrified line in the world as far as route length is concerned. It is a trans-continental road, operating electrically the single line over the Rocky Mountains, with very heavy trains on long gradients. The system employed is the direct-current at 3,000 volts on the contact line. On the down-grades regenerative braking is extensively used—on the whole, quite successfully. Both geared and gearless types of locomotive are employed.

Butte, Anaconda, and Pacific R.R.—This road is engaged principally in the haulage of copper ore by locomotives, on the 2,400-volt direct-current system. It was the original example of high-voltage D.C. applied to heavy railway operation, and served as a model for the C.M.S.P. Railway.

New York Central R.R.—The electrified portion comprises a 30-mile zone in the Hudson River and Haarlem Divisions, both locomotives and multiple-unit trains being employed. The system is D.C., 650 volts, with third rail.

New York, New Haven, and Hartford R.R.—A great deal of pioneer work has been done by this company, which now operates an 11,000-volt network running out to New Haven. The trains, hauled by locomotives or motor-coaches, run into New York over the N.Y.C. tracks (D.C., 650 volts), with the result that the equipment in many cases is very complicated.

Pennsylvania R.R.—For some years the New York terminus has been electrified at 650 volts D.C., and branch lines round Philadelphia at 11,000 volts, 25 cycles. The electrification is now in progress of the main line between New York and Wilmington, Delaware, on the single-phase system at 11,000 volts, 25 cycles.

Norfolk and Western Rly.—Mineral traffic of a very heavy character is hauled on this line. The locomotives are of the split-phase type, taking single-phase energy from the contact line at 11,000 volts, 25 periods, and supplying three-phase induction motors through a phase converter: the ease with which regenerative braking is effected is a very noticeable feature.

Great Northern Rly.—The new 8-mile Cascade Tunnel was opened in 1928, and is worked at 11,000 volts, 25 cycles. With approaches, the electrified zone is 75 miles.

Virginia R.R.—A recent contract of 15 million dollars has provided for the electrification of the Va. R.R. between Mullens and Roanoke, over a mountainous route of 135 miles. At present, with steam haulage, the company is moving 7 million tons of coal per annum, and the limit has been reached. Electrification (single-phase, 11,000 volts 25 cycles) will enable 6,000-ton

trains to be hauled up the 2 per cent. grades at 14 m.p.h. The locomotive units will weigh 200 tons each, and will provide, when coupled in sufficient numbers, 12,000 h.p. The regenerative braking of the split-phase system to be installed is of paramount importance.

Detroit, Toledo, and Ironton R.R.—This railway, in which Mr. Henry Ford has the controlling interest, has a length of route under conversion to electric traction. A 22,000-volt, 25-cycle, single-phase, contact-line pressure will be used. The locomotives will carry transformers and motor-generators for supplying normal D.C. series traction motors.

Other initiated or completed lines include Reading Co.; Boston, Revere Beach and Lynn R.R.; Delaware, Lackawana, and Western R.R.; Cleveland Union Terminal Co.; and Illinois Central.

Railway.	Section.	Route Length. Miles.	System.	Remarks.
Chicago, Milwaukee, and St. Paul Rly.	Harlowton–Avery Avery–Othello* Othello–Seattle	874	D.C., 3,000 volts	Transcontinental.
Butte, Anaconda, and Pacific R.R.	Butte–Anaconda	25	D.C., 2,400 volts	Mineral line.
New York Central R.R.	Hudson River and Haarlem division	58	D.C., 650 volts	Heavy suburban.
Pennsylvania R.R. (New York)	New York Terminal and Long Is. R.R.	103	D.C., 650 volts	Extensive suburban.
New York, New Haven, & Hartford R.R.	New York – New Haven, etc.	88	Single-phase, 11,000 volts, 25 periods.	Suburban and main line traffic.
Norfolk and Western Rly.	Bluefield–Vivian	29		Heavy coal trains.
Pennsylvania R.R.	Philadelphia–Paoli	30		Suburban.
" "	NewYork–Wilmington	—		Main-line traffic.
Virginia R.R.	Mullens–Roanoke	135		Record train weights.
Gt. Northern R.	Cascade Tunnel	75		8-mile tunnel.

Other Developments.—In many other countries the railway electrification movement has begun. In Morocco a 416-km. line is to be electrified at 3,000 volts, D.C.: motor-coaches will work at this pressure. In New Zealand, the Arthur's Pass section of the Government Railways was opened on August 6, 1923, using the 1,500-volt D.C. overhead system. Important installations have been completed, or are under construction, in Brazil (Paulista Railway, 28 miles, 3,000 volts D.C.); Chile (Bethlehem–Chile Railway, 15 miles, 2,400 volts D.C.); Japan (Tokio–Yokohama–Kozu, 65 km., 1,500 volts D.C.); and in South Africa (Glencoe–Pietermaritzburg, 3,000 volts D.C.); Java (Tanjong Priok harbour–Weltevreden, 16 km., 1,500 volts D.C.); Spain (Northern Rly., Pajores gradient, 40 miles, 3,000 volts D.C.).

Electric Locomotives.

The first electric locomotives were developed directly from the street tramway, and had bogie tracks provided with nose-suspended motors. Progress in America has in general avoided mechanisms involving reciprocating motion, and the nose-suspension and the quill drives have been widely applied to passenger, freight and shunting locomotives. On the Continent of Europe there was a widespread tendency at first to apply the steam locomotive side-rod mechanism to the electric locomotive. Many difficulties were encountered and generally overcome, but the continuous torque of the electric motor was fundamentally unsuited to the reciprocating nature of the drive, whilst the effect of steam cushioning in the steam locomotive cylinders had no counterpart in the electric drive. Subsequent designs introduced springs and similar elastic members to provide for this, and to obviate the troublesome oscillatory phenomena in side-rod drives so damaging to the bearings. The present tendency in Europe is to return to individual-axle drives with nose-suspended motors or with some form of special gearing enabling the motors to be entirely spring-borne. Among these are the Westinghouse (geared-quill, with spring connection between gear wheel and driving wheel) and the Buchli (link connection) drives, and a later form of bevel drive in which the motors have vertical shafts. Individual-axle drives allow greater freedom in the running-gear design and are adaptable to most classes of service, tending to simplification of spare parts and maintenance. The locomotives built in Great Britain have been mostly fitted with nose-suspension drives. Various types of contactor control are used.

* To be undertaken later.

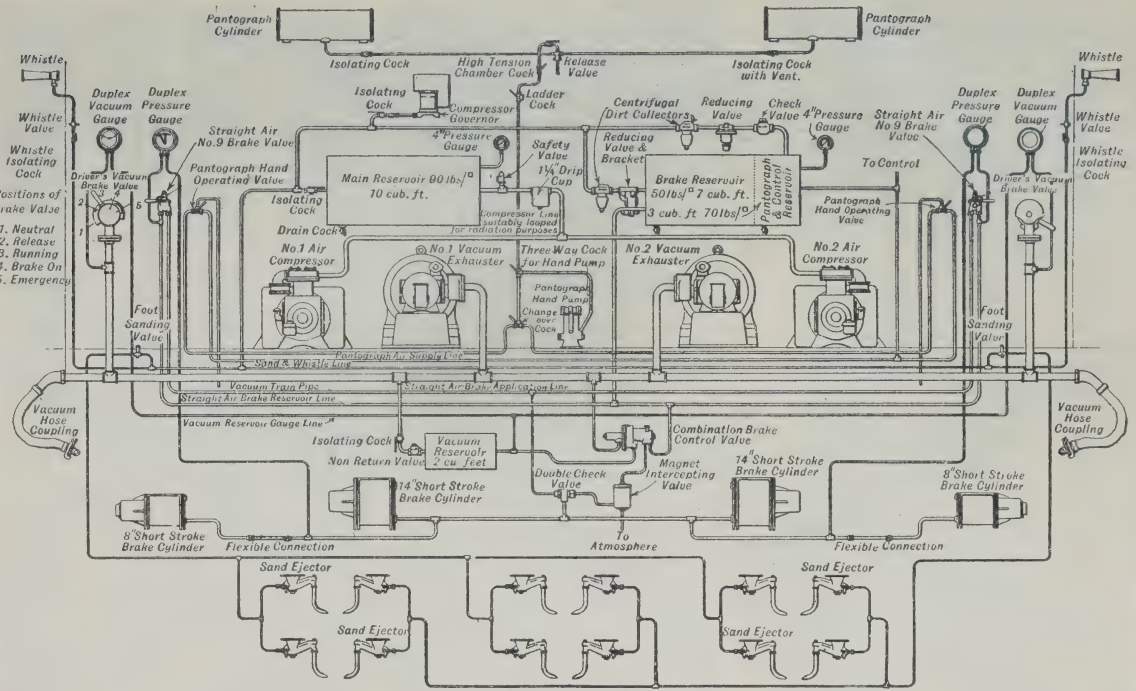

Pantograph Cylinder
Pantograph Cylinder
Isolating Cock
High Tension Chamber Cock
Release Valve
Isolating Cock with Vent.
Whistle
Whistle
Duplex Vacuum Gauge
Duplex Pressure Gauge
Duplex Pressure Gauge
Duplex Vacuum Gauge
Whistle Valve
Whistle Valve
Whistle Isolating Cock
Whistle Isolating Cock
Isolating Cock
Compressor Governor
Ladder Cock
Centrifugal Dirt Collectors
Reducing Valve
Check Valve
4" Pressure Gauge
Straight Air No.9 Brake Valve
Straight Air No.9 Brake Valve
4" Pressure Gauge
Safety Valve 1¼" Drip Cup
Reducing Valve & Bracket
To Control
Driver's Vacuum Brake Valve
Driver's Vacuum Brake Valve
Positions of Brake Valve
1. Neutral
2. Release
3. Running
4. Brake On
5. Emergency
Pantograph Hand Operating Valve
Main Reservoir 90 lbs/□" 10 cub. ft.
Brake Reservoir 50 lbs/□" 7 cub. ft. 3 cub. ft 70 lbs/□"
Pantograph & Control Reservoir
Pantograph Hand Operating Valve
Isolating Cock
Compressor Line suitably lagged for radiation purposes
Drain Cock
No.1 Air Compressor
No.1 Vacuum Exhauster
Three Way Cock for Hand Pump
Pantograph Hand Pump Change over Cock
No.2 Vacuum Exhauster
No.2 Air Compressor
Foot Sanding Valve
Foot Sanding Valve
Pantograph Air Supply Line
Sand & Whistle Line
Vacuum Train Pipe
Straight Air Brake Application Line
Straight Air Brake Reservoir Line
Vacuum Hose Coupling
Vacuum Hose Coupling
Vacuum Reservoir Gauge Line
Isolating Cock
Vacuum Reservoir 2 cu. feet
Combination Brake Control Valve
Non Return Valve
Double Check Valve
Magnet Intercepting Valve
14" Short Stroke Brake Cylinder
14" Short Stroke Brake Cylinder
8" Short Stroke Brake Cylinder
8" Short Stroke Brake Cylinder
Flexible Connection
To Atmosphere
Flexible Connection
Sand Ejector
Sand Ejector
Sand Ejector
Sand Ejector

FIG. 1.

Brakes for Electric Locomotives.

(The Consolidated Brake and Engineering Co., Ltd.)

The diagram (fig. 1) shows a brake equipment for electric passenger locomotives for heavy duty on steep grades, operating vacuum-braked trains, the locomotives being fitted with an air-pressure brake.

To synchronise the brakes on the locomotive and the train and also to give proportionate control, a combination brake-control valve is provided. This valve is governed by the vacuum in the continuous train pipe of the vacuum brake and directly admits a corresponding air pressure to pass to the air-brake cylinders upon the locomotive, giving a perfectly synchronised and balanced braking effect throughout the train and avoiding bunching at the forward end, and shocks.

The air brake can be used independently if desired, by use of the straight air-brake valve; this is useful in shunting operations. The vacuum brake can also be operated upon the train independently of the air brake, for which purpose a magnetic intercepting valve is provided.

The vacuum exhausters are of the rotary drum type directly coupled to the motors with a two-speed control. The full speed is for quickly releasing the brakes or charging up the system rapidly if time is limited. The slow speed is for maintaining vacuum and for brake control generally, the full speed only being used as circumstances require. The two speeds are suitably controlled by contacts on the spindle of the driver's vacuum brake valve, which allows slow speed in all positions except full release, in which position the exhausters run at full speed.

The motor-driven air compressors, in addition to providing compressed air for the locomotive brakes, also supply the necessary air for the sanding gear, pantograph operating apparatus, pneumatic control, and whistles.

ELECTRIC TRACTION ON SUBURBAN RAILWAYS.

Much of the electrification work hitherto has been in connection with suburban railways around large cities like London, New York, and Paris. Such lines form a most promising field for the utilisation of electric power, as they usually carry heavy and dense traffic, a condition very favourable to electric traction. Of course, on tube and similar underground railways anything but electric haulage in the light of modern requirements is unthinkable. For suburban and urban traffic the single-phase system is seldom considered unless some ulterior condition makes this advisable: in practically all cases the direct-current system is used, at voltages of 600 to 1,500, with third-rail collection. It is standard practice to generate three-phase energy in the power stations, transforming to suitable pressures for transmission to the substations. The motor-generators formerly used for A.C.-D.C. conversion are now displaced by the rotary converter, which has higher efficiency and overload capacity. The mercury vapour rectifier is also coming into use (See Section XXXV., Electrical Engineering.)

With the use of 600 volts (or more) for traction on lines of a suburban character, it is usual to replace the hand controller suitable for tramway operation by the multiple-unit control, which consists of small manual or automatic 'master-controllers' operating separate switches or 'contactors,' connected to the motors and to the tappings of the resistance grids. The contactors may be either electro-magnetically or electro-pneumatically operated. Electric coupling between the adjacent coaches permits the use of any one master-controller for the control of the train. The advantages of this so-called multiple-unit control lie in the flexibility of train length, the provision of a motor capacity always suitable to the load (for the maintenance of a uniformly fast service), and the efficient utilisation of the available adhesion (for rapid acceleration).

For leading the current to the moving trains a third rail is provided for each track, the motor coaches being equipped with cast-iron contact shoes for current collection. In England, where the Board of Trade regulations require a very small drop in the earth return, it has been necessary on occasion to add an insulated fourth rail in order to avoid the drop in the running rails which elsewhere serve as a return. This may often be of considerable advantage in cases where track-circuiting is required for automatic train-signalling. Fig. 2 illustrates a typical contact rail construction for a tube line.

On most city and suburban electrified railways dealing with heavy traffic, automatic signalling is employed; indeed for many of the congested services it is essential. Where a fourth-rail return is used, the running rails are used to convey direct or alternating current for the track circuits. Where, however, the running rails carry the power current, alternating current alone may be employed for signalling. For automatic signalling the reader should consult Section XIII., p. 645.

For lines with infrequent service the efficiency of the substations is comparatively low, while the running costs are high in consequence of the fact that the numerous substations must be capable of supplying the peak load. The transmission losses in the conductor rails are also considerable. In such cases automatic or semi-automatic substations, with higher line voltages, may be used.

Low-pressure traction motors are now standardised, as also is their control. Series-parallel arrangement of the motors of each bogie (of the motor coach) is universal practice. In special cases field-shunting is used, as, for instance, when trains make occasional runs to outlying districts. The diagram of tram-motor control given in fig. 4, p. 690, is typical also of motor coaches, except that switching operations, as already indicated, are performed by contactors operated from the master controller.

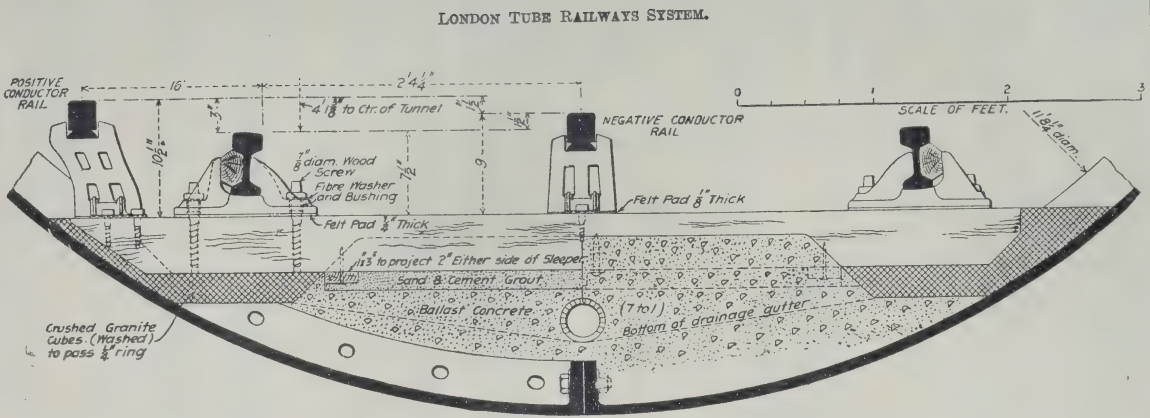


FIG. 2.—Type of Construction of Permanent Way and Conductor Rails on the tube lines of the underground railways of London.

Train units are usually made up of motor-coaches and trailer coaches. A common arrangement is one motor-coach semi-permanently coupled to one or more trailers. In such a case the end trailer is a driver trailer, and is provided with a driver's cabin with a master controller and the necessary apparatus for reversed operation. One, two, three, or more units may be coupled and controlled by one driver to form a train of any desired size according to traffic needs, and the same schedule may be maintained regardless of the train-length, since the power/weight ratio remains constant.

The following are brief details of some important suburban electrification systems:

London.—A large network of electrified lines exists in the south and west of London. The north-east district has hitherto been largely neglected. The Southern Railway has the largest suburban electrification in the world.

Line.	Route.	Route Length. Miles.	System.	Remarks.
Underground.	Metropolitan-District Rly.	28·5	D.C., 550–600 volts, third or third and fourth rail.	Surface and sub-surface.
	Bakerloo Rly.; Piccadilly Rly.; Edgware, Highgate & Morden Rly.; City Rly.	44		Deep-level tubes with surface extensions.
	Central London Rly.	11·1		
Metropolitan.	N. side of 'Inner Circle'; Harrow main line	29·5		Surface and sub-surface.
L.M. & S. Rly.	Euston and Broad St. to Kew, Watford, etc.	38·5		Surface, main-line terminus.
Southern Rly.	All sections . . .	173		Surface, main-line termini.

Paris.—The C. de F. Métropolitain and Nord-Sud operate sub-surface lines covering the area bounded by the old city walls. Stations are closely spaced, allowing of competitive services with street transport. The C. de F. de l'État (Ouest) works two suburban lines and a short local city branch to Auteuil Boulevard. The electrification of the C. de F. de Paris-Orléans embodies the main line out of Paris.

Line.	Route.	Route Length. Miles.	System.	Remarks.
Métropolitain.	Route Nos. I.–X.	62 *	D.C., 600 volts, third rail.	Sub-surface city lines.
Nord–Sud.	Clichy, St. Ouen, and la Chappelle to Porte de Versailles.	10 *	D.C., 700 volts, third rail and overhead line.	
État (Ouest).	Invalides–Versailles, St. Lazare–St. Germain, etc.	30 *	D.C., 750 volts, third rail.	More lines in process of electrification.
Paris–Orléans.	Quai d'Orsay–Juvisy, etc.	14·6	D.C., 1,500 volts, third rail and overhead line.	Part of extensive main-line electrification.

* Further lines projected.

Berlin.—The Hoch- und Untergrundbahn, serving city districts, was begun in 1902. The lines of the Reichsbahngesellschaft round Berlin, including the Stadt- und Ringbahn, are now being converted to electric haulage.

Line.	Route.	Route Length. Miles.	System.	Remarks.
Hoch- u. Untergrundbahn	City lines.	31·5	D.C., 800 volts, third rail.	Sub-surface and overhead.
Stadt-, Ring u. Vorortbahnen	Stettiner Bahnhof, Bernau, Oranienburg and Velten.	45·7 *		Extensive programme of electrification in hand.

* Further lines under conversion.

Melbourne and Sydney.—These two capital cities of Australia have modern electrification systems (complete or projected) of a very extensive nature, since far-seeing town planning has led to a wide distribution of suburban population.

Line.	Routes.	Route Length. Miles.	System.	Remarks.
Victoria Govt. Rlys.	Melbourne City and suburban lines.	154	D.C., 1,500 volts. overhead line.	
N.S. Wales Govt. Rlys.	Sydney Central – Sutherland.	23		Extensive scheme under conversion.

New York.—On account of its geographical peculiarities the city of New York has for many years been subject to traffic congestion in a north-south direction along the line of Manhattan Island. In addition, there is a heavy traffic movement east and west between Manhattan and Long Island (Brooklyn, Queens, etc.). The following are details of the principal lines:

Line.	Routes.	Route Length. Miles.	System.	Remarks.
Brooklyn Rapid Transit Co.	Manhattan–Brooklyn and Coney Island lines.	89·2	D.C., 600 volts, third rail.	Surface and elevated along main streets.
Hudson and Manhattan R.R.	Newark – Hoboken – South Manhattan.*	7·9		Surface and tunnel lines.
Interborough Rapid Transit Co.	Manhattan – Bronx and connection with B.R.T. Co.	114·7		Subway and elevated lines.

* Partly over Pennsylvania R.R. Co.'s tracks.

EQUIPMENT FOR MOTOR-COACH TRAINS.

Camshaft Control.

(The English Electric Co., Ltd.)

The main contactors in this system are operated by means of a camshaft under the control of a small pilot motor. The control circuits are operated at 120 volts, this supply being obtained from a motor generator set. The equipment includes the following essential features :—

(*a*) A camshaft composed of steel cams mounted on a mica-insulated steel shaft of square section. On rotating this shaft the cams close and open the main contactors in a definite predetermined sequence.

(*b*) A small pilot motor which drives the camshaft through suitable reduction gearing. This motor is controlled by the master controller and the position regulator, the latter being mounted on an extension to the camshaft itself.

(*c*) Two solenoid or cam-operated line breakers arranged in series to break the main circuit.

(*d*) All necessary auxiliary control gear, such as an electrically operated reverse, motor cut-out switch, overload relay, etc.

The closing of each contactor is effected against a strong spring by the rotation of the camshaft, so that the action is positive and the pressure constant, irrespective of the line voltage. Furthermore, the camshaft gives a definite mechanical interlock, controlling the sequence of operation of the contactors, and obviates the necessity for fitting electric interlocks. Since all current is broken on the line breakers, and the sequence of operation of all contactors is definite, the resistance contactors do not break any current, and are therefore not fitted with blowouts. The two contactors which deal with the operation of transition are called upon to break relatively small currents, and are fitted with blowouts. This arrangement effects a considerable reduction in the number of current-breaking contactors.

The line breakers are each fitted with a strong metallic blowout of the metallic shield type, and, in addition to their function of line breaker, are arranged to operate in conjunction with an overload relay as circuit breakers. Protection against dead short-circuits is given by special heavy duty-type cartridge fuses mounted on the roof of the motor coach. The fields of the pilot motor are separately and permanently excited at full strength from the 120 volts control circuit. The feed to the armature is taken through the camshaft motor relay, which is arranged to make this circuit when the coil is energised. Directly the coil becomes de-energised the clapper falls back, short-circuiting the armature of the pilot motor under full field, thus giving an absolutely dead beat check to the movement. The camshaft is returned to the off position by means of the pilot motor : this is effected by means of auxiliary contacts on the line breakers, which reverse the connections to the motor armature when the line breakers are in the open position. Thus, when the line breakers open, the camshaft is at once returned to the off position, and it is impossible for the motor to run in the forward direction so long as they remain open. A further interlock on the position regulator makes it impossible to reclose the line breakers until the camshaft has reached the off position. In the event of manual operation being used, the control of the notching up is, of course, entirely in the driver's hands. When automatic operation is used, the driver throws his control handle right round to the full parallel position, leaving the current limit relay to control the rate of notching up. The current limit relay is cam-operated. It is mounted on an extension to the camshaft itself and is, therefore, mechanically interlocked with the resistance contactors and position regulator.

Electro-magnetic and Electro-pneumatic Control.

In the electro-magnetic system the contactors for effecting the motor and rheostat combinations are controlled electrically from the master controller by means of magnet coils energised from the line (if the voltage be below 600 volts), or from a low (*e.g.* 32-volt) circuit, comprising a dynamotor or motor-generator or battery. Interlocking is effected by auxiliary contacts on each contactor. After being closed the control of the contactor is transfered from a 'closing' to a 'holding' coil demanding less energy. Each contactor may require several pairs of interlock contacts, according to the system or design adopted. The direction of rotation of the motors is controlled by a separate reverser. The electro-pneumatic system embodies electrically-controlled needle valves for compressed air cylinders operating the contactors mechanically. Later developments involve the use of individually-operated pneumatic contactors effecting the motor combinations, the rheostats being cut out by means of a drum-type controller operated by a differential air motor. These systems are lighter, cheaper, and less complicated for automatic control. Pneumatic control has the advantage that the control voltage is low ; a large mechanical force can be developed by a small and light compressed-air cylinder, and a spring relied upon for opening the contacts ; the mechanical operations are performed independently of the line voltage, avoiding the tendency of electro-magnetic contactors (fed from the line voltage) to vibrate and weld together when the line voltage falls to a low value.

Control of Alternating-Current Traction Motors.

The majority of single-phase motors are of the neutralised series type, and the basic method of speed control is the variation of the motor terminal voltage. Since a main transformer is invariably used, the methods available are: (*a*) Transformer tappings, controlled by contactor gear. To avoid interrupting the motor supply, a second tapping must be closed before the first is opened. The section of the transformer between the two tappings is protected by an auxiliary choking or 'preventive' coil. (*b*) Transformer tappings with sliding contact switch, developed by Brown, Boveri. The tappings are changed by means of a sliding contact switch driven by a screwed shaft operated by a control motor from the master controller. (*c*) Transformer tappings with induction regulator. This allows of fewer tappings, but owing to its greater complication has found only a restricted application. The advantage is the absence of jerky control.

Polyphase induction motors with slip-ring rotors are controlled by means of rotor rheostats (usually of the liquid type) combined with cascade connection. Motors with cage rotors are started with reduced voltage (by auto-transformer), combined with pole-changing connections.

The following details refer to typical motor-coach equipments with multiple unit control:

(*The British Thomson-Houston Co., Ltd., Rugby.*)

Details.	Metropolitan-District Rly.	L.N.E.R. (Newcastle).	L.M.S.R. & L.E.R. (Bakerloo–Watford).	Bombay–Baroda Cent. Ind. Rly.
Motors per coach .	4	2	2	2
Line voltage . .	600	600	600	1,500
Motor h.p. (1-hr.) .	195	145	235	275
Current collectors .	Shoes	Shoes	Shoes	Pantographs
Gauge . . .	4 ft. 8½ in.	4 ft. 8½ in.	4 ft. 8½ in.	5 ft. 6 in.
Length over head-stocks .	49 ft. 2 in.	55 ft. 0 in.	50 ft. 10 in.	68 ft. 0 in.
Bogie centres .	34 ft. 0 in.	40 ft. 0 in.	34 ft. 0 in.	48 ft. 0 in.
Bogie wheel-base .	7 ft. 10 in.	7 ft. 0 in.	7 ft. 0 in.	10 ft. 0 in.
Wheel diam. . .	36 in.	36 in.	40 in.	43 in.
Weight empty, tons	44·5	34	32·5	69
Seating capacity .	64	64	36	80
Control . . .	electromagnetic	electromagnetic*	electromagnetic*	electropneumatic-camshaft*

* With automatic acceleration.

ELECTRIC TRACTION ON TRAMWAYS.

The technical arrangements for electric tramways are practically standardised all over the world. Two 4-pole totally-enclosed series motors are used, either with or without interpoles. The nominal rating of each motor is 30 to 50 h.p. with double-decked cars. The current transmission is effected by means of an overhead trolley-wire and rail return, collection being performed with a wheel trolley or sliding bow-collector. The voltage is most commonly 500 to 600, although a few American inter-urban roads have adopted a pressure of 1,200 volts. For control, a drum-controller is used, allowing of series-parallel and field-weakening methods for the regulation of the speed. Braking is by air or electric devices, the latter including rheostatic and magnetic track-brakes. Hand-controlled braking is used as a standby. On tramway systems where air-brakes are not used, the hand brakes are used for ordinary service stops and the electric only when going down steep hills or for emergency stops. For tramway substations, rotary converters have replaced the older motor-generator sets, but mercury-vapour rectifiers are coming into use for modern installations (see Section XXXV, Electrical Engineering).

Tramway Tyres and Axles.

The British Engineering Standards Association has issued standard specifications Nos. 101—1929 for tramway tyres and 102—1930 for axles. The former includes particulars of the tests to be applied, and a drawing showing the shape and dimensions of the tyre of standard section. The latter specifies that the steel shall not contain more than 0·05 per cent. of sulphur or phosphorus, and defines the tests to which the axle shall be subjected.

New tyres should be turned to the same angle of inclination as the worn head of the tramway rails has assumed, and care should be taken to ensure that the tyres fixed on the same axle should be of equal hardness, in order to ensure the maintenance of equal diameters in service.

Tramway Rails.

In Great Britain the grooved girder rail is practically the only type employed for tramways constructed in public roadways.

On some inter-urban tramways, with tracks laid on private lands, ' tee ' section rails have been used, loose guard-pieces being bolted on at curves; on single lines, having passing places with automatically worked points, ' tee ' rails present practical difficulties, and the later tendency is to use a special type of the ordinary grooved girder rail for such lines.

Twenty to thirty per cent. more power is required to run cars on a grooved rail than on a ' tee' rail, due to side friction, dirt in grooves, &c.

When gauge is 4 ft. or less, in order to allow room for removing motors, special rails, without overhang, should be fixed at car pits.

Weight of straight-track rails used in this country varies from 65 lbs. per yard, as at Coventry and Norwich, to 108½ lbs. per yard, as at Sheffield, Leeds, Wigan, &c. Rails of from 90 to 105 lbs. per yard are now generally adopted, with depth from 6 ins. to 7 ins., and width of base 6½ ins. to 7½ ins. The rail now used in Glasgow is ' B.S.' section No. 8, the weight being 112·9 lbs. per yard. No. 8c, weighing 119·2 lbs. per yard, is used at curves. The depth is 7 ins. In America, rails weighing as much as 137 lbs. per yard are now being used.

The Board of Trade has made a pronouncement against the employment of rails less than 90 lbs. per yard.

For a wheel flange ¾ in. thick, groove width should be 1⅛ in. on straight, and 1¼ in. on curves less than 75 ft. radius.

It is of importance that the profile of wheel tread and flange should accord to the shape of rail table and groove.

On curves of less radius than 50 ft. there is much grinding of wheel flange against both sides of groove which quickly widens under such wear; to allow for this the guard or lip on curve rails should be made thicker and higher and rails should be preferably of manganese steel.

Curves less than 100 ft. radius should be of spiralised or other easement form.

The top of guard or lip for straight-track rails is usually made slightly lower than the table, to allow for greater wear on latter, but the effect of this is very objectionable to vehicular traffic where the rail table is not sloped or coned.

The rail web should be as nearly as practicable under the centre of rail table, so that the resultant of forces, due to rolling load of car and lateral pressure of wheel flange, shall fall well within the rail base, otherwise there is a tendency for the rail to tilt sideways.

The base should be sufficiently wide to spread the superincumbent weight on to the underbed, and to resist the side-tilting motion.

The British Engineering Standards Association has issued a series (No. 2—1927) of fully dimensioned drawings for ' British Standard ' tramway-rail and fishplate sections, and also (No. 101—1929) for corresponding standard tyre profiles for tramcar wheels. A Report (No. 79—1927) has been issued by the Association on British standard special track-work for tramways.

Fig. 3, page 685, shows a section of a 103·7 lb. (No. 7) rail.

(Expert opinion, and the actual practice of some authorities, now favours the use of a deeper groove, viz. 1¼ in., than is shown in fig. 3.)

There are four sections of rails for straight tracks, with corresponding sections for curved tracks. Their general particulars, and those of fishplates, are as shown by the table below.

The affix C to the ' B.S ' number denotes that the rail is for use on curves, the difference between the straight-track and curve rails being that the groove is ⅛ in. wider and the guard or lip thicker and higher in the latter. All the sections have heads of similar profile as regards the groove and the angle of the rail-table. This profile gives a groove 1⅛ in. wide by 1¼ in. deep, measured from the highest point of the table, which has a radius of 9 ins. The guard or lip is $\frac{3}{16}$ in. below the highest point of the table. In the curve rails the groove is 1⅜ in. wide, and the lip is brought up level with the table.

TABLE OF TRAMWAY RAILS AND FISHPLATES.

' B.S.' Section.	Rails.			Fishplates.	
	Height.	Width of Flange.	Calculated Weight per Lineal Yd.	Weight of Inner.	Weight of Outer.
Number	Ins.	Ins.	Lbs.	Lbs.	Lbs.
6 } 6C }	6½	6¾	{ 96·4 } { 103·2 }	22½	27
7 } 7C }	7	6¾	{ 103·7 } { 109·7 }	26	30½
8 } 8C }	7	7	{ 112·9 } { 119·2 }	25	30½
10 } 10C }	5	6	{ 95·4 } { 101·8 }	10½	18½

BRITISH STANDARD SECTION TRAMWAY RAIL.

'B.S.' Section No. 7.—103·7 lbs. per yd.* (Superseding 'B.S.' Section No. 4.)

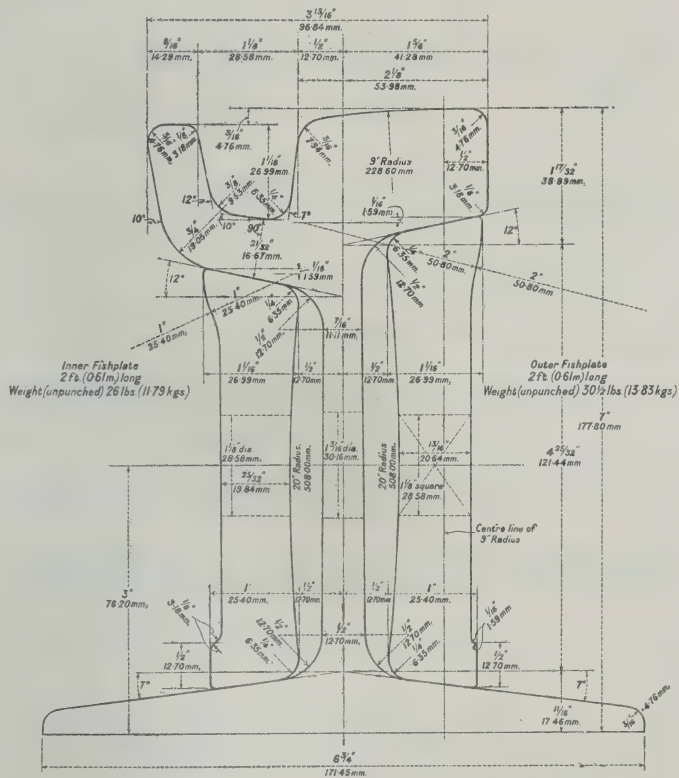


FIG. 3.

* 51·44 kgs. per metre.

Electrical resistance per yard of rail (single) = ·0000277 ohm = ·0487 ohm per mile = 11·8 times resistance of a copper conductor of same section.

The Association has likewise drawn up a Standard Specification for the manufacture of tramway rails, prescribing the chemical constituents, physical tests, etc. The principal provisions are briefly as follows:

Chemical composition.—Steel to conform to following limits:

	Bessemer Process.	Open-Hearth Process.	
	Acid.	Acid.	Basic.
	Per Cent.	Per Cent.	Per Cent.
Carbon . . .	0·50 to 0·60	0·60 to 0·70	0·60 to 0·70
Manganese . . .	Max. 1·0	Max. 1·0	Max. 1·0
Silicon . . .	,, 0·10	,, 0·10	,, 0·30
Phosphorus . .	,, 0·075	,, 0·06	,, 0·06
Sulphur . . .	,, 0·07	,, 0·06	,, 0·06

Manufacturer to furnish carbon and phosphorus determinations of each cast, and a complete chemical analysis for each rolling, or, if a rolling exceeds 100 tons, for each 100 tons or part thereof. Engineer may make independent analyses; in case of difference, samples may be analysed by an independent metallurgist, to be appointed by the Chairman of the B.E.S.A.

The following are abbreviated requirements:

Templates.—Two complete sets of templates are to be furnished, properly engraved with names of purchaser and manufacturer, 'B.S.' number of section, and date of contract.

Limits of variation.—Permissible variations are $\frac{1}{32}$ in. under or over total height, but $\frac{1}{64}$ in. only between fishing-angles.

Lengths.—Standard length of rail to be either 35 ft., 45 ft., or 60 ft., maximum proportion of short lengths being 5 and 2½ per cent. for the 35-ft., 5 and 2½ per cent. for the 45-ft., and 5 per cent. for the 60-ft. lengths.

A variation of ½ in. either way is permitted in the lengths. All short lengths to be painted at each end, the colour indicating the length.

Straightness and squareness.—All rails to be made properly straight and true in all directions when cold, and to be sawn or planed true and square, and burrs removed before leaving the works.

Branding.—A special brand is to be rolled on the web of every rail, to indicate that the rail is of 'British Standard' section and made under the conditions of the Standard Specification, also the number of section, weight, process of manufacture, manufacturer's name or mark, and year of manufacture. The number of cast or blow is also to be stamped near the end of each rail.

Falling Weight test.—A 5-ft. test-piece, head uppermost, is to be supported horizontally on steel bearings 3 ft. 6 in. apart at centres, and a tup, having 5 ins. radius face and weighing 1 ton, dropped on to it, midway between the bearings, from a height of 10 ft. If the rail fractures all from that cast may be rejected.

Tensile test.—From each 100 tons of rails a rail may be selected and a test-piece cut therefrom. It is to show an ultimate tensile strength equal to not less than 45 tons per sq. in., with an elongation of not less than 10 per cent. In case of failure the whole of the 100 tons may be rejected.

Bending.—From each 100 tons of rails a selected rail is to be tested by bending sideways to a curve of 30 ft. radius, and shall not show signs of cracking; if it does, the whole rolled from that cast may be rejected.

Holes in rails.—If required, each rail is to have holes made in it, without extra charge, for fish-bolts, electric bonds, and tie-bars. Those for the tie-bars may be punched, but all others must be drilled.

Notice and facilities for inspection.—Seven clear days' notice of the rolling of the first lot and three days for subsequent rollings is to be given; the inspector is to have access at all working hours to those parts of the mill where work under his inspection is being performed. The manufacturer is to provide all necessary apparatus in complete working order, and to provide all labour and reasonable assistance for making the specified tests. All rails accepted are to be stamped in the presence of the inspector.

Payment by weight.—Rails are to be paid for according to calculated weights before drilling or punching.

Fish-plates.—The chemical composition of the steel, tensile tests, and general conditions as to manufacture of fish-plates, are specified. Accuracy of fit between rails and fish-plates is regarded as of special importance, and sample joints are to be made whenever the engineer or inspector desires to test this.

Bending.—When cold and before being punched, the fish-plates shall be capable of being bent to a right angle round a bar 4 ins. in diameter without showing fracture.

Size.—Length for rail sections Nos. 6, 7, and 8, 24 ins., for No. 10, 18 ins., ends sawn square.

Holes are to correspond with those in the rails. All burrs are to be removed, and if fish-plates have been punched cold they are to be afterwards annealed. They are to be dipped into hot boiled linseed oil, and if required wired up in bundles. Payment is to be according to actual weight after punching.

(Copies of the complete specification and drawings can be obtained (post free, *5s. 4d.*) from the offices of the British Engineering Standards Association, 28 Victoria Street, S.W. 1.)

Notes on Chemical Constituents and Physical Tests of Tram-Rails.

Generally speaking, there are very wide variations in the specified stipulations as to the chemical constituents of steel for tramway rails, particularly with regard to the proportion of *carbon*, which may be regarded as the main hardening constituent, some engineers specifying this to be as low as from 0·28 to 0·40 per cent., while the highest extreme is from 0·60 to 0·70 per cent. It is questionable whether the latter stipulation always meets with full compliance.

Where British ores are employed, there are limitations in practicability of manufacture of very high carbon tramway rails.

If carbon is over 0·55, and especially if combined with high silicon, a somewhat brittle rail is likely to result.

Practical experiments under everyday conditions have shown that high carbon tram-rails have not given as good wear as those with moderate carbon.

Rails vary in hardness among themselves even when manufactured to the same specification and subjected to the same tests, and they show corresponding irregularities in their wear, which also varies greatly according to the situation in which rails are placed.

High carbon rails are specially liable to injury by injudicious thermal treatment in rolling.

Carbon and Manganese, particularly the latter, notably affect the electrical conductivity of steel, which in tramway rails usually varies from one-twelfth to one-fourteenth that of copper.

High manganese tends to coarse crystallisation, with resulting abrasion and flow of metal, or 'rolling out,' under heavy car traffic. This is often the cause of much trouble from hog-backing, &c.

Sulphur, when in moderate excess, is not particularly harmful, as hitherto generally supposed, but an excess of *Phosphorus* tends to brittleness.

A high percentage of sulphur mainly affects the rail-maker, causing red shortness and seamy metal with attendant difficulties in rolling.

In American standard specifications no limit of sulphur is specified.

Copper.—No limit necessary. It has been conclusively proved that it exercises no deleterious influence even when present in greater proportions than usually found in tram-rail steel.

Arsenic and Nickel.—Limiting proportions are occasionally specified, but for present processes of manufacture it is superfluous to mention these elements. There are proposals, however, to employ nickel to a greater extent in rail-steel making.

The processes usually employed in analysing the chemical constituents of steel are not so exact as for the physical tests. There is also considerable liability to personal error. The effects of segregation of the constituents often have an important bearing in the consideration of this question. Carbon is sometimes judged by the colour test, but more exactly by the combustion test.

Physical Tests.—The impact, or drop, test when conducted by a thoroughly competent inspector, is perhaps the most valuable of all tests for ascertaining the general quality and characteristics of the steel in tramway rails.

As deflection is in the nature of a measure of ductility of a rail, and as tramway rails are composed of chemical constituents with a view to securing extreme hardness in the steel—which is not compatible with great ductility—it is not usually considered desirable to exact a deflection test.

Steels of similar chemical constituents, but manufactured by different processes, will give somewhat different results in the physical tests. A high elastic limit is one of the most important properties of tram-rail steel, and can be obtained, combined with toughness, by suitable heat treatment.

The physical tests stipulated by specifications are often incompatible with the required chemical constituents of the steel and the processes to be employed in its manufacture.

The results of the chemical analyses and physical tests should be considered in conjunction.

Inspection.—It is of very great importance that there should always be careful inspection and testing of tramway rails, conducted by a thoroughly competent and reliable inspector, at the rolling-mills, before acceptance of deliveries.

Sandberg Rails.

The use of Sandberg silicon steel (silicon 0·3, carbon 0·5, manganese 1·1 per cent.) increases the life of rails by at least 25 per cent. If steel of the same composition is treated by a special process (Sandberg air-cooled steel), the life is said to be increased by 100 to 200 per cent., so far as experience has been obtained (*R. B. Holt*). The surface of rails can be hardened *in situ* by a patented process which has been adopted on several British tramways: this process diminishes the tendency to corrugation. The Sandberg sorbitic process has also been applied to tram-car tyres. The process needs to be correctly applied in order to obtain the best results.

Rolling-Mill Memoranda.

The Bessemer basic process of steel-making is now generally employed for the manufacture of tramway rails at English and Continental rolling-mills.

Rails are cut, while hot, by circular saw, to specified lengths, with a certain margin for shrinkage in cooling. They should be cut off quite square at the ends; the practice of undercutting below the head should not be permitted.

Very exact or dead lengths require to be planed in an ending or milling machine, and are usually subject to a slight extra charge.

Rails are straightened, while in a cold state, in a press.

45 ft. or 60 ft. are the most usual lengths. With the latter there are fewer joints, and they are generally preferred.

Density of Steel for Girder Tram-rails.

As a result of very careful and complete experiments and tests, undertaken at the instance of the Engineering Standards Association at the National Physical Laboratory, it was found that the average density of the steel in a large number of specimens of English-made tram-rails was 0·283 lb. per cubic inch.

Sectional area of rail in square inches=approximately one-tenth the weight of rail in lbs. per yard.

Rail Joints.

This detail is of the first importance, as the rail joints are the weakest part of the track and determine its life, being also the main factor in the cost of maintenance of the permanent way, and largely affecting the cost of upkeep of the rolling-stock and equipment.

A thoroughly strong and permanently rigid joint will maintain a smooth and quiet running surface, and will greatly lengthen the life of the rails, while assisting the electrical conductivity.

Fishplates alone, even when of the heaviest type, have proved inadequate to withstand the stress from the great vibration and pounding effect experienced from the running of heavy electrical cars, and the addition of flat plates under the rail-base has proved of no permanent advantage.

The very best type of joint should be employed, such as an efficient form of bridge joint. A type frequently adopted is the Cooper patent 'Anchor' joint, which is in the form of an inverted rail or joist riveted to the under side of the rail-base, thereby having the effect of doubling the depth of the rail about its weakest part. This device, in addition to providing an efficient connection between the rail-ends, also securely holds the rails down to the concrete foundation, thus keeping them rigidly in position and preventing springing, creep, etc. It is also of considerable value in providing a path for the electric return current.

Allied to the question of rail-joints is that of fixing down or anchoring the rails to the concrete under-bed, not only at the rail-ends, but at intervals intermediately. When rails are welded together, anchorage at close intervals is certainly a necessary accompaniment. With 'thermit welds,' anchors are placed 3 ft. from joints and then at distances of 13 ft. 6 ins. apart. At Leeds, with the 'thermit' process, the rails are anchored at 8-ft. intervals.

Taking into consideration the increased length of rails, the close-butting of ends, and other circumstances, this course is undoubtedly essential to provide against the effects from expansion and contraction, hog-backing, creep, springing, undulation, vibration, water-pumping, cold-rolling, spreading of gauge, and other causes tending to create movement in the rails.

The distance apart for intermediate anchors or holding-down devices varies from 3 ft. 6 ins., as at Hull, to 30 ft. There should be at least one intermediate with 45-ft. rails, and two with 60-ft. rails.

Rails should not be rigidly connected to each other across the track—as, for instance, by steel cross sleepers fixed under rail-base from rail to rail—as this is liable to cause waviness in the rail-table, due to unequal flexure in the rails.

Worn rails can be ground to shape *in situ* by a travelling apparatus, thus prolonging their life by several years. Worn joints, crossings, etc., can be built up to level by adding metal with the metallic arc.

Foundation.

In the most approved British methods of electric-tramway construction, the base of the rails is bedded on a foundation of concrete, composed of 4 parts of stone, 2 of sand, and 1 part of Portland cement, laid about 20 ins. wider than the gauge of the track, and 6 ins. in depth, a short length of some 3 ft. under the rail-joints being usually made from 3 ins. to 6 ins. deeper and of better quality concrete. The latter should be of clean aggregate mixed with good quality cement having somewhat quick-setting properties.

Sometimes the concrete is first laid, the rails being afterwards set, and packed to the required level with small crushed stone and cement, mixed dry. A more approved method is first to set the rails on blocks to the proper alignment and level, and to fit all the fastenings, afterwards placing the concrete in position under and around the rail-base, care being taken that there shall always be a sufficient 'head' on the upper surface of concrete above the rail-base.

At Hull, a longitudinal timber, 4 ins. thick and the width of rail-base, is interposed between the rails and the concrete. At Aberdeen, Birmingham, and Liverpool, on part of the track, cross-sleepers are used. Sleepers are no longer used in Glasgow.

The hollow on each side of the rail-web should be filled with a 'parging,' or rendering composed of 1 of cement to 4 of sand, laid to a vertical surface to pave against.

Rails should be laid so as to butt quite close at ends.

Paving.

Tar macadam and other forms of well-rolled smooth paving are coming much into use. Reinforced concrete is also being employed.

Points and Crossings.

Points and crossings should be of manganese steel throughout, or, if of the best crucible cast steel, then with specially hardened tongues. Renewable parts are of questionable value. Points should be about 12 ft. in length, having one road curved. The tongues should be at least 6 ins. longer than maximum wheel-base length of cars. Movable points should always be drained. Crossings should be of the 'leg' type, and from 7 ft. to 8 ft. long, according to angle number.

Drain-rails should never be used, but slots 9 ins. long cut in rail-groove at every low point, and trapped drain-boxes fixed and connected to sewer.

Cost of Construction.

Where the overhead running or trolley principle was adopted, the pre-war cost of construction of the permanent way varied approximately between 4,500*l.* and 6,500*l.* per mile of single track; this variation being mainly due to the different classes of paving used. Present prices are about 150 per cent. higher.

A fair general average of proportionate costs over a system may be taken as follows:—Rails and fastenings, including points, crossings and special work, 30 per cent.; paving (sett or wood) including concrete underbed, 45 (to 50) per cent.; labour, including cartage, 20 per cent.; bonding, 2 per cent.; watching and sundries, 3 per cent.

Tractive Resistance on Tramways.

The average pull required per ton of load to keep a car moving is 30 lbs. With exceptionally clean rails, as during a heavy rainfall, and with the rail grooves void of dirt, the pull may be as low as 20 lbs. per ton, or below this figure, approximating to the 10 or 12 lbs. of railway work. Again, a very dirty, clogged rail may demand more than the 30 lbs. Multiplying the pull by the speed in feet per minute gives the power in foot-pounds per minute, so that the horse-power is H.P. $= \frac{T \times W \times S}{33{,}000}$, where T is the pull per ton necessary, W is the load in tons, and S the speed in feet per minute. This rule gives the horse-power on the level. To find the power on a grade, the vertical rise of the car is found by dividing the speed of the car in feet per minute by the length of road in which there is a rise of 1 foot. The number of feet of vertical lift multiplied by the total weight in pounds gives the work done per minute against gravity, or $\frac{S \times W}{F}$, where W is the load, S is the speed, and F is the distance in which the road rises 1 foot. Thus on a 1 in 30 grade, F = 30. The addition of the two results gives the total horse-power. Track resistance is increased at curves, also increased horse-power is required during acceleration. The latter depends not only on weight of car as a whole, but also on moment of inertia of rotating parts.

Tramway Motors.

For a description of modern practice in tramway motor design and construction, see Section XXXV., 'Electrical Engineering.'

Tramway Motor Control.

Connections of Motors in Series-Parallel, with Cut-out of Resistances, on Metropolitan-Vickers Co.'s 'T' Type Tramway Controller.

The method now universally used for controlling the motors on an electric tram-car is that known as the series-parallel system. With this system the two motors are first thrown on to the line in series with one another and with a resistance. This resistance is gradually cut out by the operation of the controller until the two motors are left in series without any resistance in the circuit. The motors are then thrown in parallel with a resistance in the circuit, and when this resistance is finally cut out the motors are in full parallel.

Fig. 4 shows the various combinations as the handle of the controller is moved over.

First Point.—Current passes from the trolley through six sections of resistance, through Armature No. 1, and Field No. 1, thence through Armature No. 2, and Field No. 2 to earth.

Second Point.—Two sections of resistance are cut out.

Third Point.—Two more sections of resistance are cut out.

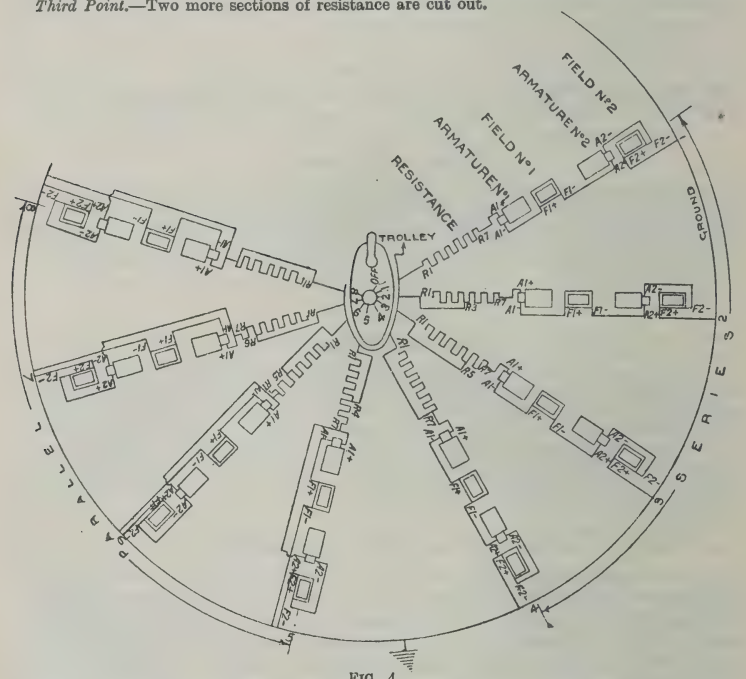


FIG. 4.

Fourth Point.—Two more sections of resistance cut out. The motors are now in full series on the line with no resistance in circuit.

Fifth Point.—Current passes from the line through three sections of resistance and thence to earth through the two motors in parallel. The armature and field of each motor remain of course permanently in series with one another on all combinations of the controller.

Sixth Point.—One of the sections of resistance is cut out.

Seventh Point.—Another of the sections of resistance is cut out.

Eighth Point.—The remaining section of resistance is cut out, leaving the two motors in full parallel on the line with no resistance in circuit.

General Electric Co.'s Tramway Controller Type KAI.

This drum type controller is arranged for the series-parallel control of two motors having a capacity not exceeding 70 B.H.P. each at the one-hour rating. It has four series, four parallels and seven braking notches, thus ensuring even acceleration and retardation. Shunt transition is employed between series and parallel. The connections are so arranged that it is possible to operate with one motor cutout. Braking may be accomplished both by rheostatic control and by electro-magnetic track brakes. The controller connections are arranged to afford an automatic run-back electric brake which comes into operation entirely independently of the action of the motor-man when the main handle is in the 'off' position. This feature prevents the possibility of the car running backwards on an incline. The controller is designed for operation on systems insulated on earth return.

Controller Drums and Handles.—The main handle is permanently fastened to the spindle of the main power drum, the reversing key only being removable. In order to render impossible improper operation of the controller, mechanical interlocking is provided which prevents the reverse spindle being moved unless the main power drum is in the 'off' position. It is impossible to move the main power drum unless the reverser is fully thrown into either the forward or reverse position. Arrangements are also provided which prevent the controller being moved on to the main power notches unless the motor cutout mechanism is positively locked in any one of its three positions.

The copper liners on the power drum are easily removable for replacement, and are provided with separately removable end segments where the arc rupturing takes place.

Contact Fingers.—The main contact fingers are of the compression spring type, the arrangement being such that the springs afford constant pressure between the contact tips and the segments of the drum while permitting of adequate variation in the depth of the stroke. This type of finger, being capable of individual adjustment, permits of the correct pressure being provided for each finger, thus eliminating undue wear and obviating heating of the contacts.

Magnetic Blow-out.—The controller is provided with individual magnetic blow-out coils to each finger. These coils are provided with steel cores which are connected to steel plates embodied in the blow-out shield partitions. These plates form part of the iron of the magnetic circuit for the blow-out coils, and as they extend throughout the greater part of the area of the shield partitions they distribute the magnetic flux between the partitions parallel to the shaft, and consequently at right angles to arcs formed between the fingers and drum segments irrespective of the position the arc may assume. This ensures the complete extinction of the arc under all operating conditions, and thereby reduces to a minimum the burning at the contacts.

Motor Cut-out.—The controller is provided with an external motor cut-out for the purpose of cutting out either motor without the necessity of removing the cover of the controller.

For this purpose an operating knob adapted to receive the reversing key is provided at the back of the controller, and by throwing this either up or down, No. 1 or No. 2 motor, as the case may be, is cut out of circuit, still leaving the other motor capable of being operated both for power and for rheostatic brake.

Weight.—The total weight of the controller complete is approximately 350 lbs., the overall dimensions being 42 ins. × 18 ins. × 9¾ ins. approximately.

BRAKING.

Electric braking consisted first of simple rheostatic braking, the drag on the motor armatures, acting as generators, causing the retarding force necessary. Resistance is inserted at first to prevent excessive currents and too severe strains on the motor windings, etc. This resistance is gradually cut out as the controller handle moves over the braking notches.

Electro-magnetic braking was added later, the braking effect being then partly electrical and partly frictional.

One weak point of electric braking was that, should the car be stopped when going up an incline, there was a tendency to run back, and then the electric braking had no effect unless the reversing handle was moved into the backward position. Omission to do this caused a number of accidents.

In the British Thomson-Houston Co.'s type B.510 controller, electric braking is obtained irrespective of the forward or backward position of the reverse handle or the direction of motion of the car. In the braking position the connections are such that the armature of No. 1 motor is in series with the field of No. 2 motor and *vice versa*.

In the English Electric Co.'s new controller, when the handle is in the off-position, electric braking immediately comes into operation if the car begins to run back.

The Metropolitan-Vickers Electric Co. have developed a new type of contactor-controller.

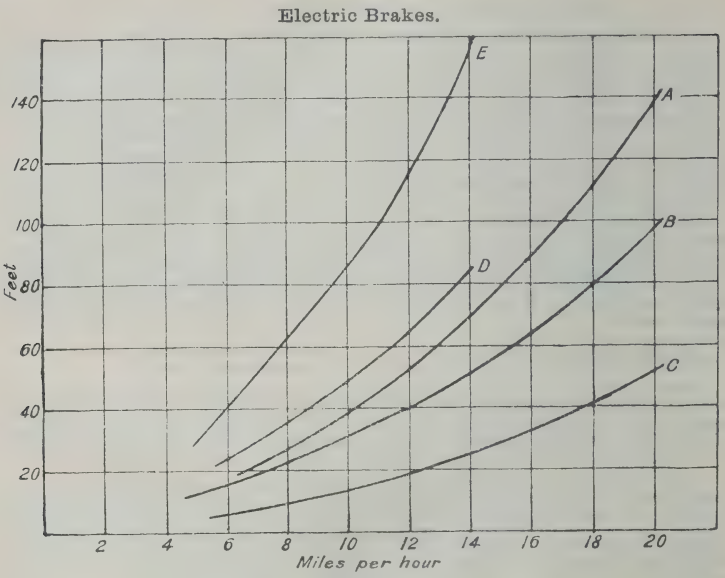


FIG. 5.

Fig. 5 shows the results obtained with tests on tramway cars fitted with various types of electric brakes. Tests A, B, and C were made under the supervision of the Special Committee on Brakes of the Tramways and Light Railways Association, tests D and E were made by Mr. A. L. C. Fell. Tests A, B, and C were made on a four-wheel car weighing 11 tons 9 cwt. loaded. Tests D and E were made on a bogie car weighing 13 tons with 6 persons. In every case the tests were made on a practically level track; in tests A, B, C, and D the rails were dry, in test E the rails were greasy. Tests A and E were made with a rheostatic brake, in tests B, C, and D the brake used was the Westinghouse electro-magnetic track brake, the track shoes being coupled to and acting with the ordinary brake shoes on the wheels. The results of each test are shown by a curve giving the speed in miles per hour at which the brakes were applied and the subsequent travel of the car in feet.

A comparison of curves D and B gives the relative effect of electro-magnetic track brakes acting under practically similar conditions on a bogie car and a four-wheel car. A comparison of curves A and B shows the behaviour of a four-wheel car under the action of a rheostatic brake and of a track brake in the case of ordinary service stops, while curve C shows the effect of a track brake on a four-wheel car in an emergency stop. In all these cases the rails were perfectly clean and dry. Curve E shows what may take place with a heavy car on a flat but greasy rail when controlled only by a rheostatic brake.

ELECTRIC EQUIPMENT OF TRAMWAYS.

Sectioning.

Trolley lines must be divided up into sections not exceeding half a mile in length by means of section insulators. The usual method of feeding the separate sections is to bring the supply cables to a small switch-board fixed in a cast-iron box near the poles carrying the section insulators. The supply cable is connected through a switch to a bus bar from which the four leads to the trolley lines are taken. Another method of feeding is to have a small section pillar in the middle of each section, into which the feeder is brought, and thence to feeder ears on the trolley wire. On the poles carrying the section insulators small iron boxes are fixed, carrying two single-pole switches connected to the section insulators. These switches are normally in the off position, so that in the event of a breakdown in the trolley line it is only necessary to go to one box in the centre of the section instead of two half a mile apart. In the event of failure on a cable, the section switches can be closed and the line run on the trolley wire alone.

All large and complicated junctions are generally isolated by means of section insulators on each line, placed as near the junction as possible, the junction being fed separately. This enables such a section to be isolated by the pulling out of one switch, and repairs can be made quickly and without shutting down the service.

Section boxes should be placed as close as possible to the poles up which the side-feed cables are carried. The simplest method of carrying the side-feeds from the box to the pole consists of a short piece of pipe passing from the section-box to the inside of the pole and held by lock-nuts to the section box. This method enables the cables to be easily withdrawn without disturbing the pavement, and also gets over the danger of the cables lying in water at the foot of the pole, as is frequently the case where they are carried underground.

The side-feed cables generally consist of rubber-insulated flexible-stranded cables, 2,500 megohm grade, and are protected by means of wooden brushes where they enter and leave the poles. The four cables should be tightly taped along their whole length to the span or bracket arm which carries them with 'P. & B.' tape or other weatherproof material. This gives them a neat appearance, and greatly increases the life of the cables. Paper-lead cables should on no account be used for side-feeds, as the lead frequently causes trouble. In some towns bare wires are used instead of rubber-covered cables, but the arrangement has an untidy appearance and possesses no advantages.

The system most in use at the present is known as the single-trolley system, the current being collected from the overhead wires and returned through the rails. In some few cases, where the earth-return current is likely to affect delicate scientific instruments, two overhead wires are carried over each track, one being the feeder and the other the return. This system, however, involves complications at junctions owing to the necessity of extra insulation between the feed and return wires, and the large number of wires required.

Trolleys.

The trolley with outside springs is the most simple form of trolley, as all springs are easily examined and adjusted. The cable is carried down the inside of the trolley arm to the bottom of the standard. The connection to the car leads is made by means of revolving rings or fixed-ring collector and a revolving brush, which allows the arm to make a complete revolution. In the Blackwell trolley with inside springs the springs are entirely enclosed, and are easily adjusted by means of one tension rod. This type has the advantage of taking up less room than the outside spring type.

The Trolley Wheel.

Trolley wheels should be of a good quality metal, but not too hard, as a worn wheel can be more easily replaced than a worn wire. Some trolley lines have been ruined by the manager, or other person responsible for the trolleys, endeavouring to get a high mileage out of the wheels.

The trolley wheel is usually about 4 ins. over all in diameter, of gun-metal or other copper alloy, and is fitted with a bush filled with some graphitic compound. The following are the dimensions recommended by the British Engineering Standards Association (No. 23—1905) as the section of the standard trolley wheel grooves (fig. 6): Width in inches = $1\frac{3}{8}$; depth in inches = $\frac{3}{4}$; radius in inches = $\frac{3}{16}$: angle in degrees = 65°.

Fixed or swivel trolley heads may be used. In Glasgow a fixed head is adopted.

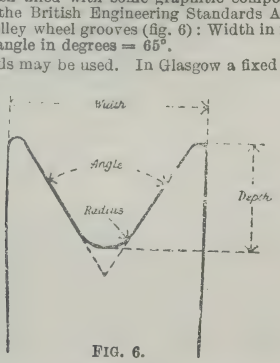


FIG. 6.

Sliding trolley heads are now being tried instead of wheels, with good results. The number of dewirements is greatly reduced, and less damage is done to the overhead equipment on dewirement. But the wear on the trolley wire is about doubled unless the wire is lubricated.

Overhead Construction.

The types generally in use are span, single-side bracket, double-side bracket, and centre pole. The span system is the best from the line point of view, as it allows the greatest flexibility, and also allows the wire to be placed centrally over each track, thus ensuring an even wear on the wire and a smaller tension on the trolleys. It has, however, the disadvantage of requiring standards

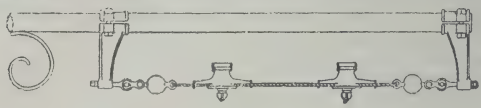


FIG. 7.

on both sides of the streets; and in many cases this is impossible owing to telegraph lines or other causes. Double bracket arms are used in some cases where the lines are laid close to the kerb on both sides and where span wires across the street would be of excessive length.

The single-side bracket is the form of construction most generally in use where economy is of importance and in cases where there are difficulties in obtaining room for structures on both sides of streets. Side brackets up to 22 ft. long have been used in some instances, but such long arms should only be used in very exceptional cases. The Board of Trade Memorandum (see page 700) limits the length of the arms to 16 ft. maximum. Where bracket arms are used, the trolley wire should be suspended by means of flexible suspensions, as in fig. 7. These consist of a short length of stranded-steel wire strained between two suitable malleable cast-iron or wrought-iron brackets bolted to the bracket arm. This length of wire should not be strained too tight, as it then becomes a practically rigid suspension; neither should it be left sufficiently slack for there to be any danger of the line being lifted by a trolley so that the insulator casting comes in contact with the bracket, and so cuts out the second insulation.

Trolley Wire Fittings.

The standard practice is to use double insulation in all cases on trolley lines, the usual method being an insulated bolt fitted into the hangers, the second insulator being at each end of the spans. All spans should be fitted either with one or two strain insulators in order to provide adjustment of the span wires. Triple insulation is now called for by the Board of Trade between any gas or electric lamp bracket attached to pole and trolley wire. Composition insulation is being largely replaced by porcelain. The most usual types of insulators are the straight line hanger and the double and the single pull-off for use on curves. The metal portion of the insulators is either of malleable cast iron or gunmetal, the latter being more expensive in first cost, but cheaper to maintain, as it does not require painting. In the Blackwell type of section insulator it is unnecessary to anchor the trolley wire over the top of the insulator.

The usual method of attaching the trolley wires to the suspension insulator is by sweating them into gunmetal ears. Where sweating is adopted the ears should be accurately milled out to take the wire, and the lips of the ears which clip round the wire should be carefully hammered over before soldering, and the ears dressed off to a feather edge after soldering, great care being taken not to notch the wire where it leaves the ear. In all cases it is absolutely essential that the wire should be held in a strong soldering clamp during the whole process of soldering until the ear is cooled. Unfortunately this practice is insisted on only in cases where the work is carried out by experienced contractors; and in many cases where corporations do their own work the sweating clamp is considered unnecessary, with the result that breakages of the wire at ears frequently occur. The sweating of the ears naturally causes the wire to be annealed, and this also is a cause of breakages.

The most satisfactory arrangement is to have the ears carefully hammered along the wire and the ends dressed off and left unsoldered. For some reason, however, this method is seldom specified, though not infrequently it is adopted in actual construction. In many cases mechanical bars are used. These are made in two halves, bolted together, and grip the trolley wire by means of a V-shaped groove drawn in the wire slightly above the centre line. This method facilitates erection, avoids softening the wire by sweating, and gives a straighter path to the trolley at the ears. It has, however, the disadvantage that the wire has to be annealed in order that the groove may be drawn into it, and the breaking strain of a grooved wire is less than that of a circular wire; there is also a tendency for the groove to turn over between the points of support, causing the wheel to run badly and frequently to leave the wire on curves.

Guards should be fixed at all frogs and on both sides of all crossings to prevent the trolleys jamming in the angles, as, in addition to their being a great safeguard, they help to preserve the wire, as the wear is taken up by the mechanical ears which support the roller bars, and these are easily renewable.

Poles.

Poles are usually 31 ft. long, and are planted 6 ft. in the ground, sufficient rake being allowed in erection so that the pole stands nearly vertical with a slight backward rake when the span and trolley wires are on. The amount of rake for a 36-ft. road in good ground should be about 6 ins., but the actual rake required varies with every class of ground, and can only be successfully estimated by experienced men. The holes should have a sufficient diameter to allow of six inches of concrete round the pole in good ground; this must be increased in bad ground or at places where additional loads are imposed due to bridles, extra long spans, etc. The poles should be planted on circular discs of concrete, or on flagstones or cast-iron plates dropped into the holes to prevent the poles cutting into the ground during erection. The holes should be filled in with 6 to 1 concrete, well rammed.

In order to prevent corrosion, all poles should be coated inside and for a distance of 8 ft. from butt outside with hot tar or Angus Smith composition.

In some coast towns corrosion, and a consequent loosening at the joints, sometimes occurs.

One-piece poles or poles welded smoothly at joints are being adopted, as in Edinburgh and Aberdeen.

In the specification for tramway poles, the light pole is suitable either for short side brackets or centre pole work, the medium pole for long brackets or span construction, and the heavy pole for anchors and pull-off poles. Three-section poles are generally used, as they are cheaper and more reliable than taper poles. Owing to the irregularity of the taper, the fitting of brackets, pole straps, etc., is extremely difficult with taper poles, and their appearance when erected hardly differs from the three-section pole.

Specification for Poles.

BRITISH STANDARD SPECIFICATION FOR TUBULAR TRAMWAY POLES.*

(No. 8—1927.) (*Abstract.*)

1. The poles shall be of wrought steel free of all defects, and shall be of four classes: light pole, medium pole, heavy pole, and extra-heavy pole.

2. The poles may be made of three separate sections (sectional poles) swaged together when hot so as to make a perfect joint; or they may be made in one piece with two reduced parallel steps (stepped poles). Poles of both types shall be made either by the hot-rolled weldless process or the lap-weld process, of steel of a tensile strength of not less than 24 tons nor more than 42 tons per square inch. The lap-welded seams in the sections shall be set at an angle of one hundred and twenty degrees (120°) to each other.

3. The overall length of poles of all classes shall be 31 ft.

4. The length of the telescope joint in the sectional poles shall be 18 ins., and the upper edge of each step shall be chamfered off at an angle of not less than 30°.

5. The visible length of the sections or stepped portions shall be :—

Top section	7 ft.
Middle section	7 ft.
Bottom section	17 ft.

6. The outside diameters of the four classes of poles shall be as follows :—

Class.	Top Step or Section.	Middle Step or Section.	Bottom Step or Section.
Light	5½ ins.	6½ ins.	7½ ins.
Medium	6½ ins.	7½ ins.	8½ ins.
Heavy or Extra-Heavy	7½ ins.	8½ ins.	9½ ins.

The variation in diameter shall not exceed $\frac{1}{16}$ in.

* By permission of the British Engineering Standards Association.

7. The minimum thickness of metal in any pole shall be as follows :—

SECTIONAL POLES.

Class.	Top Section.	Middle Section.	Bottom Section.
Light	$\frac{4}{32}$ in.	$\frac{4}{32}$ in.	$\frac{5}{16}$ in.
Medium	$\frac{5}{32}$ in.	$\frac{5}{32}$ in.	$\frac{11}{32}$ in.
Heavy	$\frac{7}{16}$ in.	$\frac{11}{32}$ in.	$\frac{13}{32}$ in.
Extra-Heavy	$\frac{9}{16}$ in.	$\frac{11}{16}$ in.	$\frac{9}{16}$ in.

STEPPED POLES.

Class.	Top Step.	Middle Step.	Bottom Step.
Light	$\frac{5}{16}$ in.	$\frac{5}{16}$ in.	$\frac{5}{16}$ in.
Medium	$\frac{11}{32}$ in.	$\frac{11}{32}$ in.	$\frac{11}{32}$ in.
Heavy	$\frac{13}{32}$ in.	$\frac{13}{32}$ in.	$\frac{13}{32}$ in.
Extra-Heavy	$\frac{9}{16}$ in.	$\frac{9}{16}$ in.	$\frac{9}{16}$ in.

The tolerance shall be 5 per cent. under the above thicknesses.

8. The completed poles shall be straight and true over their entire length to within $\frac{1}{4}$ in.

9. Five per cent. of each class of sectional pole shall, if required, be subjected to the following drop test :—

The pole shall be dropped vertically, butt downwards, three times in succession, from a height of 6 ft. on to a hard wood block 6 ins. thick laid on a concrete foundation, without showing any signs of telescoping or loosening of joints.

10. Five per cent. of each class of pole shall be subjected to the following bending tests :—

The pole shall, in each case, be fixed horizontally and rigidly supported for 6 ft. from the butt, and loaded, as a cantilever, at a point 18 ins. from the top, the load being applied at right angles to the axis of the pole. Upon the application of the following loads, the temporary deflection and permanent set, measured at the point of application of the load, shall not exceed the figures stated in the table.

Class of Pole.	Load in lbs. for temporary deflection not exceeding 6 ins.	Load in lbs. for permanent set not exceeding $\frac{1}{2}$ in.
Light	750	1,000
Medium	1,250	1,750
Heavy	2,000	2,500
Extra-Heavy	2,750	3,250

11. In the event of any pole of the above-mentioned 5 per cent. not fulfilling the test requirements, a further 5 per cent. shall be subjected to the tests enumerated. Should any further failure occur, the whole parcel from which the poles have been selected shall be liable to rejection.

APPROXIMATE WEIGHT OF POLES.

The weight of the poles referred to in the Specification may be assumed to be approximately as follows :—

	Light.	Medium.	Heavy.	Extra-Heavy.
Sectional Poles	700 lbs.	872 lbs.	1,130 lbs.	1,437 lbs.
Stepped Poles	674 lbs.	851 lbs.	1,132 lbs.	1,538 lbs.

Trolley Wire.

Trolley wire is usually 0 (·324 in.), 00 (·348 in.), or 000 (·372 in.) S.W.G., of hard drawn copper; conductivity should be 98 per cent. of pure copper; it should bear an elongation of 1½ per cent. in 10 inches, a breaking strain of 25 tons per square inch, and 20 twists in a length of 12 inches. Copper cadmium is also being used. Its conductivity is about 93 per cent. of Matthiessen's Standard and its breaking stress about 30 tons per square in. At curves and places where wear is severe, phono-electric wire is used; its conductivity is, however, low, usually 50 per cent. to 60 per cent. of Matthiessen's Standard.

The Engineering Standards Association recommends that the tensile breaking strength should be not less than 24 tons per square inch up to No. 00 S.W.G., and not less than 22 tons above that gauge.

If

L = distance in feet from pole to pole; l = actual length in feet of one span of trolley wire; d = dip in feet; T = tension in lbs. in trolley wire at point of support; w = weight in lbs. per foot of wire; then

$$d = \frac{L^2 w}{8T}\ ;\ T = \frac{L^2 w}{8d}\ ;\ l = L + \frac{8d^2}{3L}\ ;\ d = \sqrt{\frac{3Ll - 3L^2}{8}}.$$

If

e = expansion coefficient per F° = ·00001; $c°$ = change of temperature; then $l \pm c°e$ gives the new length on change of temperature = $c°$, from which the new dip d may then be calculated, or the tension T for any dip. In these formulæ the effect of wind pressure is neglected.

The lowest point of the wire should not be less than 17 feet from the road surface.

Span Wire.

The steel wire rope to form spans for carrying trolley wire is usually formed of a galvanised strand of 7 No. 12 S.W.G. or No. 7 No. 15's. The 7/12 is usually specified to bear a breaking strain of about 4,600 to 5,000 lbs. When the 7/15 wire is used it is of much harder material, and weighs 225 lbs. per 1,000 yards. The only objection to the small wire of high tenacity is the difficulty of forming the eyes over the thimbles and nozzling the ends.

Guard Wire.

Guard wire when single is No. 8 S.W.G. steel, and weighs 215 lbs. per 1,000 yards, but it is often of stranded steel. Guard wire brackets in their plainest form are T's of ¾ in. iron pipe clipped on bracket arms. In span systems guard wire is carried by special guard wire spans. The Board of Trade have an explanatory memorandum on 'Guard Wires on Electric Tramways.'

Strength and Conductivity of Wires.

Material.	Conductivity.	Breaking strain, Lbs. per Sq. In.
Commercial copper	98–99	33,000
Hard drawn „	96–97	60,000–65,000
(1) Silicon bronze	97	63,200
(2) „ „	80	76,000
(3) „ „	45	110,000
Phosphor „	26	101,000
Annealed iron	14	55,000
High carbon steel	10–12	120,000–130,000

Rail Bonding.

Except in those comparatively rare cases where a double trolley wire is employed, the rails are made use of as return conductor, and means have to be taken to connect adjacent rails so as to provide an unbroken electrical conductor. To effect this object the rails are bonded with copper, various types of bonds being employed.

The 'Chicago' Bond.

Fig. 8 shows the general design of this bond, which consists of a copper rod having tubular or thimble-shaped terminals bent at right angles to the rod, the whole being composed of one solid piece of copper. The terminals are inserted into holes in the web of the rail, and the slit in the end of the terminal is spread or clinched over the rail with a hammer and punch. A drift

pin, larger in diameter than the opening in the terminal by about one-sixteenth of an inch, is then driven in. This permanently expands the terminal and wedges it into contact with the surface

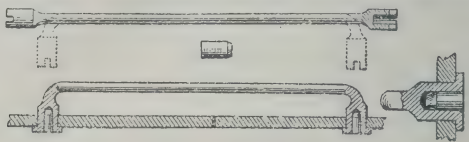


FIG. 8.—Chicago Bond.

of the hole in the web of the rail. The usual length of bond is 30 ins. The depth of the hole in the terminal in all sizes of bond is 1 inch. The size of conductor used depends upon the weight of rail and the current to be carried.

TABLE SHOWING STANDARD DIMENSIONS OF CHICAGO RAIL BONDS.

Size of Wire, B. & S. Gauge.	Diameter of Hole in Rail into which Terminal of Bond fits.	Diameter of Hole in Terminal.	Sectional Area of Rail Bond.	Diameter of Pin.
	Ins.	Ins.	Sq. Ins.	Ins.
0000	$\frac{7}{8}$	$\frac{7}{16}$	·166	$\frac{1}{2}$
000	$\frac{3}{4}$	$\frac{3}{8}$	·132	$\frac{7}{16}$
00	$\frac{5}{8}$	$\frac{5}{16}$	·104	$\frac{3}{8}$
0	$\frac{1}{2}$	$\frac{1}{4}$	·083	$\frac{5}{16}$

The 'Crown' Bond

FIG. 9.

This is a development of the 'Chicago' bond, the conductors being stranded to afford the flexibility necessary for expansion in a conductor of considerable cross-section dimensions. Fig. 9 shows two of the many different designs in which this bond is made. The terminals have the same dimensions as those in the corresponding 'Chicago' bond.

The 'Neptune' Rail Bond.

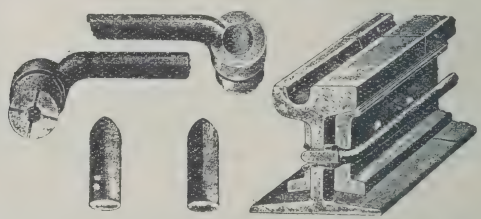


FIG. 10.

Another form of bond, the 'Neptune,' fig. 10, is similar to the 'Chicago,' but the drift pin is driven in the same direction as the bond is inserted, thus safeguarding the bond against being driven out.

The 'Columbia' Rail Bond.

A feature in this bond (fig. 11) is the few parts of which it consists. When installed, owing to the pressure exerted between the head and the thimble, and also the fact that they are of the same kind of metal, the two become one, both electrically and mechanically. It therefore has

all the advantages which may be claimed for a one-piece bond, and at the same time can be satisfactorily installed as no riveted bond can be. The contact of the rail and bond is made by a wedge expanding the thimble against the hole in the rail. By using a press, exactly the right pressure is exerted. One blow of a hammer too many or too few frequently spoils contact. As

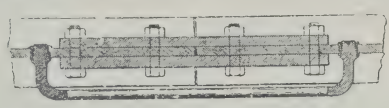


FIG. 11.

the bond is wedged both ways it cannot get loose, as, owing to the fact of its being pressed in, it is sure to make a perfect contact, and cannot be so over-stretched by hammering as to be ready to fall out at the end of a few weeks. It is essential that the hole in the rail should be clean, and the copper bright also. Bonds ought to be inserted at once after the rails are drilled, no time being allowed for rusting to commence.

WELDED BONDS.

The best of all methods of attaching bonds to rails is by electric welding. On this system short bonds can be used, welded into the head or base of the rails outside the fish-plates, or to the webs inside them; the cost of the bond and its attachment is small, and the high conductance of the joint is permanently ensured. Welding bonds to rail is found to weaken the web of rail slightly.

Rail Welding.

Tramway rails are united into continuous lengths by welding. There are several systems in use for welding rails.

THE FALK CAST WELD.

This consists in casting round the cleaned ends of the two rails a mass of cast iron from a portable cupola. Being buried up to their top table, welded rails are held against contraction from cold, and the full stress does not come upon the weld. Expansion in hot seasons is impossible, and the rails are held against lateral buckling by the support of the paving.

ACETYLENE WELDING.

Acetylene welding by the D.A. system (*The Dissolved Acetylene Co., Ltd.*, now trading as *Allen-Liversidge Ltd.*) has been made considerable use of. The method consists essentially in reinforcing the joints by means of a pressed steel shoe placed on the flange of the rails at the joint and welded to the same so as to constitute a hollow beam which increases the moment of inertia about the joint, and correspondingly diminishes the bending caused by the passage of rolling stock. Many joints of this type have been fitted in France and Algiers and in South America.

THERMIT WELDING.

In Thermit welding a mould is fitted round the rails at the joint, and into it is run molten metal from a conical crucible, which is first filled with a mixture of powdered aluminium and iron oxide. The two are intimately mixed, and heated at a point by a special ignition powder. At the temperature produced by igniting this powder, the aluminium reacts very energetically with the oxygen in the iron oxide. The reaction spreads rapidly throughout the mass in the crucible, and produces a quantity of pure superheated liquid metal, covered with a slag of alumina. The temperature set up by the reaction has been ascertained to be about 5,400° Fahr. The crucible is a lined sheet-iron cone. It is suspended over the joint to be welded, and when the reaction is complete it is tapped, the metal running into the mould, followed by the superheated liquid slag, which covers the head of the rail, and raises it to a welding temperature. The rails are then drawn together by suitable clamps, with the result that the base and web of the rails are welded by fusion with the liquid Thermit metal, and the head is butt-welded.

For extra high-carbon rails, the difficulty of butt-welding has been overcome by the use of a special shim, which is inserted between the rail-ends, and ensures a perfect butt-weld at the head.

Thermit welding is often used on new work, partly because electric power is not available.

ELECTRIC WELDING.

Electric welding has been very widely used, especially in the United States. Two methods have been developed—spot-welding and arc-welding. In the former, bars are placed on both sides of the rails at the joint, and are welded to the webs of the rails by heating them with a

very strong current at low pressure, obtained from a transformer, after which the parts are tightly pressed together. These welds are made at several points along the bars. In the latter process, steel plates are welded to the head and foot of the rails with the electric arc, taking the place of fish-plates. In both cases additional bonds are unnecessary, but one 'Neptune' rail bond per joint is often provided. Manganese steel rails can be successfully welded with the electric arc.

(See also 'WELDING,' Section XL., Part II.)

Tramway Track Leakage.

Experiments made by the U.S. Bureau of Standards show that tramway tracks laid on concrete have a low resistance to earth—from 0·2 to 0·5 ohm per 1,000 ft. of single track. With crushed stone ballast under a concrete road-bed the resistance is about three times as great, while with crushed stone ballast with 'Tarvia' finish the resistance is 2 to 3 ohms. Track laid on wooden sleepers embedded in earth has a resistance of 1 to 1·5 ohms. Open track on sleepers laid on good stone ballast has a resistance of 10 to 15 ohms or more in dry weather, 3 to 5 in wet weather, per 1,000 ft. of single track. Chemical salts used to preserve the sleepers greatly reduce the resistance.

Cables.

In English tramway work the cables must be buried. There are four systems in use for buried cables. First, the system of drawing into buried cast-iron pipes. In this system spigot and socket pipes are employed, usually 2 ins. and 2½ ins. inside diameter. A 2-inch pipe 6 ft. long will weigh 48 lbs., a 2½-inch pipe 9 ft. long will weigh 81 lbs. About 2 to 2½ lbs. of lead are required to pour each joint. The pipes terminate in buried brick chambers in which they are finished with trumpet-shaped ends to facilitate drawing in. On curves the pipes may be slightly curved with advantage, but it is usual to lay the pipes round the curve in an equiangled polygon and to use manhole boxes at such intervals that no length of pipe has a versed sine above 12 ins. Soapstone powder is used as a lubricant. The brick chambers should be built on a good concrete bed with a cement collar joint, and they may advantageously be tarred over outside to render them waterproof. The pipes are bonded across the boxes with a stout iron wire to render them electrically continuous; the boxes have a cast-iron top frame with luted cover. These boxes must be placed at such distances apart that excessive stress is not brought upon the cables when hauling them in. The cables should be gripped by their cores. In place of pipes earthenware conduits are often employed of rectangular or hexagonal shape, in short lengths and cemented at the joints, also terminated in brick chambers. In the solid system the cables are laid in troughs of wood or sheet steel, and melted bitumen is poured over them so as to completely enclose them in a casing of bitumen. Cables are sometimes simply buried in the earth. The best cables are insulated with paper and sheathed in lead about $\frac{1}{12}$ in. to $\frac{1}{5}$ in. thick, squirted on under pressure. In whatever way laid the cables are brought up in their respective casings through the base of the section box, in which they are connected to the bus bars by switches, so that the cables may be switched out of the circuit in lengths of about half a mile. In the section box are placed the switch terminals of the trolley feeders, which can also be cut out of circuit in half-mile lengths. The trolley feeder cables are carried up inside the next pole, and in the Booth-Dickinson system, brought through the top finial, then down the pole and along the span wire or bracket arm to the trolley wire terminals on the section insulator. Test and telephone wires are usually laid up in one cable and similarly laid in pipes, etc. Sometimes a telephone wire is carried overhead on insulators fixed to the poles. In laying cables and making joints the utmost care is necessary not to expose cable ends to the atmosphere excessively. Tents must be used in wet weather, but it is better to do the work in dry seasons. Cable ends must be well stoppered. It is preferable to have the cable ways under footpaths. The topmost pipe must not be nearer the surface than 18 ins. when laid in the road. Cast-iron pipes or sheet steel conduit must be dipped in Angus Smith compound of a non-brittle variety when cold, and cast-iron pipes should be dipped until they themselves are as hot as the boiling compound. The lead sheathing of cables must be electrically bonded past the manhole boxes and connected to the rails at the nearest point to the power station.

In Glasgow, cables are drawn into ducts, usually fire-clay; sets of 3 or 6. Ducts are laid in the 4-ft. way between the tracks. Manholes are not more than 400 ft. apart. All solid laid cables are gradually being replaced by duct-laid.

ABSTRACT OF BOARD OF TRADE MEMORANDUM, REGARDING DETAILS OF CONSTRUCTION AND EQUIPMENT FOR TRAMWAYS AND LIGHT RAILWAYS LAID ON PUBLIC ROADS.*

Clearance.

There should be at least 15 ins. between the sides of passing cars and also a similar space between the side of a car and any standing work, such as lamp, telegraph and trolley wire posts in a street.

* The functions of the Board of Trade are now exercised by the Ministry of Transport.

There should be at least 15 ins. between the side of a car and the kerb, whether on straight or curved roads.

The clearance between the top deck of cars and the underside of bridges should not, if possible, be less than 6 ft. 6 ins. Where this clearance cannot be obtained special precautions in working will be required, but in no case will less than 6 ft. be accepted with uncovered cars.

Posts and Brackets.

Where bracket arms 16 ft. in length will not suffice, it is desirable that span wire construction should be used.

The overhead conductors used in connection with the working of the tramways should be securely attached to supports, the intervals between which should not, except with the approval of the Board of Trade, exceed 120 ft., and the overhead conductors should be in no part at a less height from the surface of the street than 20 ft., except where they pass under bridges.

The overhead conductors should be divided up into sections not exceeding (except with the special approval of the Board of Trade) one-half of a mile in length, between every two of which should be inserted an emergency switch so enclosed as to be inaccessible to pedestrians.

Poles supporting gas lamp brackets should be efficiently bonded to the rails.

Permanent Way.

The weight of rails should not be less than 90 lbs. per yard, 100 lbs. being preferred.

The groove of the rail should not exceed 1⅛ in. in width, but a groove not exceeding 1¼ in. will be accepted on curves of less than 150 ft. radius.

Cars.

Of existing types of lifeguard, the 'trigger' is the pattern which is preferred. The hanging gate should be as close to the ground as possible, and there should be at least 3 ft. between it and the front of the guard. Both the guard and the gate should be at least as wide as the outside of the frame of the truck. The tray of the guard should be provided with a spring so as to hold the front edge down to the surface of the roadway when the tray is dropped.

All railings should be connected with earth.

All electrical conductors fixed upon the cars in connection with the trolley wheel should be formed of flexible cables protected by indiarubber insulation of the highest quality and additionally protected wherever they are adjacent to any metal so as to avoid risk of the metal becoming charged.

The trolley standard of every double-deck car should be electrically connected to the wheels of the car in such manner as either to prevent the possibility of the standard becoming electrically charged from any defect in the conductors contained within it or in the event of the standard becoming electrically charged to give a distinctive and continuous warning signal to the driver or conductor. The signal may be a red lamp bridged with a suitable fuse or (preferably) an audible signal. The signal device should be tested at least once a week.

In the Board of Trade Memorandum dealing with guard wires on electric tramways the following regulations are laid down :—

Each guard wire must be well earthed at one point at least, and at intervals of not more than five spans. The resistance to earth must be such that a short circuit between a guard wire and a trolley wire will open the circuit breaker.

The earth connection should be made by connecting the support to the rails by means of a copper bond. When first erected, the resistance to earth of the guard wires should be tested, and periodical tests should be made to prove that the earth connection is efficient.

Guard wires should be, in general, of galvanised steel, but in manufacturing districts in which such wires are liable to corrosion bronze or hard drawn copper wires should be used.

The gauge of the guard wire must be such that it will carry, without fusing, a current 50 per cent. greater than that required to open the circuit breaker, and the guard wire must be maintained in such a manner that it will at all times meet these requirements.

The supports for the guard wires should be rigid and of sufficient strength for their purpose, and at each support each guard wire must be securely bound in or terminated.

The rise of the trolley boom should be so limited that if the trolley leaves the wire it will not foul the guard wires.

A Summary of the Board of Trade Regulations for Electric Tramways.

The following is a brief summary of the principal regulations affecting electric traction :—

When any rails on which cars run, or any conductors laid between or within 3 feet of such rails, form any part of a return, such part may be uninsulated. All other returns, or part of a return, shall be insulated, unless of such sectional area as will reduce the difference of potential between the ends of the uninsulated portion of the return below 7 volts.

Where any uninsulated conductor laid between or within 3 feet of the rails forms any part of a return, it shall be electrically connected to the rails, at distances apart not exceeding 100 feet, by means of copper strips having a sectional area of at least $\frac{1}{10}$ of a square inch, or by other means of equal conductivity.

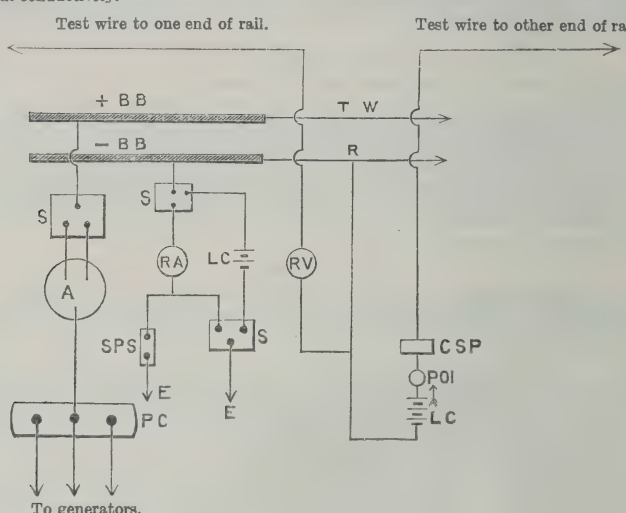


FIG. 12.—Diagram of Board of Trade Switchboard.

A, Ammeter reading $\frac{1}{10}$ — 3 and $\frac{1}{2}$ — 10 as.; B B, Bus bars; R, Return; T W, Trolley wire; S, Two-way switches; R A, Recording Ammeter 2 to 25 as.; R V, Recording voltmeter 2 to 10 v.; L C, Leclanché cells; S P S, S.P. switch; E, Earth; P C, Three-way plug connections; P O I, P.O. indicator; C S P, Contact of sensitised paper.

All uninsulated returns shall be connected to the negative pole of the generator; the negative terminal of the generator shall be also connected through a current-indicator, mentioned below, to two separate earth connections, which shall be placed not less than 20 yards apart. This clause enacts that the two earth connections may be replaced by making one connection to a water-pipe of not less than 3 inches internal diameter, provided that permission be obtained from the owner of the water-pipe and the person supplying the water.

The earth connections referred to in this regulation shall be constructed, laid, and maintained so as to secure electrical contact with the general mass of earth, and so that an electro-motive force not exceeding 4 volts shall suffice to produce a current of at least 2 amperes from one earth connection to the other through the earth, and a test shall be made at least once in every month to ascertain whether this requirement is complied with.

When the return is partly or entirely uninsulated the company shall, in the construction and maintenance of the tramway, (*a*) so separate the uninsulated return from the general mass of earth and from any pipe in the vicinity, (*b*) so connect together the several lengths of the rails, (*c*) adopt such means for reducing the difference produced by the current between the potential of the uninsulated return at any one point and the potential of the uninsulated return at any other point, and (*d*) so maintain the efficiency of the earth connections specified in the preceding regulations as to fulfil the following conditions, viz. :—

(i.) That the current passing from the earth connections through the indicator to the generator shall not at any time exceed either 2 *amperes per mile of single tramway line or* 5 *per cent. of the total current output of the station.*

(ii.) That if at any time and at any place a test be made by connecting a galvanometer or other current indicator to the uninsulated return and to any pipe in the vicinity, it shall always be possible to reverse the direction of any current indicated by interposing a battery of three Leclanché cells, connected in series, if the direction of the current is from the return to the pipe, or by interposing one Leclanché cell if the direction of the current is from the pipe to the return.

In order to provide a continuous indication that the condition (i.) is complied with, the company shall place in a conspicuous position a suitable, properly connected, and correctly marked current indicator, and shall keep it connected during the whole time that the line is charged.

The owner of any such pipe may require the company to permit him at reasonable times and intervals to ascertain by test that the conditions specified (ii.) are complied with as regards his pipe.

When the return is partly or entirely uninsulated, *a continuous record shall be kept by the company of the difference of potential during the working of the tramway between the points of the uninsulated return furthest from and nearest to the generating station. If at any time such difference of potential exceeds the limit of 7 volts, the company shall take immediate steps to reduce it below that limit.*

The current density in the rails shall not exceed 9 amperes per square inch of the cross-sectional area.

The insulation of the line and of the return when insulated, and of all feeders and other conductors, shall be so maintained that the leakage current shall not exceed one-hundredth of an ampere per mile of tramway. The leakage current shall be ascertained daily before or after the hours of running when the line is fully charged. If at any time it should be found that the leakage current exceeds *one-half* of an ampere per mile of tramway, the leak shall be localised and removed as soon as practicable, and the running of the cars shall be stopped unless the leak is *localised and removed within 24 hours.*

The insulation resistance of all continuously insulated cables used for lines, for insulated returns, for feeders, or for other purposes, and laid below the surface of the ground, shall not be permitted to fall below the equivalent of 10 megohms for a length of one mile.

List of British Standard Specifications Referring to Tramway Work.

No.	*Title.*
2—1927.	Tramway Rails and Fishplates.
8—1927.	Tubular Tramway Poles.
23—1905.	Trolley Groove and Wire.
68—1914.	Steel Conductor Rails (Resistance of).
79—1927.	Special Trackwork for Tramways.
101—1929.	Tramway Tyres.
102—1930.	Tramway Axles.
149—1922.	Wrought-Iron Wheel Centres for Electric Tramway Cars.
150—1922.	Cast-Steel Wheel Centres for Electric Tramway Cars.
173—1928.	Electrical Performance of Traction Motors.
235—1927.	Gear Wheels and Pinions for Electric Tramways.

RAILLESS ELECTRIC TRACTION.

Where the traffic conditions do not justify the construction of an electric tramway, but where electric power is available, railless traction provides an alternative to the motor omnibus. Two overhead conductors are provided, as lead and return, and the vehicles carry trolley poles with under-running trolleys collecting the current from the conductors. These trolleys permit the vehicle to deviate at least 15 ft. from the centre-line on either side, so that they can pass other vehicles; on meeting a similar vehicle moving in the opposite direction the trolleys of one must be removed from the conductors whilst the other passes. Railless tramcars compared with motor omnibuses have higher acceleration, lower working costs, longer life, and greater elasticity as regards the number of passengers carried, while they give a more reliable service and a greater annual mileage, and can climb hills with greater ease. The cost of maintenance is about 1*d.* per car mile (exclusive of tyre renewals, which amount to about 1*d.* also), and the consumption of energy is about 1 kw.-hr per car mile. The life of a trolley omnibus is ten to fifteen years, *i.e.* about double that of a motor omnibus. The vehicle costs about 2,000*l.*, and the overhead construction about 2,700*l.* a mile. The seating capacity of a single-deck car is from 28 to 36, and the speed 16 miles per hour. Double-deck cars have been built to carry 59 passengers. Many installations of this class are at work in this country and abroad.

Mr. E. M. Munro states that the approximate cost of energy per passenger mile can be ascertained from the following data :—

Type of Car.	Energy per Car Mile.	Average Passengers carried per Mile on 38% Seats.	Cost of Energy per Car Mile at 1½d. kw.h.	Average Speed m.p.h.	Cost of Energy per Passenger Mile.
Railless single decker .	1·00 kw.h.	11	1·5*d.*	8·0	0·136*d.*
„ double decker	1·66 kw.h.	20	2·49*d.*	8·0	0·125*d.*
Tramcar . . .	2·00 kw.h.	24	3·0*d.*	7·5	0·125*d.*
Petrol Omnibus (petrol) .	0·166 gals.	13	4·359*d.*	8·0	0·335*d.*

The above results are averaged from a number of examples and types examined for this purpose.

The following municipalities operate railless traction vehicles to a greater or less extent:

Aberdare U.D.C., Birmingham, Bradford, Halifax, Ipswich, Keighley, Leeds, Ramsbottom U.D.C., Rotherham, Tees-side Railless Traction Board, West Hartlepool, Wolverhampton, York.

Industrial Electric Trucks.

For moving material about railway stations, factory premises, etc., electric battery trucks have been widely adopted, capable of carrying up to 5,000 lbs. and running at speeds up to 8 miles per hour; the net weight of the trucks is 1,600 to 2,500 lbs., and the mileage per charge 15 to 25 miles. In a railway warehouse 11 tractors and 600 trailers handled 130,000 tons per annum at an average cost of 15·1*d.* per ton. (*R. J. Mitchell.*)

Electric Battery Vehicles.

Useful data on electric battery vehicles was given by Mr. C. W. Marshall at the I.M.E.A. Convention in Glasgow, 1926, based on the experience of the Glasgow Corporation Electricity Department since 1913 with almost every available type of vehicle. The present position is that the lead battery is in favour, due in a great part to the fact that it is much cheaper in initial cost and at the same time gives service at least as good as that obtained from the alkaline battery. The question of the relative merit of thin and thick plate lead batteries is still an open one. The thin plate battery lasts only 12 to 15 months and costs about 60 per cent. of the price of the thick plate battery, which lasts 30 to 36 months. It is, however, possible to get a much greater capacity for a given weight and volume with the thin plate battery, and so in some cases the vehicle becomes much more useful. For example, vehicles in the service of the Glasgow Corporation Cleansing Department which could do only one shift on refuse collection when fitted with thick plate batteries can do two shifts with thin plate. Alkaline batteries cost from 2 to 4 times as much as the best lead batteries. Although they last at least 5 or 6 years under hard service conditions, and 8 or 10 years under easier conditions, their low watt-hour efficiency and high internal resistance counteract the long life to some extent. Battery data is summarised in Table I.

TABLE I.—BATTERY DATA FOR ELECTRIC VEHICLES.

Type of Battery	Alkaline.	Lead.	
		Thin Plate.	Thick Plate.
Watt-hour efficiency in per cent.	59	75	75
Watt-hours per cubic inch	0·74	0·8	0·6
„ „ „ pound	15	12	9
Approx. initial cost in £ per kw.h.	30	4	8
Life in years	6–8	1	3
Life in cycles of charge and discharge	1800–2400	300	900

The capital and operating costs of plain electric lorries of the very best quality of load capacities of 0·5 to 5 tons are given in Table II. The total distance covered is taken to be 10,000 miles per annum and the cost of electrical energy to be 0·75*d.* per unit. The daily distance covered by a battery vehicle should not exceed 35–45 miles.

Mr. Marshall also gives the following data concerning the costs of competing transport systems. For a horse-drawn vehicle, the annual operating costs work out at 239*l.*, with an average cost per vehicle mile of 10·7*d.* A comparison between petrol, steam, and electric vehicles can be obtained from Table III.

TABLE II.—FIRST COSTS AND RUNNING COSTS OF ELECTRIC LORRIES.

	Load Capacity in Tons . .	0·5	1	2	3·5	5
		£	£	£	£	£
First Costs	Cost of vehicle	455	500	552	710	816
	,, ,, battery	120	187	208	230	250
	,, ,, vehicle and battery . .	575	687	760	940	1066
Annual charges of mileage	Driver's wages	150	150	150	150	150
	Depreciation of vehicle (6¾ per cent.) .	30	34	37	47	54
	Interest on vehicle and battery (5 per cent.)	29	34	38	47	53
	Licence	21	21	25	28	30
	Garage charges	18	20	22	24	26
	Insurance	11	12	14	16	18
Annual charges proportional to mileage.	Battery renewals	40	62	69	77	83
	Electric energy	10	15	26	40	51
	Tyres	18	21	34	48	72
	Maintenance	16	17	19	21	23
	Total running costs	343	386	434	498	560
	Costs per vehicle-mile in pence . .	8·23	9·3	10·4	11·95	13·4

TABLE III.—WEEKLY OPERATING COSTS OF PETROL, STEAM, AND ELECTRIC VEHICLES.

Load Capacity in Tons.	Petrol.	Steam.	Electric.
0·5	117*s.*	—	137*s.*
1·0	146*s.*	—	153*s.*
2·0	176*s.*	—	173*s.*
3·5	212*s.*	249*s.*	198*s.*
5·0	246*s.*	275*s.*	224*s.*

RANSOMES' ELECTRIC TROLLEY BUSES.

(*Ransomes, Sims & Jefferies, Ltd., Ipswich.*)

One of the leading makes of railless electric vehicles is the Ransomes' Electric Trolley Bus, made by Ransomes, Sims & Jefferies, Ltd., of Ipswich, who have designed two standard types of 4-wheel chassis.

The Type C Chassis is constructed primarily for the purpose of obtaining the lowest possible floor line and can be supplied as (1) a single deck saloon bus with either a front or rear entrance or both, or with a central entrance; (2) with an open or covered in double deck body, in which case the rear or loading platform would be exceptionally low. The motor has been placed above the fore axle in the position occupied by the engine in a petrol-driven vehicle.

Type D.—This design enables the entrance to be placed in front of the fore axle and gives the best possible arrangement for a one-man operated bus. It is equally suitable for use with an

additional rear entrance if a conductor is employed. The motor in this type is fitted about midway between the front and rear axles.

The back axle in both types has been specially designed to transmit the heavy power developed by the motor under the most severe conditions of overloading and steep gradients. The standard practice is to fit a single traction-type motor, but, a twin motor unit suitable for series parallel control can also be provided. The motor is of the ventilated type, special air entry and air outflow cowls being provided at each end in order to prevent splashings from the road surface entering the motor case. The motor complies with the British Standard Specification for Traction Motors in regard to rating, temperature rise, etc., and is designed to give rapid acceleration. Each rear wheel is fitted with two powerful independent brakes, each working on separate drums, and operated by foot pedal and ratchet pattern hand levers respectively. When desired, pedal-operated air brakes may be supplied, or four wheel brakes can be fitted. The steering gear is so designed that the effort required for steering is reduced to a minimum. Either pneumatic or solid tyres can be supplied. The springing of the chassis has received very careful attention and an extremely easy riding bus has been developed.

*Type D.*6.—The most recent development in trolley bus design is Ransomes' Type D.6, a 6-wheeler with double deck and top cover, the wheels all being fitted with giant pneumatic tyres. Several special features are incorporated in the design.

These vehicles are believed to be the first to be fitted with no fewer than twenty brake shoes. There is an independent brake on each of the six wheels, actuated by means of Westinghouse air system. These same shoes can be operated at will by means of the ordinary foot pedal placed in the driver's cab. The arrangement is such that, if required, the air pressure can be supplemented by foot pressure from the driver. There is also an entirely independent set of brakes on each of the four rear wheels, operated by means of a hand lever. This hand lever is electrically connected with the controller circuit, so that it is impossible for the man to operate his controller pedal with the hand brake in the ' on ' position. With such a braking system as is outlined above one would naturally expect exceptional performances, and as a matter of fact the stopping distances of the trolley buses when fully loaded with passengers are equivalent to those of the best private cars.

A very important item in the construction of the chassis is the adoption of the Thornycroft patent bogie. This offers a number of very important advantages, the greatest of which is, perhaps, the fact that the torque reactions due to acceleration and braking forces are taken care of without the need for introducing rigid torque rods, which at their best leave very much to be desired.

Another important feature of the Thornycroft suspension system is that the load on each of the bogie axles is exactly compensated so that under no condition can either rear axle take more than half the load which is on the bogie.

A special, and it is believed, a quite novel innovation is embodied in these vehicles, namely, a third differential in the transmission system. In the same way that the Thornycroft patent bogie system equalises the dead weight to be borne by each of the rear axles, so the third differential equalises the drive between all four rear wheels, each of which is free to roll freely on the road at a speed which may differ from any of the others due to varying diameters of the tyres, which latter of course depend on the wear of the tread and the comparative inflation pressure. Although there may be something to be said against the use of a third differential for vehicles which have to operate under war conditions (in which case excessive tyre wear is not an important factor) the manufacturers consider that the fitting of a third differential will materially increase the life of the tyre, reduce considerably the wear and tear, and tend to reduce power consumption under service conditions.

The chassis is fitted with a twin motor of 60–65 h.p., the two armatures being mounted in a casing which is common to both. This arrangement has been fitted so as to give all the advantages which come from series-parallel control, and tests have proved the great advantages gained by adopting this form of control in such large vehicles as those under consideration. The acceleration is smooth but rapid, and nothing short of exceptional gradients of 1 in 11 can be negotiated at speeds of 16 m.p.h. The electric motors are made by Ransomes, Sims & Jefferies, Ltd., the control gear being supplied by the Electro Mechanical Brake Company of West Bromwich.

Special attention has been given to the driving, and controls and pedals are arranged in the most convenient positions for operation. A raked steering column is provided, and this, with its double reduction gear, gives one-hand steering.

A notable departure from usual practice is the new patented system of carrying the trolley booms on the roof of the upper saloon. Messrs. Ransomes, Sims & Jefferies consider that the orthodox method of supporting the trolley base on the roof of this class of vehicle is not sufficiently robust to provide for all the stresses which are set up under service conditions. The firm, some little time ago, took out a patent covering a new form of support for the overhead gear, which relieves the roof of all stresses and transfers them directly to the pillars, which, of course, form the main stay for the sides of the bodywork. The construction permits of the roof being made lighter than hitherto, as it has been found that the new design not only relieves the roof of stresses due to overhead trolleys, but, in addition, is sufficiently robust to greatly strengthen the roof, and thus avoid any undue ' working' of the latter due to the constant variation in loads which are transmitted from the overhead gear.

The pillars, the hoop sticks supporting the roof, and the floors are reinforced with steel flitch plates. The body is thus strengthened by means of a series of steel bands completely encircling it. This has been the firm's standard form of construction for a considerable time, and has proved to have all the advantages of the complete steel or complete wood bodies, without many of their disadvantages.

Petrol-Electric Lorries.

In the petrol-electric vehicle the motive power is provided by a petrol engine, but the mechanical gear and clutch are replaced by a very flexible electric transmission. In a recent type of Stevens petrol-electric lorry of 3 tons capacity, the generator driven by the engine gives from 300 amperes to 80 volts to 230 amperes at 110 volts, and in addition to driving the vehicle it can be used to supply power for charging a battery or electric lighting, welding, etc. The electric motor is supplied through a special controller which gives forward and reverse connections with a wide range of speed variation, and when put in the neutral position enables the generator to supply electricity for external use. In the last case the generator can be given any desired characteristic—under, level, or over-compounded. A 3-ton waggon can supply three arc-welding equipments at 60 volts.

Storage Battery Locomotives.

The development of the electrification of railway termini in large cities has been retarded by the complications and expense of equipping goods yards and sidings with overhead or third rail. To meet the problem, storage battery locomotives have been designed, the storage battery being particularly well suited to improve the economy of operation where the load is of a very fluctuating nature with high peak demands but with a low average demand, as the battery is capable of discharging at very high rates for short periods of time.

Many of these storage battery locos are in service, and favourable reports are furnished by the engineers operating them. The advantages in goods yards shunting service are as follows:

(1) The locos can be operated wherever there are rails and in buildings or yards where any form of smoke or exhaust gas is objectionable.

(2) A storage battery can deliver power at very high rates; it is capable of developing great tractive effort for acceleration, which is of particular importance in shunting.

(3) Construction and operation are simple. A storage battery contains no moving parts and requires little attention.

(4) Economies will be effected in the total investment and in the cost of operation compared with that of steam locos.

The largest storage battery locomotive in the world was recently built by the General Electric Company at their Erie Works for shunting service by the Chicago and Northwestern Railways in Chicago. It is equipped with a battery of 120 Exide-Ironclad type FL-31 cells, and in addition it has an auxiliary gas engine and generator, and can operate off a third rail as a combination battery and trolley locomotive.

The power that can be developed by this locomotive at 25 per cent. adhesion is 55,000 lbs. tractive effort at over four miles per hour, which is equivalent to 600 horse-power at the wheel rim. A train of sixty 80-ton cars could be started by such a locomotive and brought up to a speed of 8 or 10 miles an hour.

Data concerning this 110-ton storage battery locomotive are as follows:

Locomotive complete	237,000 lb.	Speed at one-hour rating	10 m.p.h.
Mechanical equipment	100,700 lb.	Tractive effort 30 per cent. co-efficient of adhesion	66,000 lb.
Battery	78,960 lb.	Maximum speed	30 m.p.h.
Motor	29,800 lb.	Time motors will carry maximum tractive effort	8 mins.
Engine and generator	7,850 lb.	Ampere-hour capacity at 6-hour rate	2,700
Radiator and fan	3,500 lb.	Average volts at 6-hour rate	228·0
Control	9,150 lb.	Kw.-hour capacity at 6-hour rate	616
Brakes	5,640 lb.	Max. discharge rate in amperes.	6,500
Heater	1,400 lb.	Max kw. discharge rate	846
Length overall	52 ft. 0 in.	Approx. weight of battery	78,960 lb.
Wheel base	39 ft. 0 in.	Approx. floor space	200 sq. ft.
Rigid wheel base	8 ft. 0 in.	Approx. over-all height of battery	38 in.
Height	14 ft. 8 in.		
Width	10 ft. 0 in.		
Tractive effort one hr. capacity	20,000 lb.		

Charging Battery Vehicles.

Battery vehicles are well adapted for charging during the night hours, and specially low tariffs for this purpose are now offered in many districts. A standard Charging Plug and Socket is specified in B.E.S.A. Report, No. 74—1929.

SECTION XV

HIGHWAY ENGINEERING

ADMINISTRATION — HIGHWAY AUTHORITIES — LEGAL POINTS — HIGHWAY STATISTICS — THE NEW ROADS — COSTS OF MAINTENANCE — REGULATIONS FOR WHEELS AND LOADS — RESISTANCE TO TRACTION — ROAD CONSTRUCTION — COUNTRY ROADS — ROAD CRUSTS AND WEARING LAYERS—STREET PAVINGS—SPECIFICATIONS—CONCRETE ROADS—ROAD STONE TESTS—COSTS DATA—ROAD CRUST WORK—VEHICLES AND PLANT.

(pp. 711–762.)

(Contributed and Revised by Reginald Ryves, Assoc.M.Inst.C.E., M.Cons.E.)

(*Author of 'The King's Highway; or the Nature, Purpose, and Development of Roads and Road Systems'; 'Preliminary Studies in Bridge Design.'*)

See Addenda for recent information and data. Items as to which the Addenda pages, in brackets, should be consulted include: Heavy Vehicles on Bridges, pp. 715, 716 (762A); Regulations for Wheels and Loads, pp. 722–724 (762–762A); Superelevation, p. 728 (762B); Nature of the Tar, p. 733 (762A); Replacements and Costs, p. 746 (762C); Haulage Costs, pp. 756, 757, 759, 760 (762C).

SECTION XV

HIGHWAY ENGINEERING

ADMINISTRATION — HIGHWAY AUTHORITIES — LEGAL POINTS — HIGHWAY STATISTICS — THE NEW ROADS — COSTS OF MAINTENANCE—REGULATIONS FOR WHEELS AND LOADS — RESISTANCE TO TRACTION — ROAD CONSTRUCTION — COUNTRY ROADS — ROAD CRUSTS AND WEARING LAYERS — STREET PAVINGS — SPECIFICATIONS—CONCRETE ROADS—ROAD STONE TESTS—COSTS DATA—ROAD CRUST WORK—VEHICLES AND PLANT.

(Contributed by Reginald Ryves, Assoc.M.Inst.C.E., M.Cons.E.)

ROADS AND STREETS.

In view of the important changes in highway administration, which came into force from April 1, 1930, the related statistical matter in this section has been only slightly revised. As it stands, it may be considered to mark the end of a short era, during which the main objectives have been to effect such a classification of roads as really corresponds to the distribution of traffic, and to make the roads in every class good enough for the purposes which they serve.

This has, though not yet completely, been done, by new construction of arterial and by-pass roads, allowing of the establishment of a normal first class of main roads; by transfers, aided by grants for reconstruction; by grants for the improvement of various classes, and by annual grants for ordinary maintenance.

ADMINISTRATION.

UP TO APRIL 1, 1930.

Road administration in the United Kingdom is carried out by individual local authorities, each of which appoints its own road surveyor. The financial arrangements are, however, such as to lead to relations between the council of a county and those of districts in the same administrative county, as regards some proportion of the roads, the extent to which this has occurred depending partly upon the circumstances of the area and partly upon the policy of the councils concerned.

Recent Grants.—For distribution of grants, see p. 721.

Highway Authorities.

In all three countries the county councils are concerned with the maintenance of roads, but the arrangements differ, since there are no district councils in Scotland, while the relations between the county councils and the district councils are not the same in Ireland as they are in England and Wales. Subject to the financial modifications resulting from Government grants and the general supervision exercised by the Ministry of Transport in respect of the use of this money, the basis of highway administration is as follows:

England and Wales.—With some exceptions, applying to only a small mileage, the roads are maintained by the various authorities as follows. In cities and county boroughs all the public roads are maintained by the corporation or the borough council. In non-county boroughs and urban districts the urban council usually maintains all the roads, but the county council pays a considerable part of the cost of maintenance of such 'main' roads as pass through the area in question; and sometimes they are directly maintained by the county. With respect to rural roads it may be regarded as the normal arrangement for the more important roads, having been declared to be 'main' roads, to be maintained directly by the county council, and to be in charge of the county surveyor, while the other roads are maintained by the district councils, each council having its own surveyor. In effect there are wide differences, for the proportion of county roads to the whole may be very small or very large, the average being about 24 per cent.

In other cases the 'main' roads are maintained by the district councils under a contract with the county council; or the most important of the district roads are the subject of grants made by the county council. Since the roads directly maintained by the county must be technically 'main' roads, it follows that in counties having a large proportion of the total mileage under county management many technically 'main' roads are roads which are in fact 'secondary' in character or even less important; and conversely, there are many counties in which the main roads in fact are classed as 'assisted roads,' or are, some of them, district roads maintained at the cost of the local ratepayers.

Ireland.—The roads in Ireland are divided into 'main roads' and 'district roads.' The latter are maintained by, and at the cost of, the urban and rural districts in which they are situated, while the cost of main roads is equally divided between the district and the county. The sums for the maintenance of both classes of roads have first to be voted by the district councils, the power of the county council being almost entirely one of veto.

Administrative arrangements in the Free State are undergoing modification under the new conditions.

Scotland.—In Scotland there are no rural district councils. The urban roads within the burghs are maintained by the burgh councils and the rest of the roads by the county councils. Small counties, such as Elgin, Clackmannan, and Kinross, are each in charge of one surveyor, and Sutherland, where many of the roads are maintained by contract, has one surveyor. The larger counties are divided into districts, each in charge of a surveyor working under the district committee of the county council. Most of the counties are divided into two or three districts, but some large counties, such as Ayr, Perthshire, and Lanarkshire, into five. The mileage in a district ranges from about 120 miles in populous districts to some 500 or 600 miles in sparsely peopled districts. Each district committee of a county is composed of the county councillors representing that district. With their own officials they do all the county work for the district, and report their proceedings once a year to the county council, of which they are the component parts. In road matters the district councillors also report to the county road board, which is composed of members from each of the districts. It is recognised that the roads are better managed by persons intimately acquainted with the needs and resources of the relatively small area, and it is held that it is better to secure the services of qualified and experienced men for the direct management of the roads in each district than to employ less efficient local surveyors under a county surveyor. Midlothian is an exception to the rule in Scotland, the system of management resembling that of an English county in which a large proportion of the roads are directly maintained by the county. There are four districts, each in charge of a district surveyor, the whole being controlled by a county surveyor, through whom the reports from each district are made to the county road board. The traffic in this area is considerable and of a severe character.

Great Britain.—The Ministry of Transport contributes 50 per cent. towards the costs of maintenance of Class I. roads, and 33⅓ per cent. towards the costs of maintenance of Class II. roads; sums are also allotted in aid of the maintenance of unclassified roads of a purely rural character. In 1926 the mileage of these rural roads scheduled for grants was nearly 36,000, the 1926–1927 total being not far short of 41,000, and the March 1928 total, 55,000. This, at present, purely financial classification is independent of that under which 'main' roads are distinguished from roads not directly maintained by county councils. Since, however, the basis of it is the amount or severity of traffic, it is likely to prevail, the allotment of grants for unclassified rural roads creating in effect a third class. The result, if logical, will be four classes, unless the aided rural roads be added to Class II.

After April 1, 1930, the grants for Class I. and Class II. roads will probably be, respectively, 60 and 50 per cent.; while metropolitan and county boroughs will receive 'block' grants, based on the same percentages of recent expenditure. County councils will also receive block grants, available in aid of the maintenance of 'scheduled' roads.

For mileages, see pp. 716, 720, 722, 744.

The Local Government Act, 1929.

(Came into force April 1, 1930.)

The Act constitutes the county council, the highway authority in respect of all repairable roads in rural districts, and all present and future classified roads in urban districts and non-county boroughs with populations not exceeding 20,000. The county councils will have all functions hitherto exercised by the rural district councils under the Highway Acts, 1835 to 1885, and rural district councils will cease to be highway authorities.

County Roads.—The county council will have, with respect to the roads for which it becomes the highway authority, the like functions as with respect to main roads, enactments relating to which shall have effect as if for references therein to main roads there were substituted references to county roads.

Except in respect of urban claimed roads, every county road, its materials and drains, shall vest in the county council.

Classified Roads.—The term 'classified roads' will include roads in any class declared by the Minister of Transport to be not inferior to Class I. or II. for the purposes of the Act.

An Order creating a new urban district may provide that the county council shall continue to maintain the unclassified roads, the urban council making payment to the county council.

Transfer of other Functions.—As regards powers exercised in rural districts by the county council under the Private Streets Works Act of 1892 and sect. 150 of the Public Health Act, 1875, the county surveyor when preparing his specification shall, if and so far as the works include sewers, consult the rural district council.

Obstructions on footpaths caused by the placing of building materials, &c., will be dealt with by the county council.

Sections 21 and 22 of the Public Health Act, 1925, relating to water flowing over or soil falling on to a footpath, will be administered in rural districts exclusively by the county council; in urban districts, in respect of county roads, either by the county or the urban council.

In rural districts and, in respect of county roads in urban districts, the county council, or the district council, with the consent of the county council, may deal with street refuges, cabmen's shelters, trees in streets, and public conveniences and lavatories.

Powers Delegated: Powers Claimed: Appeals.

County councils are authorised and, unless they are able to satisfy the Minister of Transport that such a course is undesirable, are required to delegate to the rural district councils the maintenance and repair of unclassified roads. Rural district councils wishing to avail themselves of those powers must make application within three months of the commencement of the Act, the application relating to the following: all unclassified roads other than county bridges; all or any of the classified roads in the district; all or any of the county bridges. In respect of the second and third the county council will have 'unfettered discretion' to grant or refuse the application. In respect of unclassified roads, the county council must grant the application, unless satisfied that it is undesirable to do so, on grounds of efficiency and economy in highway administration.

Delegation of Powers.—Before October 1, 1929, councils had to inform the Minister of Transport how applications for delegation of powers had been dealt with and, as regards unclassified roads, the grounds of refusal had to be specified. District councils had a month in which to appeal against refusal to the Minister of Transport, who might direct the county council to grant the application. Appeals that shall have failed may be renewed every five years, or at any other time if with the consent of the Minister. A county council having delegated its powers with respect to roads or bridges may withdraw such delegation, subject to appeal by the district council to the Minister in respect of unclassified roads.

Control of Delegated Functions.—In respect of delegated powers, the district council will become agents for the county council, the execution of works, and the expenditure to be approved by the latter, who will also have some control over *personnel* and the terms of contracts. Works must be completed to the satisfaction of the county council who, in the event of non-compliance with their requirements, may themselves do what is necessary.

Urban Districts.—In regard to the right of urban districts of over 20,000 inhabitants to maintain their classified roads, those roads will all vest in the urban council, as if they were ordinary roads so vested. The cost of maintenance, repair, and reasonable improvement of those claimed classified roads will be borne by the county council, to whom detailed estimates must be presented every year. Approval of the estimates must not be unreasonably withheld, disputes as to estimates or the execution of works being settled by the Minister of Transport.

County councils will contribute towards the cost of improvements of classified roads generally, disputes being referred to the Minister of Transport, who will consider the purpose of the improvement for through traffic, for local traffic, or as a 'town' improvement.

The right of urban districts of less than 20,000 inhabitants to maintain previously 'claimed' main roads will lapse. An urban district council may, with the consent of the county council, or with an overruling decision of the Minister of Transport, relinquish the maintenance of a county road.

The clauses respecting classified roads in metropolitan and county boroughs mainly relate to grants.

Sources of Information.—The effects of the Act are explained, and action which must or may be taken by the local authorities on or before April 1, 1930, indicated, in a Memorandum issued by the Ministers of Health and Transport.

The significance of the Act, from the point of view of highway authorities, was discussed in a paper by Mr. W. J. Hadfield, at a joint meeting of the Yorkshire and the North-Western Districts of the Institution of Municipal and County Engineers, April 13, 1929.

The Local Government (Scotland) Act, 1929.

The Act constitutes the county council the highway authority for the whole county, district committees and standing joint committees ceasing to exist. The term 'county' is defined as including 'small burghs'—that is, burghs of less than 20,000 inhabitants. Excluding 'large burghs,' of 20,000 or more inhabitants, there will be a uniform road rate over the whole county.

Present and future classified roads in 'small burghs' will come under the jurisdiction of the county councils.

Contributions in lieu of rates, and the Government contribution towards the cost of maintaining roads (Act of 1889) will be discontinued.

The percentage grants for classified roads' maintenance will continue, but unclassified roads will be subject to a block grant, stabilised for a period of three years, based on the grant made for the year 1928–1929 with the addition of $\frac{1}{91}$ of 3,000,000*l.*

Delegation of Powers.—A county council may delegate the maintenance of classified roads within a 'small burgh' to the town council of that burgh.

Resulting Changes.—The number of administrative counties will be reduced from 33 to 31, and 98 district committees will be abolished. The classified roads of 173 'small burghs' will come under the control of the county councils, about 260 miles in Class I. and 84 miles in Class II.

The county councils must submit schemes for the administration of highways to the Secretary of State, on or before March 31, 1930, when it will be known which of the existing arrangements (see 'Scotland' on p. 712) is likely to predominate.

Ministry of Transport: Roads Department.

Constituted under the Act of 1919, the Ministry of Transport took over the powers and duties of Government departments with respect to (*a*) railways; (*b*) light railways; (*c*) tramways; (*d*) canals, waterways, and inland navigations; (*e*) roads, bridges, ferries, and vehicles and traffic thereon; (*f*) harbours, docks, and piers. The powers and duties of the Road Board were also transferred to the new Ministry. These include: powers to acquire land for the making of new roads and for road improvements (Act of 1909), and the duty of allotting grants to local authorities for the purpose of road improvement. These functions are carried out by the Roads Department of the Ministry, which is 'charged with dealing in the ordinary course of departmental business with road construction, improvement, maintenance, and development.' The Ministry may, by agreement with the local authority, defray half the salary and establishment charges of the engineer or surveyor to a local authority.

The Ministry is also charged with the allotment of grants for road maintenance, acts in an advisory capacity as regards both financial and technical matters, and directs the general policy in the expenditure of the large sums allotted recently for special works intended to reduce unemployment.

LEGAL POINTS FOR ROAD ENGINEERS.

The same principles of law govern all highways, whether carriage roads, bridle paths, or footpaths. The widths of some roads have been fixed by legal enactments; in England, for instance, by awards under Inclosure Acts, and in Scotland a minimum width, with powers to take a further width was prescribed by the Roads and Bridges Act of 1878. The width of a highway may be greater than that set out in an award, but *prima facie* it is not. The width has often been determined by dedication, and in some cases the dedication was recent and the intended width may be clearly inferred; but in many cases the act of dedication is presumed, or the road is a legacy from ancient times. In such cases the usual presumptions are that the abutting owner of the soil is the owner of the soil to the middle line of the road, and that the right of public passage extends from fence to fence. Road engineers should note, however—when roads require widening—that there may have been recent encroachment and that a strip beyond the fence may be really a part of the highway. There have been many such encroachments. The principle 'once a highway always a highway' should not be lost sight of. The actual width of the highway and the nature and extent of the facility to which the public has a right may depend upon the manner in which, and the extent to which, the public has used the road and its margins as well as upon the extent to which and the manner in which the owner of the soil or his tenant has exercised rights of ownership, or has done things which tend to show that the dedication of the road is subject to reservations. It is important to note that a wayside ditch, though usually a part of the highway, may be outside the highway, and, especially when a road has been set out under an award, it should not be too readily concluded that a ditch lying beyond the width of the strip awarded is a part of the highway. The highway usually extends to a depth such as may be necessary for the proper maintenance of the road and to a height required for the passage of laden vehicles. The obstructions permitted on a highway are those which are authorised by Acts of Parliament, such as telegraph posts placed by the Postmaster-General, and pipes laid by water and gas companies in defined areas. No individual or society has the right to place any post or other obstruction on the highway, except the owner of the soil where he can clearly establish

a claim to such a right. Failing this, the presumption is that he cannot place on the highway anything which amounts to an obstruction of public passage in its widest interpretation. The highway authority may erect direction posts, and, presumably, posts for cognate purposes, but it seems that such an authority may not erect lamp-posts unless they have powers as a lighting authority, even if the lamps are intended to facilitate the use of the road by the public. The highway authority has no powers enabling it to place posts or other obstructions upon the highway in the interests of certain classes of users of the road. The tendency is, however, for local authorities to go beyond their powers, and road surveyors will probably be required to carry out many works near the border line of those powers. Generally, as regards the placing of pipes, conduits, or wires along or across the highway, the persons doing so without statutory authority are likely to come into conflict with the public, or the owner of the soil, or both.

Width of Carriageway.—There is usually a presumption that the highway is dedicated in part as a carriageway, in part as footways, and grass margins which are regarded as occasional footways and rides, and in part as ditches, banks, and other enduring features necessary to the maintenance of the highway as established by use. It may, therefore, be desirable in some cases to obtain the formal consent of the owner of the soil to any radical alterations, such as the filling in of ditches. As regards trees and hedges it is important to distinguish between the powers of the highway authority as regards actual obstruction, and their further powers relating to the lopping of trees and cutting of hedges which injuriously affect the road, these powers being restricted to the months of October to March inclusive. Similar powers, not restricted to any season and applying to cases in which the tree or hedge obstructs the view of drivers of vehicles, are conferred by Clause 23 of the Public Health Act, 1925, while cases in which trees or hedges obstruct the view of drivers but do not overhang the highway may be dealt with under section 4 of the Roads Improvement Act, 1925.

The Highway Authority.—Must maintain the road in a condition fit for the passage of the public; may enforce the lopping of trees and cutting of hedges in the manner already indicated; may recover damages for substantial injuries caused by 'extraordinary traffic'; may enforce the removal of obstructions from the highway; may enforce by-laws such as those regulating the locking of wheels and the use of skidpans, and the widths of tyres in proportion to the loads carried.

The Public.—The public rights affecting the maintenance of roads include—the recovery of damages caused by 'misfeasance' on the part of the local authority, such as the placing of heaps of stone within the limits of the highway or the fixing of manholes, in such a manner as to cause accidents.

Any member of the public who finds on a highway an object which is in fact an obstruction or a nuisance may take proceedings against the person responsible for its presence, unless it is placed there under statutory authority.

Ownership.—In certain cases the highway authority becomes owner of the soil to a greater or less extent. In boroughs and urban districts, streets taken over by the local authority are vested in that authority. In rural districts 'a main road and the materials thereof and all drains belonging thereto' vest in the county council, but this does not constitute the council owners of the soil or give them powers over roadside wastes beyond those proper to their duties as the guardians of the public right of passage. As regards new roads made by the Ministry of Transport, it has the same powers and duties as those of a county council (except that of levying rates); and it is clear that if the road is on land which the Ministry has bought outright, with powers 'to sell, lease and manage' it, the Ministry becomes owner of the soil; and the Development Act does not seem to contemplate the acquisition by the Ministry of limited rights. In rural districts, highways other than main roads are vested in the owners of the soil.

Extraordinary Traffic.—It has been held that, in the circumstances of a particular case, the haulage of 100 tons a week for five weeks was extraordinary traffic.

Heavy Vehicles on Bridges.—The owner of a bridge may prohibit the crossing of the bridge by a vehicle weighing more than 3 tons, but subject to confirmation, or an increase in the limit, by an umpire or arbitrator.

Bridges as 'Streets.'—By a legal precedent of 1926 (Mr. Justice Romer), a bridge may become a street within the meaning of the Public Health Act, 1875.

The Roads Improvement Act, 1925.

A highway authority may impose restrictions or enforce changes in respect of the use of land at or near a turning or a bend in a highway, in order to prevent danger arising from the obstruction of the view of persons using the highway. Relates especially to the height and character of a wall or hedge. Provision is made for the payment, by the highway authority, of the cost of alterations and compensation for losses sustained as the direct result of those alterations.

A highway authority may prescribe building lines on either or both sides of a highway.

On classified roads the approval of the Minister of Transport is required. Building lines prescribed by urban authorities must be approved by the county council. County councils must notify the local council of building lines prescribed for main roads, and plans must be open to inspection by the public. Compensation for injury to property may be claimed.

The Public Health Act, 1925.

Highway authorities are empowered to lop trees which overhang highways; in the case of a main road the power is exercised by the county council. Appeal may be made to the Petty Sessional Court.

Local authorities are empowered to prescribe improvement lines for widening streets. Compensation may be claimed. In respect of main roads county councils are given the same powers.

See the texts of the Acts, or Mr. E. J. Stead's presidential address to the Association of Somerset Engineers ('*The Surveyor*,' January 1, 1926).

Bus-Mileage Tax.—Under the Act of 1920, the local authority may, if the Minister of Transport approve, impose a tax of ½*d.* per bus mile on regular services.

A local authority may apply to the Ministry of Transport for an order prohibiting or restricting the use of heavy motor vehicles on roads which are unsuitable for them.

Arbitration.

The settlement of disputes by arbitration is often more satisfactory than by recourse to the Courts.

The Bridges Act, 1929.

The Bridges Act, 1929, applies only to bridges which carry public carriage roads and are not maintainable by highway authorities. In most cases, the owners of such bridges have hitherto not been bound to maintain the bridge in a condition sufficient for traffic other than that of the time when the bridge was originally built.

The Act enables a highway authority, by agreement with the owner, to take over the property in any bridge to which the Act applies, or, the bridge remaining vested in the present owner, to contribute towards the cost of its maintenance, improvement, or reconstruction. In cases in which such an agreement is not reached, or a definite order of the Minister of Transport is held to be more advantageous, the owner, or the highway authority, may apply to the Minister for an order.

Such an order may require the execution of any necessary works of improvement or reconstruction, or may determine who shall be responsible for the future maintenance of the road which the bridge carries.

Subject to specified exceptions, the order may transfer the property in the bridge to a highway authority, or determine that the authority shall be responsible for the maintenance of the structure.

An agreement or an order may relate to the transfer to the highway authority of responsibility for the maintenance of the road on the bridge and its approaches, without transfer of property in the bridge or in the road over it, or of responsibility for maintenance of the bridge itself.

The Act is intended to provide for contributions by the owner, in respect of costs of maintenance, improvement and reconstruction, to the extent of his present liability. In the case of an Order made by the Minister, provision is made for the settlement by arbitration of disagreement as to the amount to be contributed by the owner. The remainder of the cost will be borne by the highway authority.

In cases in which the highway authority assumes an additional liability in respect of improvement or construction, the Minister is prepared, in approved cases, to make grants up to 75 per cent. of the approved net cost falling on the highway authority.

In respect of maintenance costs, grants normal for the class of road will be applicable.

HIGHWAY STATISTICS.

Road Mileage in Relation to Population and Area.

Road Mileage, per square mile.—Norway, 0·17; Spain, 0·174; Sweden, 0·205; New South Wales, 0·223; Hungary, 0·253; Switzerland, 0·465; Austria, 0·641; United States Rural Roads—all rural roads, nearly 1·00; surfaced roads, including sand-clay, 0·175; not including sand-clay, but more than half the mileage of gravel, 0·151; macadam and durable materials, 0·056; Italy, 0·782; Canada, 0·014; Province of Quebec, rural roads (31,000 miles), not including trails and roads in remote colonisation districts, 0·044; Peru, 0·013; Brazil, about 0·003; Denmark, 1·724; Japan, 1·743; France, 1·750; Ireland, 1·755. *Class I., or First-class Roads*; miles per sq. mile: England and Wales, 2·62; Scotland, 0·796; Great Britain, 1·98; Bulgaria, 0·314 (State roads, 0·125; communal roads, 0·188).

[To reduce to km. per sq. km., multiply by $\frac{87}{140}$, or 0·6214.]

Persons per Mile of Road: Counties (based on mileages, March 31, 1927, and census, 1921).—*Eleven*, below 61; Radnor 23, Montgomery 32, Cardigan 44, Merioneth 48, Brecon 54, Hereford 56, Devon 57, Westmoreland 57, Rutland 58, Pembroke 60, Norfolk 60. *Sixteen*, below 90; Lincolnshire, Holland, 67; Anglesey 70, Shropshire 70, Lincolnshire, Kesteven, 71; Cornwall 73, East Riding of Yorkshire 76, Carmarthen 77, Lincolnshire, Lindsey, 78; Oxfordshire 79, Denbigh 81, Suffolk, West, 83; Somerset 85, Cumberland 86, Suffolk, East, 87; Wiltshire 87, North

Riding of Yorkshire 89. *Twelve*, below 131; Gloucestershire 93, Huntingdon 97, Dorset 98, Carnarvon 101, Hampshire 105, Flint 107, Northamptonshire 114, Berkshire 118, Isle of Ely 120, Leicestershire 124, Buckinghamshire 130, Northumberland 130. *Eight*, below 200; Worcestershire 139, Sussex, West 143; Warwickshire 145, Sussex, East 151; Cambridgeshire 153, Huntingdonshire 194, Bedfordshire 199, Nottinghamshire 199. *Nine*, 200 to 300; Staffordshire 201, Derbyshire 209, Monmouthshire 217, Cheshire 223, Soke of Peterborough 223, Isle of Wight 230, Kent 230, Essex 241, Yorkshire, West Riding, 260. *Five*, over 300; Surrey 324, Lancashire 384, Durham 424, Glamorgan 450, Middlesex 927.

Population per Mile of Road.—New South Wales, 24·6; United States, whole population and rural roads only, 39·3; surfaced roads, 224; Denmark, 75·5; France, 108·0; Norway, 130; Sweden, 154·4; Japan, 210; Ireland, about 90; Italy, 451; Canada, 169; Province of Quebec, rural roads, whole population, 76; Brazil, about 3,800; Peru, about 1,400; England and Wales, 248; Scotland, 194; Great Britain, 240; Austria, 385; Bulgaria, 460; Switzerland, 504; Spain, 575·8; Hungary, 655; Nova Scotia, 1924, a mile to every seven families.

[To reduce to pop. per km., multiply by $\frac{87}{140}$, or 0·6214.]

Per Mile of Road.—Population—Acres; Bridgewater, 1214—71; Middleton, 1051—170; Burnley, 944—42; Huddersfield, 787—85; Blackburn, 772—47; Bridport, 758—87; Rochester, 917—81; Belper, U.D., 76—20; Bucklow, R.D., 107—64; Sheffield, 1067—64; Warwick, 684—280; Cambridge, 830—87; Aylesbury, 800—21.

Mileages, 1922.—Maintained by 62 county councils, 28,177; by 243 non-county boroughs, 4,656; by 812 urban district councils, 11,870; by 661 rural district councils, 94,628.

Ireland, 1926.—About 47,000 miles, of which 8,000 miles correspond to Classes I. and II. in Great Britain.

Widths of Important Roads.

Total Widths.—Essex county council has decided that, under the Roads Improvement Act and the Public Health Act of 1925, widths shall be: for Class I. roads, 60 ft. between improvement lines and 100 ft. between building lines for residences, 80 ft. for shops; Class II. roads, 50 ft., 90 ft., and 70 ft., respectively; unclassified roads, 40 ft., 80 ft., and 60 ft., respectively. *Midland* Regional Advisory Council (committee for arterial roads and main drainage) advise: main and very important secondary arterial roads, 120 ft.; other secondary arterial roads, 80 ft.; other through routes, 65 ft.; minimum building lines, width on main arterial roads, 160 ft.

Widths including Embankments and Cuttings.—It is usual to specify that the widths of embankments and cuttings shall, for widths between improvement lines, be added to the normal widths for roads on the flat.

Worcestershire.—Highways Committee recommend as least widths: between buildings, Class I. roads, 120 ft.; Class II., 100 ft.; highway widths, Class I., 60 ft.; Class II., 50 ft. *West Riding:* Highways Committee recommends, for widenings, when necessary and economically practicable, Class A, 80 ft.; B, 60 ft.; C., 45 ft.

Carriageway Widths.—Nottinghamshire, approximate average width, by calculation, of 361 miles of tarred and other surface-treated roads (in a total of 561 miles of main roads), the treated width, 18 ft. Similarly, Isle of Ely, average treated width of tarred roads, 15 ft. 8½ ins. Berkshire, Bradfield rural district, 196 miles, approximate metalled width, by calculation, 13 ft.

Norfolk.—58 miles of roads converted 1927–1928 from water-bound macadam to tar-slag macadam, this indicating their relative importance, average width of the crust, not necessarily of the visible surface, 18 ft. 3 ins., by calculation. Warwickshire, width, by calculation, of the road crusts of reconstructed and resurfaced roads, 45¾ miles, nearly 20 ft. 6 ins.; calculation of tarred surface width, nearly 18 ft. 6 ins.

Costs of New Roads.

Recent Examples.—The Glasgow-Edinburgh main road, mostly 120 ft. wide between fences; estimated cost, 69,000*l.* per mile. Kingston by-pass road, 100 ft. between fences, 30 ft. carriageway; estimated total cost, 37,000*l.* per mile; tender for construction, 22,102*l.* per mile. Sutton by-pass road, 80 ft.; estimated total cost, 42,947*l.* per mile; tender for construction, 19,290*l.* per mile. Sheffield, widening of roads to 80 ft. and 100 ft., and entire reconstruction, 32,000*l.* per mile. Great West Road, London, 120 ft. wide between fences, 50 ft. carriageway, capable of being widened to 80 ft.; all mains crossing roads carried in subways; foundations, 12 ins. of ashes and hard core; road crust, 9 ins. of reinforced concrete; carpeting, 2 ins. of rock asphalt. Total estimated gross cost, 1,012,602*l.*, or nearly 197,000*l.* per mile. Edinburgh-Glasgow Road, 100 ft. (in parts 60 ft.) between fences, 40-ft. carriageway, two 10-ft. footpaths, some heavy excavation; about 40 miles, 24 m. new, 16 m. widenings; estimate 2,000,000*l.*, or about 50,000*l.* per mile. Whitehaven Loop Road, 41,700*l.* per mile. Tyne Coastal Road,

between Heaton and Tynemouth, estimates, 175,000*l.*; length, 4¾ miles, nearly 36,800*l.* per mile. Sutton By-pass Road, 4·64 miles, 80 ft. between fences, 30-ft. carriageway, one 8-ft. footpath; estimate, nearly 44,000*l.* per mile.

Australia.—Under the Federal Aid Highway Act: graded earth roads, 5,008 miles, 1,712*l.* per mile; sand-clay, 2,541 miles, 1,730*l.*; gravel, 9,098 miles, 2,220*l.*; water-bound macadam, 608 miles, 4,500*l.*; bituminous macadam, 841 miles, 6,630*l.*; bituminous concrete, 757 miles, 7,270*l.*; Portland cement concrete, 4,075 miles, 8,570*l.*; brick, 318 miles, 10,000*l.* per mile.

State of Victoria (1928–1929).—Construction and reconstruction; Federal Aid Roads, including bridges, 2,040*l.* per mile; State highways, 156 miles completed and 34 miles cleared and formed, average 2,863*l.* per mile. Analysis of the 156 miles: of concrete, 0·3; bituminous concrete, 2·4; bituminous macadam, 83; water-bound macadam, 27; gravel, 43 miles.

Reconstruction.—Cardiganshire, including foundations and surfacing with tar macadam, 3,167*l.* per mile. Lincolnshire (Holland), 4,000*l.* Dorset, Wimborne, and Cranbourne R.D.C., 2,791*l.* A moorland road, 5,000*l.*

Jersey, Channel Islands: scarifying, cleaning old material, and mixing with 4½ galls. of bitumen to 11½ cu. ft. of stone for bottom coat of 3½ ins. (before rolling); rolling; preparing top coat of 10 galls. of bitumen to 11½ cu. ft. of dried destructor clinker, and spreading 1½ ins. thick; rolling; costs, 7*s.* per square yard. Cost of materials and their treatment for this work, 1*l.* 10*s.* 7*d.* per ton, of which labour accounts for 10*s.* 2*d.*, bitumen and its transport charges, 8*s.* 6*d.*, granite from parish quarry, 3*s.* 2*d.*, clinker from parish destructor, 1*s.* 7½*d.*, and capital charges, 2*s.* 7*d.* Relates to 1,756 tons.

Arterial and By-Pass Roads, Examples and Data, 1924–1929.

Ordnance Maps.—A special revision of roads has been made, and the results are being embodied on the 1-in. and smaller scale sheets as they become due for reprinting.

Dartford By-Pass Road: 3½ miles; carriageway, 30 ft.; 10-ft. footpaths; total width, 60 ft.; right-of-way, 100 ft.; maximum gradient, 1 in 30.

Middlesex.—Great Cambridge Road: 10 miles; total width, 100 ft.; prepared carriageway, 24 ft.; in reserve, 24 ft.; footpaths, 6 ft.; base, 3 ins. clinker; crust, 8 ins. reinforced concrete.

Great West Road, Greater London: 8½ miles; 5½ miles done; total width, 100 ft.; carriageway, 50 ft., flanked by 25-ft. grass margins, with forest trees; 10-ft. footways; parts of grass strip, available for widening of footways if needed; base, 12 ins. hardcore and ashes; strength crusts, 9 ins. reinforced concrete; carpet, 2 ins. rock asphalt.

Staines Extension: about 3 miles; total width, 100 ft.; present carriageway, 30 ft., flanked by grass strips 29 ft. each; two 8-ft. footpaths. Crust, 8 ins. reinforced concrete.

Greater London.—Works in hand or projected: 6 miles; 80 ft.; carriageway, 30 ft.; one footpath, 10 ft.; base, 6 ins. clinker; crust, 8 ins. concrete.

8¼ miles; 80 ft. average; carriageway, 30 ft.; footpaths, 8 to 10 ft.; crusts, uncarpeted, tar-macadam and bituminous clinker, in different parts of the length.

5½ miles; 80 ft.; carriageway, 30 ft.; footpaths, 8 ft. and 8 ft. Base, hardcore and flint; top course, bituminous.

2·6 miles; 80 ft.; 30 ft.; two 7-ft. footpaths.

1·5 miles; 60 ft.; 30 ft.; two 6-ft. footpaths; base, hardcore; carpeting, 2 ins. bituminous clinker.

North Circular Road, Greater London: partly carried out, includes widenings and reconstructions; widths, 100 ft. and 60 ft. New construction, 9½ miles; widenings, 5½ miles; estimated cost, nearly 1,164,000*l.*, or 77,000*l.* per mile; including crossings of 12 railways and many streams. Examples of the work: 3½ miles of widening to 60 ft.; costs, including all expenses, about 31,000*l.* per mile; widening to 80 ft. near Ealing, 55,000*l.* per mile; near Greenford, 60,000*l.* per mile. Total length will be 26 miles.

Middlesex.—Schemes adopted in past five years for about 66 miles of new arterial roads, 47 miles of reconstruction involving considerable works, and 16 miles of improvement of existing roads; estimated cost, 7,735,000*l.*; or, averaging the classes of work, 60,000*l.* per mile.

Kingston By-Pass Road: 9·6 miles, 100 ft. wide; carriageway, clinker bottom course resting on clay; reinforced concrete 8 ins., of which the lower 6 ins. is 4½ : 2½ : 1, top 2 ins., 3 : 1½ : 1 aggregate, Thames Valley gravel.

East Ham and Barking By-Pass Road: 3¼ miles, 100 ft.; carriageway, 24 ft.; footpaths, 8 ft. and 10 ft.

Gravesend-Strood Road: 5·2 miles; 80 ft.; carriageway, 40 ft.; base, hardcore and flint; crust, bituminous clinker.

Farnham-Wrotham Road: 60 miles; 80 ft.; carriageway, 30 ft.; base, hardcore and flint; bituminous surfacing.

Birmingham-Wolverhampton : 9¾ miles; 100 ft.; carriageway, 40 ft., with two optional additions of 20 ft. each; footways 10 ft. each; base, cinders and ballast; crust, 'Tarmac.'

Bolton : 2½ miles; 70 ft.; carriageway 50 ft.; base, hand-picked stone; strength crust, 6 ins. concrete; paving, 6 ins. granite setts.

County of Durham.—New Coast Road; minimum width, 60 ft.; carriageway, 24 ft.; crust, 4 ins. tar macadam; base in cuttings and on embankments, 12 ins. punning, 4 ins. ashes. Cost, about 37,000*l.* per mile.

Glasgow : 4½ miles; two carriageways of 30 ft. each, with middle grass strip, 40 ft.; footpaths, 12 ft. each; grass margins, 8 ft. each.

Leeds Ring Road : 30 miles; 10 miles made; right-of-way, 100 ft.; initial carriageway width, 24 ft.; bottoming, 4 ins. ashes; base, 12 ins. pitching; intermediate layer, 4 ins. 'dross'; carpet, 3 ins. tar-macadam.

Liverpool, 28 miles; 120 ft.; a middle grass strip, with tramway, flanked by privet hedges; a 25-ft. carriageway on both sides of the middle strip, and 10-ft. footways outside; partly, 4½ ins. pitch-grouted macadam, without foundation; partly a 9-in. concrete strength crust paved with 4-in. cubical setts of Welsh stone. Cost about 57,000*l.* per mile.

Nottingham : 2½ miles; 70 ft.; carriageway, 20 ft.; and 2 miles; 80 ft.; carriageway, 54 ft.; base, 12 ins. of slag on 2 ins. of ashes; crust, 4 ins. tar-macadam. The works include a 3-span bridge over the Trent.

Southport : 1½ miles; 75 ft.; carriageway, 27 ft.; footpaths, 7½ ft. each; bottoming, 3 ins. clinkers; base, 9 ins. pitching; strength crust, 2½ ins. tar-macadam; carpet, 1½ in. asphaltic.

Other cases : Surrey East and West Road—nearly 5 miles; 60 ft.; carriageway, 30 ft., extendable to 40 ft.; base, 3 ins. to 6 ins. clinkers; crust, 8 ins. reinforced concrete.

Hartlepool : 1 mile 3 furlongs, and 1 mile; base, 4 ins. clinkers; intermediate layer, 9 ins. slag; crust, 4 ins. tar-macadam.

St. Austell By-Pass Road, 1¾ miles.

Essex : London-Yarmouth Road, 1¾ miles.

Mid-Glamorgan : Inter-valley road scheme: three portions of 2 miles apiece, and one portion of 1 mile; form a part of a 40-mile scheme; bottoming, ashes; base, 12 ins. of hand-picked stone; crust, 6 ins. water-bound mountain limestone; width, 30 ft.; carriageway, 20 ft.; footpath, 5 ft.

Whitby-Sandsend Road.—About 2 miles; 40 ft.; carriageway, 20 ft.; tar-grouted macadam: works include a cutting 125 ft. deep, an embankment 75 ft. high, and a bridge of 41 ft. span. Cost about 25,000*l.* per mile.

Sidcup By-Pass Road.—2½ miles; 80 ft.; carriageway, 30 ft.; two 10-ft. footways; 6 in. of clinker, 9 in. of hardcore; wearing carpet 3 in., 2-coat synthetic asphalt; cost about 50,000*l.* per mile.

Morecombe Ring Road.—2·21 miles; carriageway, 30 ft.; foundation, 14 ins. of hand-pitched rubble on ashes and clinker; cost about 41,000*l.* per mile.

London-Folkestone Road.—Maidstone to Charing; 12½ miles, 60 ft.; carriageway, 30 ft.; one 6-ft. footway; three-quarters of width is old carriageway, the remainder new, in concrete; bituminous carpet; estimated cost about 55,000*l.* per mile; estimated cost of the whole work, 400,000*l.*

Aberdulais-Resolven Road.—A county main road; 4·4 miles; 30 to 36 ft.; carriageway, 25 to 31 ft.; one 5-ft. footway; steepest gradient, 1 in 11; foundation, 8 to 12 ins. of ballast topped with mountain limestone; wearing crust, 4 ins. of tar macadam; cost about 10,000*l.* a mile.

Birmingham-Coventry Road.—Widening to 80 ft. or 60 ft., 10 miles; a spur, 6 miles to Castle Bromwich.

Newcastle-Tynemouth Road.—Nearly 4·7 miles; about 43,000*l.* per mile, or, not including the cost of four large bridges, about 31,700*l.* per mile. Half the 102 ft. width made; 30-ft. carriageway; reinforced concrete 7½ ins. at sides, 10½ ins. at centre, on flat bed of ashes; top 2 ins. blue whinstone chippings; single reinforcement.

Liverpool-Manchester Road.—27 miles; estimated to be finished by the end of 1932; 120 ft.; 40-ft. carriageway to be provided in the first instance; designed for two more (20-ft.) carriageways; and four 10-ft. footways; circuses of 240 ft. diameter at crossings of classified roads; estimated cost, 3,000,000*l.* Liverpool section, 2 miles, opened May, 1929.

Aberdeen.—Outer Ring and By-pass Road; including a bridge, 41,500*l.*

West Riding.—Barnoldswick; 70 ft.; 1¾ miles; 100,000*l.*

County Down.—Newry to Warrenpoint; 4¾ miles; estimate, 56,500*l.*

Welwyn By-Pass Road.—One mile; formation 60 ft.; large reinforced concrete bridge; 60,000*l.*

Daventry By-Pass Road.—A 20-ft. reinforced concrete carriageway; 3 in. clinker bottom course; concrete 6 ins. of 4 : 2 : 1, gravel, and 2 ins. of 3 : 1½ : 1, granite; bays 22 ft. 6 ins. × 12 ft.; reinforcement square mesh 7 lbs. per sq. yd.; one layer on excavation and surface road bed; two layers on fill; granite concrete kerbs; 1⅞ miles; nearly 21,000*l.* per mile; 60 ft. total width; maximum depth of cutting, 14 ft.; maximum height of embankment, 9 ft.

Bexley Heath By-Pass Road.—Nearly 5·7 miles; normal width, 80 ft.; carriageway, 30 ft.; concrete surface; four new bridges; about 54,000*l.* per mile; including bridges.

York Ring Road.—Six-inch reinforced concrete with (1) 4-in. tar macadam; (2) 2-in. clinker asphalt.

Inverness-Perth Road.—Completed to Blair Athol 1928. Grant of 600,000*l.* Length, 78 miles, and 25 miles of widening; carriageway, 18 ft. between 'stone haunches'; 3-in. gravel bottoming; 9-in. stone; tar-macadam carpet, 3 in.; 35 new bridges.

Construction across a bog; reinforced concrete raft 21 ft. wide and half a mile long; expansion joints 50 ft. apart; in the form of a trough, with sides 10 ins. deep and 6 ins. thick; floor 8 ins. thick, with upper and lower reinforcement.

Blackburn Arterial Road.—4¼ miles; (1) 120–100 ft.; two 30-ft. carriageways, separated by a grassed strip, two footways; (2) 60 ft.; one 30-ft. carriageway, two footways. Completed 1928, with one 20-ft. carriageway, sufficing for present traffic.

Market Weighton By-Pass Road.—In the East Riding; about 3¾ miles; estimate, 7,630*l.* per mile.

PROPOSED: 1929.

Colchester: West to East By-Pass Road.—3½ miles; estimated cost, 245,000*l.*

Porlock Hill By-Pass Road.—Steepest gradient 1 in 12; 3 miles; estimated cost, 30,000*l.*

Costs of Road Maintenance.

(Per Mile.)

ENGLAND AND WALES.

(1927–1928.)

Main Roads.—Hampshire, 654 miles; direct control 585 miles. Northamptonshire, 336 miles, 654*l.*; cost to the county, 351*l.*; directly maintained, not including improvements, 648*l.* per mile; cost to the county, 340*l.* Warwickshire, 580 miles; direct control, 512 miles, about 444*l.* Durham, 387 miles; directly maintained, 359 miles. Herefordshire, 513 miles. Somerset, rural, 924 miles (24 miles directly maintained), about 322*l.* per mile.

Classified Roads (1927–1928).—Estimated expenditure per mile, on the maintenance and repair of Class I. and Class II. roads and bridges.

Class I.—County councils; England and Wales, 17,102 miles, 539*l.*; Scotland, 5,531 miles, 302*l.*; urban and rural district councils, 790 miles, 754*l.*; county borough councils, 1,276 miles, 1,135*l.*; metropolitan borough councils, 252 miles, 2,317*l.*; Scottish burghs, 188 miles, 894*l.*

Class II.—County councils; England and Wales, 7,766 miles, 320*l.*; Scotland, 3,595 miles, 169*l.*; urban and rural district councils, 3,565 miles, 375*l.*; county borough councils, 592 miles, 938*l.*; metropolitan borough councils, 118 miles, 1,373*l.*; Scottish burghs, 50 miles, 540*l.*

All Class I., 25,139 miles, 544*l.*

All Class II., 15,686 miles, 330*l.*

Henley and Hambledon Rural District.—Omnibus routes, 9 miles, 239*l.*; Grade I. roads, 24 miles, 115*l.*; Grade II. roads, 24 miles, 75*l.*; Grade III. roads, 33 miles, including roads to single farms or houses, 2*l.* 10*s.*: total, 138 miles.

(1928–1929.)

Hampshire.—662 miles; direct control, 593 miles, the latter costing 458*l.* Holland, Lincolnshire, 218 miles, including unemployment relief works, 666*l.*

Northamptonshire.—All urban and rural main roads, 349 miles, 672*l.* per mile; 510 miles under direct control.

Somerset.—943 miles, rural, 313*l.*

Berkshire.—402 miles.

Herefordshire.—513 miles: Class I., 257½; Class II., 181; unclassified, 74½.

Sheffield.—All highways; 402 miles, 234*l.*; average of last five years, 224*l.*

OTHER COUNTRIES.

Federated Malay States.—Average, 1,474 dollars (172*l.*) per mile. 1928: 'arterial' roads, 2,600 miles, including streets in towns; average cost, 1,483 dollars, or 173*l.* per mile; most costly road, 2,004 dollars, or 233*l.* 1*s.* 6*d.* per mile.

Welland County, Canada.—About 204*l.*

Italy.—Typical costs per mile: 1st class, 4,025 lire; 2nd class, 3,220 lire; 3rd class, 2,415 lire; 4th class, 1,610 lire.

Report on the Road Fund, 1927–1928.

For full particulars as to finance, mileage, grants, road schemes, execution of works, etc., see the Annual Report on the Administration of the Road Fund, Ministry of Transport, price 1*s.* from H.M. Stationery Office, Adastral House, Kingsway, London, W.C. 2.

Finance.—(1) Road Fund receipts and payments. Receipts; issued from the Consolidated Fund (Motor Taxation Receipts), 19,665,580*l.* Total receipts, 20,348,503*l.* Payments: net payments under grants, 18,743,371*l.*; transfer to the Exchequer, 12,000,000*l.*; payment to Irish Free State, 200,000*l.* Total net payments, 31,703,364*l.*

(2) Net amount paid to Exchequer, 24,518,000*l.*; net amount paid by Exchequer to Road Fund, 19,665,580*l.*

(3) Revenue remitted to the Motor Tax Account by local authorities, 22,502,403*l.*; licences issued by Post Office, 2,335,000*l.*

Grants.—Maintenance and improvement of Class I. and Class II. roads and bridges, 9,770,092*l.* Maintenance of selected 'unclassified' roads in rural areas, 1,661,076*l.* Unemployment schemes (including trunk road reconstruction, 723,042*l.*; new bridges, 342,423*l.*; Mersey Tunnel, 1,226,821*l.*), 3,629,950*l.* Improvement works in rural areas, 1,228,180*l.* Other improvement schemes, widenings, reconstructions, bridges, etc., 1,918,215*l.* Total grants, 18,830,278*l.*

Distribution of Grants: Classified Roads.

England and Wales.—Main roads, Class I., 5,424,061*l.*; Class II., 959,126*l.* District roads; urban, Class I., 231,623*l.*; Class II., 245,312*l.*; rural, Class I., 108,405*l.*; Class II., 279,746*l.* Roads in county boroughs: Class I., 838,044*l.*; Class II., 218,021*l.* London roads: Class I., 342,194*l.*; Class II., 65,204*l.*

Scotland.—County roads: Class I., 587,107*l.*; Class II., 176,554*l.* Burgh roads: in seven larger burghs; Class I., 98,143*l.*; Class II., 10,086*l.*; in all other burghs: Class I., 158,079*l.*; Class II., 40,162*l.*

Great Britain.—Class I., 7,787,656*l.*; Class II., 1,994,211*l.*; both classes, 9,781,867*l.*

Special Grants.

Scotland.—Grants to highway authorities in areas where the highway rates are exceptionally high and long stretches of Class I. and Class II. roads run through sparsely peopled districts: up to 75 per cent.; total sum, 369,409*l.*

Rural Areas.—Towards the cost of reconditioning important roads, 1,250,000*l.* (as in two preceding years; in 1924–25, 2,250,000*l.*; in 1920–24, 1,150,000*l.*).

Grants to Counties, All Purposes.

Exceeding 400,000*l.*—Devonshire; including 139,536*l.* unemployment relief works, 489,857*l.* Essex; including 475,759*l.* maintenance Class I and Class II roads and bridges, 649,578*l.* Kent; 425,261*l.* Lancashire; including 259,616*l.* in respect of works other than maintenance and unemployment relief works, 735,618*l.* West Riding of Yorkshire, 715 525*l.*

Exceeding 200,000*l.*—Buckinghamshire; including 116,823*l.* unemployment relief works 357,154*l.* Cheshire, 301,204*l.* Cornwall, 204,928*l.* Derbyshire, 251,562*l.* Durham, 247,931*l.* Gloucestershire, 228,598*l.* Lincolnshire (Lindsey), 236,121*l.* Middlesex, 327,148*l.* Norfolk, 289,925. Northamptonshire, 274,089*l.* Northumberland, 239,700*l.* Shropshire, 222,871*l.* Somerset, 334,638*l.* Southampton, 284,228*l.* Staffordshire, 272,352*l.* Surrey, 391,959. Sussex, East, 244,521*l.* Warwickshire, 224,080*l.* Wiltshire, 227,037*l.* North Riding of Yorkshire, 266,574*l.* Glamorgan, 218,472*l.*

Totals for Counties.—11,852,773*l.*, including, in respect of unemployment relief works, 743,252*l.*

Grants to County Boroughs.

(*(a)* total; *(b)* in respect of unemployment relief works.)

Exceeding 100,000*l.*—Birmingham *(a)* 130,544*l.*; *(b)* 15,297*l.* Newcastle-upon-Tyne *(a)* 174,539*l.*; *(b)* 142,354*l.* Sunderland *(a)* 155,983*l.*; *(b)* 149,713*l.*

Exceeding 50,000*l.*—Bournemouth *(a)* 60,139*l.*; *(b)* 30,032*l.* Bristol *(a)* 89,781*l.*; *(b)* *nil.* Kingston-on-Hull *(a)* 67,395*l.*; *(b)* 56,447*l.* Leeds *(a)* 79,399*l.*; *(b)* *nil* (maintenance, 24,065*l.*), other purposes, 55,334*l.* Liverpool *(a)* 50,705*l.*; *(b)* *nil.* Rotherham *(a)* 58,832*l.*; *(b)* 49,965*l.* Sheffield *(a)* 52,157*l.*; *(b)* 17,242*l.*

Total for county boroughs *(a)* 1,998,149*l.*; *(b)* 502,504*l.*

GRANTS TO LONDON AUTHORITIES.

In respect of maintenance.—On roads and bridges in Classes I and II; to the London County Council, 47,771*l.*; to Metropolitan Borough Councils, 407,398*l.*; on scheduled roads, *nil*; in respect of other purposes, 35,906*l.* Total, 443,304*l.*

OTHER UNEMPLOYMENT RELIEF GRANTS.

North Orbital Road, London, 50,310*l.*; Unemployment Grants Committee (Loan Charges), 962,718*l.* River Mersey Road Tunnel, 1,226,821*l.*

GRANTS TO LONDON AUTHORITIES.

London County Council, 49,513*l.*; Metropolitan Boroughs, 393,791*l.*; Total, Administrative County of London, 443,304*l.*; including, in respect of maintenance, Class I and Class II roads and bridges, 407,398*l.* Scheduled roads, *nil*; unemployment relief works, *nil*; other purposes, 35,906*l.*

MILEAGES OF ROADS: MARCH 31, 1928.

Class I.—England and Wales, 19,420; Scotland, 5,719; Great Britain, 25,139.

Class II.—England and Wales, 12,041; Scotland, 3,645; Great Britain, 15,686.

Both Classes.—England and Wales, 31,462; Scotland, 9,364; Great Britain, 40,825.

Scheduled, or third class roads: England and Wales, 42,249; Scotland, 12,804; Great Britain, 55,053.

Classified and Scheduled Roads.—England and Wales, 73,710; Scotland, 22,169; Great Britain, 95,879, or 53·6 per cent. of all roads.

England and Wales.—All main roads, 31,200; all district roads, 108,988; county borough roads, 11,439; London roads, 2,212. Total, England and Wales, 153,838.

Scotland.—Counties, including burgh roads in charge of counties, 22,586; other burghs, 2,490. Total, Scotland, 25,076.

Grand Total, Great Britain: 178,914 miles. An increase of about 130 miles.

Increase in mileage of main roads in England and Wales, 1925–1926, 1,100; decrease in mileage of district roads, 950.

REGULATIONS FOR WHEELS AND LOADS.

(See Addenda for new regulations.)

Traction Engines.

The regulations as to tyres of traction engines cited in the preceding issues of the Year-Book may be regarded as obsolete. The concerted action now being taken to encourage the use of tyres which will not damage the roads is leading to good results. New regulations will probably be withheld until some defined policy as regards the classification of these vehicles can be adopted, especially as relating to agriculture on the one hand and long-distance transport on the other. The tendency of practice as regards transport of goods is towards the use of small tractors with rubber tyres for work, such as local furniture removals, and the use of heavy vehicles of one of the two 6-wheeler types for higher-powered transport over long distances, except where ordinary steam wagons and lorries suffice. The scope of the old-fashioned heavy traction engine is becoming much reduced.

Heavy Motor-Cars.

This term applies to motor vehicles weighing more than 2 tons unladen, and refers to three distinct classes: (1) Motor wagons or lorries; (2) traction engines not exceeding 5 tons in weight and in other respects fulfilling the requirements; and (3) delivery vans and omnibuses, provided with rubber tyres and allowed to travel at 12 miles an hour.

Maximum Weight Heavy Motor-Cars.—The vehicle unladen, 7½ tons; vehicle and trailer, unladen, 9¾ tons; 'axle-weight' (any pair of wheels with their full load), 8 tons; axle-weight of trailer, 4 tons. An amending order of 1922 which, to meet special cases, allows 22 tons for the total of axle-weights and 6½ ton trailer axle-weight, will probably soon be superseded, but the necessary increases for tyre widths, in the table, can easily be calculated for such a case. This Order allows for a speed of 12 miles per hour if all the wheels have soft tyres, and the total length does not exceed 33 feet.

Tyre Widths.—These are: minimum for the vehicle itself, 5 ins.; minimum for the trailer, 3 ins. All wheels: ½ in. for each 'unit of axle-weight,' which is, 7½ cwt. for a 3-ft. diameter wheel, 1 cwt. more for every 12 ins. more, and 1 cwt. less for every 6 ins. less diameter. These rules do not apply to wheels with rubber tyres.

Heavy Motor-Car (Amendment) Order, 1927.

The chief provisions of the Heavy Motor-Car (Amendment) Order, 1927, may be summarised as follows :

Overall length of the vehicle not to exceed 27 ft. 6 ins. A vehicle more than 26 ft. in overall length may not draw a trailer. (See below.)

In the case of public service vehicles, one brake shall be applied by a pedal, and the operation of either brake shall not in any case disengage the engine from the transmission gear.

Public Service Vehicles : an axle-weight must not exceed 5½ tons, and the sum of such weights 9 tons. For other heavy motor-cars the figures remain 8 tons and 12 tons respectively. (See below.)

Six-wheeled Rigid Frame Vehicles : maximum overall length, 30 ft. ; greatest unladen weight, 10 tons.

Axle-weights : the sum of the registered axle-weights shall not exceed 12 tons for a public service vehicle, nor 19 tons for other vehicles ; individual axle-weights shall not exceed 4½ tons for public service vehicles, nor 7½ tons for other vehicles.

Heavy Motor-Car (Amendment) Order, August 9, 1928.

The effect of this order is : speed limits (now superseded by the Road Traffic Act, 1930 ; see Addenda) ;

Heavy Motor-Car without Trailer.—(*a*) With pneumatic tyres, 20 miles per hour ; (*b*) with solid rubber tyres or partly with solid rubber tyres, and partly with pneumatic tyres, 12 miles per hour ; (*c*) wholly or partly with metal tyres, unladen weight not exceeding 3 tons, and registered axle-weight of any axle not exceeding 6 tons, 8 miles per hour ; (*d*) any other case, 5 miles per hour.

Heavy Motor-Car with Trailer.—(*a*) Articulated vehicle fitted throughout with pneumatic tyres or solid rubber tyres, 12 miles per hour ; (*b*) vehicle with ordinary trailer : Vehicles and trailer fitted entirely with pneumatic tyres, 12 miles per hour ; vehicle and trailer fitted with solid tyres, or partly with solid rubber and partly with pneumatic tyres, 8 miles per hour ; any other case, 5 miles per hour.

Tyre Width, Camber, and Pressures.—The value of width of tyre in reducing the pressure of the wheel upon the road crust is not fully operative even on flat roads, since the wider the tyre the more frequent are the occasions on which it fails to bear for its full width on the slightly uneven surface of the road. The effect of width in reducing pressure has a more definite limit on hard roads with a sharp camber, unless the road is wide, with sloped and not barrelled camber, and the vehicle travels wholly to one side of the middle line. On a barrelled camber, the vehicle does not rest upon the full width of its tyres even when keeping to one side of the road. Traction engines traversing narrow roads tend to wear them into a pair of parallel flat strips, with an exaggerated cambered part in the middle. Since the reduction of bearing width is a serious matter, and, apart from unusually troublesome vehicles and difficult roads, applies to many classes of vehicles on properly made roads, and even to tyres no wider than 3 ins. or 4 ins., it is necessary to take it into account in the development of vehicles and roads. In the first place, it is desirable to find what is the average relation between diameter and tyre width as regards pressure, a relation which depends upon the modulus of elasticity of the crust and other factors. Since, however, wheels of large diameter punish the road less than wheels of small diameter, even when the actual pressures (load per sq. in. of surfaces in contact) are somewhat higher, the position generally may be no worse even if we fail for some time to establish the true relation and make the tyres of large diameter wheels somewhat too narrow. A general increase in the diameters of the wheels of many vehicles is, then, to be aimed at. The principle is recognised in the regulations for heavy motor cars. Next, as regards tyres which are necessarily very wide, attempts should be made to design the vehicles so that some duly constrained side play, angular or bodily, is permitted to the wheels. The side play which the wheelwright, with centuries of experience behind him, gives to cart and wagon wheels, greatly tends to economy in the wear and tear of the vehicles and the road, side play taking place to a large extent between the greased axle and the hubs of the wheels instead of occurring as a rubbing of the wheels on the road. The play is partly axial, but to a small extent angular, and allows of a slight adjustment to the cambered road. Thirdly, dead flats on roads (no gradient) should, as opportunity arises, be converted to gradients of 1 in 200, or steeper, and the camber reduced. On steeper gradients, and with some road crusts and subsoils, a nearly flat camber is suitable, and in other cases a slight tilt to one side only may be given to the whole crust.

Tyre Widths for Horse-drawn Vehicles.—Under the Highways and Locomotives Amendment Act, 1878, a county authority may make regulations for tyre widths of vehicles drawn by animal power, in proportion to the loads carried. The following examples of such regulations are taken from 'The Surveyor and Municipal and County Engineer,' April 21, 1911.

Tyre Widths for Certain Loads: Horse-drawn Vehicles.

Four-Wheeled Vehicles.—Tyre width, 2½ ins., about 3 tons in several counties. Width, 3 ins., 2 tons 11 cwts. ; 2 tons 19 cwts. ; 3 tons 1 cwt. ; 3 tons 11 cwts. ; width, 3½ ins., 3 tons 1 cwt. ; 3 tons 11 cwts. ; 3 tons 19 cwts. Width, 4 ins., 3 tons 1 cwt. ; 3 tons 11 cwts. ; 3 tons 19 cwts. ; 4 tons 1 cwt. Width, 4½ ins., 3 tons 11 cwts. ; 3 tons 19 cwts. ; 4 tons (summer ⊗). Width, 6 ins., 4 tons 1 cwt. (winter ⊗) ; 5 tons 1 cwt. ; 6 tons 1 cwt. Width, 9 ins., 5 tons 1 cwt. ; 6 tons 1 cwt.

Two-Wheeled Vehicles.—Tyre width, 2½ ins., 1 ton 1 cwt. Width, 3 ins., 1 ton 1 cwt.; 1 ton 11 cwts.; 1 ton 11 cwts. (winter); 1 ton 17 cwts. Width, 4 ins., 1 ton 11 cwts.; 2 tons 1 cwt.; 2 tons 11 cwts. Width, 4½ ins., 1 ton 11 cwts.; 1 ton 17 cwts. (summer); 2 tons 1 cwt.; 2 tons 11 cwts. (summer). Width, 6 ins., 2 tons 11 cwts.; 2 tons 17 cwts. (summer); 3 tons 1 cwt. Width, 8 ins. and 9 ins., 3 tons 1 cwt.; 9 ins., 2 tons 17 cwts. (winter).

Table I.—Legal Speed Limits of Motor Vehicles, 1931.

Abbreviations.—a.w. = all wheels. Fitted with: p.t. = pneumatic tyres, s.e.t. = soft or elastic tyres. N.T. = when not drawing a trailer. D.T. = when drawing a trailer.

Class of vehicle.	Miles per hr.
Passenger Vehicles.—(1) if a.w.p.t. and N.T.:	
(*a*) if not adapted to carry more than 7 passengers exclusive of the driver and is not a heavy motor car nor an invalid carriage	no limit
(*b*) if a heavy motor car or adapted to carry more than 7 passengers exclusive of the driver	30
(2) In any other case, including invalid carriages	20
Goods Vehicles.—(1) N.T. (*a*) motor cars; a.w.p.t., and (*b*) heavy motor cars constructed or adapted for the conveyance of horses and their attendants and used solely for that purpose; if a.w.p.t.	30
(*c*) (i) Motor cars, a.w. not p.t. but are s.e.t.; and (ii) heavy motor cars a.w.p.t.	20
(*d*) heavy motor cars a.w. are not p.t. but are s.e.t.	16
(2) D.T. (*a*) if a.w. of the vehicle and trailer are p.t., or if the trailer is attached to the drawing vehicle by partial superimposition in such a manner as to cause a substantial part of the weight to be borne by the vehicle and a.w. of the vehicle and trailer are s.e.t.	16
(*b*) if a.w. of vehicle and trailer are not p.t. but are s.e.t.	8
(3) In any other case	5
Locomotive and Motor Tractors.—(1) *Heavy Locomotives.* (*a*) within any city, town, or village	3
(*b*) elsewhere	5
(2) *Light Locomotives.*—When N.T. or not more than two trailers, if a.w. of the locomotive and of any trailer drawn by it are s.e.t.	8
(*b*) In any other case	5
(3) *Motor Tractors.*—(*a*) N.T., if a.w., s.e.t.	16
(*b*) D.T. of a.w. of the tractor and any trailer are s.e.t.	8
(*c*) In any other case	5

French Road-Surface Tests.

The following figures are given as those experimentally determined by the Paris police for the distances in which a motor-car weighing a ton and three-quarters, travelling on the level at 19 miles an hour, can stop on various kinds and conditions of road surface:—

Dry granite, 26 ft. 3 ins.; very wet granite, 32 ft. 10 ins.; greasy granite, 65 ft. 7 ins.
Dry wood, 26 ft. 3 ins.; very wet wood, 46 ft.; greasy wood, 65 ft. 7 ins.
Dry asphalt, 32 ft. 10 ins.; very wet asphalt, 59 ft.; greasy asphalt, 82 ft.

Table II.—Steam Rollers.

(Ordinary Loads and Dimensions in Working Order.)

Description of Roller.	Driving Wheels.				Front Rollers (as a whole).				Wheel Base.		Width Rolled.	
	Diam. in Ft.	Tread of each, Ins.	Load on both, Tons.	Load per In., Cwts.	Diam.	Tread in Ins.	Load in Tons.	Load per In., Cwts.				
					Ft. In.				Ft. In.		Ft. In.	
'Ten-ton'	5¼	16	7·25	4·5	4 0	49	3·75	1·6	11	2	6	4
'Twelve-ton'	5¼	16	7·9	4·9	3 10	49	4·3	1·7	11	3	6	4
'Fifteen-ton' or 'Sixteen-ton'	6	20	9·75	4·9	4 6	54	6·5	2·4	12	8	7	4
		18		5·5								

Diesel Engined Rollers.

Tandems: weights, empty and ballasted, 6 and 8 tons, 8½ and 10 tons.

Three-Rolls: weights, empty and ballasted, 11½ and 13¾ tons; front roll, 4 ft. diam. by 4 ft. 2 ins. wide; rear rolls, 5 ft. 6 ins. diam. by 17 ins. wide; rolled width, 6 ft. 9 ins.

Six-wheeled Steam Wagon (Garrett): capacity, 12 tons; body, 22 ft. by 6 ft. 9½ ins. First wheel base 13 ft. 4 ins.; second wheel base, 3 ft. 10½ ins., the four driving wheels being driven by one differential gear.

Six-wheeled Omnibus (Daimler): rigid type; covered top; wheel base to midway between rear axles, 17 ft.; rear axles, 3 ft. 6 ins. apart; total wheel base, 18 ft. 9 ins.; rear overhang, 8 ft. 3½ ins.; maximum outside width of body, 7 ft. 6 ins.; turning circle, 60 ft.; rear axle clearance, 6½ ins.; passengers, 104, 45 on lower and 59 on upper deck. (Used for transport of works staff.)

Passenger Chassis (Dennis): frame height, 24½ ins. ('one-step' type); wheel base, 16 ft.; rear axle clearance, 7 ins.; passengers, with omnibus body, 33, or less; with coach body, 30, or less.

Low-Loading Vehicle.—Two-tonner; hand gear end-tipping; height from roadway, 5 ft. to top of side, about 4 ft. to top of half side (flap down); tips to 45 deg.; 140 cu. ft. capacity, or 5·2 cu. yds.

Caterpillar Tractors, 10-ton and 5-ton: in use with street scrapers; as at Blackpool and at Lichfield, the latter drawing a train of three self-loading scrapers.

Pneumatic Tyred.—Public service vehicles; cesspool emptier (Dennis); 1000 gal.; 6-ton chassis. Cleansing wagon (Halley), 7 cu. yd. capacity. Also many other vehicles.

RESISTANCE TO TRACTION.

TABLE III.—INFLUENCE OF DIAMETER OF WHEELS.

The following figures are taken from the results of experiments by General Morin:—

Pull in lbs. per 2,000 lbs., at 3 miles an Hour.

Nature of Road.	Cart without Springs. 4½-inch Tyres. Diameter of Wheels in Inches.		Wagon without Springs. 4½-inch Tyres. Diameter of Wheels in Inches.	
	64	80	36 and 60	44 and 78
	Lbs.	Lbs.	Lbs.	Lbs.
Earth road, very good, nearly dry	55	44	74	63
Gravel road, loose gravel on firm earth	143	114	190	163
	168	134	225	192
	180	144	241	206
Broken stone road:—				
In good condition, dry	30	24	40	34
Firm, little soft mud	55	44	74	63
Portion worn out, thick mud	80	64	107	92
Stone block pavement, fair condition, dry	25	20	34	29
Thick oak bridge floor	35	28	47	40

Influence of Width of Tyres.

Under some circumstances narrow tyres offer less resistance to traction than wide tyres, but these circumstances are unusual, or in themselves undesirable, or do not hold in all kinds of weather. For instance, narrow tyres will cut through stiff mud on a hard bottom better than tyres which, though wider, are not wide enough to ride over the mud by pressing it in a rut of moderate depth. Wheels will run along ruts made by wheels with wider tyres, but not in ruts made by narrower wheels. Narrow tyres will run through a layer of loose stones on a hard road better than wide tyres, and they allow of the vehicle being pulled out of a slough which would hold down wider felloes. Generally, however, tyres classed as wide offer less resistance than tyres classed as narrow, and the more so when their cumulative effects on soft roads are considered, as well as the comparison of the moment. It is important to note that a vehicle which, on soft roads, can be pulled more easily than a similar vehicle with narrower tyres, will still show some advantage on a hard and firm road, or will not be at a disadvantage; and it must not be overlooked that in hot weather the tractive effort with narrow tyres is very high on asphalt pavements and even considerably increased on tarred roads.

The following are extracted from a table in which 1½-in. tyres and 6-in. tyres are compared, the figures given being 'resistance to traction in lbs. per ton' (*Ira O. Baker*).

TABLE IV.—INFLUENCE OF WIDTH OF TYRES.

Description of Road.	Width of Tyres. 1½-inch.	6-inch.
	Lbs.	Lbs.
Broken stone road, hard, smooth, no dust, no loose stones, nearly level	121	98
Gravel road, hard and smooth, few loose stones, size of black walnuts	182	134
Gravel road, new gravel, not compact, dry	330	260
Gravel road, wet, loose sand, 1 in. to 2½ ins. deep	246	254
Earth road, loam, dry, hard, no dust, no ruts, nearly level	149	109
Earth road, loam, stiff mud drying on top, spongy below	497	307
Earth road, clay, dry on top, but spongy below	618	464
Earth road, clay, stiff, deep mud	825	551
Earth road, loam, mud, 2½ ins. deep, very sticky, firm below	251	325
Earth road, clay, sloppy mud, 3 to 4 ins. deep, hard below	286	406
Earth road, loam, dry, loose dust, 2 ins. to 3 ins. deep	90	106

Tractive Effort on Different Pavements.

These experiments (*Ira O. Baker*) provide important comparisons between pavements differing only in condition as well as between pavements of different kinds. For instances the pulls in lbs. per 2,000 lbs. were: on considerably worn wood blocks, 36; on asphalt at 42° F., 34; on oak plank, nearly new, 32; and on smoothly-dressed granite blocks, 30; but the 32 for oak plank rose to 38 when it had been worn down ¼ in. in many places; and the 34 for asphalt rose by 3 lbs. with a rise of 10° in temperature, and was 70 lbs. when the temperature was 84° F. On cedar blocks, a pull of 50 lbs. when they were covered with ¼ in. of crushed gravel, rose to 90 lbs. with a ½ in. covering of silica pea gravel. On a brick pavement, ¼ in. of dust raised the pull from 17 lbs. to 31 lbs. On a steel plateway a pull of 19 lbs. was raised to 28 lbs. by ⅛ in of fine dust, and to 40 lbs. by ¼ in. of powdered stone. Out of a number of brick pavements the best required a pull of 17 lbs. per 2,000 lbs., and a granite macadam crust only 18 lbs.

Ministry of Transport, 1928; concrete surfaced roads; solid rubber tyres; 40-in. diameter wheels, 100; 21-in. diameter wheels, 90; relative tractive efforts.

At 15 miles per hour: 40-in. diameter solid rubber tyres, 33 lbs. per ton.

Rubber and Iron Tyres.

Experiments made in 1895 showed that the tractive effort is less with pneumatic tyres than with solid rubber or iron, and that the advantage increases with increase of speed. Iron tyres were better than solid rubber on good or ordinary pavements, but required a somewhat greater pull on bad, irregular pavements.

Effect of Speed.—Experiments by Morin showed an increase of tractive effort with increase of speed. The differences between the pulls required at a walk and those at a fast trot were considerable, as were those between the pulls at a 'trot' and 'a fast trot.' Other experiments showed a very small increase with pneumatic tyres and much greater increases with solid rubber and with iron tyres.

Undulations.—The effects of the passage of large numbers of heavy or fairly heavy vehicles of identical form, and the less serious effects of the same nature caused by the passage of many vehicles of similar weights and similarly sprung, have led to the forming of undulations of the same generic class as strings of puddles. Diversity in design and springing is favoured by road engineers, and 'fleets' of exactly similar vehicles are regarded as specially injurious. Direct action is hardly practicable, but the tendency to favour diversity seems likely to gradually effect some practical results.

ROAD CONSTRUCTION.

(By Reginald Ryves, M.Cons.E., Assoc.M.Inst.C.E.)

Country Roads.

Widths of Roads.—The total width given to a new road should include provision for: (1) the carriageway; (2) the supports for the edges of the crust, such as rough stone or gravel shoulders or 'water tables'; (3) such further strips of grass or other margin as may be necessary to hold drifted snow, to provide for the admission of air currents, or to keep sub-soil water at a suitable distance from the crust; (4) footways; (5) side ditches where necessary; (6) catch-water drains where necessary; (7) fences, and, where other parts of the width are not available, places for planting shade and shelter trees.

The engineering cost of making a road, especially where there are considerable earthworks, may be reduced by locating the footway outside the main formation width. The ground surface may lend itself to this; or the footway may be on a berm of the side of a cutting; not on the slope of an embankment, unless a high and dense hedge is between the footway and the carriageway. Overhedge location is very economical, especially if the path need not be fenced off from the field.

Width of Carriageway.

Width of Carriageway.—Maintenance is the dominating factor where traffic is light. Under the item 'carriageway' in the provision for total width allowance may be made for probable future widening, having regard to the relative economy of widening alternative routes or making new roads. In hilly country, and especially on sidelong slopes it is usually better to allow no such provision between fences, since the future can best be cared for by acquiring land only, the cost of obtaining a given width of flat strip by two earthworks being usually very much less than that of obtaining the same width by one earthwork. For the one wide strip the quantity of earthwork is much greater and the rate for cutting may be higher, while the retaining wall will cost much more than the two smaller ones. With flat earth slopes the total width required may be as great or greater.

Width in Relation to Maintenance.

Width in Relation to Maintenance.—The best width, having regard to the nature and amount of the traffic, will depend upon the materials available and their cost. One width will give a minimum maintenance cost with a hard and costly stone, a greater width may be the most economical for a softer stone. First decide the economical width for each material, and then compare costs. When the traffic is moderate the 'weathering' factor may be important and on little-used roads it may predominate. Usually the economic width for sandstone and flint crusts is greater than that for limestone crusts, partly because the more porous crusts made with sandstone and flint may have flatter cambers, tending to a good distribution of the traffic, and partly because the quartzy materials wear into large grains which are only removed by the heavier rains, while limestone crusts wear into fine stuff which is readily removed. The exposure of a wide surface is, therefore, usually undesirable with limestone crusts. Similar comparisons are furnished by harder stones, such as quartzites, compared with granites which contain much felspar. The nature of the subsoil affects the width, especially if the cost of construction or reconstruction has to be considered, because with soft subsoils, not tolerant of high pressures under heavy wheel roads, it may be better to reduce the pressure by making a deep crust than to reduce rutting or surface wear by making a wide crust.

Width and Traffic.

Width and Traffic.—The width may be decided, partly or wholly, from the traffic point of view. In making a mountain road, or widening a narrow lane, the ruling width may be that necessary for the passage of two loaded vehicles, and, as a matter of safety, a road with walls may be made somewhat wider than a similar road with hedges. In such cases it is important that the edges of the crust should not be steeply cambered or weak, since such conditions reduce the effective passing clearance. On roads of this kind the carriageway is that imposed by the total width, but it is often good economy to carry the actual crust right across, with such change in the nature of the materials near the edges, as may be suggested by the relative importance of the margin as a support to the crust or as a surface drain. Apart from such cases, and taking that of the normal carriageway on a road with suitable margins, the widths suitable are, for a very large mileage of country roads generally, either that required for one vehicle at a time, but sufficient for the cautious passing of two vehicles, or that required for the frequent and convenient passing of two vehicles. In both cases, it is desirable to give ample support to the edges, either by under-trenching, with a filling of large-gauge stone, or by continuing the crust under the turf beyond the incidence of wheel loads. Using the word as applying to the measurement from surface margin to surface margin, the following widths are regarded as suitable. For single-line traffic, 12 ft. is a width largely adopted in many countries. Ten feet or even 8 ft. may be sufficient for roads with very little traffic if the soil is suitable and the margins readily accessible. A gravel soil, especially if these strips be sifted a little when the road is made, offers this advantage, as does, under some conditions, a sandy soil. On clay soils, the width of the crust itself should be such as to admit of the passing of vehicles, or the edges of the crust may be broken down by rutting of the clay just beyond. Clay is not easily brought back to its proper level. On account of its hardness in dry weather, however, it makes good margins for narrow 'fair weather' roads, which may be made with a very narrow metalled or gravelled width. For roads on which the passing of two vehicles must be readily accomplished, a width of 16 ft. is very usual, and 15 ft. is enough in countries in which the ordinary vehicles are narrow. In France a considerable mileage of important roads has a carriageway of about 16½ ft., but this may, in some cases be the economical maintenance width. The usual width of the carriageways of the French Routes

Nationales is from 5 to 7 metres, or say, 15¾ to 23 ft. An important system of English county roads has an average width of 17½ ft. On very steep gradients it is often desirable to have very wide carriageways.

Except where local conditions demand a greater width, 30 ft. is a sufficient width for the carriageway of any but important arterial roads, other roads locally carrying very much traffic and parts of roads where vehicles are often standing. The best widths for traffic are not multiples of some unit even when such a unit can be decided upon. The best allowances for clearance depend upon the nature of the road. The necessary dimensions for clearances between vehicles depend on the nature of the margins and, in the case of a mountain road, may reasonably be small, having regard to the cost of construction and the moderate speeds which are proper to traffic on such roads. In the case of a road carrying much traffic, there is danger in having a carriageway less than some dimension depending upon how many lanes of traffic are provided for; but there may also be danger in providing a carriageway of a width considerably greater than that which is suitable. Thus, of two roads which are similar in other respects, the more costly may be the less efficient. The matter of width is one for judgment rather than for rules or formulas.

Camber.

Camber.—The amount of camber may be stated as the total rise in the width of carriageway or as the average slope of the sides. A rise of one-fiftieth is often recommended, or slopes of 1 in 25. Slopes of 1 in 32 have been recommended as an average. Actually, the amount of camber depends upon the nature of the crust, the gradient, the nature of the traffic, the widths and the climate, and no general rules can be laid down. It may be noted, however, that a rise of more than 1 in 50 should not be given without careful consideration to roads with much motor traffic, while there are some roads, especially those with quartzy crusts on steepish gradients, which require hardly any camber. The barrelled camber has the advantage that the support which the earth margins give to the edge of the crust is less nearly horizontal than in the case of a sloped camber (two slopes joined by a small arc in the middle of the road), and the barrel form resists somewhat better a tendency to the formation of ruts on each side of the middle line. The former advantage disappears, however, if the edges of the road crust are suitably designed, and the barrelled camber has the serious disadvantage that the slope becomes steeper near the edges where cyclists ride when they are being passed by motor cars. On roads with much motor traffic the mud is soon shifted to near the edges, this part being often somewhat slippery. An elliptic camber has sometimes the disadvantage that it is too flat in the middle, especially on roads with much motor traffic and of such width that a strip of about 5 ft. width in the middle gets the main part of the wear of the off-side wheels of cars going both ways. The camber with two slopes seems suitable for a greater range of roads than the barrelled form, and the advantage of a tilt to one side only must not be overlooked in the special cases to which such a device applies.

Super-elevation.

There is no theoretically correct degree of super-elevation at a bend on a road. Super-elevation for maximum speeds would render the road unsafe in respect of encounters of vehicles at bends, besides causing inconvenience and danger in the case of a vehicle travelling slowly. Even if attempts were made to impose degrees of super-elevation calculated for some moderate speed the results would, at some places, be unpractical. Successful application has been made of the principle of adopting a moderate speed, or a small range of speeds for primary calculations, with a limit of super-elevation expressed either as an angle or in inches. It may be desirable to use both limits, in order the better to deal with the great range in the degrees of curvature.

There remain cases in which super-elevation cannot be applied, as at snake bends, or can only be applied in an unsatisfactory manner, as to half a snake bend, or, in the case of a very short bend, inconveniently as regards the merging of the cross slope into the normally cambered straight road.

On some roads, properly applied super-elevation is a great improvement, especially the slight super-elevations suitable for tortuous mountain roads.

Drainage.

Specific works may come under one or more of the four heads: (1) interception of surface waters which would flow across the road, or along it, or would flood it; (2) interception of seepage water; (3) surface drainage of rain-water from the road and its margins; (4) under drainage of the road bed and crust. In a soil neither very stiff nor very porous, a side ditch may perform all four duties. When on the uphill side of a road it acts as a catch-water drain. It also acts as a channel to carry to suitable outlets the rain-water gathered by the road. It may also intercept seepage water from the hillside or the land above a sunken road; and, if near enough to the edges of the crust, and deep enough, it provides for the establishment of an hydraulic gradient from the crust and the road bed to the ditch bottom or water-level. On a stiff soil it does not perform this function, except to a very limited extent and when deep and close to the crust.

In a stiff soil, therefore, a deep ditch is not needed for this purpose, the water being dealt with by surface flow, aided by the filling of all hollows, and by evaporation. A deep ditch may, however, play the part of an interceptor of seepage water, forced through a fairly stiff soil from wet places on the hillside above. Apart from such a purpose a ditch on a stiff soil may be shallow if the necessary capacity can be given by the width available. The stiffer the soil the deeper the drain, up to a point at which it will no longer drain the road bed and may be quite shallow. It is then very important to fill hollows on all parts of the roadside which are above the level of the underside of the crust.

Catchwater Drains.—It is often desirable to have a catchwater drain clear of the uphill fence so that the bank may not be torn by water running down it. The outlets may be clear of the road, or lead to the larger cross culverts. Such a drain and, when necessary, side drains, may be provided with 'drops' or falls of hard materials, to reduce velocity and scour. Catchwater drains are much needed on many English roads.

Side Drains.—For a given soil there is a slope at which water hardly moves through it. To provide for flow, a slightly *steeper* slope, carried from the bottom of the side ditch or shoulder drain, should reach the edge of a waterproof crust, or the middle of a porous crust. This determines the depth of the ditch, or the depth at which the pipe drain in the shoulder trench lies; or it decides which of them will be the cheaper to make and maintain. On porous soils the natural slope of the land may provide drainage; but even then care must be taken that when the road runs directly down the slope there is some easier way for the water to flow than down the road bed and out through the crust; and sudden changes in the soil must be noted and provided for. On a long slope of this kind, on a porous soil, some diversion of subsoil water by trenches filled with clay may be useful, where the water can be turned to a depression in the ground outside the road. Pipe cross drains may be used for the same purpose, and if backed by clay on the downhill side will be very effective.

Drainage of the Crust.—This comes under one or two of the main divisions of drainage; surface drainage being the more important on stiff soils, and under-drainage on porous soils, while it may be necessary to make the most of both on soils such as those marls and silts which are at once receptive and retentive of water. If the crust is more porous than the road-bed and subsoil it will probably be desirable to provide a drainage course, or bottoming, such as a telford pavement. If the soil is more pervious than the crust such a provision is not likely to be necessary for drainage. When the crust or its bottom course allows of drainage to the sides, but not very rapid drainage, this may be aided by giving a camber to the road-bed, which may also be desirable so as to provide sufficient thickness of crust where it is most wanted, towards the edges. Cross-drains are seldom economical. If they are farther apart than the width of the road they are less effective than shoulder drains: if they are nearer together than the width of the road, they cost more than one shoulder drain, or, as they have to be placed deeper, as much as both shoulder drains. Cross drains require many more outlets than shoulder drains. Camber is given to a road so that water may drain from the surface of the crust. Too steep a camber may defeat its object by driving traffic to the middle, so that ruts are formed. When a road is puddled by much traffic in drizzly weather, camber is ineffective as the little ridges of mud hold up the water. On a slope of 1 in 24 a ½-in. ridge of mud holds up a foot-wide puddle; on places worn flat very much wider puddles. Under such conditions the best drain, on an impervious road, is a broom. When the crust is porous and self-draining a very small camber is sufficient, but the outlets to the side ditches must be sufficiently numerous and effective, or the road will be flooded during rain. On a road with no gradient it may be necessary to have flush edges, that is, no raised water-tables; but with a very small gradient this is unnecessary as the water will flow to the outlets. To prevent scour and the cutting back of their beds till the crust and road-bed are attacked, the outlets through turf margins must not carry the water with too high a velocity. The formation of a delta with the road washings may be allowed, the stuff being cleared away when the level of the delta becomes too high and the outlet is ineffective. Generally, a porous road crust need not have a steep camber if any of the following conditions are fulfilled: (1) good drainage of the crust, with shoulder drains if necessary; (2) a porous subsoil with a low, normal subsoil water level; (3) a porous subsoil on an embankment, or drained by effective side ditches. An *impervious* crust need not have a steep camber—(1) on gradients, especially if fairly steep; (2) if kept clean; (3) in dry and in showery climates or where the rains are usually heavy, as opposed to damp and drizzly climates.

Drought.

Owing to the presence of clayey matter as a constituent of many important road stones the weakness of road crusts when too wet has received more attention than their weakness when too dry. Sandy materials, on the other hand, resist wheel loads and disturbance generally better when damp than when dry. Drought, therefore, injures such roads, including crusts made with quartzite, sandstones, and flints (apart from dust as a nuisance). Other roads may be seriously affected by droughts, which cause a crumbling of the margins, or the gaping of a clay soil, the effects being observed when heavy rains occur. Drought may be combated by shade, either of trees or hedges, and by care in carrying out drainage so that the soil is not made too dry. Dead leaves are beneficial on sandstone and flint roads, and various materials, more or less imitative of the gummy decay of the leaf, have been used to lay dust, or to bind road crusts of a sandy character.

Roadside Trees.

For a fuller presentment of the subject see *The Surveyor*, August 9, 1907; and a summary of main points with notes relating to road surfacings of the new types, in the same journal, June 5, 1925.

The chief and prevailing influence exerted by trees on the road crust is a great reduction in daily or annual ranges of crust temperature, which facilitates design in some particulars. The effect in reducing maximum temperatures, separately considered, is next in importance in some cases. In other cases, notably with broken-stone, water-bound, and gravel or flint crusts, the effect of the trees in screening the road from loss of heat by radiation is the more important.

Shade in Summer.—The value of shade, apart from comfort, is in a reduction of evaporation from macadam and flint roads, reducing the tendency to disintegration in dry spells. On tarred roads it prevents licking up of the tar and the formation of ruts, and prolongs the life of the film. In the case of concrete roads the reduction of maximum crust temperatures reduces the tendency to the forming of cracks. On asphaltic and bituminous roads it either reduces the rutting and wearing effects of traffic, or, when its effect is taken into consideration in designing the crust or carpet, it is possible economically to construct a more efficient carpet, less hard and less slippery than one intended to withstand the effects of a whole day of hot sunshine.

Screening Effects.—While some trees, such as the elm, have a greater frost-screening effect than others, such as the ash, all have an important effect in decreasing the range of temperature to which the road crust is subjected, in one day or in one year. Generally, they reduce very much indeed the number of occasions on which a road is slippery because frozen. By lessening the severity of freezing and by afterwards screening the frozen surface from the full effects of sunshine, trees save macadam and flint roads from most of the licking-up after frosts. They also, by mitigating the effects of severe frosts, save such roads from disintegration due to the formation of ice within the crust. Slipperiness due to hoar frost, or to the freezing of condensed mist or drizzle on a cold road, is much worse on the new surfaces than it is on water-bound macadam, so that, in this respect, the value of the tree is increased by the new conditions in winter and early spring.

General Policy.—The significance of trees in the above respects is now widely recognised. Further, there is a reaction against the planting of trees in too formal a manner, as, for instance, in avenues which do not suit the locality or express any characteristic of the stretch of road. Clump treatment is being advocated by a number of surveyors, and it is realised that the principle may be applied broadly in respect of the planting or treatment of roadside trees and hedges.

For a recent study of the road and roadside in relation to the landscape, see *The Surveyor*, March 8, 1829 (A. D. C. le Sueur), and editorial comment March 15.

The Strength Crust and the Wearing Layer.

The word ' foundation ' is often misapplied. It should properly be used for the special bottoming that is necessary on soft soils, and sometimes on clayey soils, in order that a distinction may be made between such preparatory work and the construction of the actual ' crust,' which plays the principal part, or is the sole agent, in distributing the load. The essential nature of the crust is the same, whether it distributes the wheel loads upon the natural soil or upon a bottoming of hard material, and whether it receives the wheel loads in their full intensity or partly distributed by a carpet of considerable strength and thickness. Thus, the so-called ' concrete foundation ' of a wood-paved road is really the strength crust, and the wood-paving is the carpet or wearing layer, and on a bad natural soil the concrete crust will need something in the nature of a foundation proper.

These considerations are of special importance at the present time, because recent developments point to the value of an ordinary waterbound broken stone crust, as what is called ' a foundation ' for a carpet of bituminous materials, or stone setts, wood blocks, or bricks; and it must be remembered that the extent to which the loads are spread by any of these carpeting materials is so small that the old crust should be brought up to its full strength, as a strength crust sufficient for the traffic. Further, in using materials with bituminous binders, it is possible to effect so good a bond with water-bound materials in the lower course that the whole thickness acts as one crust, the square of the depth of this crust being the measure of its beam or slab strength as well as of its effect in spreading the load as a cone. It is, therefore, very necessary that such a condition should not be confused with the condition when the wearing layer of relatively costly materials is quite thin, only thick enough to serve as a wearing carpet. There is a strong tendency towards the adoption of this plan, because the cost of making one set of materials quite effective in wear-resistance and another set effective in strength and resiliency tends to be less than that of using materials which have both qualities.

The Waterproof Carpet.

If a road-bed is so situated that no water will reach it from beyond the edges of the crust, and if a layer is spread over it, effective in preventing the access of water to it but not preventing the gradual escape of all moisture, the crust will become dry and be no longer water-bound. Tarred roads which are thus rendered very dry sometimes compare unfavourably as regards corrugations with other roads which have been less effectively waterproofed, or which are kept moist by the creep of water in the road-bed.

Water-bound Broken-stone Crusts.

In the ordinary water-bound, broken-stone road crust the wearing layer is simply the top part of the strength crust. Usually, the stone in this top portion is harder or tougher than that in the rest of the crust and differently graded, and it is therefore necessary to take care that the change in materials as the crust is built up from the first course of stone is not accompanied, at any stage, by the formation of a plane of division. There must be no hard surface and no parting layer within the crust when it is finished.

Binding Material.

One of the most frequent causes of the too rapid wear of both new crusts and repairs, especially repairs, is lack of sufficient binder. The use of too much earthy binder instead of a gritty binder, such as stone dust, is also a cause of too rapid wear. The former weakness, lack of binder, is, however, a much more serious and widespread weakness than is usually supposed, especially on secondary roads in some areas and by-roads and lanes in other areas. If no suitable binder is available when holes are filled with loose stones, it is better to use mud than to use nothing at all, and mud is better than dry earth, because the water comes to the top.

Segregation.

It is very necessary to avoid segregation. A mixture of stone and binder, supplied in the proper proportions, is ineffective, and fails to form a tough and durable crust if there are patches in which one size or another is in excess of its proper proportion. This applies also to the finishing of a road surface by filling considerable hollows with chips before the final rolling. The crust when finished should be practically mass concrete with its own true surface. The final touches should be of the nature of grouting, not rendering.

The Telford Road Crust.

In the case of a crust of the Telford type, in which large stones are placed with their lengths transversely to the road, the spaces packed with chips or gravel, and a layer of broken stone and binder formed over this, it is necessary either to treat the lower course as a foundation and to make the upper course adequate as the strength crust, or, which is usually better, so to form the upper course that its stones penetrate the gravel or chips of the lower course, and the whole thickness forms one strength crust. On very bad, wet ground the former plan is the better, that is, to treat the lower course as a foundation only, and make it not too compact for effective drainage.

Single Crust Roads.

In addition to the preceding notes, it may be pointed out that if the single crust is to have a fair chance of success, the road-bed should be carefully prepared. It should be brought to the correct shape, usually cambered, and it should be made of the same degree of hardness throughout, by ramming, and adding earth and ramming again, and, where necessary, by loosening before ramming. When a bottoming of gravel, or large stones, or a mixture of both, is used there should still be some preparation of the natural soil if it is of varying degrees of density; and it should always be brought to about the right camber. Otherwise, material is wasted, and the degree of resistance in the foundation varies, tending to the development of humps and hollows in the crust. Hollows in the road-bed should not be filled with the foundation material, but with material as similar as possible to that in the road-bed itself. Road crusts should *not* be thinner near the edges than they are in the middle part.

Footways.

Location.—Apart from the main engineering aspect the advantages of locating footways aloof from the carriageway are obvious. On new roads there should be a strip of turf of ample width between the footway and the carriageway. In recent cases, avenues of trees have been planted in such strips, but clumps of smaller growth, or hedges interrupted by clumps and by gaps, would, in many cases, be more appropriate and much more useful. Relatively small trees are valuable for this purpose, such as maple, Cornish elm, bird cherry; also dogwood and allied species.

Where width between fences is restricted, as in many cases of road improvement, overhedge footpaths offer many advantages, a narrow footway alongside the carriageway being provided where expedient. Such overhedge footpaths are usually very well placed in respect of drainage, but suitable cross drainage works should be provided where needed. The footpath should be given a considerable camber, or single cross slope, in most cases.

Construction.—The road-bed of a footway should be carefully prepared. A foundation of clinker (unless destructor or other hard clinker) or one of ashes, may be worse than useless. Only in rare cases is under-drainage of the crust necessary. The crust should be made with as much care as are the carriageways, preferably, as a rule, in the same way as a good broken-stone road crust, and with heavy rolling by a narrow roller, or a rammer. The softer classes of road metal provide the best materials. Most gravels are to be avoided. Cinders and ashes are

abominations. The material should, in any case, be graded, and contain no large lumps. The surface should be that of the crust itself; if dressed with tar or bitumen, the dressing should be applied sparingly. No gritting.

TAR MACADAM.

There are in ordinary use two ways of forming a bituminous-bound road crust. If the stone is coated beforehand with the tar, and allowed to stand until the coating is tacky, the crust can be 'consolidated' with a moderate amount of rolling. The extent of consolidation will partly depend upon the extent to which the stone is graded. As a rule, very little grading is regarded as necessary, the surface being closed by rolling in ¾-in. to ½-in. material. The other kind of crust, formed by 'pitch grouting,' usually consists of stone graded through a wider range, the smallest material being sand, added with the pitch, which is poured in hot. The surface is closed with small material. For tar macadam the 'general directions' issued by the Road Board advised at least 60 per cent. of 2½-in. stone and not more than 30 per cent. of 2½ to 1¼ in., the remaining 10 per cent. not being in the body of the crust but being used for closing (¾ to ½-in. stuff). For pitch-grouting, the sizes are: 60 per cent. of 2½ ins. and 35 per cent. of sizes *graded* from 2½ ins. to 1¼ in. In addition 5 per cent. of chips are to be used for closing after the pitch has been grouted in. The body of the crust will, however, contain a proportion of sand, equal, by measure, to that of the pitch. The quantity of pitch varies according to circumstances, but ordinary quantities, noted in the 'general directions,' are 1¼ gall. per sq. yd. for a 2-in. layer, 1½ gall. for a 2½ in. layer, and 2 galls. for a 3-in. layer.

The pitch used (Road Board specification) is to be thinned with tar oil in the proportion of 10 to 12 per cent. (of the whole). The pitch itself should not have more than 22 per cent. and must not have more than 24 per cent. of free carbon. The tar oil must have a specific gravity between 1·065 and 1·075 at 20° C. and must be commercially free from light oils and water.

Tar-macadam road crusts have been successfully made with stone of considerably smaller average sizes than the above.

Cost of Plant.—A plant at Stourbridge, including a 2-ft. revolving cylinder, 10 ft. long, inside a 3-ft. shell, and driven by a 5½ h.p. gas-engine, cost 350*l.*, and with it some 40 to 50 tons can be heated and mixed by three men in one day. The proportions of cost shown by a recently erected plant at Belfast are: broken stone and screenings, 60·67 per cent.; tar, etc., 11·49 per cent.; wages and cartage, 21·25 per cent.; tools, oil, etc., 1·94 per cent.; repairs and fittings, 4·65 per cent.

Limiting Gradient.—For ordinary tar macadam about 1 in 25 is regarded as the limiting gradient, but 1 in 30 or even 1 in 35 is sometimes preferred.

Cold Emulsions.—Emulsions of tar, of bitumen, or of both together, applied cold, are now being used to a considerable extent for both grouting and surface treatment. There are two kinds of emulsions, the natures of which have been studied. By one method there is produced an unstable product which, however, may be suitable if it is made for use on a specific job. By another process, or processes depending on one and the same principle, there is produced a material in which each particle of tar or bitumen is coated with a skin of some substance which isolates it, so that, for a very long time, the particles all remain suspended in the water without segregating. This is the kind to buy for general use. The material as supplied by the manufacturers is sometimes intended to be used neat, in which case two or more strengths may be supplied for grouting and for surface treatment. Other manufacturers supply a material which before use is to be mixed with water in a proportion depending on the purpose for which it is to be used. The former practice is likely to prove to be the better. These emulsions can be used for grouting and for surface work on damp or even wet crusts. A damp crust is better than a dry one, on the principle of the damp swab, resistance to penetration being less than when the materials are dry. If, however, a road crust is full of water it should be allowed to drain to a sufficient extent before being grouted. In the case of surface work it will be sufficient if the crust has begun to drain, being somewhat pervious, or if, it being impervious, water has ceased to drain from it and heavy rain is not immediately expected. Emulsions may, therefore, be used all the year round, saving in long periods of frost or wet. In the spring, summer, and autumn they may be used on many days on which the hot treatment is impossible. Their adoption must depend considerably on costs, but when the cost is higher than that with the hot treatment they may be used as a stand-by in order that work may be less often interrupted; thus, possibly, reducing the total cost or sparing the public from inconvenience. In grouting work the material, if not to be diluted, may be used directly from the barrel, which is then the unit related to road surface area; or it may be poured from cans. It is brushed into the road surface in order to effect an even distribution. The surface is then blinded with chippings which, as in other cases, should be of more even and somewhat smaller gauge than is usual in such work. The quantity to be used depends, as in the case of other grouting, upon the proportion of voids in the crust, and upon its thickness. If the crust is continuous and thicker than the depth which has to be grouted, precautions must be taken to prevent the absorption of too much emulsion per square yard. In the case of surface treatment, the procedure to be followed and the quantities per square yard may be found after a few trials by any person accustomed to hot tarring work. Before using an emulsion which is to be diluted, the latest information should be considered.

A special purpose for which cold emulsions may be used is that of gradually strengthening an absorbent road crust, by small doses repeated at intervals. No generally conclusive results have yet been obtained in this practice (1929).

SURFACE DRESSING

Including Treatment with Tar, taken as the Typical Case; with Bitumen; and with Mixtures of Tar and Bitumen.

General Policy.—At the present time the best results seem to follow the practice of tarring once a year only, except where the tarred surface is completely worn off before the autumn, and allowing the tar and some of the stony material of the crust to wear down by the winter. This prevents the road from becoming jolty and provides good foot and wheel-hold for the winter. Tar which remains below the surface is not harmful, but excess of tar, producing a greasy emulsion on a hard road, or thick, greasy mud in damp places, is to be avoided.

Conditions affecting Results.—Wind and free play for local air currents are needed to keep tarred roads cool. Roads exposed to wind and air currents will suffer little from sunshine; roads in airless places suffer much from sunshine. Roads in cuttings, and those flanked by walls and close palings, and especially if they run across the direction of prevailing winds, suffer from high temperatures in summer and from greasiness due to damp conditions in winter. Shade in itself is beneficial, both as regards the effects of sunshine and with respect to the frequency of frosts. Porous road crusts absorb more of the tar than impervious crusts, which reduces difficulties arising from high temperature or from greasy conditions. A dry and showery climate is much more favourable than a damp and drizzly climate, as regards both the work of tarring and the results obtained.

Cost of Tarring.—An ordinary cost is from 2*d.* to 3*d.* per square yard, in many cases about 2½*d.*, and in not a few well above 3*d.* In the year 1910–1911 the Buckinghamshire roads, 117 miles, cost ⅔*d.* per square yard. Main roads often cost about 150*l.* a mile per annum for tarring alone. One gallon of tar is usually sufficient for spraying 5½ square yards of road surface, and a gang of six men with a 320-gallon boiler will tar-paint 5,000 square yards in an ordinary day's work. For recent figures see p. 755.

Nature of the Tar.—Tar of the right consistency can be got by adding pitch to crude tar. It is better, however, to get the proper consistency by heating the tar until the lighter constituents are driven off, since their presence is in itself a source of trouble. The Road Board specifications describe two tars, No. 1 and No. 2, the latter which is the denser, for use only on warm, dry roads, and suitable for painting worn places. Tar No. 1 has a specific gravity of 1·16 to 1·22 at 15° C. (1·19 recommended). It must have not more than 1 per cent. of water and ammoniacal liquor. The free carbon should not exceed 16 per cent. by weight. Tar No. 2 has a specific gravity of 1·18 to 1·24 at 15° C. (1·21 recommended), and must be free from water. The free carbon should not exceed 18 per cent. by weight. Various bituminous preparations may be used instead of tar on most roads, except water-bound roads, for which tar is more suitable.

Some recent specifications allow as much as 21 or 22 per cent. of free carbon. In two such specifications, (*a*) and (*b*), the following are included: Products distilling at less than 100° C., (*a*) and (*b*) not to exceed 1 per cent; products distilling between 170° and 270° C., (*a*) 12 to 24, (*b*) 16 to 26 per cent.; between 270° and 300° C., (*a*) 4 to 12, (*b*) 3 to 10 per cent.

Method of Application.—The road should be dry and warm. The surface should be brushed hard to remove all dust and loose particles, the second brushing being in some cases done with soft brooms. The tar in the boiler should be at about boiling point, say 220° to 240° C. Its temperature when delivered on to the road will depend on the mode of application. Machines resembling water-carts do best on smooth roads which are true to shape. The spray, a fine rose at the end of a flexible tube, is used by hand, with the advantage that the quantity of tar applied to different parts may be varied if necessary. The spray is very effective in coating the stones with a clinging film, after which the required thickness can be sprayed on as desired. It works well on uneven roads. Tar can be successfully applied by hand, poured on and brushed in.

Blinding the Surface.—When the freshly tarred surface can be left for some hours undisturbed, this is usually done. On porous roads, or where the quantity used is small, it may be sufficient to blind this somewhat hardened surface with dust only; clean dust from the preparatory sweeping or destructor dust. In other cases, varying quantities of sand are added, or stone chips, the quantities and method of application depending upon the extent to which these materials are intended to form a mat with the tar, or to act as a temporary protection. Chips may crush to a kind of filling coat, or they may be used so as to form a definite mat. Sand may be added so as to form a mat, or may be strewn a little at a time so as to avoid the forming of a too sandy and brittle mat, but at the same time to act as a parting layer between the tar film and the wheels of vehicles. Some of the sand, if not removed by rain, may be swept off when the tar has hardened and the road is becoming dusty. The true granites are not amongst the best of materials for blinding the surface.

Hot Bitumen Treatment.—The preceding paragraphs apply, substantially, to treatment with hot bitumen. The choice of tar or bitumen may depend upon costs, the nature of the road crust, or personal preference. Some surveyors prefer tar for the treatment of water-bound broken stone roads. Tar is often preferred for the first treatment in French practice, bitumen for subsequent treatments. The Northamptonshire County Highways Committee concluded, 1928, that bitumen is more durable and ultimately less costly than tar, but that tarred surfaces

are less slippery ; the county surveyor might therefore, at his discretion, prefer tar, except near fisheries and streams.

Effects on Fish Life.—Tar is more likely than is bitumen to harm fish life if rain comes soon after it has been applied. Generally, if safeguards are not fully to be relied upon, bitumen is preferred to tar for roads the washings of which may pollute ponds or streams. It has been shown (*Butterfield*) that after the primary risk has passed or been avoided by applying the surface treatment in fine weather, ill effects upon fish or water-plant life may result from the hashing up of the treated surfaces and the wearing down of the material to a fineness such that it is washed away by rain.

Safeguards.—The chief safeguards are : suitable weather for the work ; use of pure materials, properly specified ; prevention of hashing up of the film ; passage of the road washings through settling pools or ditches and through filtering materials ; dilution with purposely diverted clean surface waters.

Dust.—There may be ill effects (*Butterfield*) from dust produced from surfaces treated with tar or bitumen, either as such or as material removed by rain after a dry spell.

Cold Emulsions (see under 'Tar Macadam,' p. 732).—For surface dressing, cold emulsions are largely employed, in direct competition with hot applications, in which case there may be reasons for preferring one or the other. Cold emulsions can also be used, with full efficiency, when the roads are damp and, at the discretion of the surveyor, in rainy weather. In Herefordshire cold emulsions are used for the constant repairing of potholes (*G. H. Jack*); all noticeable potholes having disappeared after 12 months of this work.

SLIPPERINESS.

Some of the work done recently, on main roads especially, has produced very slippery surfaces. In some cases the road is regarded as dangerously slippery except when quite wet or quite dry, while there are some roads which are not satisfactory under any weather conditions. In the case of a surface which is, in the ordinary course, tarred from time to time in order to preserve it, it may be desirable to incorporate finely crushed flint or quartz with the tar ; in other cases the rolling in of crushed flint in hot weather may be desirable. The treatment depends upon the nature of the crust or carpeting, and no general directions can be given. To prevent slipperiness by a suitable choice of materials and methods, care should be taken to avoid unsuitable grading, the use of too much very smooth stone in the aggregate, the use of an unsuitable bituminous or asphaltic binder, and, generally, the employment of materials which either separately or in combination tend to form a grease under traffic. The crust or carpet should be reasonably resilient, a surface which rings musically under blows being avoided.

Very good results in producing non-slippery surfaces have been reached by Mr. E. J. Lovegrove, who puts down crusts of a carefully graded aggregate of crushed destructor clinker, the bituminous binder being made of a substance having a somewhat low 'penetration,' under the usual test. The materials should be so chosen and so compacted that they will not wear too slowly. In the case of water-bound roads it has been found necessary to use a very gritty binder of siliceous material in order to correct the tendency towards the formation of a greasy slime, which is a feature under road conditions, of some limestones and most granites, including all true granites.

Non-Slippery Road Crusts.—Non-slippery road crusts can be made in the tar macadam and tar-slag macadam classes, using coated material or grouting with hot or cold preparations. Also in the cement grouted class, with a fairly wide range in the choice of stone and using rapid-hardening cement.

See p. 753, 'Gladwell System.'

Silicated Roads.—Another class, of growing importance in this and other respects, is that of silicated roads—that is to say, roads in which the sodium silicate effects the binding of the whole crust. This method seems to be at present restricted to areas in which soft limestone is available, but it may be found worth while to modify it by importing the necessary proportion of such material to bind other stones. The French engineer Jacques-Thomas found that the proportion of silica to soda, initially 3·15 in a solution, was reduced by penetration into a piece of limestone after 24 hours soaking ; at 1 to 2 cm. within the stone, 3·0 ; between 2 and 4 cm., 2·9 ; between 4 and 6 cm., 1·5. (For data relating to the method, see p. 742.)

Rubber Mud.—After a long spell of dry weather on roads in Norfolk (S. H. Warren) skidding was frequent. The dust formed during the dry spell was analysed and found to consist largely of rubber, which formed a greasy mud with the water of the first shower.

The remedy for this would seem to be to sweep the roads at threat of rain after a long, dry period.

To reduce the effects of collisions due to skidding, there has been introduced the policy of providing guide posts which snap off very near the ground if struck hard by a vehicle. The principle, inversely stated, is that of refraining from putting up resistant fences or walls, instead of ordinary hedges, at dangerous bends and turnings. The guide posts, which are the first on this principle, have been designed for the R.A.C. and accepted by the Minister of Transport.

CONCRETE ROADS.

As regards country roads of ordinary width serious difficulties in the use of a concrete crust which is also the wearing layer, are: the necessity for allowing time to set; the practical difficulty in keeping one-half of the road closed for a month or more; and the resulting joint of hard to fresh concrete.

Alternatively, there is the difficulty, both in law and fact, of completely closing a road in long stretches for a period of a month *plus* the time it takes to get through with the concreting of the stretch of road in question.

The use of one of the rapid hardening or high alumina cements (such as 'Ferrocrete' and 'Ciment Fondu') with which rapid hardening concrete can be made has overcome this difficulty, and concrete strength crusts for arterial and by-pass roads are steadily growing in favour. The naked concrete road crust, though regarded as experimental in many specific cases, must now be considered as competing with other kinds of road surfacing.

Generally.—The weakness of joints, as regards strength and wear, is one of the chief difficulties in the adoption of the concrete wearing crust. It has been found that joints become 'frayed,' and open up. As in the case of strength crusts of concrete under paving or carpeting, the whole crust of concrete suffers from the effects of street openings and local repairs, but in a greater degree. These difficulties are not insuperable, but they are serious especially as regards country roads.

When joints are not provided, or are insufficient, concrete roads crack very badly, as has been the case to a large extent in the United States.

REINFORCED CONCRETE ROADS.

When the slab is definitely a foundation, under a strength crust, light reinforcing is sufficient and usually single reinforcement. In this case the lengthwise reinforcement may be more than the crosswise reinforcement in order to prevent the forming of temperature cracks across the road. When the slab is to be the strength crust, or the principal strength crust, double reinforcement is usually to be preferred, and in most cases is the proper method structurally. When the purpose of the slab is to extend the supporting area of an insufficiently firm but equally resisting foundation, the lower reinforcement may be stronger than the upper—more or thicker rods. When the purpose of the slab is to combine the unequal resistances of different parts of an unequally resisting foundation the upper and lower reinforcement should be about equally strong. Because the little piece of the slab under the wheel is supported from athwart the wheel as well as from fore and aft, the slab should be reinforced across the road about as strongly as it is along the road. Because a wheel close to the edge of the road slab is unsupported from one side, the slab must either be reinforced as a cantilever near the edges, or the lengthwise reinforcement must be increased to the actually necessary extent by closer spacing of the rods. There are alternatives, applicable under special conditions, such as extending the slab under the footway, or converting the edge of the slab into a kerb with the proper reinforcement in the vertical plane. When the calculations show that it it demanded, there should be diagonal reinforcement in the vertical planes of a road slab. A reasonable base should always be provided on very soft or treacherous ground, including otherwise firm ground which may suffer denudation by the action of trickling water. A slab cannot float as a raft nor sustain its own weight and the live loads if the average resistance of the soil is unequal to these. A slab provided with parapets, of a piece with it, may, however, be worth considering as a means of carrying a road across a bog.

Depth of Slab.

Much thinner slabs than those usually employed would be theoretically sufficient in most cases, and usually more economical of materials, but practically it is found to be better to have thicker slabs for the following reasons: (*a*) With a deeper spacing between the upper and lower grilles, less steel is needed, the mesh can be larger, larger stones may be used in the aggregate, and the filling in of the concrete is done more easily. (*b*) Any fairly large piece of road metal, left accidentally on the foundation or in the aggregate, may lead to the formation of a tension crack on the under side of the slab. (*c*) In some cases the depth which must somehow be filled is not sufficient to make it worth while to provide a firmly rolled hard-core foundation, the cheaper solution being to employ a relatively low-stressed slab, with a somewhat reduced proportion of cement and relatively light reinforcement. (*d*) Lastly, 7 ins. of fairly tough concrete, with large-mesh reinforcement, is more easily cut through, for excavations, than 4 ins. of very tough concrete with a smaller mesh reinforcement.

The advantages of having two layers of reinforcement are now being recognised, the value of the steel in two layers being greater than that of the same quantity in one layer. In some cases there is also reinforcement in vertical planes: an example of this being vertical triangular reinforcement, between an upper and a lower grille, in Paddington.

NAMES OF ROCKS USED FOR ROAD METAL.

When a single word is employed as the name of a rock used for road metal or for paving setts the word may be definite in itself, denoting a rock of a particular composition which does not, as a rule, vary much from the average. In other cases, the general word, or word denoting the normal material, is compounded with another word, usually the name of a mineral which is present in an unusually large proportion, or which is an unusual constituent or one of several, any of which may be present. Thus, a diabase is usually composed chiefly of labradorite and augite, and sometimes olivine; and dolerite, though different in texture, is similar in composition. An olivine dolerite is a dolerite in which olivine is present in a considerable proportion, and a quartz-enstatite diabase is a diabase in which both quartz and enstatite (minerals, not rock names) are present in noteworthy proportions.

The following classification was suggested some years ago (*vide* 'The King's Highway') as a tentative grouping. It is here slightly altered.

Granites: True granites, horneblende granites (micro-granites).

Felstones: (*a*) Basaltic felstones:—Basalts, diabases, dolerites.

(*b*) Hornblendic felstones:—syenite, diorite, quartz-syenite, quartz-diorite.

Traps: (*a*) Whinstones—basic traps.

(*b*) Elvans—acid traps, quartz-felsites, quartz-porphyries (micro-granites).

Quartzites: both metamorphic and sedimentary.

Limestones: compact limestones, crystalline limestones.

Rags: calcareous sandstones, siliceous limestones.

Sandstones: true sandstones, grits.

Flints: flints, chert.

Pebbles: large broken pebbles to be named according to their class, small pebbles classed as gravels.

SOME IMPORTANT WORDS IN THEIR NARROWER APPLICATION.

The following are definitions of certain important rocks:—

Granite.—A coarsely-grained, wholly crystalline rock, mainly composed of alkali felspar, quartz, and a ferro-magnesian mineral.

Gneiss.—A metamorphic (altered) rock, usually resembling granite, but showing a foliated or banded structure.

Syenite.—A wholly crystalline rock, composed of orthoclase felspar, a ferro-magnesian mineral (usually augite or horneblende), and a small proportion of quartz.

Diabase.—A compact, dark rock, mainly composed of labradorite, augite, and sometimes olivine; is thoroughly crystalline and often ophitic (ophitic: a structure in which the crystals of one mineral inter-penetrate or are moulded upon those of another).

Dolerite.—Often classed as basalt, or as diabase. (The terms 'diabase' and 'dolerite' are often used synonymously.) Is more crystalline than basalt.

Diorite.—Consists essentially of oligoclase, andesine or labradorite felspar, and horneblende, augite, or biotite.

Basalt.—A black compact lava, basic (alkaline) in character. Consists essentially of plagioclase, augite, and olivine, is sometimes porphyritic, and often contains uncrystallised glass substance in the ground mass.

Porphyry.—Contains large crystals, usually of potash-soda felspar, in a ground mass of felspar, quartz, and other minerals.

Andesite.—A lava, intermediate between acid and basic, consisting principally of plagioclase, felspar, and ferro-magnesian minerals, usually without olivine.

Quartzite.—Composed almost entirely of close-fitting quartz grains, the cementing material being itself quartzose, distinguishing it from sandstones.

Whinstones.—In its narrower and more correct use the term is applied to dyke or trap rocks. A typical whinstone may be classed as an andesitic dolerite, containing augite, or as a quartz diabase.

Bituminous Macadam Roads.

(Abbreviated from a description by A. W. Dean, Chief Engineer, Massachusetts Highway Commission.)

A gravel, or suitable sand, foundation may not be needed. If a stone foundation is made on a clay subsoil, the interstices should be filled with gravel to prevent the clay working up, but if one is used care should be taken to use suitable materials, especially the gravel (see p. 389).

First course.—The thickness of the first course, of broken stone, depends on local conditions. For instance, if a 10-in. foundation is necessary, the first course of the crust need only be about 2 ins. thick. If no foundation is necessary, the first course of the crust should be not less than 4 ins. after rolling, or 6 ins. for heavy loads. The stones of this course may vary from ½ in. to 3 ins. in longest dimensions, but the percentage of the small size should be very small. For a course of 4 ins. or less a good size is 1½ ins. to 2½ ins. In a 6-in. course the maximum size may be 3 ins.

The voids must be well filled with fine material to prevent loss of bitumen from the upper course. The surface must be even. All surplus dust and fine material must be swept off, leaving the upper stones bare to receive the second, or bituminous, course.

Bituminous course.—The second, or bituminous, course to be 2 ins. thick, or 3 ins. for heavy vehicles in large volume. The stone must be of good quality and uniform in character, and the sizes not segregated. The best results are obtained with stone from ¾ in. to 2½ ins., the larger sizes predominating, although the course be only 2 ins. thick. The rolling should be thorough.

Application of bituminous material.—This is applied at a temperature of 250° to 300° F., with a mechanical distributor, with a force equivalent to that obtained under a pressure of 30 lbs. per square inch, or more. The amount per square yard is from 1¼ gallons on a 2-in. course to about 2 gallons on a 2½-in. course. It should be very evenly spread.

Surfacing.—The surface should be lightly but completely covered with pea stone (½-in. stone) and thoroughly rolled with a 15-ton roller, if available, and if the stone is of hard quality, and a lighter roller for a poor quality stone.

Seal coat.—A seal coat of the same quality of bitumen is applied, half a gallon per square yard, covered with pea stone, and finished off with a heavy roller. When the bitumen is applied the stone must not be wet but may be moist. It must be clean and free from dust.

Bituminous Carpeting.

(Abbreviated from a description of work carried out in Kent. County Surveyor of Kent, H. T. Chapman, M.Inst.C.E.)

The work is done in two operations. The base coat of local stone, flint, scarified material, or clinker, mixed with bituminous material, is laid hot to form a nearly voidless concrete, not less than 3 ins. thick. For the carpet, sand, grit, or selected crushed clinker is graded between a 200 and a ¼-in. mesh and mixed with from 13 to 15 per cent. of bituminous material, and laid at a temperature of 300° to 350° F.

In a specific case, 1914, the base coat was 3 to 4 ins. thick, of crushed shingle and flint, 1½-in. to ½-in. gauge, coated with a pitch and tar mixture, 8 parts soft pitch to 3 parts tar. The carpet was 1½ ins. thick, of grit 31 per cent., sand 56 per cent., and bitumen 13 per cent. All these ingredients were heated to 350° F. and mixed. The carpet was first cross-rolled and then consolidated with a 10-ton roller.

Costs.—Costs were, base coat, 2*s.* 6*d.*, carpet, 2*s.* 7*d.*, total, 5*s.* 1*d.* per square yard, the prices of the materials being at practically pre-war rates.

The results have been very satisfactory.

According to the remarks of several speakers at a meeting of the Institution of Municipal and County Engineers, asphalt macadam, as a wearing surface for roads should be laid in one coat. It can apparently be laid in one thickness of 3½ ins. quite satisfactorily, whereas if put down in two layers the upper part creeps or peels off under traffic.

Concrete Roads, Including Wearing Surface.

Chelsea Embankment.—Much traffic, including very heavy vehicles. A naked concrete roadway, in slabs about 50 ft. long and 4 in the width of the carriageway. Upper and lower grille reinforcement. Lengthwise joints formed by laying fresh concrete against hardened concrete, and run with bitumen. Narrow transverse space joints filled with bitumen. Fine cracks appeared in a few slabs in the first cold weather after placing the concrete. These gave no trouble except where across the sharper than right-angle corners formed by the skewing of the transverse joints, but this tendency to breaking off of the corners was checked by the use of bitumen, which has also been used to protect all joints and cracks. No other maintenance measures needed. Condition very good after 40 months, slight depressions at a few corners, but no further breakage. One of the best surfaces in London.

Steel-Mesh Road Reinforcement (Proprietary).—A series of metal strips bent into zigzag shape, laid on edge, side by side, and locked together by means of pins into a continuous mat of steel squares. Has been used on a tramway route in Birmingham. Is expected to be useful at omnibus stopping places. Imbedded in naked concrete at certain spots on Chelsea Embankment, its value cannot yet be estimated because that concrete crust is, so far, generally in good condition; but no signs of the steel-mesh doing any harm can be seen.

Life of Concrete Road Crusts.—New King Street, Deptford; laid 1917; 9-in.; double reinforcement; heavy traffic; in serviceable condition, 1927. Tilbury Dock Road: laid in 1917, in alluvium; 7-in.; single reinforcement; in good condition, 1927. Canmore Street, Dunfermline: laid 1916; in satisfactory condition, 1927. Southwark: 130 roads and streets laid since 1918 give satisfaction. Main Bridgewater Road, laid 1920: no expenditure on maintenance and very slight signs of wear, 1927.

Points from Ordinary Practice.

When making a reinforced concrete road slab in half-widths at a time, the transverse rods of the first half must be left projecting sufficiently to be effectively bonded into the second half. Unless there is a considerable margin of strength, the weakness across the middle line should be countered by providing additional lengthwise reinforcement on each side of it. The road-bed should be drenched before the concrete is laid. If it be clay, or a clayey soil, hard-core should be rolled in. The concrete must not be in the least sloppy; if excess water be removed by rolling, this should be done transversely, with a light roller. If the slab provides the wearing surface a tamping finish may be sufficient, and floating should be done very sparingly; the belt finish is better. The concrete should be protected by a tarpaulin for a day or two, and, anyhow, be amply protected from frost. It should be matured by waterings for ten to fourteen days, and not opened to traffic for two or three weeks under very favourable conditions, being kept closed for much longer periods under ordinary or adverse conditions. For the sake of economy of concrete the road-bed should be brought to the same camber as the surface. The proper proportions for the concrete can only be decided by the designer, but in order to provide a starting-point for the calculations, it may be noted that a fairly ordinary proportioning for slabs of about 6 ins. in thickness is 1 : 2 : 4, with 1 : 1¾ : 3 for the surface, or, if the fine material be graded from ¼ in. to dust, 1 : 1½ : 3 throughout. Reinforcement should be accurately placed in position before concreting, and should be, preferably, not more than 2 ins. nor less than 1½ in. from the surface or bottom of the concrete. On clay soils a bottoming, such as clinkers, should be provided, and, generally, a suitable bed prepared for the concrete. Expansion joints should be provided at intervals of 50 ft. or less if the slab is reinforced, or at not more than 30 ft. if it is not reinforced.

Analysis of Costs.

Billingham, urban district (*G. H. Hill*).—Main roads on building estate; shillings per sq. yd., stripping, forming, and laying ashes, manual labour, 1·57; team, 0·61; concreting the carriageway, manual labour, 1·03; team, 0·11; cement, 1·59; gravel, 1·91; reinforcement, fabric, 0·84; petrol for mixer, 0·04; hire of mixer, 0·25; use of tools, 0·12. Total, 8·07 shillings per sq. yd. of the 6-in. slab of 1 to 6 gravel concrete.

Drying and Mixing Plant.

The principal essential parts of a Standard Plant are: (1) elevating skip, (2) drying drum or drums, (3) furnace, (4) flue and fan, (5) mixer, (6) driving pulley and gear, and (7) underframe, which also forms the operating platform. The plants can be hauled from place to place by a steam-roller or traction-engine, and they are almost invariably belt-driven by the same engine—a pulley being fitted for the purpose.

The smaller portable plants (Urban and No. 0) are sometimes fitted with a direct-coupled petrol engine, mounted on a platform, but the largest portable plant (No. 1) is driven by a separate power unit—usually the tractor that hauls it. The stationary plants are invariably belt-driven by a separate steam, petrol, or gas engine or electric motor though a direct-coupled power unit could be fitted to the platform if required.

The aggregate to be dried is either fed by chute direct into a fixed measuring hopper or shovelled or tipped into the skip, which measures a definite batch. The skip is hoisted by power, strikes the auto-knockout gear, and quickly and cleanly discharges the whole batch into the fixed receiving hopper without spilling.

In the Urban plant this receiving hopper has a counterbalanced door which automatically opens and passes the batch into the drying drum and then closes again. This means that once an Urban batch has been discharged from the skip, it must pass into the drum. On all the larger sizes the door of the receiving hopper must be opened by a hand lever, so that a batch can be kept in the receiving hopper any length of time, thus making it possible for *five* separate batches to be handled by the plant at the same time—one each in skip, receiving hopper, drying and heating drums, and mixer.

From the receiving hopper the batch is passed into the drying drum or drums; then, when all the moisture has been evaporated and the aggregate heated to the required temperature, it is

passed by hand lever into the mixer, from which, after being *visibly* thoroughly coated with the hot binder that is poured in, the whole batch is discharged into the receptacle beneath.

The Urban plant takes 12 to 15 b.h.p., which is ample to drive the plant and any auxiliary equipment, such as a tar or bitumen pump.

TABLE V.

APPROXIMATE AVERAGE OUTPUTS PER HOUR OF TAR AND BITUMINOUS MACADAM, GIVEN BY BATCH-TYPE COMBINED DRYING AND MIXING PLANTS.

Type of Plant	Portable or Stationary			Stationary only	
No. of Drying Drums	Single Drum	Double Drum		Double Drum	
Size of Plant	Urban	No. 0	No. 1	No. 2	No. 3
Batch Capacity	6 cu. ft. or 0·17 cu. metres	6 cu. ft. or 0·17 cu. metres	11·5 cu. ft. or 0·33 cu. metres	18 cu. ft. or 0·5 cu. metres	27 cu. ft. or 0·76 cu. metres
Tar Macadam :—	Output per Hour.	Output per Hour.	Output per Hour.	Output per Hour.	Output per Hour.
2½ in. . . 5·7 c.m. . .	4 tons or 4,064 kilos.	6 tons or 6,096 kilos.	12 tons or 12,192 kilos.	16·5 tons or 16,764 kilos.	24 tons or 20,400 kilos.
1½ in. . . 3·8 c.m. . .	3·5 tons or 3,556 kilos.	5 tons or 5,080 kilos.	10 tons or 10,160 kilos.	14·5 tons or 14,732 kilos.	22 tons or 22,352 kilos.
¼ in. . . 0·635 cm. .	3 tons or 3,048 kilos.	4·5 tons or 4,572 kilos.	9 tons or 9,144 kilos.	12 tons or 12,192 kilos.	19·5 tons or 19,812 kilos.
Bituminous Macadam	*Up to about 25 per cent. less than above figures given for Tar Macadam.*				

The above output figures are based on aggregate weighing 100 lbs. or 45 kilos. per cu. ft., and an output of one batch every 2½ minutes.

BRICK ROADS.

(By Alfred B. Searle, Author of 'Clays and Clay Products,' etc.)

Bricks used for roads should be so hard and tough that they will resist the crushing, vibration and abrading actions of the traffic. They must be made of suitable materials, such as marls and shales, which yield dense, heavy and impervious product when burned. The clays used for common building bricks are quite unsuitable, being too readily disintegrated. Bricks for roads should consist of a tough skeleton, the interstices of which are filled with a tough slag-like mass of great viscosity when molten. When broken, such bricks show a dull, glassy structure, not readily affected by vigorous blows from a hammer. The modulus of rupture R should be at least 1,500 lbs. per sq. in. It is determined by supporting a brick on two knife edges 1 in. apart, and applying a pressure or load of W lbs. by means of a central knife edge. If the brick is b in. wide and d in. deep,

$$R = \frac{3Wl}{2bd^2}.$$

As bricks vary considerably, the test should be made on at least six bricks. When ten bricks are placed in a rattler with 10 cub. in. balls each weighing 7½ lbs. and 245–260 smaller balls of 1 lb. each (making a total of 300 lbs. of balls) and the rattler is rotated 1,800 times at 30 revs. per min., the pieces of brick more than 2 ins. diameter should not weigh less than 75 per cent. of the original weight of the bricks.

The size of bricks used for roads is of minor importance, provided that all the bricks used on a job are sufficiently uniform, i.e., with a variation of not more than ¼ in. in length and ⅛ in. in depth and width. The size usually preferred is 9 ins. by 4½ ins. by 3 ins.

The best results are obtained by using bricks with lugs (A, A, figs. 1 and 2) and a double bevel or bulge of $\frac{3}{32}$ in. at each end to ensure a joint at least $\frac{3}{16}$ in. wide at the top and the bottom of the brick. These are no more expensive to make and they 'key' better than plain bricks. *Pressed* bricks should not, as a rule, be used for roads, as they are less durable than those made by the wire-cut process. Bricks should be laid with the rough, wire-cut sides vertical and in contact with the filler so as to ensure the latter being retained properly. This is almost impossible with the smoother surface possessed by pressed bricks.

The *foundation* of the brick road must be good and well drained. A 1 : 3 : 6 concrete foundation 6 in. thick is best, but an old macadam road, if well graded and rolled, is satisfactory.

The *cushion* or *bedding course* may be of (i) sand, (ii) sand and dry Portland cement, or (iii) mortar. A *sand bed* should not be more than 2½ ins. thick and consist of uniform grains, and the sand used must be quite dry. It may be consolidated by rolling under a pressure of 15 lbs. per sq. in. A *cement-sand bed* should consist of 1 part of dry Portland cement to 3 parts of sand, well mixed in a dry state and spread on the foundation to a thickness of ½ in., any surplus material being removed with a straight-edge. This bed must not be rolled. The bricks are laid dry on this cushion and are then saturated with water so as to cause the cement beneath to set.

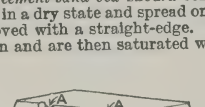


FIG. 1.

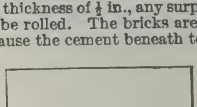


FIG. 2.

The *joints* should be filled with a 3 : 4 mixture of Portland cement and sand, the latter passing completely through a No. 20 sieve, but being wholly retained on a No. 100 sieve. The cement and sand should be mixed in the dry state, the mixture delivered to the road and mixed with water in a concrete mixer as required. The resultant slurry should be received in spouted cans and poured from these in a thin stream between the joints of the bricks, the fluid being well brushed into the joints. Two or even three applications of slurry may be needed, and are better than filling the joints at once. The second and subsequent quantities of slurry should be more viscous than the first one, and between each application any surplus slurry should be pushed (not drawn) into the joints with a rubber squeezer. The men should not walk or stand on the finished roadway when at work.

Maturing is effected by covering the road with sand which is kept well wetted for several days, so as to allow the cement to set properly. If this precaution is not taken the cement will be weak through a deficiency of water. The sand then may be allowed to dry, after which it can be used on another section of the road.

The road should not be used until three weeks after filling, as if the grouting is disturbed before it has hardened sufficiently the joints will be weak and the road will not be durable. If a road must be open for traffic in a much shorter time, a cement filling is little or no better than plain sand which is merely brushed into the joints, a small surplus being left on the surface of the road to be worked into the joints by the traffic; or a hot mixture of tar and sand may be poured into the joints. If time is less important, the cement-filled joints are far the best.

NOTES ON BRICK ROADS.

Experiments are being made in London with the surfacing of concrete road foundations with bricks, in order to carry heavy road traffic. The first bricks supplied from France for this purpose were laid in October 1927 at the Great Western Railway, Westbourne Park Goods Yard, and the entrance to the Municipal Depot of the Chelsea Borough Council has now also been paved with these bricks. In all cases the bricks will be subjected to exceptionally heavy traffic.

The bricks used are reddish-brown externally, while the interior is blue or mottled red. They are tough and strong, heavy and compact. Their weight is, on an average of ten bricks, 136·5 lbs. per cubic foot, and their absorption is less than 1 per cent. When laid on the flat and crushed, they split vertically, and have as high a crushing strength as first-class Staffordshire blue bricks. The characteristics which make them particularly suitable for road surfaces are: (1) Regularity of shape and size; (2) an even grain at right angles to the wearing surface; and (3) a texture which affords a firm foothold.

The method of construction which has been found to be most satisfactory, according to experience in France, is as follows: Fine, damp sand, free from small stones, is laid on a properly-cambered and matured concrete foundation. The sand is screeded to a uniform depth of ¾ in., and then pressed down by hand. The bricks are laid on this, as close as practicable to one another, in the manner of wood-block paving. They are then rolled by a three-ton roller. If any of the bricks break under this treatment, they can be pulled out quickly by a special pair of tongs and replaced by others. The joints are then filled with Texaco, or some other suitable asphalt, at a temperature of 350° F. This is poured in from a melting kettle. An alternative

method is to use a grout made with quick-hardening cement for the joints. Traffic can be allowed on the road a few hours after the asphalt joints are finished, or in a few days when cement is used.

Brick paving is used to a considerable extent in the United States, where both bitumen and cement filling have been used. While the former has mostly proved satisfactory, the latter has, under certain climatic conditions, proved quite unsuitable, actual explosions of paving having occurred, with considerable damage to passing traffic.

The following conclusions have been drawn as the result of tests made by the United States Bureau of Public Roads: (1) Bricks should be bedded on sand in preference to a cement-sand bed. (2) The thickness of the bricks should be at least 2½ ins. and, to allow for wear, should be preferably thicker. (3) The super-elevation on curves should be sufficient for a reasonable speed in order to prevent excessive wear on the track by the outer wheel. (4) Considerable care should be taken to obtain suitable vertical fibre bricks of high strength and uniform quality.

It is generally considered that this class of work is suitable for heavy traffic and where renewals must be effected quickly. The fact that some two million square yards of this type of paving have been laid in the United States since 1905, and that it continues to be adopted, indicates that it has been found to be satisfactory.

POINTS FROM SPECIFICATIONS.

From Road Department Specifications.

(The first six paragraphs apply to the re-surfacing of water-bound macadam.)

Thickness of Crust.—On a good natural foundation, old and new crusts together not less than 4 ins.; nor under ordinary circumstances less than 5 ins.; on yielding subsoils not less than 9 ins. New coatings of more than 3 ins. should be applied in two courses, separately rolled.

Cross-fall.—The finished surface should have a cross-fall of 1 in 24.

Stone and Screenings.—The stone should be broken as cubically as possible. About 70 per cent. should pass through a 2½-in. ring, and not through a 1¾ in. ring. About 20 per cent. should pass through a 1¾-in. ring, but not through a screen with rods ¾ in. apart. The screenings from this screen should be kept for top-dressing.

Rolling.—The roller should weigh about 10 tons, and be worked from the sides towards the centre, the correct cross-fall being attained before adding water or binding.

Binding.—The binding should be the best obtainable. It should be either of the same material as the new coating or of granite, limestone, or slag chippings, or failing these, suitable pit-sand or gravel, and the largest stone in it should not exceed ¾ in. in its greatest dimension.

Re-rolling.—In some cases it is advisable that a steam-rolled water-bound macadam surface should be lightly watered and re-rolled from a week to a fortnight after the first rolling.

Surface-tarring a Water-bound Macadam Road.—The road should be scraped or brushed in the early months of the year, in wet weather. Tar No. 1., Road Department Specification, should be used and be heated to between 220° and 240° F. The spouts of pouring cans should be not less than 1½ inches in diameter at the orifice. A gallon should cover from 5 to 7 square yards. Delay before gritting is advisable. On roads with heavy traffic it is desirable to tar the road again, or a central strip of 9 to 12 ft., two or three months after the first tarring; a gallon to cover 8 to 10 square yards.

Pitch-grouted Macadam.—The finished thickness should be 2½ to 3 ins., or on very light trafficked roads 2 ins., for single grouting, and 4 to 4½ ins. for double grouting. The cross-fall of the finished surface should be 1 in 32. Of the broken stone aggregate, 60 per cent. should be 2½ ins. and 35 per cent. graded 2½ to 1½ ins. For closing down after grouting, 5 per cent. of chippings, ¾ to ⅜ in. The pitch to comply with the Road Board Specification, the quantities applied being: for a 2-in. coating, 1½ gallon per sq. yd.; for a 2½-in. coating, 1⅔ gal. per sq. yd.; for a 3-in. coating 2 gals. per sq. yd., but the quantities needed may vary, and the voids must be filled. The pitch must be raised to a temperature of 300° F. and the sand to 400° F., and the matrix formed by mixing the pitch and the sand in equal parts, and kept stirred till it is poured. The final rolling to begin immediately after the pouring of the matrix and carried on rapidly. The 5 per cent. of graded chippings should be spread in part previously to the rolling, and the remainder during its progress.

Double Pitch Grouting.—The lower layer should be the thicker and consist of large stones, 3-in. to 2-in. gauge; a cheap local stone suitable for foundation work may be used. No chippings are required for the bottom layer. The upper layer aggregate of hard road stone 1½ in. gauge. The chippings should be graded from ½ in. to ¼ in. The quantities of pitch required are 3¼ gals. per sq. yd. for 4-in. work and 3½ gals. for 4¼-in. work.

Points from other Specifications.

Clinker Asphalt Paving (Sheffield: Mr. W. J. Hadfield).—Hard-burnt destructor clinker is used, crushed and screened; ½-in. mesh for the bottom coat and ¼-in. mesh down for the top coat. The rejections are used for ordinary tar macadam. The material is heated to 300° C. There is about 10 per cent. bitumen in the bottom coat and 17 per cent. in the top coat. The bottom coat is spread to rather more than 1 in. in thickness and slightly rolled. The top coat has a thickness of rather less than 1 in. when rolled, giving a total of 2 ins. Rolling is continued till no further impression can be made on the material. A 12-ton roller is preferred.

Manufactured Asphalt (A) ('Highway Engineering,' Blanchard and Drowne).—This is a combination of native bitumen and refined asphaltic oil. Specific gravity at 25° C. between 0·965 and 1·000. Ninety-nine per cent. soluble in carbon disulphide. To contain between 18 and 25 per cent. of bitumen insoluble in 88° Baumé paraffin naphtha. Between 5 and 10 per cent. of fixed carbon. Penetration between 150 and 200, with a No. 2 needle, weighted with 100 grammes; 5 seconds, 25° C.

Manufactured Asphalt (B) (Blanchard and Drowne).—This is manufactured by fluxing a refined native bitumen. Specific gravity 1·010 to 1·045 at 25° C. Ninety-two per cent. soluble in carbon disulphide. Between 18 and 25 per cent. of bitumen insoluble in 88° Baumé paraffin naphtha. Between 9 and 13 per cent. of fixed carbon. Penetration 100 to 200, with a No. 2 needle, weighted with 100 grammes; 5 seconds, 25° C.

Asphaltic Carpet.—British report to Fifth International Road Congress; type specification; bitumen soluble in carbon disulphide, 12 per cent.; aggregate passing: 200 mesh, 17; 100 mesh 15; 80 mesh, 15; 50 mesh, 25; 40 mesh, 8; 30 mesh, 4; 20 mesh, 2; 10 mesh, 2 per cent.

Granited Rock Asphalt.—(Same report): Type specification of gauging; fine powder, passing 100 mesh, 30 to 40; passing 50 mesh, 20 to 30; passing 30 mesh, 10 to 20; passing 10 mesh, 5 to 15 per cent.

Single-coat Asphalt-Macadam.—(Same report): Type specifications; (*a*) 3-ins. thick on poor foundation for moderate traffic; 1-in. stone, 35; ½-in. stone, 35; graded sand, 18; filler (such as Portland cement), 5 per cent.; bitumen, 7 per cent.; (*b*) 2-ins. thick on sound foundation for heavy traffic; ½-in. stone, 33; graded sand, 45; filler, 10; bitumen, 12 per cent.

Two-coat Asphalt-Macadam.—(Same report): For total thickness 3 ins.; binder course 2¾-ins., wearing course 1-in. Binder course, bitumen 5, sand 15, broken stone all passing 1½-in. ring and not more than 5 per cent. passing 10 mesh sieve, 65 per cent.; wearing surface, all passing ⅜-in. mesh sieve; passing 10 mesh and retained on 40 mesh, 10 per cent.; 40–80 mesh, 20; 80–200 10; passing 200 mesh, 12; bitumen soluble in carbon disulphide, 10·5. For 4½-in. total thickness as above, but 3½ and 1½-in. courses; binder course, 7 per cent. bitumen; sand, 25 per cent.; the broken stone, 85 per cent.; wearing surface, all passing ⅜-in. mesh; 10–40 mesh, 35; 40–80, 55; 80–200, 30; passing 200, 20 per cent.

Concrete Road Crust.—Epsom R.D.C., Class B road, 2,752 tons per day; reinforcement to have ultimate resistance across the road of 12,500 lbs. per foot; slab, 7 ins. of shingle concrete finished with 2 ins. of granite concrete; shingle portion: aggregate, by weight, 54 per cent. passing 1½ to ¼-in. square mesh, 24 per cent. passing $\frac{5}{16}$ square mesh, 22 per cent. passing $\frac{3}{16}$ mesh; sand, all passing $\frac{3}{16}$, not more than 5 per cent. passing 100 mesh sieve; cement, rapid-hardening ('ferro-crete'), 370 lbs. to each wetted and mixed cubic yard of concrete; surface layer, 1½ in. dolerite containing sufficient ¼-in. to ⅜-in. chippings to reduce the voids to 30 per cent. by volume; of this 3 parts; sand 1 to 1½ parts; rapid-hardening cement, 550 lbs. per cubic yard. This cost 9*s.* 8½*d.* per square yard for carriage way 20 ft. wide. Cost, including excavation and surface treatment with bitumen, 9,086*l.* per mile.

Concrete Road Crust.—Southwark; lower course, contains 18 per cent. of stone ¾-in. to 2-in. gauge; upper course, 3 of crushed gravel to 1 of cement; the gravel, 8 of ⅜ in. to ¼ in., 17 of ¼ in. to ¼ in., 75 of ¼ in. to dust, per cent. Such a crust, on roads with 22,000 tons of traffic per week, is treated with silicate of soda, with good results.

Silicated Road Crust.—French practice (Jacques-Thomas): aggregate, hard or semi-hard Jurassic limestone or similar porous limestone; gauge, passing a ring of 1½ in. to 2 in. diam.; coarse sand (the finer kinds, having a tendency to form pasty mortars, are to be avoided). About 40 litres of silicate and ½ cub. m. of sand to each cub. m. of broken stone; say, 6¾ gals., ½ cub. yd., 1 cub. yd. Maximum, of silicate per cub. m., 55 litres, say 9½ gals. per cub. yd. Proportion of silica to soda in the solution, 3·0 to 3·5, the lower figure for porous materials (see p. 734).

Bitumen.—For use in road construction (suggested specification 'Modern Road Construction,' Francis Wood). The bitumen should not contain more than 15 per cent. of fixed carbon. The portion insoluble in carbon disulphide at air temperature shall wholly consist of mineral matter and not more than 4 per cent. carbonaceous matter. Not less than 60 per cent. nor more than 75 per cent. shall be soluble in 88° naphtha at air temperature. It shall be soluble in carbon tetrachloride to the extent of not less than 1½ per cent. less or 1½ per cent. more than the solubility in carbon disulphide. It shall not contain more than 2½ per cent. of paraffin scale.

Nature of Pavings used in London.

Based on the Metropolitan Paving Committee's Annual Reports up to and including the 22nd Report, 1929. The streets, except in the City of London, are classed as First Class, Second Class, and other streets.

The London area offers a favourable field in the study of street pavements. The climate does not approximate to any of the extremes of the British Isles, the gradients are variable, but seldom very steep, and the market for materials is a good one. The traffic conditions which have to be faced include heavy dock traffic, very busy city traffic, that of residential roads, motor omnibus routes, and the great main roads leading to the country. Lastly, the reports of the Metropolitan Paving Committee provide the materials from which deductions can be made as to the value and importance of different pavements, and including comparisons as to wearing qualities and costs.

Percentage Mileages of Different Pavings, 1927–1929.

Table VI. has been prepared on the basis of a table given in the Committee's Twenty-Second Annual Report, 1929, from which the percentages have been calculated.

Cost of Maintenance.—Found by dividing the sum of annual cost of maintenance and annual loan charges for repairing carriageways, by the mileage. Where no entry is found under annual charges, the cost of maintenance is divided by the mileage.

There is no entry under loan charges for the City of London, Fulham, St. Pancras, Southwark, or Woolwich.

The entry for Holborn is 'nil.'

Lewisham.—The cost per mile is calculated from the figure obtained by deducting 21,277*l.*, income, from 89,412*l.*, total expenditure including loan charges. The cost per mile without that deduction is 636·4*l.*

Creosoted Softwood.—Creosoted softwood, chiefly deal and fir, is largely employed, especially for main thoroughfares, and continues to prove satisfactory. In the earlier work a 6-in. 'foundation' of Portland cement concrete was usually provided, but this thickness is not now found sufficient for heavy traffic. Concrete 8 ins. thick is sometimes put down, but for heavy traffic it is usually from 9 to 12 ins. The thickness of the wood-paving is usually 5 ins., but often 4 ins. The blocks are often grouted with pitch, or dipped in a mixture of tar and pitch when they are being laid, but the most usual recent practice is to run the joints with a mixture of hot pitch and creosote oil. The following sizes of blocks are used: length, width, depth; $9 \times 3 \times 5$, $9 \times 3 \times 4$, $8 \times 3 \times 5$, $8 \times 3 \times 4$.

This is now the principal paving for First Class streets; occasionally used in Second Class and other streets.

Life of Softwood Paving.—Edgware Road, 11 years, to 1905; Cannon Street, 16 years, 1906–22; Ludgate Hill, 17 to 20 years; Queen Victoria Street, 17 years, 1908–25; Edgware Road, 17 years, 1905–22; Fleet Street, 19 years, 1903–22; many streets over 20 years (O. E. Winter); Haverstock Hill, paved 1901, 26 years, so far (O. E. Winter).

Strength Crusts for Softwood Paving.—Softwood paving demands a better strength crust to support it than do some other carpetings, and it repays in a greater degree than those the care and money expended in providing the desired strength and the greatest possible trueness of surface. Floating is, however, not desirable, the smaller roughness of the solid concrete being unimportant if the surface is and remains true. For really heavy traffic, double reinforcement is necessary, since single reinforcement, unless at half-depth, only adds to the strength here and there where the conditions happen to suit the disposition. In London, some of the single reinforcement or plain concrete strength crusts recently provided for softwood paving are not good enough to do justice to the qualities of the wood.

Hardwood.—Hardwood pavings have proved very lasting, but are nearly superseded by softwood or other pavings. *Jarrah* has lasted 10 years in the Euston Road, very heavy traffic; 19 years in the Streatham High Road; 10 to 14 years in heavy traffic thoroughfares in Hampstead; 26 years in a first-class street in Hackney; and in other streets 13 to 17 years, and 19 years. *Karri* has lasted up to 21 years. These woods are inferior to softwoods as regards even wearing, and do not so readily take up the armouring of grit which so greatly aids in preventing slipperiness of softwood pavings. In 1915, *sectional Jarrah* blocks, which seem to be better than ordinary blocks of hardwood, were laid in pitch, on a 9-in. concrete 'foundation,' and in some cases re-cut *Jarrah* blocks were re-laid. Since then these woods have been hardly at all used, but some re-cutting and relaying is still carried out.

Wood-paving is now often laid on a thin layer of asphaltic or bituminous material, between the wood blocks and the concrete, but many road surveyors prefer to lay them directly on the concrete.

Asphalt.—Compressed asphalt on a concrete strength crust or 'foundation' is one of the most important heavy traffic pavings in London. The thickness of concrete required varies as in the case of wood-paving, and is practically the same for similar traffic. The thickness of the asphalt carpet is usually 2 ins. or 1½ ins. Compressed rock asphalt is sometimes used for 'other' streets, being a very sanitary paving.

TABLE VI.—LONDON CITIES AND BOROUGHS: PERCENTAGES OF PAVINGS, 1927–1928.

City or Borough.	Total Mileages of Streets.	Percentages of Different Pavings, by Mileages.								Cost of Maintenance per Annum per Mile.
		Wood.	Asphalt.	Granite Setts.	Macadam, Water-Bound, etc.	Tar Macadam.	Concrete Paving.	Bituminous— -grout.	Bituminous— -macadam.	
Chelsea	33·2	23·9	5·0	3·0	45·5	6·7	1·2	—	14·6	—
Deptford	53·8	11·1	5·13	7·06	58·5	13·8	0·28	3·4	—	824·5
Fulham	65·2	15·3	29·4	—	55·2	—	—	—	—	354·7
Greenwich	71·5	5·24	11·5	6·64	71·0	3·84	1·75	—	—	—
Hackney	110·1	10·47	11·38	2·32	74·7	1·22	—	—	—	829·6
Hammersmith	65·0	17·3	0·96	0·77	8·095		—	—	—	595·9
Holborn	25·0	13·5	46·0	13·0	20·5	7·0	—	—	—	408·8
Lambeth	139·5	17·10	12·99	4·81	59·4	2·37	0·04	2·88	—	499·0
Lewisham	140·5	11·25	2·63	1·5	72·6	11·82	0·21	—	—	484·9
Poplar	63·0	12·7	22·22	23·80	41·27	—	—	—	—	1003·5
St. Marylebone	59·0	—	—	—	—	—	—	—	—	553·7
St. Pancras	88·6	12·09	3·25	28·29	45·60	(10·79, including asphalt mixtures)				477·3
Southwark	64·8	16·17	30·92	13·30	16·71	23·17	—	—	—	304·2
Stepney	91·4	4·39	3·29	61·30	30·65	—	—	—	—	1266·2
Westminster	99·1	38·08	27·91	10·22	15·66	4·25	—	—	—	—
Woolwich	98·1	4·09	6·14	6·14	83·73		—	—	—	528·3
City of London	48·5	(199)	(225)	(49·4)	(—)	(2·32) thousands sq. yd., also, rubber (0·76) and mastic on granite (2·4).				783·3

Asphaltic macadam and asphaltic clinker macadam are used in 'other' streets, and sometimes, especially the latter, in Second Class streets.

Bituminous layers.—Bituminous materials have been employed, in some cases, as carpeting layers only, but in other cases in crusts of sufficient thickness to take a share in distributing the loads. In 1915, only materials of this kind were used for street surface renewals in *Fulham*, usually 2½ to 3½ ins. of 'bituminous macadam' on a 12-in. foundation of hardcore, the surface being treated with liquid bitumen and granite chips or with a 1-in. carpet of asphalt. This construction is very interesting, as it embodies (*a*) a wearing crust which partially distributes the load; (*b*) a foundation which is also a strength crust; (*c*) a wide spread on the road bed and low stresses in this crust, these results being obtained by the use of a considerable depth of relatively cheap material; and (*d*) the initial spread of the load by the wearing crust. In *Hampstead*, 'asphaltic macadam' 1½ ins. and 2 ins. thick was laid, and in *St. Marylebone*, 'bituminous pavements' 1½ ins. to 3½ ins., some of these on old macadam crusts. 'Asphaltic graded granite' 3 ins. thick was put down in *St. Pancras*, and 'asphalt macadam,' 4 ins. deep in two layers on existing macadam, in *Woolwich*.

Bituminous crusts and carpets are now largely employed for 'other' streets, destructor clinker being sometimes used as the aggregate.

Tar macadam.—The following were laid in 1915. Tar macadam (granite) 3 ins. thick on old macadam: tar macadam 4½ ins. thick: tarred slag macadam, 4 ins. thick on a foundation of hardcore, and 4½ ins. thick on 'an old foundation.' Since then the use of this form of paving has extended. Work noted in the 1925 report includes: 4-in. 2-coat work on hardcore foundation; seven roads of the 'other' street class in Hammersmith, 4 ins. tar-slag macadam; 3½ ins. tarred limestone macadam on existing macadam; and 4 ins. tar-slag macadam, replacing granite sett paving.

The 1926 report describes work with tar-slag macadam and tarred limestone macadam.

Stone paving.—Considerable areas were paved, in 1915, with small granite setts, the usual depth being 4 ins.; also, Aberdeen granite setts, 4 × 6 ins. on a 12-in. crust of cement and clinker concrete. Re-dressed Aberdeen granite setts 3 ins. by 6 ins. were laid on 6 ins. of concrete; fine dressed 'Norwegian' granite setts were laid in cement and sand grout on 12 ins. of concrete; and setts, 4-in. cubes, laid in 1 in. of ashes and grouted with pitch grout, on a macadam crust.

The 1925 report gives particulars of the use of setts in Deptford; in Tunnel Avenue, Greenwich, traffic 75–100 tons per yard width per hour; in Lambeth, a First Class street, and an 'other' street; in Whitefriars Street; in Poplar; and in Stepney, several streets. The part played by stone setts, now no longer used for main thoroughfares as such, is indicated by the nature of the traffic in the localities mentioned.

The 1926 report refers to stone sett paving in Stepney, St. Pancras, and Greenwich.

The trend of practice, as shown by the later reports, may be studied under 'Replacements,' p. 746.

See also p. 737, 'Chelsea Embankment.'

Filler for Concrete Pavements.

A cement-coloured filler for filling cracks and joints in concrete pavements, which does not give the unsightly appearance of the black lines resulting from the use of the bituminous fillers commonly employed consists of approximately twelve parts rosin, one part crude rubber, with sufficient barium sulphate to give the desired colour. In preparation the rubber is dissolved in petrol, and the rosin is then mixed in with an application of heat, the colouring material being added as required. Any desired consistency can be obtained by varying the proportions. A material with remarkable adhesive properties is produced, and it can be heated so that it will flow into the cracks and joints.

Recent Practice and Costs of Pavings in London.

Foundations.—Recent foundation work includes 11 ins. 6 to 1 concrete with 'fabric' reinforcement; 8 ins. 'Ciment Fondu' 6 to 1 concrete with 'fabric' reinforcement; 9 ins. 6 to 1 concrete, similarly reinforced, several cases; 12 ins. concrete; 12 ins. clinker concrete; 9 ins. clinker concrete; 6 ins.-clinker concrete ('other' street); 6 ins. concrete, a notable case, as this (Mile End Road) is a First Class street; 4 ins. reinforced concrete ('other' street); 9 ins. reinforced concrete ('other' street); 6 ins. concrete, a notable case, Commercial Road, East. Both the First Class streets noted as having been provided with 6-in. concrete foundations are sett-paved; but as this paving has no crust strength, the difference in thickness of foundation compared with other streets is remarkable.

Rapid hardening cements ('Ferrocrete' and 'Ciment Fondu') have been used, for the whole foundation or for floating, in many cases. Much foundation work in London is without reinforcement. That usually employed is a single layer with different spacings of lengthwise and transverse rods.

APPROXIMATE GRADING ANALYSIS.

Base.		Wearing Surface.	
45 per cent.,	1½ in. stone	200 mesh,	18 per cent.
25 „	½ in. chippings	100 „	17 „
25 „	sand	80 „	17 „
5 „	200 mesh	50 „	35 „
		30 „	6 „
		20 „	4 „
		10 „	3 „
100 „			100 „

By weight the amount of bitumen would be 9 per cent. and 14 per cent. respectively.

The cost of wood paving in Fulham may be gathered from the fact that the Council allotted 10 years ago the sum of 4,000*l.* per annum to maintain 160,000 sq. yards; this has now been increased to 202,268 sq. yards without increase in allotted sum. Any surplus not spent is placed to a reserve; this reserve at the end of 1916 was 12,000*l.*, sufficient to repave one-seventh of the whole area. The average cost is therefore 5*d.* per sup. yard per annum; the cost of the heaviest trafficked road does not exceed 9*d.* per sup. yard, even allowing for the concrete foundation. It has been stated in the Committee Rooms of the Houses of Parliament that local authorities have roads which cost over 2*s.* and which are of macadam.

The motor bus damages the road to no greater extent than the steam lorry, if as much, as many of these have steel tyres.

(Francis Wood.)

ANNUAL COSTS PER SQUARE YARD.

In the cases of some of the road crusts or pavings which were replaced or renewed, the annual cost of maintenance of the worn-out or superseded crust or paving is given:

Water-bound Macadam, sprayed or unsprayed; 'other' streets, considerable traffic, 6½*d.*, 3½*d.*

Compressed Rock Asphalt.—First-class streets; very considerable traffic, 1*s.* 10*d.*; considerable traffic, 1*s.*; 'other' streets, 8*d.*

Softwoods.—Creosoted deal, Westminster Bridge Road, 8½*d.*; yellow deal, Regent Street, laid 1906, 10½*d.*; softwood, Northumberland Avenue, a very wide street, not heavy traffic, 6*d.*; second-hand, reconditioned blocks, laid 1917, 1*s.* 2½*d.*

Wood.—Presumably hardwood, Aldersgate Street, laid 1874, 1*s.* 0½*d.*

Asphaltic Macadam, laid 1917, Vauxhall Bridge, very heavy traffic, 1*s.* 3*d.*

Granite Setts.—Replaced, 1927, by asphalt, average annual cost of the sett paving, 2*d.* per sq. yd.

REPLACEMENTS AND COSTS: TWENTIETH AND TWENTY-SECOND REPORTS.

(Numbers higher than (24) relate to the Twenty-Second Report.)

Water-bound Macadam: (1) Replaced by bituminous-bound clinker and bituminous carpet, cost, using old materials, 10*s.* 8½*d.*; (2) by 2½-in. bituminous carpet on the regraded surface, cost, 6*s.* 4*d.*; (3) by 4-in. tar-slag macadam, cost, including strengthening of foundations, 8*s.* 8*d.*; (4) by 3½-in. bituminous-grouted macadam, cost, of preparation, 1*s.* 7*d.*, of the paving, 7*s.* 1*d.* (2 years' free maintenance); (5) by 3-in. asphalt macadam, cost, of preparation, 5*s.* 5½*d.*, of the paving, 10*s.* 6*d.* (5 years' free maintenance); (6) by 4-in. tar-slag macadam, cost, of preparation, 1*s.* 11*d.*, of the paving, 8*s.* 4*d.* (5 years' free maintenance); (7) by bituminous paving, 2¾-in. asphalt concrete, 1½-in. carpet, cost, of preparation, 2*s.* 11½*d.*, of the paving, 14*s.* 3½*d.* (5 years' free maintenance); (8) by 3-in. bituminous macadam, cost, of preparation, 4*s.*, of the paving, 7*s.* 11*d.* (3 years' free maintenance); (9) by 2-coat asphalt macadam, 5-in., cost, of preparation 4*s.* 10*d.*, of the paving, 12*s.* 6*d.* (5 years' free maintenance); (10) by clinker asphalt, 1½-in. base, 1-in. carpet, hardcore foundation regraded, cost, of preparation 4*s.* 2*d.*, of paving, 7*s.* 4*d.*; (11) by creosoted deal on 8-in., 6 to 1, rapid hardening cement concrete, cost, of foundation, 16*s.* 11*d.*, of the paving, 13*s.* 4*d.*; (12) by Trinidad Lake asphalt macadam, cost, of preparation, 1*s.* 4*d.*, of the paving, 7*s.* 6*d.*; (13) by 3-in. granite concrete; 1 'Ferrocrete,' 1 sand, 1 of ⅜-in. granite chippings, 4 of 2-in. granite, surface dressed with silicate of soda, cost, including reshaping the old crust, 11*s.* 3*d.*; a similar case, 17*s.* 6*d.*; (14) by granite setts, 5-in. deep, on 6-in. reinforced concrete, cost, 39*s.*; (25) a number of cases in which the old surface has been prepared, or the old material used, for bituminous surfaces, as (*a*) cold as asphalt ('Carpave') 2-coat work; total cost, 6*s.* per sq. yd.; (*b*) grouting with cold emulsion, preparing old surface 9*d.*, surfacing 9*s.* 6*d.*; (26) by 2 ins. of

compressed asphalt on 9 ins. of Thames ballast concrete; cost of foundation, 14*s.* 6*d.*, of the paving 10*s.* 9*d.*; (27) by destructor clinker asphalt (Fulham), 1-coat work, 2½ ins., on regraded hardcore foundation; cost, 6*s.* 7*d.*; (28) by redressed setts, cement grouted, on 6 ins. of 6 to 1 Thames ballast and rapid hardening cement concrete; cost of foundation and surfacing, 24*s.*

Tar-slag Macadam, (15) replaced by 8-in. concrete with a granolithic surface, cost, 29*s.* 6*d.*; (29) by whinstone treated with bitumen, 4 ins., 2-coat work, on rolled bottom, 9*s.*; (30) by granite rapid hardening cement concrete; cost of foundation and surfacing, 14*s.* 9*d.*

Tar Macadam, (16) replaced by creosoted deal, laid dry and close, grouted with pitch or bitumen; on 12-in. or 9-in. lightly reinforced concrete foundation, cost, of foundation, 14*s.* 5*d.*, of the paving, 10*s.* 4*d.* (2 years' free maintenance); (35) by a surfacing of crushed granite, sand, and bitumen, 2 ins. on 9 ins. concrete foundation, cost of foundation and surfacing 23*s.* per sq. yd., with 5 years' maintenance; (36) by 2½ ins. asphalt, laid hot on existing surface; cost of preparation, 11*d.*, and of the surfacing, 8*s.* 3*d.*

Asphaltic Macadam, (17) replaced by softwood; cost of paving, 13*s.* 3*d.*; (35) by 2-in. compressed asphalt on 9-in. concrete; cost of foundation, 14*s.* 9*d.*, of the paving, 9*s.* 9*d.*

Jarrah, laid 1912, (18) replaced by creosoted softwood; cost of paving, 15*s.* 1*d.*; (19) by creosoted deal on 12-in. clinker concrete foundation, 1-in. floating, cost, of foundation, 21*s.* 5*d.*, of the paving, 12*s.* 1*d.*; (20) a similar case, cost of paving, 9*s.* 6*d.*; Jarrah blocks laid in 1901, by (34) creosoted deal on 12-in. concrete foundation, cost of foundation 23*s.* 2*d.*, of the paving, 15*s.* 6*d.* Another similar case, Jarrah laid in 1922; cost of new work, 34*s.* 2*d.*

Compressed Rock Asphalt, etc., (21) laid 1911, replaced by softwood, cost of the wood paving, 13*s.* 2*d.* (2 years' free maintenance).

Norwegian Granite Setts, (22) laid 1911, replaced by Aberdeen granite setts, 8 ins. by 5 ins. by 4½ ins., fine dressed, cost of the paving, 42*s.*

Granite Setts, (23) replaced by 2 ins. compressed rock asphalt on 9-in. concrete, cost of foundation 14*s.* 11*d.*, of the paving 17*s.* 10*d.* (5 years' free maintenance); (31) by mastic rock asphalt, 2 ins., made with granite chippings on 6 to 1 Thames ballast and rapid hardening cement concrete, cost of foundation 9*s.* 8*d.*, of the paving, including 2 years' maintenance, 10*s.*; (32) by pitch grouted creosoted deal, on 9 ins. of 6 to 1 Thames ballast and rapid hardening cement concrete, cost of foundation 25*s.* 6*d.*, of the paving 11*s.* 3*d.*; (37) by compressed asphalt on 6-in. concrete, cost 21*s.* 2*d.*, with 5 years' maintenance.

Creosoted Deal, laid 1914, replaced by (24) Swedish granite setts, bitumen-grouted, cost of the paving, 33*s.* 8*d.*; (33) by clinker asphalt, 3½-in. base and 1½-in. topping, rolled hot (Fulham), cost of foundation and surfacing, 12*s.* 6*d.*

Experimental Rubber Paving.

Various types of rubber paving have recently been put into service, on trial. In *Southwark* (Mr. A. Harrison) two kinds have been put down: (*a*) slabs, 9 ins. by 3 ins. by ½ in., vulcanised on expanded metal and anchored to concrete: (*b*) composite blocks, 8 ins. by 4 ins. by 4 ins., laid on a concrete foundation, like wood blocks. The second type is regarded as likely to prove successful.

In *Glasgow* and *Edinburgh* (1923) a different method has been adopted (described by Mr. E. J. Edwards). The rubber block, 9 ins. by 4½ ins. by 2 ins., is fixed by means of an embedded steel spike into a concrete base of slightly smaller dimensions. These blocks were laid on a concrete foundation, and were grouted in with pitch or cement. It is thought desirable to lay this paving when the concrete foundation is still green; if not, the blocks should be laid with bitumen and grouted in with pitch and bitumen. The top layer of the blocks laid in Edinburgh and Glasgow contained 30 per cent. of plantation rubber. The present cost (Mr. E. F. Spurrell) of a rubber paving includes about 4*l.* per sq. yd. for the rubber alone.

Matters demanding attention are: the question whether green concrete will set properly under the impervious paving; whether oil spilt on the roadway will seriously harm the rubber; and whether rubber paving can be effectively restored after excavation.

A rubber *footway* laid down in Edinburgh twenty years ago has been found to have lost only a fraction of an inch of its thickness.

London, 1926.—An experiment is being tried in New Bridge Street, Blackfriars, where rubber as a road surface is to be subjected to a very severe test.

Heavy terra-cotta blocks (known as 'Gaisman' blocks), 10⅛ ins. by 8 ins., are laid as a base to hold a superimposed rubber cap, ⅝ in. in thickness. This is put on by a process which, it is claimed,

will withstand the pounding and spreading force of the heaviest traffic. The blocks with these rubber caps are laid on a 12-in. concrete bed, with a cushion of sand ⅜ in. thick. They are set in by means of joints, composed of a special bitumen and rubber mixture. During 1927 this paving showed no sign of wear until about November, when flaking off of considerable fragments of the rubber was seen to be taking place here and there, probably due to the sudden 'braking' of vehicles on dry days, and the pressure of grit. It is possible, however, that the blocks which have suffered may have been defective. It is more slippery than the adjacent wood paving, but less slippery probably, than asphalt. Greasiness in wet weather is partly due to pulverized gritting material brought on to it from adjacent wood paving.

By the end of 1928 the blocks adjacent to grids, manholes, and, in some places, alongside the strip of other material separating them from the kerb, had suffered considerably. The fairer test, the condition of the blocks in the remainder of the area, was inconclusive, the wear not being sufficient to indicate approaching failure, nor so little as to suggest successful competition with other pavings as regards cost. Except for some further wear, and evidence of tendency, in a proportion of the blocks, to surface peeling, these remarks apply to the condition of the paving at the end of 1929.

Bradford, 1927.—A small area of experimental paving consisting of wooden blocks, in each of which there is a number of metal caps, each containing a rubber 'egg.' The cups and 'eggs' are replaceable. (Inventor, P. Henry.)

London, 1928.—Laid in January (*J. S. Cowper*), in Thurloe Place, close to South Kensington Station. A 2½-in. carpeting of 'Cowper' blocks on a concrete strength crust, single reinforcement. Each block, in plan, two 9-in. × 4½-in. areas joined by half-length overlap. No fixing devices; middle third provides tongue and groove joint ½-in. deep; lugs and slots in ends of bottom third; made to the camber; tapped into place; no jointing material; surface finely striated to give initial rugosity. Top layer 'durable rubber,' shock absorbing; bottom layer soft enough to sit tightly on the concrete; middle layer harder material, with a high coefficient of elasticity. After 9 months, in perfect condition, the fine striations being hardly dimmed. Condition very good at the end of 1929.

Glasgow.—A street crossing, 40 ft. × 10 ft.; experimental. Paved May 1923 with blocks 9 in. × 4½ in. × 4 in. deep (2½ins. of rubber fixed with steel binders to a 2-in. block of concrete); supplied by North British Rubber Company. Cost of blocks, 5*l*. 11*s*. 7*d*. per sq. yd.; cost of laying, about 10*s*. per sq. yd. In November 1928, no appreciable sign of wear, good condition very heavy vehicular traffic.

Edinburgh.—The same paving, also laid in 1924, was, in 1929, reported by the city engineer to have given every satisfaction and to be in appearance as good as when laid; medium traffic.

London, Fresh Wharf.—400 sq. yds. on a gradient of about 1 in 24. T-irons are embedded in the concrete foundation, from kerb to kerb at intervals of 12 to 15 ft., to prevent creep of the blocks. These are 8 ins. by 4 ins. by 3 ins. deep, with vertical sides. The lower 2½ ins. is of hard rubber, on which a ½-in. cap. of soft rubber is mounted. The blocks were dipped in hot bitumen when being laid. Supplied by the Leyland and Birmingham Rubber Company. Considerable traffic, mainly heavily laden vans and lorries. Reported as not creeping and showing very little sign of wear in 1929.

Effects of Motor-Omnibus Traffic and other Heavy Traffic.

Some years before the war reports of the Paving Committee showed that heavy motor traffic had increased the cost of maintenance of macadam roads generally. As regards paved streets and roads, the cost of maintenance had increased in those areas in which the stage of strengthening and improvement had been reached, but had increased only slightly or not at all where this stage had been passed. When the cost of reconstruction had been taken into account on the one hand, and actual increases in the passenger-mileage and ton-mileage on the other, it seemed that the economics of road maintenance in London had not been changed for the worse, though the cost of cutting through the greater thickness of concrete provided a new argument in favour of the scheme for providing subways for the various conduits under the main streets.

Surfaces for Steep Gradients.

Bituminous-bound Destructor Clinker.—Muswell Hill 1 in 10; varied traffic, including London omnibuses.

Broken-stone, Cement Grouted.—Rapid hardening cement; used for gradients of 1 in 15 and 1 in 10.

Tar-Macadam.—Aberdulais-Resolven Road, 1 in 11. In Blackburn (A. T. Grosseman) gradients up to 1 in 6.

Asphaltic-Slag Macadam.—Gradients up to 1 in 12.

Tests of Road Stones.

(See Report to the Third International Road Congress, T. E. Stanton and R. G. Batson.)

The equipment of the National Physical Laboratory, Teddington, enables the following test of road stones to be made :—

(1) A dry attrition test (resistance to wear); (2) a wet attrition test; (3) an abrasion test (for hardness); (4) a repeated impact test (toughness); (5) a test of the cementation value: (6) an absorption test.

Attrition test.—The machine is of the 4-cylinder Deval type, as used in France and in the United States. The cylinders are $7\frac{1}{2}$ ins. in diameter and 14 ins. long, set at 30° to the axis of rotation. About fifty pieces of the rock, weighing eleven pounds, are placed in one of the cylinders, and the machine revolved 10,000 times at about 30 revolutions per minute. Only the material which will pass a sieve of $\frac{1}{16}$ mesh is counted in measuring the amount of wear, which is recorded both as a percentage and by the French coefficient $\frac{40}{\text{percentage of wear}}$.

In the wet test 1·1 gallon of water is used with 11 lbs. of stone. The wear is usually greater in the wet than it is in the dry test; but not always.

Repeated impact test.—This test is made with the Page impact machine, which has a hammer weighing 4·4 lbs., delivering the blows through a plunger, the surface of which, in contact with the specimen, is spherical to a radius of 0·4 inch. The test piece is a cylinder, 1 in. by 1 in., ground to the true diameter, and cut to length by a diamond saw. The test consists of a series of blows, beginning with a fall of 0·4 inch, which is increased by 0·4 inch with every successive blow until the test piece is broken. The toughness is measured by the number of blows. Tests can also be made with a constant height of fall.

Abrasion test.—The test piece is prepared in the same way as for the impact test and is of the same size. The machine is of the Dorry type, and consists of a cast steel disc revolving in a horizontal plane. The test piece is held, with its axis vertical and with a load equal to 3·5 lbs. per square inch, against the surface of the disc 26 cms. from the centre. Standard sand of 30 to 40 mesh is fed continuously through funnels at a definite rate, the same in each case. After about 1,000 revolutions of the disc, at the rate of 28 per minute, the test piece is weighed and the loss of weight noted. The test is repeated on the other end of the test piece, and the average loss of weight by the two runs is recorded. The hardness is measured by $(20 - \frac{1}{3}W)$ where W is the loss of weight in grammes per 1,000 revolutions.

Cementation test.—Coarsely crushed rock, weighing 1·1 lb., is ground in a ball mill with 0·02 gallon of water, the balls being steel shots (two) 5·1 ins. in diameter. The material is ground for two and a half hours at the rate of about 2,000 revolutions per hour. The resulting paste is moulded into briquettes, 1 in. diameter and 1 in. long, the maximum pressure being 1,880 lbs. per square inch. After twenty-four hours, the briquette is dried for twenty hours in the air, and for four hours in a hot air bath at 200°F., cooled for twenty minutes in a dessicator, and then tested. It is placed on the anvil of the machine, with its axis vertical, and struck through a plunger by a small hammer with an effective drop of 0·4 in., the rebound being measured by the movement of a lever attached to the hammer, and holding a pencil against a paper-covered revolving drum. This records the number of blows required to destroy the resiliency of the test-piece, after which there is no rebound. The mean of six tests is taken as giving the cementing value of the material.

Crushing strength.—The specimen is a cylinder 1 in. in diameter by 1 in. long, ground out of a rough sample by emery-wheels. The cracking stress and crushing stress are measured.

Specific gravity.—Three samples are taken from each batch of stones, and the average specific gravity recorded as that of the batch.

Absorption of water.—A stone of about 13 lbs. is weighed in air and then in water. It is kept in water for three days and is then weighed in air. If A=weight in air, B=weight in water immediately after immersion, and C=weight in water after immersion for three days, the number of pounds of water absorbed per cubic foot of rock is nearly

$$\frac{C - B}{A - B} \times 62{\cdot}5.$$

Petrological analysis.—The classification of the rock, in the terms of geological nomenclature and as regards its constituents, is made in the Geological Museum, London, S.W.

Samples.—The minimum number of stones sent in the case of a rock to be used as a road stone, if suitable, should be 250, broken to the shapes and sizes usual in road work. For the abrasion and impact tests eight stones should be sent, about $1\frac{1}{2}$ ins. by $1\frac{1}{2}$ ins. by 3 ins. Samples should be sent in strong wooden boxes or in separate compartments of cases. The position of the quarry must be so described that its locality can be identified by the Geological Survey map; or a piece of the map with the position marked on it may be sent.

TESTS OF BITUMINOUS MATERIALS.

Penetration Test.—The material to be tested is usually brought to a definite temperature, which for standard comparisons is 77° F. (25° C.). The penetrometer consists essentially of a weighted needle attached to the 'hand' of a dial by a simple mechanism so that the hand on the dial registers the distance moved by the needle. The point of the needle is brought to the surface of the specimen, the dial reading noted, the needle released for one second or sometimes longer, and then held. The difference between the two readings on the dial measures the penetration. It is desirable to make tests with different weights on the needle and at different temperatures.

Viscosity Tests.—Materials too soft for the needle penetration test may be tested by several methods, including the following: The *Crosby Consistometer* consists of a balanced wheel, grooved at the circumference and provided with a device to measure the arc through which it turns. Wheels of 4 ins. and 8 ins. diameter have been used. At one end of a thread which passes round the circumference of the wheel is a small hook, to which plungers can be attached, and at the other end is suspended a pan for weights. An excess load of 0·3 gramme is required to overcome friction and inertia. The machine measures the distance moved through by a disc of known area under a known load, through a substance contained in a box of fixed dimensions, during five seconds. The plungers are round brass discs with spindles in the centre. The box is 2½ ins. in diameter and 1 in. deep for ordinary materials, 5 ins. deep for very fluid materials. The plunger is immersed, the weight placed in the pan, partly counterbalancing the weight of the plunger and leaving a known downward force which is left to act for 5 seconds, and moves the plunger through the material under test for a distance which can be accurately read on the measuring device. The *float apparatus* of the New York Testing Laboratory measures the time taken before a plug of the substance under test becomes warm enough to flow under given conditions. The plug of the substance is in a brass collar attached to a float or saucer which is floating in water. The temperature of the water is maintained constant, at 90° F., for instance, until the plug is sufficiently soft to allow the water to enter the saucer and sink it. The temperature of the plug at the beginning of the test is fixed by keeping it for 15 minutes in cold water at a constant temperature.

Solubility Test.—Solubility in carbon disulphide. A small quantity is placed in a weighted flask containing 100 c.c. of carbon disulphide, and afterwards decanted through an asbestos mat and the precipitate dried and weighed.

Other Tests.—Appropriate methods are employed for determining the specific gravity, the melting point, and the coefficient of expansion.

ROAD CRUST WORK AND STREET PAVING.

Quarrying.—Ayrshire, quarrying road metal, 3*s.* 2*d.* per ton; quarrying and tarring, 9*s.* 4*d.* per ton. Herefordshire, 13*s.* 3*d.* per ton, including depreciation of plant.

Power Cost Data (E. Dalton).—Quarrying: oil-engine compressor, drills, etc., 100 h.p.; fuel oil nearly 0·4*d.* per brake horse-power-hour; average load factor, 50 per cent.; fuel oil and lubricating oil, 0·75*d.* per ton of rock quarried, the calorific value of the fuel oil being 19,400 B.Th.U., and lubricating oil 3*s.* 6*d.* per gallon.

Stone-breaking Plant.—Output 70 to 80 tons of road metal per day; cost about 2,000*l.*

Road Metal.—Warwickshire, granite and basalt, 23*s.* 8*d.* per ton; West Riding, granite, 17*s.*; Lichfield, gravel crushed at the pit, 7*s.* 10*d.* per cub. yd., or 6*s.* 8*d.* per ton; other cases, 19*s.* to 23*s.* per ton. Truro, roadstone, 4*s.* 10*d.* per ton. Somerset, limestone, 4*s.* 6*d.* per ton on lorry at quarries; tar-macadam, 14*s.* 8*d.*

Quantities used in a Year.—680 miles of road, 216,000 tons, including 64,000 tons of tar-macadam; 284 miles of main road, 38,000 tons; 634 miles of road, 132,000 tons, including 32,000 tons of granite coated with tar and bitumen. Per mile, in a year, crust renewals, 269 tons to 360 tons.

About 610 miles; nearly 44,000 tons of granite chippings treated with tar and bitumen in the county hot mixing plants. About 1,400 miles, nearly 944,000 galls. of tar and 259,000 galls. of bitumen used for treating about 6½ million sq. yds. of road surface. 812 miles, 255,000 tons of road materials, 959,000 galls. of tar and other surface dressings; 550 miles, 16,000 tons of tarred slag, 5,400 tons of tarred limestone, 203 tons of tarred granite.

On 662 miles of county roads; 94,700 sq. yds. of bituminous coatings, 20,305 tons of tarred macadam (slag 14,689, limestone 3,660, granite 1,956), 3,529 tons of limestone and granite chippings used in semi-grouting work. For about 218 miles of roads, 47,000 tons of materials purchased.

On 513 miles of county roads; 19,268 tons of Clee Hill stone; 16,161 tons of limestone; 878 tons of slag; 11,967 tons of tarred macadam; 566,247 gal. of bitumen and tar.

Consolidating Road Metal.—Several counties, about 1*s.* 10*d.* per ton ; also 2*s.* to 2*s.* 3*d.* ; lowest figure noted, 1*s.* 1½*d.*, 1925. Staffordshire, 1927, carting materials and forming the road crust, 29*s.* 10*d.* per ton, or 3*s.* 9*d.* per sq. yd. ; consolidated thickness, 4 ins., 1929. Northamptonshire, 1929, 11·65*d.* ; Holland, Lincolnshire, 8·83*d.*

Tar-Macadam Plant.—Durham County Council, output 1,000 tons of tar-macadam and 1,000 tons of all-grade dry slag per week.

Tar-Macadam Crusts.—Various counties : 4-in., about 7*s.* 3*d.*, using old stone, about 6*s.* 2*d.* per sq. yd. ; per ton, 36*s.* ; Midlothian, 3*s.* 7*d.* per sq. yd. Dorking : analysis of making up roads with tar macadam, including deep scarifying and strengthening of haunches (S. S. Gettings)—Main roads : materials, 73·2 per cent. ; manual labour, 9·2 per cent. ; team labour, 6·3 per cent. ; scarifying and rolling, 10·1 per cent. ; tradesmen's bills, 1·2 per cent. 'Assisted' roads : materials, 66·6 per cent. ; manual labour, 9·7 per cent. ; team labour, 13 per cent. ; scarifying and rolling, 8 per cent. ; tradesmen's bills, 2·7 per cent. Herefordshire : tarred material, 3-in. coatings, mixed hot with tar, squeegeed with bitumen, 6*s.* per sq. yd., or, if tarred chips are used instead of the bitumen, 5*s.* 6*d.* per sq. yd. London : 3½-in. tarred limestone, 7*s.* 1*d.* Sunderland : tar macadam, 4 ins. thick on 10-in. punning, 8*s.* 9*d.* per sq. yd. Another case, a 4-in. crust, 8*s.* 6*d.*, or with 12 ins. of hardcore base, 14*s.* 6*d.* per sq. yd. Morecombe Ring Road : tar-macadam top course (see 'New Roads'). London (Wandsworth) : trial stretch of tar macadam ; traffic 24,000 tons per day. Tar-macadam metal, per ton, 36*s.* Watford By-pass road, 17,500 sq. yds., at 7*s.* 6*d.* Newport, Mon. : 200 tons used. Tar-slag, per ton, 22*s.* 6*d.* ; dry slag, 12*s.* 6*d.* Somerset : main roads, 1929, 2½-in., 4*s.* 2*d.*

Reinforced Tar-Macadam.—A road near Hull (T. W. Chapman) : traffic, 4,000 tons per day and upwards ; reinforced tar-macadam crust had been practically untouched after three years' service. Leeds : reinforced tar-slag macadam. Warwickshire : experimental, reinforced tarred slag, 6 in. by 6 in. mesh, 6*s.* 3*d.* per sq. yd.

Tar-Macadam.—Recent data ; Ossett, laid on 10 ins. of local stone pitching ; Sidmouth, most of the local roads are of tar-slag macadam ; Swaffham, Urban District : usual practice, re-surfacing with tarred materials with concrete margins for horse traffic. Swindon, tar macadam replacing water-bound crusts.

Totnes : 2 in. to ½ in. stone, one coat, 3 ins. thick ; covered with granite chippings before rolling ; surface sealed with cold emulsion and ⅜-in. granite chippings ; cost, 3*s.* 2*d.* to 3*s.* 7*d.* per sq. yd.

Semi-grouted Crust.—Isle of Wight ; using crushed limestone, 2 ins. to ¾ in., produced at 5*s.* 6*d.* per sq. yd. ; crust formed by wet rolling ; bitumen emulsion applied, 1 gal. per sq. yd. Surfaced with ¼-in. chippings, 140 sq. yds. per cu. yd. ; rolled ; sealed, ½ gal. per sq. yd. ; blended with ¼-in. chippings ; cost about 3*s.* 4*d.* per sq. yd.

Grouted Limestone.—Chard ; main road programme, 1928, 24,000 sq. yds. of grouted limestone 1½ in. thick.

Tar-bitumen Grouting.—Hot application ; crust 5 ins. before rolling ; from 2½ gal. to 1 gal., usually 1¼ gal., per sq. yd. ; cost about 3*s.* 3*d.*, or from 1*s.* to 1*s.* 6*d.* more than the costs of the same crusts waterbound.

Bituminous Tarmacadam.—Sandstone, or gritstone, broken 2 ins. to ¾ in. ; 10 gal. of 60 per cent. bitumen and 40 per cent. tar mixture ; 3-in. crust ; sprayed and gritted ; contract price, 4*s.* 7*d.* per sq. yd. ; tarred every year. Roads so made up cost 260*l.* per annum for maintenance. Similar crusts cost 5*s.* 8d., 4*s.* 8*d.*, 6*s.* 4½*d.* per sq. yd.

Tar-slag Macadam.—Recent data ; five roads leading to Queensferry Bridge, surfaced with tar-slag macadam ; Norfolk, 73 miles of roads converted to tarred slag in one year ; Warwickshire, main road reconstruction, 3-in. to 5-in. tarred slag, 2-coat and 3-coat, on slag foundation.

Bristol-Avonmouth road, between Bristol and Shirehampton, nearly all tarmac. Norfolk, 1923, 17½ miles of tar-slag (see 'New Roads' and 'London'). Laying 3-in. of tar-slag macadam on a prepared foundation, about 5*s.* per sq. yd. Strengthening a water-bound crust and surfacing with tar-slag, 5*s.* 4*d.* per sq. yd. London, 2-in. bottom course, 1½-in. second course, ¾-in. to ½-in. topping, 7*s.* 2*d.* per sq. yd. ; 4-in., 8*s.* 8*d.* per sq. yd., with 3 years' maintenance ; Hampshire : county main roads, 1925-1926, tarred slag used, 15,000 tons.

Tarmacadam Footpaths.—Market Harborough : reconstruction ; new materials, tar, ashes for bedding, and slag dust for top-dressing. A portable 'cracker' used for breaking up the old material. Ten gal. of tar mixed with 7 barrows of the broken material, by hand ; rolled with an 8-cwt. roller. Cost per sq. yd. : labour, 6*d.* ; haulage, 1½*d.* ; tar, at 7½*d.* per gal., 5¼*d.* ; slag dust, ¼*d.* ; petrol and oil, ⅛*d.* ; surface tarring, 3½*d.* Total, 1*s.* 4½*d.*

Asphalt.—Hendon, main roads, 2-in. carpet on reinforced concrete; Northants., main roads, 2-in. carpets; Wrexham, main streets, 2-in. carpets on reinforced concrete. Asphalt carpets laid in 1927 included stretches of roads in Essex, Lincolnshire, Warwickshire, Hertfordshire, and Dumbartonshire, and on the Great North Road.

Synthetic Asphalt Plant.—Outputs of three sizes, tons per hour, 4 to 6, 8 to 10, 12 to 15; corresponding to square yards of carpet 2 ins. thick per ten-hour day, 400 to 600, 800 to 1,000, 1,200 to 1,500.

Asphalt Macadam.—Trinidad asphalt macadam, 2-coat work, 3-in. and 1½-in., 15*s.* 6*d.* per sq. yd.; 4-in. 13*s.*, with two years' maintenance. Warwickshire, 2-coat, 2-in. and 1-in.

Asphaltic Macadam.—Two-coat; 2 ins. and 1 in.; 11*s.* per sq. yd.; no appreciable wear in 3 years; deepest depression, ⅜ in.; usual 'wave depth,' ⅛ in. to ¼ in.

One-coat asphaltic (Trinidad) macadam: 2¾ in.; stone graded, 1¼ in. to ¼ in.; 10 per cent. cement filler; 10 per cent. bitumen; 25 per cent. sand; cost, 10*s.* per sq. yd.; 'no appreciable' deterioration after 3 years.

Asphalt Carpet.—On concrete strength crust; Northern Ireland: 17*s.* 10*d.* per sq. yd., with 10 years free maintenance. Also, 2-coat asphalt 9*s.* 6*d.* per sq. yd., 5 years free maintenance.

Clinker Asphalt.—London, 2½-in., 7*s.* 10*d.*, with five years' maintenance; 6,000 tons, most clinker asphalt, laid in Huddersfield, 1925, 7*s.* 4*d.* per sq. yd. Cheltenham, clinker asphalt for the roads is made at the destructor works; York, 2-in. clinker asphalt on 6-in. reinforced concrete foundation; Sheffield, about 800,000 sq. yds. of streets, clinker asphalt.

Clinker Asphalt Plant.—Huddersfield, 45 tons per eight-hour day.

Bituminous Macadam.—County Londonderry: 4 ins. of coreing, blinded with gravel and consolidated; 4-in. coat of hot bituminous macadam, graded 2 ins. to ¾ in.; rolled; sealed with Road Board tar mixed with 30 per cent. of bitumen; gritted with ¾ in. to ¼ in. whinstone chippings; rolled. Also, on good waterbound macadam crusts, 2-in. carpet of hot-laid bituminous macadam.

'Bituminous' Work.—London, 2-in. carpet, stone-filled; bituminous paving, 5-in. 12*s.* 1*d.*, with one year's maintenance; bituminous grouted macadam, 3½-in., 11*s.* 7*d.*, 2 years' maintenance; 2-in. carpet, 7*s.* 11*d.*, with five years' maintenance; 4-in. paving, 11*s.* 8*d.*, with five years' maintenance; 2-in. carpet, aggregate of graded granite, 8*s.* 6*d.* per sq. yd., with five years maintenance. Blackpool, 2-coat surfacing, 9*s.* 6*d.* per sq. yd. 'Bituminous asphalt,' 2-in. 3*s.* 3*d.* to 3*s.* 10*d.*; and 3-in., 5*s.* 7*d.*; 3½-in., 6*s.* 1½*d.* per sq. yd. Hampshire, 2½-in. carpet 7*s.* 8*d.* per sq. yd., 1 ton covering 7·3 sq. yds.

Black-base.—Largely used in the Western States, U.S.A. A generic term for lower courses or strength crusts with stone, slag, or gravel aggregate and 4 to 6 per cent. of bitumen, usually mixed hot. Five states have 2,500 miles of such lower courses.

Water-bound Crusts.—Herefordshire, good limestone crusts for light and medium traffic, tar-sprayed twice, spread with chips, 3*s.* per sq. yd. (not including foundation work). Cornwall, water-bound surfacing, 1*s.* 11*d.* per sq. yd.; Basingstoke, reforming old crust and adjusting camber, 2*s.* 6*d.* per sq. yd. Durban, resurfacing with macadam, 5*s.* 1*d.*

Strength Crusts.—Durban, 9*s.* 10*d.* per sq. yd.; 14-in. of large stone pitching, hand-packed, filled with spalls; on this 2-in. of whinstone, rolled, finished with 3 to 4 ins. of 2-in. whinstone; ready for carpeting after three months.

Sett Paving.—Guildford High Street in the London-Portsmouth main road. Laid November 1867: lasted till 1927, being turned after 48 years' service.

Concrete.—Surrey: reinforced concrete, 8 ins. thick, 5 ins. between the upper and lower reinforcement, bars ½-in. diameter (usually) and spaced 9 ins. both ways in the grille; cost about 10*s.* per sq. yd.

Barnoldswick, West Riding (1928 programme): 16 streets in reinforced concrete; also 17 private streets. Clacton: in one year, 50,000*l.* spent on concrete roads, private streets. Thurnscoe, Yorkshire: concrete carriageways, 26,000 sq. yds.; cost, nearly 8*s.* per sq. yd. Example—layer of ashes; bottom course, 4 ins. of 1 : 2 : 4 gravel concrete; upper course, 4 ins. of 1 : 1½ : 3 granite concrete. Main roads reinforced with 9-in. or 12-in. square mesh, $\frac{5}{16}$-in. steel rods. Irish Free State (*J. Quigley*): concrete, one-coat, 6 ins., reinforced with wire mesh 4·8 lbs. per sq. yd. Proportions, 1 : 1½ : 3. Cost, 12*s.* per sq. yd. In good condition after three years; only maintenance treating joints with bitumen. County Wexford; Enniscorthy and Wexford Road: 11½-in. concrete.

Ilford, 1929: carriageways 20 ft. wide between kerbs, reinforced concrete slabs 50 ft. long and 22 ft. 6 ins. in bottom width; single reinforcement; laid on 3 ins. of clinkers. Cost per sq. yd., excavation, 1*s.*; clinker, 6*d.*; concrete and reinforcement, 7*s.* 6*d.*; treating the surface with silicate of soda, 6*d.*; total, 9*s.* 6*d.*

Concrete Foundation Work.—Lytham St. Anne's, housing roads over very bad ground, peat subsoil; excavation 6-ins. below crust proper; 6-ins. of destructor clinker placed to contour; 4-ins. of 2-coat concrete, 1 : 2 : 4, of 1½ to ⅝ Millstone grit and ¼-in. down pit sand; ordinary cement, fabric reinforcement; two treatments with silicate of soda. Bideford, Northern Road deviation, 7-in. local gravel concrete, 2-in. wearing surface of basalt chipping concrete, fabric reinforcement, top and bottom in some places.

Cement Grouting.—Rapid hardening cement; the dry spreading and watering system, on an upper course 2½ ins. to 3 ins. of local gravel, 1*s.* 5½*d.* per sq. yd. Another case, same system, 3-in. crust of hard broken limestone, 2*s.* 4*d.* per sq. yd.

Broken stone crust, 4 ins. to 4½ ins., cement grouted and rolled; cost, 3*s.* 4*d.* per sq. yd.; good results. Buxton Hill, cement grouted crust, gradient 1 in 7.

Gladwell System (Application of).—A method which, in principle is that known as the 'Gladwell System,' has been followed at Kirkcaldy, using cement mortar (*A. Walker*). Road scarified to 3 ins. if necessary; rolled and formed to 1 in 32; a 2½-in. layer of whinstone, 1½ to 2½ in. gauge, is rolled, the roller passing twice over every part; a mortar, 2 parts of clean sand to 1 part of rapid hardening cement, spread to about 1½ in. thick; a 2½-in. layer of stone spread over it; the whole rolled until the mortar comes to the top; surplus slurry brushed forward. This forms a good holding surface for horses. For 200 sq. yd., 26 tons of cubical stone, 1½ ins. to 2½ ins., 5 cu. yd. sand, 52 cwt. rapid hardening cement. Cost of the surfacing only, 3*s.* to 4*s.* 6*d.* per sq. yd.

Brick Paving (H. Boot, 1924).—Typical case with bricks at 6*l.* per 1,000 and bituminous joints, 8*s.* 9*d.* per sq. yd.; bricks, 5*s.* 9*d.*; jointing, 1*s.*; labour and other charges, 2*s.*; cost at Keighley, 8*s.* per sq. yd.

Brick Paving, experimental. Wolverhampton-Walsall main road, concrete foundation, blue brick on edge. North Midlands, (1) Birmingham, wire-cut blue brick on concrete strength crust, floated ¾-in. sand-cement; joints grouted 1½ to 1 mortar; laid Sept. 1922; (2) Stone-Newcastle, Class I. main road, very heavy traffic; 7-in. to 8-in. 6 to 1 concrete strength crust; 9 ins. by 4½ ins. by 3 ins., common blue wire-cut bricks on edge, laid 1922; (3) Wolverhampton-Stourbridge, Class I. main road, two test lengths laid 1923 and 1925; (4) Walsall-Wolverhampton, Class I. main road, 3,000 sq. yds. blue brick grouted with cement, on concrete strength crust; (5) West Bromwich, Class II. road; exceptionally heavy steel-tyred traffic. From a report by J. E. Swindlehurst, M.Inst.C.E., on inspection in the summer of 1926. Relating to (1) little signs of wear under heavy traffic; (2) speaking broadly, surface good after 3½ years; (3) generally in a good state; (5) indications of wear after 3 years.

Cold Emulsion Work. *Oxfordshire* (C. E. Dolphin).—Surfacing main roads with field flints and gravel, the flints from the Chiltern Hills, the gravel from the Thames Valley; flints, 3-in. downwards, hand-picked from fields, cost 5*s.* per cu. yd., laid to depth of 3 in. on the scarified road crust, all shapes together; watered and rolled; bound by grouting with cold emulsion, ¾ gallon per sq. yd., or 1 gallon per 1½ sq. yd.; covered with ⅜-in. shingle, 160 sq. yd. per cu. yd.; traffic for a few days, then a sealing coat of emulsion, ¼ gallon per sq. yd.; blinded with fine sand and rolled. Costs, per sq. yd.: flints 8·5*d.*, shingle 0·75*d.*, sand 0·54*d.*, roller and horse hire 2·41*d.*, labour 0·3*d.*, grout 8·25*d.*, sealing coat 2·75*d.*; total, 2*s.* 2·2*d.* A similar but cheaper process is also employed. For roads carrying up to 600 tons per day, a gravel, shingle, sand, grout, sealing coat road costs 1*s.* 11·3*d.* per sq. yd.

Hampshire.—Emulsified bitumen, 2·42*d.* per sq. yd., cost of grouting a limestone road; total average cost, including the sealing, 4*s.* 1¼*d.* per sq. yd.

Herefordshire.—Cold emulsion used all the year round for repairing potholes. Also used for surface dressing. Hot materials employed also.

Somerset.—Waterbound flint crust; swept clean; patched with cold dressing; tarred topping spread, 22 sq. yds. per ton; bituminous emulsion applied, ½ gall. per sq. yd.; blinded with ½-in. chippings; cost of this work on the existing flint crust, 3*s.* 0⅝*d.* per sq. yd.

Metallic Powder.

A finely ground metallic powder ('Sal-ferricite'), largely composed of pure iron, has recently been employed for hardening roads, the cost of surface hardening at Bexley Heath and Teddington being about 8*d.* per sq. yd. It has also been used to harden a surfacing about 2 ins. thick in Barcelona.

General Data.

Cheshire (W. Holland, 1924).—30-ft. carriage-way only, per mile: granite paving on concrete, 30,000*l.* (life, 30 years or more). Asphalt on concrete, 20,000*l.* (life of surface, 20 years). Two-coat bituminous surface on existing foundations, 13,000*l.* (life, 20 years). Reinforced concrete, 10,000*l.*; surface limited life, foundations, 30 years or more. Tar macadam, tar-sprayed, 5,280*l.*; life, 10-12 years. Water-bound macadam, 2,200*l.* (life with light traffic, 12 years or more).

Generally (E. S. Sinnott, 1924).—Per sq. yd.: 4½-in. water-bound limestone macadam, 2*s.* 4*d.*; 4½-in. water-bound granite macadam, 3*s.* 9*d.*; 4½-in. tarred slag or stone macadam, 5*s.* 2*d.*; 4½-in. tarred bituminous macadam, 5*s.* 6*d.*; 6 to 9 in. concrete, plain or reinforced, 9*s.* 6*d.* to 12*s.* 6*d.*; 2-in. asphaltic carpet, 6*s.* 6*d.* to 8*s.* 6*d.*; 4½-in. asphaltic macadam, 9*s.* 6*d.*; 4½-in. asphaltic part base and part carpet, 13*s.* to 16*s.* Asphaltic carpet, 2-in., on concrete or reinforced concrete base, 16*s.* to 20*s.* Wood block paving on concrete, 22*s.* 6*d.* to 31*s.*

Top Courses and Carpets.—British report to Fifth International Road Congress: bituminous and asphaltic work; ordinary costs, with five years' maintenance included: 2-coat work, 4-in., 11*s.* 6*d.* to 13*s.*; single-coat, stone and sand aggregates, 3-in., 9*s.* to 10*s.*; 2½-in., 8*s.* to 9*s.* 3*d.*; 2-in., 6*s.* 6*d.* to 7*s.* 6*d.*; clinker, 2-coat, 3-in., 7*s.* 6*d.* to 8*s.* 6*d.*; single coat, 2-in., 6*s.* 3*d.* to 7*s.* 3*d.*; mastic, 2-in., 8*s.* 9*d.* to 10*s.* 6*d.*; compressed rock asphalt carpet, 12*s.* to 15*s.*

Proportions of Surfacings.—Somerset: Classified roads and urban main roads; asphaltic and bituminous, 13 miles; tarmacadam, 155 miles; waterbound granite, limestone, and other stone, so classed, 540 miles; waterbound flint or gravel, 37 miles; other materials, 12 miles.

Surface Dressing.

Area covered.—The area covered by a gallon of tar is seldom less than 6 or more than 8 sq. yds. The cost varies with the local price of tar, an ordinary rate being 2*d.* to 3*d.* per sq. yd.
Norfolk, 5·6 sq. yd.; bitumen, 4·0 sq. yd. Hampshire, hot bitumen, 7·32 sq. yd.

Related Data.—Hampshire county roads: surface tarring, hot; 8·74 sq. yd. per gal., 1·850*d.* per sq. yd. Area, sq. yd., per man-hour, tarmacadam surfaces, 90·8; other previously treated surfaces 86·3, surfaces not previously treated 44·5; average, 83·9. Cold emulsion of bitumen, 3·82 sq. yd. per gal. Kenilworth (1928–1929): tar-spraying; 6 sq. yds. per gal.; 125 sq. yds. per ton of granite chippings, rolled in; 3½*d.* per sq. yd.

Surface Tarring.—Evesham and Pebworth rural district, 3·6*d.* per sq. yd. Somerset, county roads, 3·3*d.* Essex, average of costs of dressings, 2·82*d.* Warwickshire, tar-spraying, 2·87*d.*, 7·38 sq. yd. per gal. Staffordshire, 2·92*d.* London: average of 9 boroughs, 2·4*d.*; greatest, 3·37*d.*; least, 1·63*d.* Hammersmith (6·94 sq. yd. per gal.; cost 2·51*d.* per sq. yd. in 1920); 1926, 8·4 sq. yd. per gal., cost 1·87*d.* per sq. yd.; 1927, 1·63*d.* Durban, 6*d.* per sq. yd. Port Adelaide, tar-dressing: roadways, 4*d.*; footpaths, 2½*d.* Hampshire: tar, 1·9*d.*; bitumen, 2¾*d.* 1928–1929: Holland, Lincs., 3·51*d.*

Northants, surface dressing, 3·8*d.* Deptford, 2·51*d.* Warwickshire, 1927, 161*l.* per mile of road, surface dressing. Hampstead, hot bitumen dressing, 2·71*d.* per sq. yd. Taunton Rural District, surface dressing, contract, 3¼*d.*

Glasgow, 1927–1928, surface treatment, with chippings; 60 per cent. No. 1 tar and 40 per cent. bitumen; 5·29*d.* per sq. yd.; also, 60 per cent. No. 2 tar and 40 per cent. bitumen, 5·15*d.* per sq. yd. 1928–1929: 60 per cent. No. 1 tar and 40 per cent. bitumen, 4·54*d.*

State of Victoria: double coating, 8·59*d.*; single coating, 5·28*d.*; average on various surfaces.

Analyses of Tarring Costs.

Dorking.—Tar-spraying: manual labour, 17·3 per cent.; team labour, 16·7 per cent.; tar, at 7½*d.* per gallon, 33 per cent.; grit (at about 1 ton per 30 gallons of tar, cost 14*s.* 1*d.* per ton), 25 per cent.; tradesmen's bills, 8 per cent.

Norfolk, urban, 1924 (F. Hall Smith).—Manual labour, 0·43*d.*; tar, at 7*d.* per gallon, 0·82*d.* (covering 7½ square yards); gravel, 1·48*d.* (covering 150 sq. yds. per load); tools and establishment, 10 per cent. Total, 1·85*d.* per sq. yd.

Consett, Durham, 1929 (T. Hutton): Hot work; tar at 5½*d.* per gal., 5 sq. yds. per gal.; limestone chippings at 14*s.* 1*d.* per ton, 127 sq. yds. per ton; cost: chippings, 1·34; tar and bitumen, 1·12; labour, 0·72; transport, 0·21; plant, etc., 0·05; total, 3·44*d.* per sq. yd.; Cold work: emulsion at 9d. per gal., 4½ sq. yds. per gal.; chippings, 115 sq. yds. per ton; cost: chippings, 1·44; emulsion, 2·31; labour, 0·52; transport, 0·21; plant, 0·05; total, 4·53*d.* per sq. yd.

1927–29.

Deptford, Urban, 1927 (H. Morley Lawson).—Pence per sq. yd.: Tar, or other dressing material, 0·286; chippings, 0·503; wages, 0·314; cartage, 0·325; coke, oil, repairs and sundries, 0·078; total, 1·506.

Analysis of Cost, 1924 (A. H. Prince).—Per cent., tar or bitumen, 53; haulage, 25; labour, 12; grit, 8; sundries, 2.

Areas and Mileages, in a year:—

Kent.—1924–1925: the county spent 48,000*l.* on surface treatment, besides contributing 16,000*l.* towards the costs of similar work by urban authorities. *Worcestershire.*—Spent 35,394*l.* on tar painting. *Norfolk.*—5,337,000 sq. yds. treated with tar and 1,180,000 with bitumen, using nearly 944,000 gallons and 259,000 gallons respectively. *Essex*, used 959,000 gallons of tar. *Hampshire*, 392 miles of water-bound main roads are surface dressed.

Emulsified Bitumen.—Applied instead of tarring: Hampshire, 3·28*d.* per sq. yd.; Axbridge rural district, (*a*) first dressing, October, 3·9*d.*, second dressing, in the following May, 3·7*d.*, with chips; (*b*) first dressing, in January, 5·0*d.*, second dressing, with chips in the May following, 5·8*d.*; Bedford, single dressing, 1·86*d.*

Road Machinery.

Concrete Mixers.—From makers' descriptions. Portable concrete mixers, rated as 5 cwt. or 6 cu. ft. capacity, petrol engine, loading skip; output, 25 tons per day or more. Portable mixer, rated as 7 cu. ft., chain drive, interchangeable axles, 3½ h.p. engine. A 5-h.p. mixer, output up to 80 cu. yd. per day. A tilting drum mixer, paraffin engine, 30 to 32 batches of 3½ cu. ft. per hour.

Tar-Macadam Plant.—Drying and mixing plants, 34 to 200 tons per day, range of sizes; others, up to 200 tons per day.

Asphalt-making Plant.—Average on one job, 6½ tons of 2-in. carpet; best day's work, 7½ tons per hour. A 10-ton portable plant, output 1,000 sq. yds. of 2-in. bituminous carpeting in an 8-hour day.

Roadside Excavator.—For road widening work; cut, 18 ins. wide up to 2 ft. high; speed, 30 linear yds. per minute (presumably average soil); may be adjusted to work below road level, or above (up to 18 ins.), and below at the same time; can be fitted to any tractor or roller.

Pneumatic Spade.—A tool on the principle of a pneumatic riveting and chipping machine is the pneumatic spade.

Tar-spreading Machines.—Barnes machine, 1,000 and 700 gal. sizes; brushes made to move to and fro while spreading the tar. Clayton tar-spreading steam wagon, 6-ton, 1,000 gal., 4 to 5 miles per hour speed, interchangeable bodies.

Road Gritting Machine.—Towed by tar-spraying machine; carries 2 to 3 tons of grit; made for various widths, normally 2 to 6 ft.; up to 9 ft. if required.

Street Sweeping Attachment.—About 2½ h.p.; speed, 6 to 8 miles per hour; 6-ft. track; brush 7 ft. long by 20 in. diameter; 70-gal. water tank; two sprinkler nozzles.

Compressed Air Data.—At pressures up to 120 lb. per sq. in.; 100 cu. ft. of air per minute will operate any of the following: 2 rock drills, 2 road chisels, 5 riveting hammers.

Tar-spraying.—Steam vehicles, for whole process, capacities up to 6,000 yards per hour.

Breaking up Concrete.—Machines for breaking up old concrete foundations; 6 to 10 cubic yards per hour on a 12-in. crust.

Crawler Excavator.—Weight, 15 tons; tools; shovel, dragline, grab, back-acting trencher skimmer, scoop, crane; petrol-paraffin engine, 4-cyl., 32 b.h.p. (or steam or electric), 1 to 1¼ gal. of paraffin per hr.; 3 digs per minute when working shovel.

Trench Pump.—With suction head 15 ft., 7,000 gal. per hr., with total head 50 ft., 5,000 gal. per hr.; 4-in. suction hose, 2½ h.p.

Petrol Hoist.—Petrol or paraffin; 4-cyl., 20 h.p., pulls 4,000 lbs. at 125 ft. per min. on each drum.

Road Rollers.

Cardiff R.D.C.—Steam roller, 10-ton, purchased 1897, sold in working order 1924; cost per annum, including overhaul and all additions and repairs, 58*l.* Depreciation, 3·6 per cent. of original cost and cost of scarifiers (1903); maintenance, 8 per cent.; total cost per annum, 11·6 per cent. Steam roller, 12-ton, bought 1905, sold 1924; depreciation 3·05 per cent.; maintenance, 7·87 per cent. Steam roller, 10–12 ton, 772*l.* Motor roller, 773*l.*

Working Costs.

Somerset County Council (E. Stead).—Steam rollers, 1*l.* 9*s.* 9*d.* per day. Analysis, percentages: wages, 40·47; coal, 31·04; oil, 2·38; repairs and renewals, 15·57; sundries and insurance, 1·31; depreciation, 9·30. Petrol rollers, 1*l.* 1*s.* 2*d.* per day. Analysis: wages, 49·26; petrol, 24·68; oil, 2·77; repairs and renewals, 11·66; sundries and insurance, 1·03; depreciation, 10·60.

Warwickshire.—Steam rollers; 13 rollers; average days worked, 231; cost per day, *not* including depreciation or interest, 1*l.* 7*s.* 3*d.* Hired rollers, consolidated road metal at the rate of 1*s.* 11·6*d.* per ton. Holland, Lincs., cost per day, 1*l.* 8*s.* 9*d.*

Northants.—Cost of consolidation of road metal by steam rolling, per ton: 1926–1927, 1*s.* 4*d.*; 1927–1928, 1*s.* 1½*d.*; 1928–1929, 11·65*d.*

Holland, Lincs.—1928–1929, 8·83*d.*

Horse Haulage Costs.

Southport.—Refuse collection, cost per house (or other premises) cleared, per annum, 11*s.* 6*d.*

Warwickshire.—Haulage from wharves and railway stations, 6*s.* 5*d.* per ton; 2*s.* 3*d.* per ton-mile.

Hampshire.—14*d.* per ton-mile.

Camberwell.—4 tons per day, distance 10 miles, cost per ton-mile, 8·7*d.*

Newcastle-on-Tyne.—Refuse collection, cost per bin emptied, 2·31*d.* (compared with steam 2·16*d.*, and petrol 2·03*d.*).

Lincolnshire, 1925.—Cart, horse, and driver, 11*s.* per day. Durham 13*s.* Another county, hire of horse and wages of carman, 4*l.* 4*s.* per week.

Durban.—Single-horse cart, 9*s.* 6*d.* per day; 3-mule carts, working under different conditions, 18*s.* per day; up to 1 mile hauls both cost about 1*s.* 4*d.* per ton.

Haulage Costs.

(See also p. 759.)

From wharves to jobs, country road work, 5*s.* 10½*d.* per ton. Steam wagons, 9·7*d.* per mile (compared with petrol wagons, 9·3*d.* per mile).

Refuse Collection, 1923.—Working cost 4*s.* 3*d.* per ton when total cost was 7*s.* 10*d.* per ton. A comparison (in the same town): *horsed vehicles*, by day 8*s.* 7*d.*, by night 8*s.* 7½*d.*: *motor vehicles*, by day 7*s.* 11½*d.*, by night 8*s.* 10½*d.*, all per ton. 1924. Per ton: motor vehicles 12*s.* 4*d.*; horsed vehicles 13*s.* 8*d.* Under some conditions the difference is greater. In a number of cases under different conditions to these horsed vehicles cost less. 1926–1927. Leighton Buzzard, 4*s.* 10½*d.* per load, 3*s.* 8*d.* per ton. Deptford: public cleansing vehicles, all 3–4 ton end tipping, six comparable vehicles selected, costs ranged from 10*d.* to 1*s.* 8*d.* per ton-mile; mean, 14*d.* nearly. At Coventry, the 'Pagefield' system, cost 10*s.* per ton, has saved 2,000*l.* in one year, working 118 streets per week.

Deptford, 1927–1928 (H. Morley Lawson).—Road sweepings, 3–4 ton tipping lorry; average weight per load, 3 tons; mileage, 6,932; petrol per day, 4·49 gal.; miles per gal., 5·23; cost per day, 2*l.* 5*s.* 4*d.* Dust and road sweepings: 3-ton low-loading lorry; average weight per load, 3 tons 2 cwt.; mileage, 1,774; petrol per day, 2·22 gal.; miles per gal., 6·72; cost per day, 2*l.* 15*s.* 6*d.* Road materials: 3–4 ton tipping lorry; average weight per load, 2 tons 6 cwt.; mileage, 5,863; petrol per day, 4·23 gal.; miles per gal., 4·59; cost per day, 2*l.* 5*s.* 4*d.* Dust, road sweepings, and materials, five vehicles, 3–4 ton 'Halley' tipping lorries; average weight per load, 2 tons 14 cwt. to 3 tons 1 cwt.; mileages, 1,877 to 2,472; petrol per day, gal., 4·29, 4·70, 4·73, 4·83, 6·41; mean, 5·0. Miles per gal., mean, 3·1; cost per day, mean, 2*l.* 7*s.* 0*d.*; costs per ton-mile, pence, 11½, 14½, 16¾, 19¾, 26; mean, 17·7.

HAULAGE OF ROAD MATERIALS.

Hampshire.

(W. J. TAYLOR, M.Inst.C.E., County Surveyor: Annual Report.)

(The 'legal loads' were not exceeded.)

Hydraulic Tipping Steam Lorries.—Four of 3¼, 3½, 3¾, and 4¾ tons legal loads. Average ton-mileage, 13,740; average cost per ton-mile, 8·93*d.*; highest, 10·34*d.*; lowest, 7·25*d.*

Screw Tipping Steam Lorries.—Four of 3½, 3¾, 3¾, and 3⅞ tons legal loads. Average ton-mileage 14,500; average cost per ton-mile, 7·82*d.*; highest, 9·1*d.*; lowest, 6·6*d.*

Non-Tipping Steam Lorries.—Three of 4, 4, and 4⅜ tons legal loads. Average ton-mileage, 13,307; average cost per ton-mile, 8·91*d.*; highest, 10·9*d.*; lowest, 6·5*d.*

Eleven Steam Lorries.—As above, average ton-mileage, 13,897; average cost per ton-mile, 8·265*d.*

Petrol Lorry, Non-Tipping.—3½-ton, ton-mileage, 20,125; cost per ton-mile, 7·46*d.*

One-ton Petrol Lorries.—Three, average ton-mileage, 8,150; average cost per ton-mile, 8·23*d.*

The costs per day for steam vehicles ranged from 1·31*l.* to 2·17*l.*; average, 1·62*l.* For the 3½-ton petrol vehicle, 2·28*l.*; for the 1-ton trucks, 1·11*l.*

NOTE.—The full table in Mr. Taylor's Report should be consulted, or p. 550 of the July 1926 issue of the *Surveyor*. The figures show in a remarkably clear manner the effects upon costs of the factors:—

(*a*) Ton-mileage and (*b*) empty mileage to full mileage. Considering the range of vehicles and of loads, the degree of conformity with these factors included is notable. No exhaustive study has been made to find the critical figures of basic comparison, but there is a special significance in a portion of the cost, amounting to about 4·5*d.* per ton-mile. Taking this figure and a basic ton-mileage of 14,000, the probable cost in pence per ton-mile for any of these vehicles is indicated by some such formula as:

$$\left(4{\cdot}0 + 4 \times \frac{14{,}000}{\text{ton-mileage}}\right) \times \frac{\text{empty mileage}}{\text{full mileage}}$$

This tentative formula, or qualifier, gives the following results: 9·8 against 10·34 actual; 7·4 against 7·25; 7·03 against 9·1 (relatively few working days); 6·6 against 6·6; 10·5 against 10·9; 7·03 against 6·5; total for steam vehicles, 7·82 against 8·27 actual; petrol vehicle 6·78 against 7·46; 1-ton trucks, 8·16 calculated against 8·23 actual.

See WORKING COSTS DATA, p. 756.

Public Service Vehicles: Prices, 1926–1928.

Motor Fire-engines.—1,400*l.*, 1,125*l.*, 1,050*l.*, 725*l.* Motor-cycle fire engine outfit, 300*l.*

Motor Ambulances.—770*l.*, 700*l.* 20–25 h.p., 710*l.* Others, 650*l.*, 255*l.*

Tipping Wagons.—Steam, 3-way, 945*l.* Clayton 6-ton, end-tipping, 885*l.*

Refuse-Collecting Vehicles.—Several types, 580*l.* to 600*l.*; 560*l.*; 1,070*l.*

Gully-Emptying Vehicles.—Steam, 1,050*l.*, several types about 1,000*l.*

Motor Cesspool Emptier.—1,390*l.*

Motor Lorries.—425*l.*; 1-ton, 240*l.*; 1-ton Ford, 225*l.*; light motor van, 200*l.* Dennis, 3½-ton, 735*l.*; 2½-ton, 700*l.*; 'Pagefield' refuse-collecting lorry, with 5 containers, 2,046*l.*; tipping lorry, 30-cwt., 350*l.*; 6-ton motor wagons, 1,130*l.*

Steam Wagons.—700*l.*; 6-ton, 745*l.*

Freighters (S. D.).—572*l.*; dustcart, 'dustless' type, 7 cub. yd. capacity, 683*l.*; end-tipping, 634*l.*

Tractor and Trailer.—For refuse collection, hydraulic tipping tractor, 8½ cub. yd. capacity, 477*l.* for the unit; extra trailers, each 190*l.*

Street Sweepers.—Rotary sweeper, 578*l.*; 487*l.* Sweeper, sprinkler, and collector, 1,000*l.*; 984*l.* Sweeper and sprinkler, 850*l.* Street washing machine, 1,300 gal., 1,075*l.*

Motor Omnibuses and Coaches.—Urban type, 1,250*l.*, 1,144*l.*; single-deck, 6-wheeled, 1,200*l.*; 26-seater, saloon, 1,150*l.*; 20–24 seater, 1,070*l.*; 20–26-seater, 995*l.*; 16-seater, 795*l.*; chassis for 32-seater, 780*l.*; chassis, 14-seater, 375*l.* 18-seater, 535*l.*; 20-seater, 735*l.*

Tram-cars.—Single decker, 2,455*l.*

Footpath Roller.—Motor roller, 1¾-ton, convertible to lawn mower, 262*l.*

Vehicles and Road Plant for 442 Miles of Main Roads, Worcestershire.

(*C. B. Hammond.*)

(*To deal with heavy post-war arrears of work.*)

Fifteen 5-ton steam waggons; two 3-ton petrol waggons; one light works lorry: six motor cycles and side-cars, for divisional surveyors; five trailer waggons for the 5-ton lorries; thirteen steam rollers, 8-ton and 10-ton; two light patching rollers, with trailers; twenty-eight sleeping vans; eighteen water carts; seventeen horse-drawn tar-painting machines; ten hand-drawn tar-painting machines; thirteen horse brooms.

Equipment for Refuse Collection.

Gosport (population 3,400, area 3,998 acres). Vehicles, one Fordson tractor; three covered, 2-wheeled, 2-ton trailers; one 1-ton Ford van, sundry horses—method, horses collect only, tractor plies with trailers to and from the destructor. Cost per ton, 7*s.* 8*d.* Collection twice a week.

New Road Construction: Rates.

(*W. P. Robinson, Surrey.*)

Surface excavation not exceeding 12 ins. deep and run to tip, 8*d.* per sq. yd.; excavation to formation, hauling any distance, shaping, formation and trimming slopes, 2*s.* 8*d.* per cub. yd.; reinforced concrete post and wire fence, 4 ft. 6 ins. high, including straining posts and all fittings, 3*s.* 3½*d.* per lin. yd.; provide, spread and roll clinker, 3 ins. thick, to formation, 10*d.* per sq. yd.; concrete in carriage-way, 8 ins. thick, excluding reinforcement, laid and finished, 7*s.* 9*d.* per sq. yd.; single layer reinforcement, weight 5 lbs. per yd. laid, and including all jointing, laps, etc., 1*s.* 3*d.* per sq. yd.; double layer reinforcement, weight 10 lb. per yd., laid and including joints, laps, etc., 2*s.* 2*d.* per sq. yd.; 10 in. by 6 in. concrete kerb bedded and jointed in cement mortar to carriage-way, 5*d.* per lin. yd.

Tree Planting.

(*R. M. Morton.*)

Rosyth.—Average cost of the tree, 1*s.* 2½*d.*; limes, 1*s.* 7*d.*; chestnuts, 1*s.* 6*d.*; Lombardy poplars, 9*d.* Handling, transport and planting, 3*s.* 3½*d.* per tree; total cost per tree, 4*s.* 6*d.* Relates to 2,721 trees.

Dunfermline.—Total cost per tree, 2*s.* 7*d.* Relates to 210 trees.

Essex.—Programme of tree-planting on arterial roads; estimate, 6,000*l.*; probably at the rate of 500*l.* per annum.

Standard Trees for Roads and Hedges.

[Based on information supplied by Lt.-Col. H. V. Lane, Lane's Nurseries, Berkhampstead.]

For Roads.—(i) Planted 60 ft. apart. Height of tree: Common ash, common beech, chestnut, lime, plane, poplar, sycamore, 8 to 12 ft.; oak, 7 to 10 ft.; elm, 6 to 10 ft.; walnut, 6 to 8 ft.; purple beech, 5 to 7 ft.

(ii) Planted 20 ft. apart. Height of tree: Maple and mountain ash, 8 to 10 ft.; acers and thorns, 6 to 7 ft.

For Hedges.—(iii) Planted 1 ft. apart, 3 to 3½ ft. in height: beech, hornbeam, privet. (iv) Planted 7 to the yard: thorn (quick), 2½ ft. high. (v) Planted about 2 ft. apart, 3 to 4 ft. high, holly, yew, box, and cypress.

Costs per Mile of Road.—Basis for independent calculation: per penny per tree, 7*s.* 4*d.* per mile; per shilling per tree, 4*l.* 8*s.* (vi) Single line of trees, stakes at 9*d.* each: at 2*s.* 6*d.* per tree, 14*l.* 6*s.*; at 3*s.* per tree, 16*l.* 10*s.*; at 3*s.* 6*d.* per tree, 18*l.* 14*s.*; at 4*s.* per tree, 20*l.* 18*s.*; at 4*s.* 6*d.* per tree, 23*l.* 2*s.*; at 5*s.* per tree, 25*l.* 6*s.*; at 5*s.* 6*d.* per tree, 27*l.* 10*s.*; at 6*s.* per tree, 29*l.* 14*s.*; at 6*s.* 6*d.* per tree, 31*l.* 18*s.*; at 7*s.* per tree, 34*l.* 2*s.*; at 7*s.* 6*d.* per tree, 36*l.* 6*s.*

(vii) *Hedges*, such as (v) (above), taken as 2 ft. apart; as stakes are not required, other spacings are in proportion: for *single hedge*, at 2*s.* 6*d.* per tree, 330*l.*; at 3*s.*, 396*l.*; at 3*s.* 6*d.*, 462*l.*; at 4*s.*, 528*l.*; at 4*s.* 6*d.*, 594*l.*; at 5*s.*, 660*l.*; at 5*s.* 6*d.*, 726*l.*; at 6*s.*, 792*l.*; at 6*s.* 6*d.*, 858*l.*; at 7*s.*, 924*l.*; at 7*s.* 6*d.*, 990*l.*

(viii) *Hedges*, such as (iii) (above), taken as 1 ft. apart, other spacings in proportion; double the values in (vii).

Quickset Hedge (thorn).—Taken as 7 to the yard: at 5*l.* per 1,000, 61*l.* 12*s.*; at 7*l.* 10*s.* per 1,000, 92*l.* 8*s.*; or, with a per yard and b shillings per 1,000, the cost of *one* hedge is $£\frac{a \times b \times 88}{1000}$.

Watford By-pass Road.—Planted with beech, birch, and sycamore. The choice of beech and sycamore, especially the latter, has been adversely criticised.

Charing, Kent.—North and south by-passes: mountain ash, wych-elm.

Hampshire.—Main road, 1928–1929: 800 trees planted at widenings and on treeless stretches of road; species, the same as those growing near; regular rows avoided, planting in irregular clumps.

WORKING COSTS DATA FOR VEHICLES.

(See also p. 756.)

Steam Wagons and Lorries.

Steam Wagons.—Per day, 2*l.* 15*s.*; per loaded mile, 5*s.* 7*d.*; per mile, 2*s.* 9*d.*; per ton of load, 5*s.* 1*d.*; per ton-mile, 1*s.* 2*d.* Depreciation, 14 per cent.; wages, 40 per cent.; coal, 15 per cent.; repairs and renewals, 12 per cent.; tyres, 8 per cent.; sundries, 6 per cent.; oil, 5 per cent. Miles travelled in year, loaded, usually 1,500 to 2,000; unloaded, about the same. Working days: four vehicles exceeded 180, three exceeded 120 (out of nine vehicles)—these are 5 to 6-ton vehicles. Average load per journey, about 4½ tons; average distance per journey, three vehicles about 5 miles, three vehicles about 3½ to 4 miles, the other three less.

Liverpool (see 'Electric' and 'Petrol').—Steam wagons, costs: per ton, 5*s.* 5*d.*; per mile, 3*s.* 9*d.*; per ton-mile, 1*s.* 6½*d.*

Camberwell.—Steam wagon, under favourable circumstances, 1922: Running costs, fuel, 28 per cent.; oils, tools, and sundries, 5½ per cent.; licence and insurance, 4 per cent.; repairs and fittings, 5½ per cent.; wages, 53 per cent.; depreciation, 15 per cent.; tyres, 3 per cent.; or, including extra labour for loading, about half these percentages, and extra labour nearly 50 per cent. Total cost, 392*l.*; mileage, 857; cost per mile, 9*s.* 2*d.*; cost per mile without extra labour for loading, 4*s.* 7*d.* Hauling, with trailer, 26 tons per day, distance 10 miles: cost per day, 3*l.* 3*s.*; extra labour for loading, 3*l.* 2*s.*; cost per ton-mile, 5·7*d.*

Godalming.—Oil-fired steam lorries, heavy oil at 10½*d.* per gallon; six ton-miles per gallon.

Warwickshire.—Per working day, 2*l.* 17*s.* 4*d.*; tons hauled per annum, 5,273; cost per ton delivered on to the road, 2*s.* 6¾*d.*

Nottinghamshire.—For steam wagons: average age, 7 years; average load, 6·5 tons; average mileage, 447; average loaded mileage, 202·5; costs: per day, 3*l.* 7*s.* 3*d.*; per mile, 6*s.* 5½*d.*; per ton-mile, 1*s.* Depreciation at 12½ per cent. has already cleared off the cost of two of these vehicles. About three months after the end of the financial year, 1924–1925, the whole capital sum had been discharged, and the working cost per ton-mile reduced to 10*d.* 1926–1927: steam wagons and trailers; average days worked, 192; tons per journey, 6·56; cost per day, 2*l.* 14*s.* 6*d.*; cost per ton-mile, 10·2*d.*

Durban.—6-ton steam wagons, 1*s.* 4*d.* per ton-mile.

Somerset County Council (W. J. Taylor).—Steam wagons, 7 Foden and 10 Sentinel: costs, 1927–1928: per working day, 2*l.* 12*s.* 11¾*d.*; per mile, 1*s.* 7¾*d.*; per loaded mile, 3*s.* 2¾*d.*; per ton carried, 2*s.* 11¾*d.*; per ton-mile, 7·45*d.* (2*d.* less than in 1925). Working costs, percentages: wages, 46·16; coal, 13·98; oil, 3·39; repairs and renewals, 14·32; tyres, 5·01; sundries, insurance, and licence, 11·99; depreciation, 5 per cent., 5·15.

Berkshire County Council (J. F. Hawkins), 1927–1928.—Steam wagons, 3 Clayton and Shuttleworth: miles, loaded, 6,719; unloaded, 6,945; total, 13,664 (one vehicle sold, January 1928), total miles for the other two, 10,558; the three; tons hauled, 11,903; days worked, 662½; days standing by for repairs or delays, 117½; loaded journeys, 1,932; empty, 1,993; ton-miles, 41,396; average, load per journey, 6·16 tons; loaded journey, 3·48 miles. Working costs, all charges: per working day, 2*l.* 9*s.* 8*d.*; per mile, 2*s.* 5*d.*; per loaded mile, 4*s.* 11*d.*; per ton carried, 2*s.* 9*d.*; per ton-mile, 9½*d.* Total cost, 1,644*l.* 0*s.* 3*d.*; including, wages, 829*l.*; fuel, 259*l.* 5*s.*; repairs and renewals, 240*l.* 13*s.*; depreciation, 119*l.* 15*s.*

Surrey County Council (W. P. Robinson).—Four Foden steam wagons; 5-tonners; miles loaded, 15,950; unloaded 12,282; tons hauled, 11,299; days worked, 954; ton-miles, 74,307; average costs, per day, 3*l.* 16*s.* 3*d.*; per ton-mile, 11·74*d.* (see p. 760 for Surrey petrol vehicles). Reduced, relative costs, 5-ton vehicles per ton-mile; steam, 5·65*d.*; petrol, 4·87*d.* (see p. 760).

Holland, Lincs. County Council (W. A. Rogerson).—5-ton rubber-tyred steam and petrol wagons, with mechanical tipping gear; cost per ton, 2*s.* 9½*d.*; cost per ton-mile, 10·15*d.*

Electric Vehicles.

Worcestershire.—County council lorries (all costs except capital charges, depreciation, and insurance), 14·9*d.* per ton-mile.

Willesden.—2½-ton vehicles; capacity, 6 cubic yards; units per mile, 1·223; tons lifted per man-day 2·29; tons per vehicle per day, 6·28; cost (total) of collection, 11*s.* 2*d.* per ton.

Liverpool (see under 'Steam' and 'Petrol').—Cost per ton, 4*s.* 6*d.*; cost per mile, 5*s.* 9*d.*; cost per ton-mile, 3*s.* 4*d.*

Canterbury.—2½-ton vehicles: cost of electricity, per mile, 7·23*d.*; per ton, 13·7*d.*; cost of electricity, oil, water, grease, and repairs, 20·3*d.* per ton-mile. Total cost per ton-mile, including capital charges and insurance, 2*s.* 9*d.*; per working day, 6*l.* 17*s.*

Wolverhampton.—(A) Vehicles 1½ years old: kilowatt-hours, per mile, 2·22; costs—capital charges, 22 per cent.; battery, 21·6 per cent.; tyres, 2·6 per cent.; insurance, 1·6 per cent.; wages, 30·7 per cent.; repairs and maintenance, 5 per cent.; electrical energy, at 2¼*d.* per unit, 16·5 per cent.; cost per ton-mile, 10·2*d.*; cost per hour in use, 5*s.* 9*d.* (B) Vehicles 3½ years old: kilowatt-hours, per mile, 1·87; costs—capital charges, 21 per cent.; battery, 15·5 per cent.; tyres, 1·8 per cent.; insurance, 0·7 per cent.; wages, 41·6 per cent.; repairs and maintenance, 9·9 per cent.; electrical energy at 2¼*d.* per unit, 9·5 per cent.; cost per ton-mile, 1*s.* 4*d.*; cost per hour in use, 7*s.* 11*d.*

Nottingham.—October 1915 to December 1924: 3-ton electric vehicles; refuse collection. Total costs: current, 4,802*l.*; tyres, 848*l.*; battery upkeep, 453*l.*; maintenance, 3,905*l.*; licences, insurance, etc., 1,678*l.*; depreciation, 13,893*l.*; total upkeep, 25,597*l.*; wages, drivers and mates, 46,341*l.*; grand total, 71,920*l.*; mileage, 322,797; tons, 17,631; cost per ton, 8*s.* 2*d.* Now twenty-eight such in service.

Petrol Vehicles.

Liverpool (see 'Steam' and 'Electric').—Costs: per ton, 6*s.* 7*d.*; per mile 2*s.* 8*d.*; per ton-mile, 1*s.* 7*d.* *Test*, 5½-ton lorry and trailer; tare of lorry, 4 tons 7 cwt.; of trailer, 1 ton 13 cwt.; average speed, 12 miles per hour; petrol, 5·9 miles per gallon; with nearly 11½ tons load, 60 net ton-miles per gallon; with nearly 4½ tons load, 28·5 ton-miles per gallon; average, 46 ton-miles per gallon; oil, 516 miles per gallon. Another place, 3½-ton lorry, 11 miles per gallon of petrol.

Surrey County Council (W. P. Robinson).—Six 5-tonners; miles loaded, 34,569; unloaded, 24,484; tons hauled, 21,177; days worked, 1,680; ton-miles, 162,548; average costs, per day, 2*l.*; per ton-mile, 4·95*d.* One-tonner, 14·06*d.* 1½-tonner, 17·54*d.* per ton-mile (see p. 759 for Surrey steam vehicles). Reduced, relative costs, 5-ton vehicles per ton-mile; petrol, 4·87*d.*; steam, 5·65*d.* (see p. 756).

Worcestershire.—Ford landaulette, one year, mileage, 6,844; cost per mile, 1*s.* 5½*d.* (this cost includes 'rent,' insurance, 'uniform,' etc.).

Middlesbrough.—Street-watering van: interest and redemption, 18·4 per cent.; tyres, 3·6 per cent.; repairs and renewals, 7·2 per cent.; driver, 22·6 per cent.; oils and grease, 1·4 per cent.; insurance, 1·4 per cent.; petrol (at 2*s.* 2*d.* per gallon), 38·9 per cent.; taxes, 4·3 per cent.; sundries, 2·2 per cent.; cost per day, about 2*l.* 16*s.*

Motor Haulage.—By contract, 13·57*d.* per day; by council's department, 13·40*d.* per day.

Middlesbrough.—Street-sweeping machine: interest and redemption, 10·5 per cent.; tyres, 3·4 per cent.; repairs and renewals, 6·8 per cent.; driver, 17·5 per cent.; oils and grease, 1·8 per cent.; insurance, 1·1 per cent.; petrol (at 2*s.* 2*d.* per gallon), 36·8 per cent.; refills for brooms, 17 per cent.; taxes, 3·3 per cent.; sundries, 1·8 per cent.; costs, about 7*s.* per mile, or 4⅜*d.* per 1,000 sq. yds. Street-sweeping machine, 20*s.* per mile.

Street Sweeping.—By motor sweeper, 7*d.* per 1,000 sq. yds.

Refuse and Sweepings Collection.—1*s.* per ton-mile.

Various Vehicles.

Per Day, 1925.—Steam wagons (Staffordshire County Council), 3*l.* 4*s.* 9*d.*; Nottinghamshire, 2*l.* 14*s.* 9*d.*; Somerset, 2*l.* 13*s.* Steam rollers: nine steam rollers, Warwickshire County Council, working costs only, 29*s.* 6*d.* Nottinghamshire, steam rollers, including the hauling of water, 1*l.* 14*s.* 6*d.* Somerset: steam rollers, 10-ton, working costs, 1928, 1*l.* 9*s.* 1*d.*; petrol rollers, 6-ton, 19*s.* 2*d.* Another case, 1*l.* 13*s.* 7*d.* Durham, 1*l.* 14*s.* 4*d.*

1925–1926.—Cardiff R.D.C., 1*l.* 18*s.* 6*d.*; West Riding C.C., 1*l.* 11*s.* 4*d.*; petrol rollers, 1*l.* 19*s.* 1*d.*; Somerset C.C., 10-ton steam, 1*l.* 9*s.* 6*d.*; 6-ton petrol, 1*l.* 8*s.* 6*d.*; steam wagons working costs only, 2*l.* 11*s.* 6*d.*; rotary sweepers (motor), 1*l.* 15*s.* 2*d.*; gulley emptiers, 2*l.* 17*s.* 3*d.* (per gulley, 2*s.* 9*d.*; watering van, 1*l.* 19*s.* 9*d.*; steam wagons, 3½ to 4¾ tons lading, hydraulic tipping, 1*l.* 12*s.* 6*d.*; screw tipping, 3½ to 3¾ tons, 1*l.* 14*s.* 4*d.*; non-tipping 4-ton, 1*l.* 10*s.* 5*d.*; petrol 3½-ton vehicle in the same 'fleet,' 2*l.* 5*s.* 7*d.* Torquay: refuse collection, 5*s.* 4*d.* per ton, with current at 2*d.* per unit. See under preceding classified headings.

Per Hour.—One-ton horse carts, 3*s.*; 10-ton steam rollers, 8*s.* 7*d.*; 10-ton motor rollers, 8*s.* 7*d.*; motor gully cleanser, 9*s.* 3*d.*; motor broom, 3*s.* 10*d.*—all including attendants. Electric vehicles, 3½-ton, 1½ years old, 6*s.*; the same, 3½ years old, 8*s.* 1923: steam roller (Kent), 6*s.* 3*d.* Vehicles of some years' service (Kent): steam tractors, 7*s.* 7*d.*; petrol lorries, 7*s.* 5*d.*; Foden lorries, 6*s.* 6*d.*

Kent: working expenses per hour, 4*s.* 4*d.*; steam tractors, 5*s.* 4*d.*; petrol lorries, 5*s.* 3*d.*; steam lorries, 5*s.* 9*d.*

Per mile.—Lorries, 9·44*d.*; 1-ton Ford trucks, 7¼*d.* (12·2 miles per gallon); rotary motor sweepers, 1*s.* 9*d.*; watering van, 1*s.* 11*d.*; tipping lorry, 4*s.* 2*d.*; the same, hauling road materials only, 3*s.* 2*d.*; steam waggons, 1*s.* 10*d.*, 1*s.* 9*d.*

Per ton.—Worcestershire county lorries, 2*s.* 11½*d.*; steam waggons, working costs only, 2*s.* 6½*d.* From wharves and stations to jobs, County Council, 3*s.* 4½*d.*; tipping lorry average, 5*s.* 1*d.*; hauling road materials, 3*s.* 6*d.*, 3*s.*, 2*s.* 9*d.*

Per ton-mile.—Tipping lorry, 4–5-ton, average, 1*s.* 8*d.*; hauling road materials, 1*s.* 2*d.*; steam lorry, 8½*d.*; steam lorries, capital charges paid off, 10*d.*; steam wagon, hydraulic tipping, 8·9*d.*; screw-tipping, 7·8*d.*; non-tipping, 8·9*d.*; non-tipping petrol lorry, 7·5*d.* (larger mileage).

Pembrokeshire county steam waggons (A. H. Thomas), 9·3*d.*

Birmingham (A. Baker).—Operating costs: motor omnibuses, 12 months, 15*s.* 4*d.*; tramcars, 12 months, 14*s.* 8*d.*; trolley omnibuses, 6 months, 12*s.* 6*d.* per mile.

Traffic Statistics.

Paisley-Glasgow Road, 6,796 tons per day; eight roads radiating from Liverpool, over seven million tons (less than one million in 1913); rate for July over nine million tons per annum. London, millions of tons per annum: Hyde Park Corner, estimated, 48; Hammersmith Broadway, 19; The Angel, Islington, 16. Railway statistics suggest that about 34 million tons per annum of traffic has been diverted to the roads. (*Lord Montagu of Beaulieu*, '*The Times*,' 1922.)

Main Route Traffic: Tons per Day.—The three consecutive figures are, in every case, pre-war (1910, 1911, or 1912), 1922, 1925: London-Edinburgh, near Framwellgate Moor, Durham, 636, 3,346, 5,348; London-Dover, near Newington, Kent: 568, 1,943, 4,623; London-Portsmouth, near Pease Marsh, Surrey: 790, 3,809, 6,417; London-Holyhead, near Redbourne, Herts., 508, 2,649, 4,166; Liverpool-Sheffield-Lincoln, near Sankey Bridges, Lancashire, 1,167, 8,250, 11,606. The pre-war figures are averages for the year (presumably), those for 1922 and 1925 being in August (British Report on Bituminous and Asphaltic Roads: Fifth International Road Congress, 1916). A road in Norfolk, 4,752; another, 2,326.

Christchurch, New Zealand.—1922 figures compared with 1915; per cent.; light motor vehicles, increase 37·9 in number and 58·5 in weight; heavy motor vehicles, 5·8 and 33·1; bicycles, 42·5 and 4·6; decrease, horse-drawn vehicles, horses and cattle, 13·8 and 88.

Wolverhampton.—1925-1926, Municipal transport services: tramcars, 30; motor-omnibuses, 27; trolley-buses, 16; passengers carried: trams, 15·24; motor-buses, 5·62; trolley-buses, 4·67 millions per annum.

Scotland.—Traffic census, 1922: tons per day, industrial areas, maximum 5,542: rural areas, maximum 739.

Great Britain.—About 14,000 steam waggons and road locomotives, using a million tons of coal and coke per annum.

London.—About 7,000 motor-omnibuses per day, using the Strand, Oxford St., Piccadilly, Regent St., and Parliament St., taxed at 6*d.* per bus-mile, amounting to 7,000*l.* a mile of these routes (*Shrapnell-Smith*).

Paris.—At four cross-roads, 3 p.m. to 7 p.m., during one week: averages per hour; year 1910, 1,240, 2,544, 2,613, and 2,168, respectively; year 1925, in the same order, 2,510, 4,407, 3,677, 3,005.

Average Weight.—Average weight of traffic units: summer, 1922, 1·10 tons; 1925, 1·29 tons; winter, 1925, 1·74 tons; Great Britain.

Heavy Loads.—Sheffield, 100 tons on 4 wheels, 'commonplace'; Sheffield to Birmingham, 75 tons (W. J. Hadfield).

Rate of Increase: Motor Vehicles.—Number of motor vehicles licensed, exceeded the number of the preceding year by, per cent.; 1922 to 1928; 12, 16, 17, 15, 11, 10, 8.

Transport Ministry's Return Relating to Tramway Undertakings.

A return issued (November, 1928) by the Ministry of Transport giving particulars of the tramway undertakings in Great Britain for the year ended December 31, 1927, for companies, and March 31, 1928, for local authorities, shows that during the period covered there were 233 tramway undertakings, of which 167 were owned by local authorities and 66 by companies. The route mileage worked amounted to 2,514 miles, a decrease of 40 miles compared with the previous year.

The total capital expenditure at the end of the period under review amounted to 99,913,760*l.*, an increase of 1,303,255*l.* in the year. The total gross receipts during the year were 27,751,504*l.*, and the total working expenditure 21,943,147*l.*, leaving net receipts amounting to 5,808,357*l.*

The number of passenger journeys during the year was 4,705,842,932, an increase of 5·51 per cent. compared with 1926–27. Car miles run amounted to 396,554,545, an increase of 4·70 per cent. compared with the previous year.

The return also gives particulars of the working of trackless trolley undertakings in Great Britain. At the end of the period the undertakings of twenty local authorities and one company were open for traffic, and the length of route operated amounted to 111 miles, an increase of 24 miles. The number of passengers carried increased by about 59 per cent. to 80,112,970.

ADDENDA.

District Roads: Delegation. (See p. 713.)

The number of rural district councils to which the management of unclassified roads has been delegated is 262, out of a possible 643.

The Road Traffic Act, 1930.

Generally, the Act repeals all legislation which might have been incompatible with its provisions in respect of the control of traffic and regulations for vehicles and loads.

New Classification of Motor Vehicles.

Motor vehicles are now grouped in the classes:

Heavy Locomotives, not load carrying, the unladen weight exceeding 11½ tons.

Light Locomotives, not load carrying, unladen weight 7½ to 11½ tons.

Motor Tractors, not load carrying, unladen weight not exceeding 7½ tons.

Heavy Motor Cars, not being vehicles classified under the section as Motor Cars, unladen weight exceeding 2½ tons.

Motor Cars, not being motor cycles nor invalid carriages: (1) constructed solely for the carriage of passengers and their effects; adapted to carry not more than 7 passengers besides the driver; fitted with tyres of the prescribed type; weight not exceeding 3 tons; (2) in any other case, not exceeding 2½ tons.

Motor Cycles, having less than 4 wheels, the unladen weight not exceeding 8 cwt.

Invalid Carriages, specially designed as such and solely so used, unladen weight not exceeding 5 cwt.

Regulations.—Provision may be made by regulations (1) for sub-division of the above classes and making different provision for each sub-division, varying, as respects any class, the maximum or minimum weight.

Regulations under the Act.

The Motor Vehicles (Construction and Use) Regulations, 1931, and the Motor Vehicles (Authorisation of Special Types) Order (No. 1), 1931; in force from January 15, 1931.

The Regulations repeal 26 Orders made under the Locomotives on Highways Act, 1896, relating to heavy motor cars and motor cars, and supersede the Locomotives Acts in respect of the construction of locomotives. Manufacturers and owners should refer to the Orders for detail as to time limits and exemptions.

Provisions as to Tyres.

Subject to a few exceptions:

(1) On and after January 1, 1933, locomotives and tractors and their trailers must be fitted with pneumatic or with rubber tyres; (2) heavy motor cars and motor cars exceeding one ton in weight unladen; (*a*) registered after January 1, 1933, must be equipped with pneumatic tyres, (*b*) must all be so equipped by January 1, 1940; (3) every trailer constructed after January 1, 1933, drawn by a motor car or heavy motor car, must be equipped with pneumatic tyres, and all such must be so equipped by January 1, 1940. Exemptions, (*a*) land implements, agricultural trailers; (*b*) constructed before January 15, 1931, and (1) designed for horses or cattle; (*c*) (ii) designed for furniture and similar household effects.

The maximum width, 7 ft. 6 ins., of a heavy motor car registered on or before January 1, 1932, may be exceeded by 6 ins. if this increase is necessarily due to conversion to use with pneumatic tyres.

Locomotives.—If the soft tyres obligatory after January 1, 1933, are in sections, these may not be separated by more than ¾ in., and the aggregate of spaces must not exceed 6 ins. This regulation does not apply to agricultural locomotives having smooth-soled steering wheels tyres, not less than 5 ins. tread, and driving-wheel tyres not less than 12 ins. wide, smooth-soled, or shod with diagonal bars not less than 3 ins. wide, nor more than ¾ in. thick, nor more than 3 ins. apart; nor to agricultural tractors, similarly, but steering wheel tyres 2½ ins. and driving-wheel tyres 6 ins.

Maximum Dimensions.

(Not verbatim, but sections given for reference.)

(5) Four-wheeled vehicle, 27 ft. 6 ins.; more than 4 wheels, 30 ft.; articulated vehicle, 33 ft.; 8-wheeled articulated vehicle registered before January 1, 1931, 36 ft. (20) Locomotive, overall width 9 ft. (26) Motor Tractor, width 7 ft. 6 ins. (27) Overhang, 6 ft. (31) Heavy Motor Car, width 7 ft. 6 ins. (39) Motor Car, width 7 ft. 2 ins. (32) and (40a) Overhang in both cases seven twenty-fourths of length. (51) Trailer, length, exclusive of draw-bar, 22 ft., unless normally used for indivisible loads of exceptional length, or being a land implement or part of an articulated vehicle. (52) Overall width (other than a land implement), 7 ft. 6 ins.; a trailer in use by a travelling showman before January 15, 1931, 8 ft. 9 ins.

Maximum Weights.

(21) *Locomotive.*—Unladen, 15½ tons; additional 1½ ton if having a permanent fitting such as a crane or winding drum, and, independently, 2 tons if fitted with soft tyres. Not more than three-quarters of the total weight may be carried by any two wheels.

(33) *Heavy Motor Cars.*—Four-wheeled, 7½ tons; 6-wheeled, 10 tons; more than 6 wheels, 11 tons (permanent fitting allowance as (21)).

Other Regulations.—(56) The maximum weight of a locomotive must not exceed the unladen weight by more than 3 tons.

(57) The maximum weight of all the trailers drawn by a locomotive may not exceed 40 tons.

(58) The sum of the weights transmitted to the road by the wheels of a trailer and of a tractor or heavy motor car drawing it must not exceed 22 tons.

(59) *Heavy Motor Cars.*—Maximum weights: of a single wheel in a transverse line, 4 tons; of any two of four, 8 tons; of any two of more than four, 7½ tons. Total weights: all wheels, 4-wheelers, 12 tons; 6-wheelers, 19 tons; limit, 22 tons.

Heavy Vehicles on Bridges.

The bridge authority of any bridge may, by means of conspicuous notices, prohibit the use of the bridge: (*a*) by a vehicle exceeding a defined maximum laden weight, not less than five tons; (*b*) in respect of the laden weight of a vehicle when travelling at more than a defined speed. Provision is made for appeal to the Minister, who may order a prohibition or restriction to be removed, or may refer the matter to an arbitrator.

RECENT TECHNICAL INFORMATION.

Road Tars: British Engineering Standards Association, Specification, 76—1930, and Part 2—1931.

Requirements.	Tar No. 1.	Tar No. 2.	Tar No. 3.
Specific Gravity at 15 degs. C. not less than	1·140	1·150	1·160
(See Note A) nor more than	1·225	1·240	1·260
Water or ammoniacal liquor not more than, per cent. by weight	0·5	0·5	0·5
Other distillate (light oils) below 200 degs. C., not more than, per cent., by weight	1·0	1·0	1·0
Distillate between 200 degs. C. and 270 degs. C., within the range, per cent., by weight	9·5 to 21·0	8·0 to 16·0	6·0 to 12·0
Distillate between 270 degs. C. and 300 degs. C. (heavy oils) within the range, per cent. by weight	3·5 to 12·0	3·5 to 12·0	3·0 to 8·0
Phenols or crude tar acids, not more than, per cent. by volume	5·0	4·0	2·0
Naphthalene, not more than, per cent. by weight	6·0	5·0	3·0
Free carbon, per cent. by weight	Max. 20	6·0 to 21·0	8·0 to 22·0
Consistency or viscosity Hutchinson, at 25 degs. C., within the range seconds	10–40	40–125	70–120
(But surveyors may specify higher consistencies when required.)			

Note A.—Tar produced from Scottish coals and carbonised in vertical retorts may have minimum specific gravities 1·110 for Tar No. 1 and 1·120 for Tar No. 2, provided such tar complies with the specification in all other details.

Points from Road Tar Specifications and Analyses.

Belgium, Germany, France, Sweden; percentages.—On the whole, the specifications, and the analyses of consistent and approved road tars, follow the British standards. A tar found satisfactory in Sweden has sp. gr. 1·1 and water 0·3, a Paris tar sp. gr. 1·19 and water, min. 1·0. The French departmental specification allows up to 0·75 water, also, distillate 170 to 270 degs. C., min. 12, max. 24. German road tar No. 1, phenols, max. 3·0; free carbon 5 to 16; crude anthracene, max. 3·0; viscosity, Hutchinson, for that tar 8 to 15, for a German tar called 60/40, 20 to 70; Belgium, tars 1 and 2, as the British, the Swedish tar, 52. Drop points are, Belgian 'Wetterteer," laboratory tests, 34 deg. C., German 'Wetterteer,' prescribed, min. 30 degs. C. Distillate below 170 degs. C., in Belgian 'Wetterteer,' nil, in Belgian, German and French specifications, five tars, max. 1·0. In the Swedish tar, analysis 0·2; in the Paris tar, analysis 0·6. The German specifications allow 9 to 17 and 1 to 10, distillate 170 to 270 degs. C., the analyses of the 'Wetterteers' giving, Belgian 1·9, German 1·0.

Road Planning and Design.

Building and Improvement Lines.—The adoption of 120 ft. spacing for building lines and 60 ft. for improvement lines seems to be nearly general. In some counties building lines only, and in others improvement lines only, have been prescribed.

Road Widths.—A Ministry of Transport memorandum gives, as widths to be aimed at: (*a*) for Class I roads, 60 ft.; for Class II roads, 50 ft.; but greater widths should be provided in many cases; (*b*) Normal unit width for carriageways, 10 ft., for dual roads, initial minimum unit width, 10 ft., increments 9 ft.; (*c*) footpaths, minimum 4 ft. in the open country, elsewhere, 5 ft.

Radius at Bends.

The Ministry of Transport Memorandum of August 1930 advises that where conditions are favourable, a minimum radius of 1,000 ft. should be aimed at; failing which, the width of the carriageway at the bend should be increased. Normal practice in open country in the United States is to provide 5 degs. (1,146 ft. radius) or 6 degs. (955 ft.) curves. The German reporters to the Sixth International Road Congress prescribe a minimum radius of 300 metres in level and rolling country.

The Main Highways Board of New Zealand recommend, in easy country, 6, 4½ and 3 chains (100 ft.) radius, for first, second and third class roads; in difficult country, 2, 1½ and 1 chain. Additional widths, on the inside of the bend, are given, up to 3 chains, 5 ft.; 4 chains, 4 ft.; 5 chains, 3 ft.; 10 chains, 2 ft.

Superelevation at Bends.

A convenient approximate formula for balance of centrifugal and gravity accelerations, or of centrifugal force and component of weight is:

$$\frac{h}{l} = \frac{v^2}{15 \times r}$$

where h is the height of tilt, l the horizontal width, in the same units; v the speed in miles per hour, r the radius in feet.

The reporters to the Sixth International Road Congress, October 1930, recommend the following superelevations: for radius 50 to 100 ft., 1 in 10; 200 to 500 ft., 1 in 12; 500 to 800 ft., 1 in 16; 800 to 1,000 ft., 1 in 24. The German reporters indicate 4 in 100 as normal, 5 in 100 on sharp curves and 6 in 100 where the curve is on a steep gradient. The French reporters give, for roads in Morocco, a limit of 1 in 10 on curves of short radius, and on mountain roads, where the radius r corresponds to superelevation 1 in 10 the superelevation of a curve of radius R is

$$d = \frac{r}{\mathrm{R} \times 10}$$

The Main Highways Board of New Zealand employ a formula:

$$\mathrm{C} = \frac{6 \times \mathrm{S}^2}{1000\ \mathrm{R}}$$

in which C is tilt in inches per foot, S, speed in miles per hour and R, radius in 100 ft. chains. It gives superelevations three-quarters of those for balance, and is a modification of h in inches $= l$ in feet $\times \frac{\mathrm{S}^2}{\mathrm{R}} \times \frac{4}{5}$ in which R is in feet.

The Board recommends tilts in inches per foot: 2 chains, 1½; 3 to 7 chains, 1; 8 to 15 chains, ¾; 16 to 40 chains, ½; high-class pavements not more than 1 in.

Visibility at Summits.

The Ministry of Transport Memorandum of August 1930 advises that curves of parabolic form shall be provided at all changes of gradient and be such as will provide a clear view of 50 yds. on each side of a summit to drivers of mutually approaching vehicles. In many States (U.S.A.) it is required that a 500-ft. line of sight must be provided 5 ft. above the crown of the road.

London Pavings: Recent Replacements.

From the Metropolitan Paving Committee's Report for 1929–1930.

Wood Blocks.—Presumably soft wood: (1) Creosoted deal on concrete, replaced by 2½ ins. hard clinker asphalt on 10 ins. reinforced concrete; cost of carpet and strength crust, 17*s.* 10*d.* per sq. yd. (2) 'Wood Blocks,' replaced by 3 × 6 × 5 ins. Guernsey granite blocks on 9 ins. of concrete; cost 40*s.* 6*d.* (3) Creosoted deal, laid 1915, replaced 1928 by Swedish granite setts, 5 × 5 × 4 ins. deep, on repaired old concrete foundation; cost, foundation 9*s.* 9*d.*, paving 25*s.* 5*d.* (4) 'Wood Blocks,' replaced by 2 ins. natural rock asphalt on new 5 ins. concrete; cost, foundation 6*s.* 10*d.*, carpet 14*s.* 5*d.* (5) 'Softwood,' laid 1899, replaced 1929 or 1928 by 2 ins. compressed natural rock asphalt on 10 ins. concrete, cost, foundation 14*s.* 8*d.*, carpet 10*s.* 6*d.*

Hardwood Blocks.—(1) Jarrah, 3 ins. sectional blocks, laid 1913, replaced 1928 by creosoted softwood on 12 ins. concrete; cost, of foundation 21*s.* 9*d.*, paving 15*s.* 6*d.* (2) 'Hardwood,' laid 1911, replaced 1928 by creosoted deal on 8 ins. reinforced concrete; cost, of foundation 15*s.* 2*d.*, of paving 11*s.* 4*d.* (3) Jarrah blocks, laid 1907–8, replaced 1929 by creosoted deal blocks on 9 ins. of concrete; cost of the paving 12*s.* 8½*d.*

Stone Setts.—Granite setts, laid 1910, replaced by creosoted deal blocks; cost of the paving 13*s.*

Tarmacadam and Tarslag Macadam.—(1) Replaced by 3 ins. Guernsey granite concrete on 6 ins. Thames ballast concrete; cost, the lower layer 3*s.* 8*d.*, the upper layer 11*s.* 10*d.* (2) Tarslag macadam, laid 1913, replaced by 2 ins. asphaltic macadam on 9 ins. reinforced concrete, on a bed of 3 ins. of hard furnace clinker. (3) Tarmacadam, replaced by creosoted deal blocks on 9 ins. concrete. (4) Tarmacadam, replaced by 2 ins. asphaltic carpet on the hardcore foundation; cost of carpet, 4*s.* 6*d.* per sq. yd., with 3 years free maintenance.

Waterbound Macadam:

Carpeted.—(1) With 2 ins. asphalt macadam; (2) with 4 ins. asphalt macadam; (3) with 3½ ins. bituminous grouted macadam; (4) with 2 ins. bituminous macadam; (5) with 2½ hot-laid asphalt; (6) and (7) with 3 ins. asphaltic concrete; (8) with 3 ins. asphaltic macadam; (9) with 3½ ins. tarred macadam.

Replaced.—(1) By a cold grouted limestone crust of 2½ ins. to ¾ in. grade material, cost, with foundation, 7*s.* 6½*d.*; (2) by 4 ins. tarslag macadam, on the old subcrust consolidated, cost, preparing foundation 3*s.* 7*d.*, surfacing 6*s.* 5*d.*, with three years free maintenance.

Haulage Costs. (See pp. 759–61.)

Per Ton-Mile.—Holland (Lincs.) 9*d.*; Worcestershire, 5·8*d.*; Essex, 6-ton steam wagons, 8¼*d.*, 5-ton steam wagons, 10¾*d.*, petrol lorries, 6-ton, 7¼*d.*, 5-ton, 9¾*d.*; Surrey, 5-ton steam wagons, 8·4*d.*, 5-ton petrol lorries, 5·36*d.* (average ton-mileages, steam 22,932, petrol 27,640); Aberdeen, average, 5·9*d.*; Bristol, steam, 5·2*d.* (average haul 5·56 miles, average load 4·76 tons), electric, 8·4*d.* (average haul 5·0 miles, average load 2·36 tons).

Per Vehicle-Day.—Surrey, steam, 3·137*l.*, petrol, 2·391*l.*; Essex, steam, 6-ton 2*l.* 19*s.* 2*d.*, 5-ton 2*l.* 13*s.* 5*d.*, petrol, 6-ton, 2*l.* 11*s.* 7*d.*, 5-ton, 1*l.* 12*s.* 8*d.* Other cases, 5 and 6-ton vehicles, 2*l.* 16*s.* to 2*l.* 18*s.*

Per Ton Hauled.—Holland (Lincs.), 2*s.* 3*d.*; Worcestershire, 2*s.* 1¼*d.*; county road work; steam wagons, usually well within the limits, 2*s.* 2*d.* to 3*s.*

Per Mile.—Surrey, steam, 6-ton, 1*s.* 9¾*d.*, 5-ton, 2*s.* 2¾*d.*; petrol, 6-ton, 1*s.* 5¾*d.*, 5-ton, 1*s.* 7¾*d.* Aberdeen, 1*s.* 5*d.*

SECTION XVI

ARRANGEMENT AND CONSTRUCTION OF ENGINEERING WORKSHOPS — WORKSHOP PRACTICE

(pp. 765–780)

(With Contribution by H. N. Allott, M.Inst.C.E.)

SECTION XVI

ARRANGEMENT AND CONSTRUCTION OF ENGINEERING WORKSHOPS — WORKSHOP PRACTICE

ENGINEERING WORKSHOPS.

(Contributed by H. N. Allott, M.Inst.C.E.)

For ordinary engineering purposes the arrangement of shops on one level is most convenient, as this allows of better supervision, better access between the various shops, and better light than is possible where the works are arranged in buildings of more than one storey. Multiple storey buildings are generally only used in cases where the work and the machinery are of a light description and where the cost of land outweighs the increased cost of buildings and the advantages of the single-storey shop.

It is advisable to arrange the position of the office buildings so that the entrance or exit to offices and works, both for persons and goods, may be controlled from one time and enquiry office. For small work where heavy cranes are not required, and where the type of machinery employed is small, the saw-tooth type of roof will be found most convenient; whilst for shops where the machines and the work are of a heavier description, and the spans consequently greater, a ridge roof is the best arrangement.

Figs. 1 and 2 show typical buildings comprising heavy machine and erecting shops, together with light machine and fitting shops.

The heights shown in the buildings are 30 feet for the erecting and heavy machine shops and 15 feet for the light machine and fitting shops. It will be noted that the roofs of all the bays are of the ridge type. The smaller bays of roofing are also of this description, since saw-tooth roofing should not generally speaking be of a greater span than 25 feet.

The widths of the bays depend on the machines employed and the character of the work to be carried out, and where these matters can be arranged to suit a bay of not more than 25 feet in width a roof of the saw-tooth type is undoubtedly the best. If the bays are of a greater width than 25 feet, the increased length of the glazing bars necessitates the use of a centre purlin and also increased strength in the rafter back.

The arrangement shown in figs. 1 and 2 is laid down on the supposition that the flow of work in the small machine bays will be parallel to the line of the larger bays, the machines and cranes where required being arranged in lines in this direction.

The roof principals are arranged at right angles to the length of the shop, so that the shafting, heating, and lighting pipes may be conveniently carried from them, and the span of the roof is accordingly 30 feet. Should, however, the character of the work to be carried out make it convenient to arrange the machines and cranes in the smaller bays at right angles to the length of the shop, the roof principals for the smaller bays will run in the contrary direction, and the span would be reduced to 25 feet, when a saw-tooth roof may be used with advantage. In tropical countries the use of saw-tooth roofing is almost compulsory as the rays of the sun are almost vertical.

Saw-tooth roofing where possible is usually arranged with the glazing facing north so as to prevent the direct glare of the sun through the glass. The principal advantage of the saw-tooth roof is that by its use a more evenly lighted and cooler shop is obtained.

The amount of glazing to be allowed for in a roof of the ridge type should be not less than about 50 per cent. of the area of covering, and in town districts with a smoky atmosphere this may be increased with advantage to 60 per cent. or 70 per cent.

The glass should be equally distributed on both sides of the roof, as by that means a more even illumination is obtained than when all the glass is placed on one side of the roof.

Another advantage of distributing the glazing is that the direct glare of the sun and consequent heating of the shops in summer is minimised.

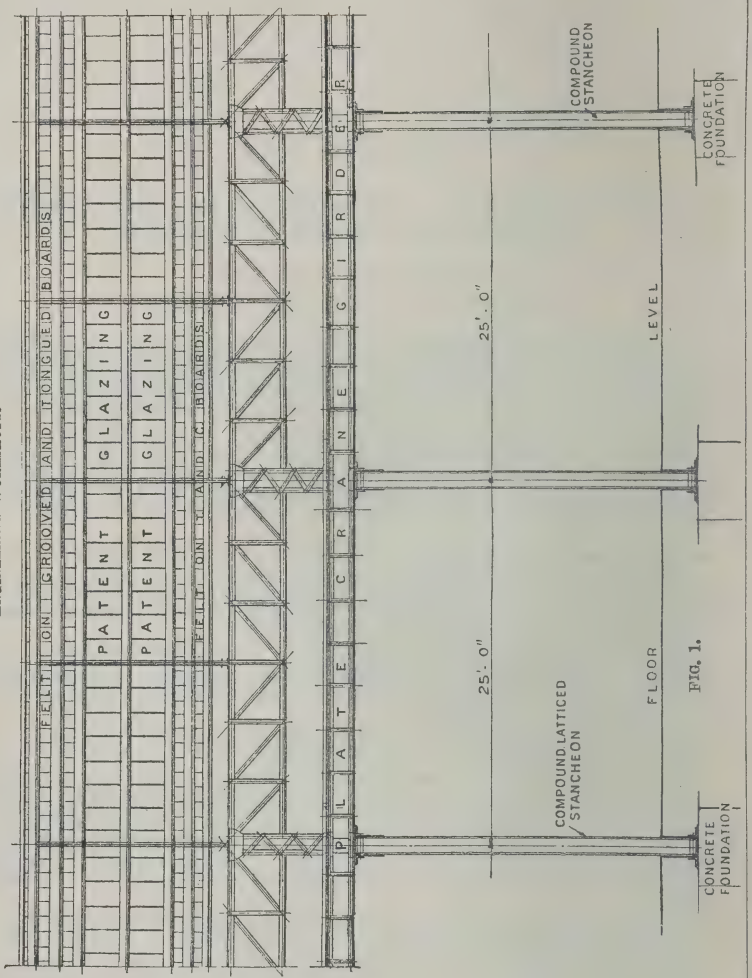

FELT JOINT GROOVED AND TONGUED BOARDS
PATENT GLAZING
PATENT GLAZING
FELT JOINT TONGUED BOARDS
PLATE LATTICE GIRDER
PLATE LATTICE GIRDER
25'. 0"
25'. 0"
COMPOUND LATTICED STANCHEON
COMPOUND STANCHEON
FLOOR LEVEL
CONCRETE FOUNDATION
CONCRETE FOUNDATION

FIG. 1.

ENGINEERING WORKSHOPS.

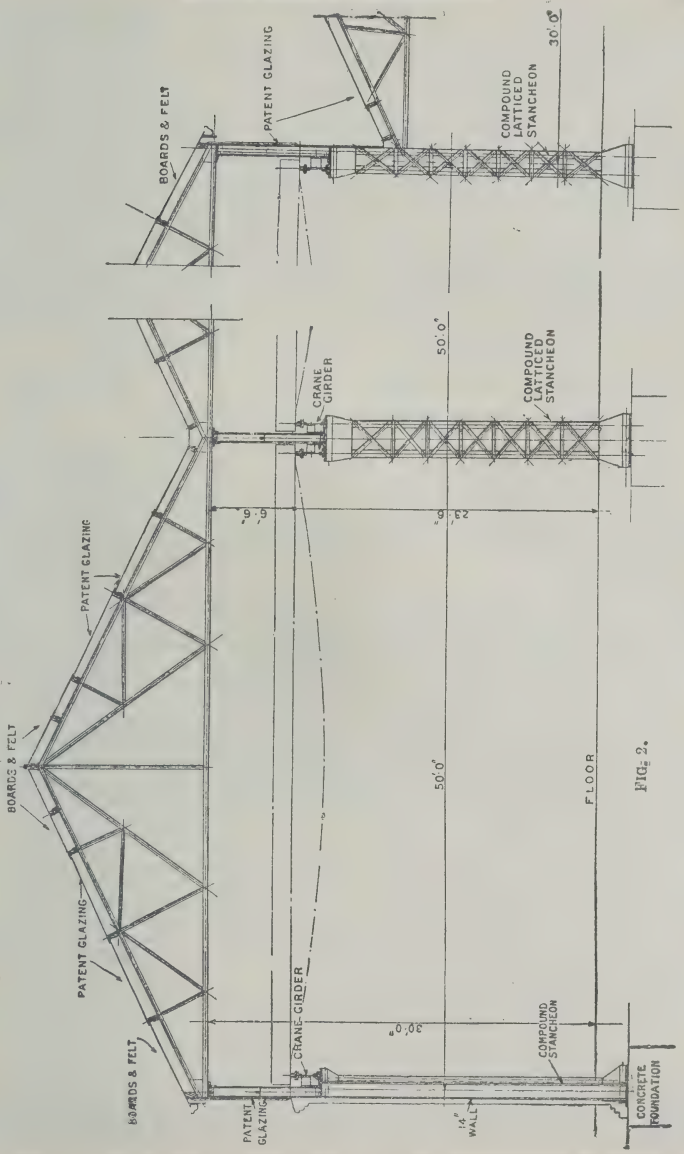


FIG. 2.

The roof covering in this country usually consists either of slates, asphalt felt, or corrugated iron.

For slates or corrugated iron the roof generally is given a rise of not less than quarter of the span, for if made of a flatter slope rain and snow are liable to blow in at the joints.

For felted roofs the roof may be practically flat if the joints are properly made with mastic, and it is only necessary to give a fall of one or two inches to the foot to allow for drainage.

Galvanised corrugated iron is usually only used in this country for buildings required for temporary purposes, and when used should be carefully painted before and after fixing, and afterwards properly maintained, as otherwise its life is only short.

Slating, when used, may be laid on battens nailed to spars, or to boarding nailed to the purlins. When laid on spars the slates should be pointed on the underside with hair mortar. When laid on boarding the boarding should be covered underneath the slates with sarking felt.

An alternative to natural slates is the use of asbestos slates. These are best laid either with the diagonal parallel to the slope of the roof, or in what is termed 'honeycomb' fashion. The point of the slate should be cut off, as it is otherwise liable to be broken by any one walking on the roof. It is necessary in all cases to see that proper lap is given to the slates, and this is especially the case where asbestos slating is used, as owing to its smooth surface the roof is more liable to leakage, where proper attention is not given to this.

An alternative to galvanised corrugated iron sheeting is the use of corrugated sheets of asbestos composition, which can be obtained up to 10 ft. in length and 27½ ins. in width.

Asphalt felt laid on thicknessed, grooved, and tongued boarding is cheaper than slating, and is not damaged like slating by men walking on the roof.

The use of felt of good quality is permitted by the building departments of most local authorities, although one or two treat it as not being sufficiently incombustible.

The felt is laid in mastic and nailed to the boarding, the joints being lapped about 4 ins. One thickness of stout felt will make a satisfactory job, if properly laid and if proper attention is given to maintenance, though the roof is generally covered with an extra thickness of a thinner quality of felt where a thoroughly good job is required. One advantage of the use of felt is that a considerable saving can be effected by forming the gutters in felt, instead of using cast iron gutters; this is especially the case in roofs where large valley gutters are required. A considerable amount of lead required for flashing at various parts of the roof can also be saved.

The only attention required to be given to felted roofs is a coat of varnish at intervals of about five years.

The roof glazing is best carried out with patent glazing, the alternatives to patent glazing being wood or tee steel glazing bars, the glass being puttied in.

The patent glazing is preferable, as with puttied bars the putty becomes defective and the glazing leaky, unless systematic attention and painting is given.

The best systems of patent glazing consist of a steel bar, to which is fixed a lead flashing, which is worked down on to the glass to form a weather-tight joint.

In most cases the glazing bar is sheathed all over in a lead casing, the ends of which are soldered up, thus obviating the necessity of painting. The bars should be galvanised before the sheathing is put on. Except in the case of vertical or steep-pitched glazing, as in the case of a saw-tooth roof, it is not advisable to joggle the glazing bar to form a lap in the glass. If this is done on a flat-pitched roof leakage is liable to occur, owing to capillary attraction between the two panes of glass. Where it is necessary to form a joint in the glazing on a roof of ordinary pitch, it is better to step the glazing bars.

The roof glazing is best carried out with wire wove glass, which holds together and does not drop if a pane is broken.

In a machine shop the number of cubic feet of air space is generally sufficiently large, in comparison with the number of men employed, to render extensive roof ventilation unnecessary, and sufficient ventilation may be provided by opening sashes in the glazing to a saw-tooth roof, or by ventilators at intervals along the ridge in the case of a roof of the ridge type.

The ventilators for a ridge roof to a machine shop are best provided with opening shutters so that the ventilation may be stopped in the winter or in driving rain or snow if required, or if preferred, exhaust ventilators can be used.

In the case of a foundry it is usual to provide ventilators with continuous louvres for the whole length of the shop. These are best made of wood with a rebated fillet screwed on the back to stop rain or snow from driving in as far as possible.

The roof principals are best formed in steel of angles, tees, and other rolled sections. A steel principal offers less obstruction to the light and harbours less dust than a timber principal, and can generally be designed at very little greater cost, if any.

For roofs of moderate span up to, say 50 feet, economical spacing may be taken as from 10 feet to 12 feet, which allows purlins of angle section to be employed. For roofs of larger spans it is frequently more economical to space the roof principals farther apart, and to use purlins of channel, zed, or rolled joist sections.

It is usual to arrange the roof so that the purlins are carried at the panel points of the truss, *i.e.*, at the intersection points of the rafter with the other members. If this is not done the strength of the rafter back must be increased to provide for the bending moment due to the load coming on to it from the purlin, in addition to the compression due to the internal stresses in the framing of the roof principal itself.

A factor of safety of 4, which is equivalent to a tensile stress of about 7 tons per square inch, is commonly used for buildings of a permanent description. In the case of buildings where a long life is not required this figure may be increased to 9 tons per square inch.

The stanchions and girders carrying the roof and gantries are now commonly made of steel of various sections in place of cast iron as in the older shops. A steel stanchion can usually be designed more economically than one of cast iron, is not so liable to damage from blows, and attachments can more easily be arranged for girders to carry shafting, jib cranes, etc.

Round cast-iron columns are sometimes used to carry the roofing of small spans for light machine shops, and light foundries where cranes are not required.

In cases where the crane loads and lengths of the stanchions permit, cheap and convenient forms are provided by the use of rolled steel joists of suitable section, with flat bars or steel channels riveted on to give additional flange area (figs. 3, 4, and 5). The section formed of a rolled steel joist and two channels is a very convenient one, and allows of the easy attachment of brackets, etc., in both directions.

FIG. 3.

FIG. 4.

FIG. 5.

Types of Compound Stanchions to carry Light Cranes or for Multiple Storey Buildings.

Where the shafting is carried from the crane girders, as is sometimes the case, the deflection and stiffness of the girders under load must be inquired into to ensure that the proper running of the shafting is not interfered with.

The economical spacing of the stanchions is determined by the total cost of the stanchions and foundations and the girders carried by them, but this must also in many cases be affected by the spacing of the machinery. In the small machine bays the tools will usually be arranged in long rows parallel to the lines of stanchions and the width of the bay will be a sum of the machine widths and alley ways. The spacing of the stanchions longitudinally will, however, not be affected by the machines, as these will usually be clear of the line of stanchions. For light work, bays arranged with the stanchions placed in squares of 20 feet will usually be suitable and economical in structural cost.

In the case of the large machine bays, the details of the machines will affect the centres of the stanchions longitudinally, as it will be frequently necessary to arrange for these to be placed partly between the stanchions. The width of the bay will be determined by the depth taken by the machines and the space required for assembling work and storing material. The spans and widths of bays shown will be generally convenient for a shop doing ordinary engineering work, but each case requires consideration and should be decided on its merits.

Provision must be made to carry the shafting on the stanchions for the larger bays, where the machines are not independently driven by motors. This is most conveniently arranged for by fixing four longitudinal channels braced together or two rolled steel joists at a convenient height.

The stanchions should be stayed transversely by knee bracing, or by lateral bracing, along the plane of the bottom member of the roof principals. In the case of very high buildings it may be necessary to use both methods.

The roof should also be braced diagonally along the planes of the rafters at the ends of the building, and in the case of very long buildings at intervals in addition.

The foundations for the stanchions and walls are best constructed of cement concrete, mixed six to one. No bedstones are necessary for the stanchions, which can be wedged up to the required level, and grouted with cement and sand in equal proportions. It is advisable to have one or two holes drilled in the stanchion base to vent same, and so allow of the escape of air when grouting. The walls are best constructed of brickwork, in hydraulic lime mortar, or may be framed and constructed of reinforced concrete, about 3 ins. to 4 ins. in thickness.

By the use of special metal lathings, which can be plastered on both sides with cement mortar, it is possible to obtain a weatherproof material with a thickness of 2 ins. If this material is used, a waterproofing medium should be added, in mixing, to the mortar for the outer covering.

This latter method of construction is specially applicable to the filling in to the sides of the main bay, above the roof of the small machine shop, and to the ends of ventilators. It may be also used for a roof covering, where it is necessary that no inflammable material should be used.

Where it is desirable to erect a building with more than one floor, or where galleries are required in a machine shop, these are best constructed with a main framework of steel. The stanchions should be composed of rolled steel joists, or of compound sections, as shown in figs. 9 and

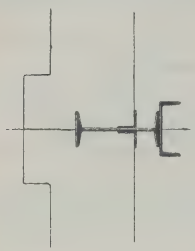


FIG. 6.

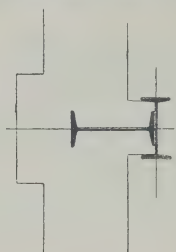


FIG. 7.

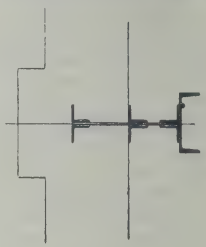


FIG. 8.

Types of Wall Stanchions to carry Cranes.

10, the main floor girders being rolled steel joists or plate girders, as required by the loads coming upon the floor. The depths of the main joists and girders should not be less, if possible, than $\frac{1}{12}$ of the span.

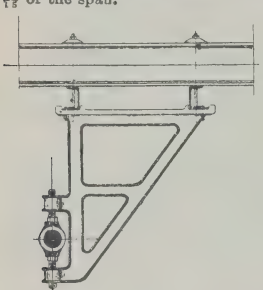


FIG. 9.

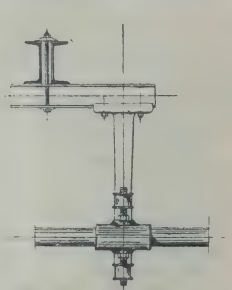


FIG. 10.

Detail Connections for carrying Shafting.

The use of a steel frame construction is especially desirable in a building of more than one storey, as the lower floors are dependent for light on the windows, which are placed in the side walls. If steel framing is used, the covering material of brickwork or concrete can be reduced to a minimum, and the window area is correspondingly large.

Floors.

Floors are now generally made of concrete, either with or without a covering of timber.

When the wearing surface is of concrete it should be finished with granite sand and cement, in equal proportions for the thickness of one inch. Concrete is not so suitable as wood for the wearing surface of a floor. It is always impossible to ensure freedom from cracks in concrete over large areas, it is tiring to walk on, is dusty, and liable to damage tools and fragile articles dropped upon it.

If a wood floor is used it is usually laid on a foundation of cement concrete, usually from 4 ins. to 6 ins. in thickness, this being a variable dimension depending on the loads to be put upon the floor, and on the nature of the subsoil.

For floors where the work is of a heavy description the best surface is one of wood blocks 4 ins. deep, which may be formed by sawing off 3-inch deals to lengths. To prevent damp from rising through the concrete the upper surface of the foundation should receive a coat of coal tar pitch about $\frac{1}{4}$ inch thick applied hot, and the blocks should be dipped in the hot coal tar and bedded whilst the pitch is hot, any open joints being afterwards run in with pitch. Where the work is of a lighter character, the most satisfactory wearing surface is grooved and tongued maple planking, nailed either to 4-inch by 3-inch battens laid in the concrete or to planks laid transversely.

The battens should be creosoted, or dipped in 'Jodelite' or 'Sideroleum,' or some similar wood preservative, and the planking laid in hot tar. It is very important that the whole surface of the floor should be thoroughly tarred before the wood blocks or planking are laid, and that any wood blocks or pegs used for levelling purposes should be removed. Where the loads are very light and the foundation good, tar concrete may be substituted for the cement concrete, the upper surface being tarred as before.

The supported floors of a multi-storey building, or of galleries, are usually constructed of concrete where a fire-resisting floor is required, or of timber in other cases. Where a concrete floor is used it may be constructed of reinforced concrete, or preferably of rolled steel joists and concrete, as the more or less concentrated loads from machines and materials coming on the floor can be more easily arranged for.

In this case the floor can be constructed of small steel joists from 4 ins. to 7 ins. deep, according to the span and load, spaced about 2 feet apart, and finished with a flat ceiling. If the concrete is arched or recessed between the joists, hangers can be easily attached by hook bolts to carry countershafting. Another method where the loads are considerable is to use joists of heavier section, and to form an arched floor of concrete between these.

The upper surface of the floor can be finished with wood planking, as previously described. If the floor is constructed of timber, wood joists bedded on the steel main girders can be used, with wood planking laid on top. It is well, if this form of floor is used, to finish the floor with boards from 1 in. to 1½ in. in thickness, nailed transversely to the planking. Both the planking and the boarding should be thicknessed and laid with grooved and tongued joints.

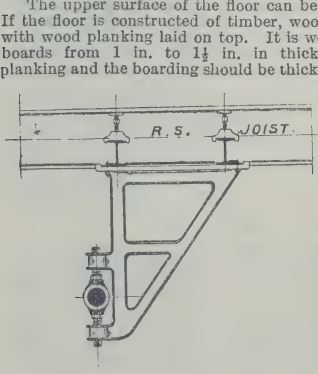
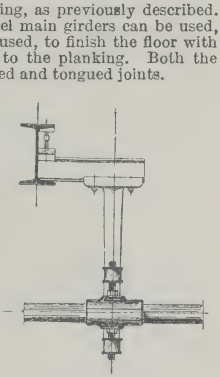


FIG. 11. Detail Connections for carrying Shafting. FIG. 12.



Window sashes are best formed of steel, which if properly painted at intervals is as durable as wood, offers less obstruction to the light, especially in windows of large size, and can be obtained at the same or less cost. The heads to windows may be brick arches, but are preferably formed of concrete, reinforced with small rolled joists or tees.

The position of the sanitary conveniences should be arranged so that undue time is not wasted by a long journey from the shop and back. The minimum accommodation is fixed by the Factory Acts, and the Bye-laws of the local authorities. A regulation of the Factory Acts which is strictly enforced by most local authorities, requires the conveniences to be cut off from the main building by a free air space, and this is undoubtedly desirable from a sanitary point of view.

The foreman of a shop should have a desk and office in that shop. The position should be arranged so as to give him a convenient oversight over the whole of the men employed.

The office may consist of a portion of the shop railed off or of a small wood building glazed all round and raised above the general floor level so as to overlook the shop.

Dry grinding machinery is always to be exhausted by a fan.

Smith work is best done in a separate building. Light smiths' fires are supplied with exhausting fans to clear the smoke.

Wood-working shops may be fitted with exhausters to remove all waste, which is best delivered to the boiler house and destroyed.

As no work can be done cheaply without thorough gauging arrangements, the modern method is to use the limit gauge system, for which purpose every works must have a full set of accurate high-class gauges to be used with special care in testing the working gauges, which may be made of cheaper material, and rejected as wear places them out of the limit of accuracy required.

Finished parts should be inspected as made, and finished machines when ready for shipment.

The tool room is an essential department in a modern workshop. In it are kept all tools and gauges required, and every tool given out for use should have a check in its place in the tool room. Fixtures, belts, and other requisites for different tools should have a place on a shop stand, which may be simply a shelf or two of plate iron, supported by pipe framing. No accumulations must ever be allowed on the floors and in corners. Conical pointed buckets provided for dirty waste cannot be left standing about, but must always be in place in their ring stands.

Provision must be made for the prompt removal of all chips, especially if greasy. Where automatic machinery is employed, oil is pumped over the work by a pump in the machine itself. In some shops oil is supplied in pipes, and returns to the pumps through pipes by way of filters. Machines very freely supplied with oil for the tools should be stood on lead-protected floors to prevent oil soakage. Greasy chips can be purified of oil in a centrifugal machine.

In a large works a tool room in each room or on each floor may be required; but no more than actually needful should be provided. Preferably rooms should be as large and as few as possible. The tool room fully carried out includes much more than mere storing and providing tools; its duty is to lay out systems of doing work, and to collaborate with the drawing office in regard to materials and stores. It is in a sense not merely a part of the works: it is also a part of the managerial department.

The drawing office also should be more than a mere room for drawing. It is, or should be, the very centre of management, for from it should emanate design, orders for materials, and the intellectual work of the whole establishment, so that design may be kept in touch with materials, market prices, etc.

Where tools are belt driven, it is usual to employ a countershaft between line shaft and tool, but if the line shaft runs at 250 r.p.m., as a modern shaft may well do, the countershaft may be omitted; there would then be a fast-and-loose pulley on the machines themselves, and no cone pulleys, their place being taken by the all-gear headstock in the lathe.

Heavy lathes may be driven from built-in motors set across the bed, and driving through chains a worm shaft with a right and left worm driving worm wheels on each side of the face plate wheel, and pinions in the worm shafts driving this wheel.

The introduction of high-speed steel, with its great capacity for removal of metal and its large absorption of power, has completely changed conditions. In the lathe the tendency is to abandon the stepped belt cone and employ a single belt pulley, only driven, it may be, directly off the line shaft, speed variations being obtained by gearing. Motors are still frequently belted to a machine, but the tendency is towards making the motor an integral part of the machine it drives. Small tools are more economically group driven from a shaft and a single motor. Whether tools shall be driven in groups or by a single motor to each tool can only definitely be decided by the actual conditions of the case. Where every machine has its own motor the aggregate power of all the motors will be two or three times that of the current generating dynamo.

The application of electrical methods has tended to advance gas and compressed air and mechanical hammers to displace steam hammers; but for larger work the steam hammer is still imperative, being superior to any other hammer. But for the very largest work of forging, the hydraulic press now does the best work, and steam hammers are not made equal to dealing with the largest work now undertaken.

In place of hooks for lifting, shop cranes are sometimes fitted with electro-magnets for lifting plates or such articles as can be thus easily held. By this means time is economised, for there is no use for slings, and the load can be put down by the lift man without help from labourers to detach, simply by the turn of a switch.

For much work that is too heavy for easy movement, the machine tools are carried to the work. In this system the floor is of heavy cast-iron girder plates grooved like the platen of a planing machine. Work is laid upon this floor, and machine tools are bolted to the floor, and operate on the work. This is rendered possible by a system of electric driving, the motors being part of the machine. Machine tools may be attached to the work, and the designer should take this into consideration in his design, since by slight modifications the work may be made so as to facilitate such fixing of the tools.

FLOOR SPACE FOR MACHINE TOOLS.

Size.	Machine.	Greatest No. of Machines per 1,000 sq. ft.
12 ins. to 20 ins.	engine lathes, faced same way	20
12 ins. to 20 ins.	engine lathes, back to back	24
22 ins. to 24 ins.	planers, heads and tails	10
22 ins. to 24 ins.	planers, faced same way	9
28 ins. to 30 ins.	planers, faced same way	5
14 ins.	multi-spindle drills	40
16 ins. to 20 ins.	upright drills	36
26 ins. to 30 ins.	upright drills	12
Small milling machines		36
Ordinary milling machines and automatic screw machines		20
Hand screw machines and plain and universal grinders		15
Polishing machines		24

The above areas are exclusive of main passage-ways, and if it is decided to run a 6 ft., 8 ft., or 10 ft. passage the entire length of building this area must be added.

Every tool in a workshop should be drawn in outline as regards its leading dimensions, and the drawing pasted on a suitable board together with the overall dimensions of the work it will take, the speeds, feeds, and other information for the drawing office.

The drawing office should be the repository of all information as to stock material; it should really include the stores and purchase department, and work in progress in the shop, and during erection should be visited by the draughtsmen chiefly concerned, so that needless costs and charges in manufacture and erection may be avoided.

Concrete Floors.

To deaden the noise from trucks rumbling over a concrete floor, use a heavy tar paper pasted to the floor by paint. The floor to be protected is first given one coating of grey cement paint. On the following day, when the paint is thoroughly dry, a second coat is applied. At the same time one side of a five-ply tar paper is painted; and when both paper and floor are still wet the paper is carefully laid, wet side down, on the floor and rolled with either a roller or wide-tired truck until all signs of air-pockets beneath the paper disappear. The surface improves with age, and very effectively reduces noise at a low cost.—*Contract Record.*

Oil stains on concrete floors may be removed by using a mixture of 1 lb. of oxalic acid in 3 gallons of water, with enough wheat flour added to make a paste that can be applied with a brush. Allow the application to remain for two days, and then remove it with clean water and a scrubbing brush. A second application may be necessary in stubborn cases.—*Building News.*

To remedy dusty concrete floors, a surface treatment of some kind should be applied to remove the surface skin which does not exceed $\frac{1}{32}$ in., or properly to treat it. One way is to grind the surface by hand or power as a terrazzo floor is polished. Other methods are to apply liquid coatings of various kinds. One of the most satisfactory treatments is with boiled linseed oil thinned with petrol to such a consistency that it will penetrate the pores of the surface. If one coat does not cure the trouble generally two will. A second method is to clean the surface thoroughly and then apply a coat of silicate of soda or water glass, 40 deg. Baumé, diluted one part to six of water, to seal the surface. Then wash off the excess which opens out the pores again and apply a second coat.

See also p. 393.

For making slippery iron floor plates reasonably safe under ordinary shop conditions, mix a small quantity of powdered sal ammoniac thoroughly with a large quantity of fine sand. Sprinkle the mixture over the floor so that all parts are covered, and damp the surface with a watering can and let it lie over a week-end, or even overnight, then sweep the sand up. The rust coating formed by the treatment will be found to last for several weeks, after which the process must be repeated.—*Ironmonger.*

Care of Factory Windows.

A number of tests of the reduction of daylight in factories has been made by the engineers of the Detroit Steel Products Company, Michigan. It was found that with vertical glass the transmission of light through the windows decreased rapidly during the first 20 days after cleaning, from a normal transmission of 80 per cent. for clean glass to 65 per cent. The decrease continued to 34 per cent. at the end of 120 days. Sloping glass collected more dirt. At slopes of 15°, 30°, and 45° the transmission of light decreased from 80 per cent. to 60, 55, and 50 per cent. respectively, in the first 20 days, and to 25, 19, and 16 per cent. at the end of 120 days. Whilst at first sloping windows on a roof admit more light than an equal area on the vertical, this advantage is lost at the end of six months. Owing to wind and rain about one-fourth of the dirt is found on the outer surface and three-fourths on the inner. Factory windows should therefore be washed more often on the inside than on the outside. The autumn, not the spring, is the right time for washing, as during the dark winter months the windows will have been recently cleaned. So far as collection of dirt is concerned there is not much difference between various types of glass, but ribbed and figured glasses are much harder to clean. The roughened side of the glass should be outside, as less dirt collects on that side.

(*W. C. Randall and A. J. Martin*, '*American Ill. Eng. Soc.*,' vol. 22, 1927, pp. 239–253.)

Electric Power Driving.

Where many small machines are placed near together and driven more or less intermittently, one small line shaft and one motor will drive all the little machines, and the motor will be of far less cost and power than the aggregate of motors if applied one to each machine. But for large machines, single motors to each machine are a decided economy.—*C. H. Benjamin.*

The Use of High-Pressure Gas in Works.

A satisfactory arrangement is to have a shop supplied with gas at 200 ins. pressure, to be reduced at the various furnaces, etc., by means of governors or valves, to the predetermined pressure corresponding with the temperature it is desired to maintain in each furnace. Pressures suitable for typical work are as follows:—For melting tin or solder a pressure of 65 ins.; for small furnaces for hardening, etc., the same pressure may be used; for blueing steel, from 65 ins. to 80 ins., the flames playing horizontally under the lime-tray set in a brick frame; for the lead-pot for tempering steel the pressure should be 150 ins.; while for melting metals at from 2000° to 2000° F. it should be 200 ins. Soldering-irons may be heated with gas at 65 ins.; two burners, consuming 6 ft. of gas per hour, and playing direct on to the irons, being sufficient to keep a workman going without delays. For lighting, gas at 65 ins. may be used.

With gas at 2*s.* per 1,000 and coal at 20*s.* per ton there is little to choose between gas and coal firing as regards cost. As regards cleanliness, uniformity, etc., the advantages are, however, in favour of gas, and in many cases with properly-designed furnaces it is greatly superior to coal, not only from the point of view of cost, but from convenience and ease of manipulation. A properly-designed gas-furnace will enable the time of preparation from the cold to be greatly shortened.—*A. W. Onslow.*

WORKSHOP PRACTICE.

Limits and Limit Gauges.

There are four usual 'fits' in practical mechanics, namely—

The *force* fit, which requires a screw, a hydraulic press, or the use of heat on one piece to get the fits together. The *drive* fit, which can be done by a hammer. The *push* fit, which can be pushed together by hand, and may even be revolved slowly, but would heat and seize if run for any time; and the *running* fit, which will run indefinitely if lubricated, and varies with the class of work. Running fits may be divided into several classes, and are classed in three divisions by the Newall Engineering Co.

The provision of a separate external limit gauge for each and every diameter used—and for each variation in the quality of fit required on such diameter—would entail in most cases an expenditure so heavy as to practically prohibit the installation of a full equipment; yet in modern commercial production efficient means of gauging work are imperative, and, to overcome the necessity for the multiplicity of external limit gauges required, an adjustable type was specially designed and introduced by the Newall Engineering Co. These being adjustable, they are not affected by wear as are all solid gauges.

The gauge has two fixed anvils on one jaw and two movable anvils (adjusting screws) on the other jaw, forming two pairs of measuring faces, the front pair being the 'go' and the back pair the 'not go' points; the adjusting screws are securely locked in position by means of cotters and nuts of special design. Each gauge, by the length of travel of its adjusting screws, covers a range of sizes, and can be easily and quickly set up to such diameter and limits within its range as may be wanted.

There are two methods of setting up or adjusting the measuring points. The first is by means of limit reference bars ground to the 'high' and 'low' limits for the different allowances prescribed in standard tables. The other method is with standard reference bars and 'setting dials'; the adjusting screws of the external limit gauges have taper holes in their inner ends, and the dials are provided with taper shanks which fit into these holes. The 'go' and 'not go' points of the gauge are set to a standard reference bar (or other reliable standard) of the nominal size required; the adjusting screws are then locked and the dial and index inserted at zero. The screw carrying the dial is released and can be adjusted to the desired allowance, either above or below 'standard,' by rotating the screw in the plus or minus direction, as shown by the arrows on the index. This screw is then locked, the dial and index transposed, and the process repeated with the other screw. For English sizes the dials are divided to read by 0·00025 in., and for metric sizes by 0·01 mm. One pair will serve for all gauges up to 6 ins., or 150 mm.; gauges over these sizes require a larger pair.

Definitions as given by the British Engineering Standards Association:—*Tolerance*—a difference in dimensions prescribed in order to tolerate unavoidable imperfections of workmanship. *Allowance*—a difference in dimensions prescribed in order to allow of various qualities of fit.

British Standard Limits and Fits for Engineering.*

(No. 164—1924.)

(*Abstract.*)

In 1906, Report No. 27 on British Standard systems for limit gauges was issued. In this Report the shaft was recommended as the constant member, but the recommendation was not generally adopted by the engineering industry. In 1917, therefore, a Committee was appointed to revise the Report, and to consider the whole question of limits for cylindrical work for the engineering industry in general. The present Report is the work of that Committee and is in substitution of Report 27—1906.

The Committee has unanimously decided to reverse the previous decision and to recommend the adoption of the hole as the constant member.

This is generally referred to as the 'hole basis' when speaking of a system of limits, and the principal table of standards will give the tolerances appropriate to holes of different sizes and different grades of workmanship, so that all holes of the same nominal size and same specified grade will be interchangeable.

* By permission of the British Engineering Standards Association.

TABLE I.—BRITISH STANDARD LIMITS AND FITS FOR ENGINEERING. UNILATERAL, BILATERAL, AND OVERSIZE HOLES. (British System—Inches.)

All Dimensions except Nominal Sizes are expressed in thousandths of an inch.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26
Size Multiplier.	Nominal Sizes.	UNILATERAL HOLES. (In which the Low Limit of the Hole is Nominal Size.)								BILATERAL HOLES. (In which the Nominal Size of the Hole lies between the High and Low Limits.)								OVERSIZE HOLES. (In which the Low Limit of the Hole is larger than the Nominal Size.)						Non-Mating Holes and Shafts.	
		B		U		V		W		K		X		Y		Z		A		G		H		J	
m	Inches.	H	L	H	L	H	L	H	L	H	L	H	L	H	L	H	L	H	L	H	L	H	L	H	L
3	0 to 0·29	+0·3	0	+0·6	0	+1·2	0	+ 2·4	0	+0·1	−0·2	+0·3	−0·3	+0·6	−0·6	+1·2	−1·2	+1·2	+0·6	+ 1·8	+1·2	+ 2·4	+ 1·8	+ 3·0	− 3·0
4	0·3 „ 0·59	+0·4	0	+0·8	0	+1·6	0	+ 3·2	0	+0·2	−0·2	+0·4	−0·4	+0·8	−0·8	+1·6	−1·6	+1·6	+0·8	+ 2·4	+1·6	+ 3·2	+ 2·4	+ 4·0	− 4·0
5	0·6 „ 0·99	+0·5	0	+1·0	0	+2·0	0	+ 4·0	0	+0·2	−0·3	+0·5	−0·5	+1·0	−1·0	+2·0	−2·0	+2·0	+1·0	+ 3·0	+2·0	+ 4·0	+ 3·0	+ 5·0	− 5·0
6	1·0 „ 1·49	+0·6	0	+1·2	0	+2·4	0	+ 4·8	0	+0·3	−0·3	+0·6	−0·6	+1·2	−1·2	+2·4	−2·4	+2·4	+1·2	+ 3·6	+2·4	+ 4·8	+ 3·6	+ 6·0	− 6·0
7	1·5 „ 2·09	+0·7	0	+1·4	0	+2·8	0	+ 5·6	0	+0·3	−0·4	+0·7	−0·7	+1·4	−1·4	+2·8	−2·8	+2·8	+1·4	+ 4·2	+2·8	+ 5·6	+ 4·2	+ 7·0	− 7·0
8	2·1 „ 2·79	+0·8	0	+1·6	0	+3·2	0	+ 6·4	0	+0·4	−0·4	+0·8	−0·8	+1·6	−1·6	+3·2	−3·2	+3·2	+1·6	+ 4·8	+3·2	+ 6·4	+ 4·8	+ 8·0	− 8·0
9	2·8 „ 3·59	+0·9	0	+1·8	0	+3·6	0	+ 7·2	0	+0·4	−0·5	+0·9	−0·9	+1·8	−1·8	+3·6	−3·6	+3·6	+1·8	+ 5·4	+3·6	+ 7·2	+ 5·4	+ 9·0	− 9·0
10	3·6 „ 4·49	+1·0	0	+2·0	0	+4·0	0	+ 8·0	0	+0·5	−0·5	+1·0	−1·0	+2·0	−2·0	+4·0	−4·0	+4·0	+2·0	+ 6·0	+4·0	+ 8·0	+ 6·0	+10·0	−10·0
11	4·5 „ 5·49	+1·1	0	+2·2	0	+4·4	0	+ 8·8	0	+0·5	−0·6	+1·1	−1·1	+2·2	−2·2	+4·4	−4·4	+4·4	+2·2	+ 6·6	+4·4	+ 8·8	+ 6·6	+11·0	−11·0
12	5·5 „ 6·59	+1·2	0	+2·4	0	+4·8	0	+ 9·6	0	+0·6	−0·6	+1·2	−1·2	+2·4	−2·4	+4·8	−4·8	+4·8	+2·4	+ 7·2	+4·8	+ 9·6	+ 7·2	+12·0	−12·0
13	6·6 „ 7·79	+1·3	0	+2·6	0	+5·2	0	+10·4	0	+0·6	−0·7	+1·3	−1·3	+2·6	−2·6	+5·2	−5·2	+5·2	+2·6	+ 7·8	+5·2	+10·4	+ 7·8	+13·0	−13·0
14	7·8 „ 9·09	+1·4	0	+2·8	0	+5·6	0	+11·2	0	+0·7	−0·7	+1·4	−1·4	+2·8	−2·8	+5·6	−5·6	+5·6	+2·8	+ 8·4	+5·6	+11·2	+ 8·4	+14·0	−14·0
15	9·1 „ 10·49	+1·5	0	+3·0	0	+6·0	0	+12·0	0	+0·7	−0·8	+1·5	−1·5	+3·0	−3·0	+6·0	−6·0	+6·0	+3·0	+ 9·0	+6·0	+12·0	+ 9·0	+15·0	−15·0
16	10·5 „ 11·99	+1·6	0	+3·2	0	+6·4	0	+12·8	0	+0·8	−0·8	+1·6	−1·6	+3·2	−3·2	+6·4	−6·4	+6·4	+3·2	+ 9·6	+6·4	+12·8	+ 9·6	+16·0	−16·0
17	12·0 „ 13·59	+1·7	0	+3·4	0	+6·8	0	+13·6	0	+0·8	−0·9	+1·7	−1·7	+3·4	−3·4	+6·8	−6·8	+6·8	+3·4	+10·2	+6·8	+13·6	+10·2	+17·0	−17·0
18	13·6 „ 15·29	+1·8	0	+3·6	0	+7·2	0	+14·4	0	+0·9	−0·9	+1·8	−1·8	+3·6	−3·6	+7·2	−7·2	+7·2	+3·6	+10·8	+7·2	+14·4	+10·8	+18·0	−18·0
19	15·3 „ 17·09	+1·9	0	+3·8	0	+7·6	0	+15·2	0	+0·9	−1·0	+1·9	−1·9	+3·8	−3·8	+7·6	−7·6	+7·6	+3·8	+11·4	+7·6	+15·2	+11·4	+19·0	−19·0
20	17·1 „ 18·99	+2·0	0	+4·0	0	+8·0	0	+16·0	0	+1·0	−1·0	+2·0	−2·0	+4·0	−4·0	+8·0	−8·0	+8·0	+4·0	+12·0	+8·0	+16·0	+12·0	+20·0	−20·0
21	19·0 „ 20·99	+2·1	0	+4·2	0	+8·4	0	+16·8	0	+1·0	−1·1	+2·1	−2·1	+4·2	−4·2	+8·4	−8·4	+8·4	+4·2	+12·6	+8·4	+16·8	+12·6	+21·0	−21·0
22	21·0 „ 23·09	+2·2	0	+4·4	0	+8·8	0	+17·6	0	+1·1	−1·1	+2·2	−2·2	+4·4	−4·4	+8·8	−8·8	+8·8	+4·4	+13·2	+8·8	+17·6	+13·2	+22·0	−22·0
23	23·1 „ 25·29	+2·3	0	+4·6	0	+9·2	0	+18·4	0	+1·1	−1·2	+2·3	−2·3	+4·6	−4·6	+9·2	−9·2	+9·2	+4·6	+13·8	+9·2	+18·4	+13·8	+23·0	−23·0
RANGE FACTOR, r		+·1	0	+·2	0	+·4	0	+·8	0	+·05	−·05	+·1	−·1	+·2	−·2	+·4	−·4	+·4	+·2	+·6	+·4	+·8	+·6	+1·0	−1·0

H = High Limit of Tolerance; L = Low Limit of Tolerance; the sign + or − below the symbols H and L indicates whether the limits are positive or negative.

TABLE II.—BRITISH STANDARD LIMITS AND FITS FOR ENGINEERING. STANDARD SHAFTS.
For use with either Unilateral, Bilateral or Oversize Holes. (British System—Inches.)
All Dimensions except Nominal Sizes are expressed in thousandths of an inch.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
Size Multiplier.	Nominal Sizes.	F		E		D		C		B		K		L		P		M		Q		R		S		T		TT	
m	Inches.	H	L	H	L	H	L	H	L	H	L	H	L	H	L	H	L	H	L	H	L	H	L	H	L	H	L	H	L
3	0 to 0·29	+1·2	+0·9	+0·9	+0·6	+0·6	+0·3	+0·4	+0·1	+0·3	0	+0·1	−0·2	0	−0·3	−0·2	−0·5	−0·3	−0·6	−0·5	−0·9	−0·9	−1·5	−1·5	−2·4	−2·4	−3·6	−3·6	−6·0
4	0·3 „ 0·59	+1·6	+1·2	+1·2	+0·8	+0·8	+0·4	+0·6	+0·2	+0·4	0	+0·2	−0·2	0	−0·4	−0·2	−0·6	−0·4	−0·8	−0·6	−1·2	−1·2	−2·0	−2·0	−3·2	−3·2	−4·8	−4·8	−8·0
5	0·6 „ 0·99	+2·0	+1·5	+1·5	+1·0	+1·0	+0·5	+0·7	+0·2	+0·5	0	+0·2	−0·3	0	−0·5	−0·3	−0·8	−0·5	−1·0	−0·8	−1·5	−1·5	−2·5	−2·5	−4·0	−4·0	−6·0	−6·0	−10·0
6	1·0 „ 1·49	+2·4	+1·8	+1·8	+1·2	+1·2	+0·6	+0·9	+0·3	+0·6	0	+0·3	−0·3	0	−0·6	−0·3	−0·9	−0·6	−1·2	−0·9	−1·8	−1·8	−3·0	−3·0	−4·8	−4·8	−7·2	−7·2	−12·0
7	1·5 „ 2·09	+2·8	+2·1	+2·1	+1·4	+1·4	+0·7	+1·0	+0·3	+0·7	0	+0·3	−0·4	0	−0·7	−0·4	−1·1	−0·7	−1·4	−1·1	−2·1	−2·1	−3·5	−3·5	−5·6	−5·6	−8·4	−8·4	−14·0
8	2·1 „ 2·79	+3·2	+2·4	+2·4	+1·6	+1·6	+0·8	+1·2	+0·4	+0·8	0	+0·4	−0·4	0	−0·8	−0·4	−1·2	−0·8	−1·6	−1·2	−2·4	−2·4	−4·0	−4·0	−6·4	−6·4	−9·6	−9·6	−16·0
9	2·8 „ 3·59	+3·6	+2·7	+2·7	+1·8	+1·8	+0·9	+1·3	+0·4	+0·9	0	+0·4	−0·5	0	−0·9	−0·5	−1·4	−0·9	−1·8	−1·4	−2·7	−2·7	−4·5	−4·5	−7·2	−7·2	−10·8	−10·8	−18·0
10	3·6 „ 4·49	+4·0	+3·0	+3·0	+2·0	+2·0	+1·0	+1·5	+0·5	+1·0	0	+0·5	−0·5	0	−1·0	−0·5	−1·5	−1·0	−2·0	−1·5	−3·0	−3·0	−5·0	−5·0	−8·0	−8·0	−12·0	−12·0	−20·0
11	4·5 „ 5·49	+4·4	+3·3	+3·3	+2·2	+2·2	+1·1	+1·6	+0·5	+1·1	0	+0·5	−0·6	0	−1·1	−0·6	−1·7	−1·1	−2·2	−1·7	−3·3	−3·3	−5·5	−5·5	−8·8	−8·8	−13·2	−13·2	−22·0
12	5·5 „ 6·59	+4·8	+3·6	+3·6	+2·4	+2·4	+1·2	+1·8	+0·6	+1·2	0	+0·6	−0·6	0	−1·2	−0·6	−1·8	−1·2	−2·4	−1·8	−3·6	−3·6	−6·0	−6·0	−9·6	−9·6	−14·4	−14·4	−24·0
13	6·6 „ 7·79	+5·2	+3·9	+3·9	+2·6	+2·6	+1·3	+1·9	+0·6	+1·3	0	+0·6	−0·7	0	−1·3	−0·7	−2·0	−1·3	−2·6	−2·0	−3·9	−3·9	−6·5	−6·5	−10·4	−10·4	−15·6	−15·6	−26·0
14	7·8 „ 9·09	+5·6	+4·2	+4·2	+2·8	+2·8	+1·4	+2·1	+0·7	+1·4	0	+0·7	−0·7	0	−1·4	−0·7	−2·1	−1·4	−2·8	−2·1	−4·2	−4·2	−7·0	−7·0	−11·2	−11·2	−16·8	−16·8	−28·0
15	9·1 „ 10·49	+6·0	+4·5	+4·5	+3·0	+3·0	+1·5	+2·2	+0·7	+1·5	0	+0·7	−0·8	0	−1·5	−0·8	−2·3	−1·5	−3·0	−2·3	−4·5	−4·5	−7·5	−7·5	−12·0	−12·0	−18·0	−18·0	−30·0
16	10·5 „ 11·99	+6·4	+4·8	+4·8	+3·2	+3·2	+1·6	+2·4	+0·8	+1·6	0	+0·8	−0·8	0	−1·6	−0·8	−2·4	−1·6	−3·2	−2·4	−4·8	−4·8	−8·0	−8·0	−12·8	−12·8	−19·2	−19·2	−32·0
17	12·0 „ 13·59	+6·8	+5·1	+5·1	+3·4	+3·4	+1·7	+2·5	+0·8	+1·7	0	+0·8	−0·9	0	−1·7	−0·9	−2·6	−1·7	−3·4	−2·6	−5·1	−5·1	−8·5	−8·5	−13·6	−13·6	−20·4	−20·4	−34·0
18	13·6 „ 15·29	+7·2	+5·4	+5·4	+3·6	+3·6	+1·8	+2·7	+0·9	+1·8	0	+0·9	−0·9	0	−1·8	−0·9	−2·7	−1·8	−3·6	−2·7	−5·4	−5·4	−9·0	−9·0	−14·4	−14·4	−21·6	−21·6	−36·0
19	15·3 „ 17·09	+7·6	+5·7	+5·7	+3·8	+3·8	+1·9	+2·8	+0·9	+1·9	0	+0·9	−1·0	0	−1·9	−1·0	−2·9	−1·9	−3·8	−2·9	−5·7	−5·7	−9·5	−9·5	−15·2	−15·2	−22·8	−22·8	−38·0
20	17·1 „ 18·99	+8·0	+6·0	+6·0	+4·0	+4·0	+2·0	+3·0	+1·0	+2·0	0	+1·0	−1·0	0	−2·0	−1·0	−3·0	−2·0	−4·0	−3·0	−6·0	−6·0	−10·0	−10·0	−16·0	−16·0	−24·0	−24·0	−40·0
21	19·0 „ 20·99	+8·4	+6·3	+6·3	+4·2	+4·2	+2·1	+3·1	+1·0	+2·1	0	+1·0	−1·1	0	−2·1	−1·1	−3·2	−2·1	−4·2	−3·2	−6·3	−6·3	−10·5	−10·5	−16·8	−16·8	−25·2	−25·2	−42·0
22	21·0 „ 23·09	+8·8	+6·6	+6·6	+4·4	+4·4	+2·2	+3·3	+1·1	+2·2	0	+1·1	−1·1	0	−2·2	−1·1	−3·3	−2·2	−4·4	−3·3	−6·6	−6·6	−11·0	−11·0	−17·6	−17·6	−26·4	−26·4	−44·0
23	23·1 „ 25·29	+9·2	+6·9	+6·9	+4·6	+4·6	+2·3	+3·4	+1·1	+2·3	0	+1·1	−1·2	0	−2·3	−1·2	−3·5	−2·3	−4·6	−3·5	−6·9	−6·9	−11·5	−11·5	−18·4	−18·4	−27·6	−27·6	−46·0
RANGE FACTOR, r		+·4	+·3	+·3	+·2	+·2	+·1	+·15	+·05	+·1	0	+·05	−·05	0	−·1	−·05	−·15	−·1	−·2	−·15	−·3	−·3	−·5	−·5	−·8	−·8	−1·2	−1·2	−2·0

H = High Limit of Tolerance; L = Low Limit of Tolerance; The sign + or − below the Symbols H and L indicates whether the limits are positive or negative.

TABLE III.—BRITISH STANDARD LIMITS AND FITS FOR ENGINEERING.

FITS OF STANDARD SHAFTS IN "U" AND "X" HOLES. (British System—Inches.)

All Dimensions except Nominal Sizes are expressed in thousandths of an inch.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
Size Multiplier.	Nominal Sizes.	STANDARD SHAFTS IN UNILATERAL "U" HOLES.														STANDARD SHAFTS IN BILATERAL "X" HOLES.													
m	Inches.	UF	UE	UD	UC	UB	UK	UL	UP	UM	UQ	UR	US	UT	UTT	XF	XE	XD	XC	XB	XK	XL	XP	XM	XQ	XR	XS	XT	XTT
3	0 to 0·29	−1·2	−0·9	−0·6	−0·4	−0·3	−0·1	0	+0·2	+0·3	+ 0·5	+ 0·9	+ 1·5	+ 2·4	+ 3·6	−1·5	−1·2	−0·9	−0·7	−0·6	−0·4	−0·3	−0·1	0	+0·2	+ 0·6	+ 1·2	+ 2·1	+ 3·3
		−0·3	0	+0·3	+0·5	+0·6	+0·8	+0·9	+1·1	+1·2	+ 1·5	+ 2·1	+ 3·0	+ 4·2	+ 5·6	−0·6	−0·3	0	+0·2	+0·3	+0·5	+0·6	+0·8	+0·9	+1·2	+ 1·8	+ 2·7	+ 3·9	+ 5·3
4	0·3 „ 0·59	−1·6	−1·2	−0·8	−0·6	−0·4	−0·2	0	+0·2	+0·4	+ 0·6	+ 1·2	+ 2·0	+ 3·2	+ 4·8	−2·0	−1·6	−1·2	−1·0	−0·8	−0·6	−0·4	−0·2	0	+0·2	+ 0·8	+ 1·6	+ 2·8	+ 4·4
		−0·4	0	+0·4	+0·6	+0·8	+1·0	+1·2	+1·4	+1·6	+ 2·0	+ 2·8	+ 4·0	+ 5·6	+ 8·0	−0·8	−0·4	0	+0·2	+0·4	+0·6	+0·8	+1·0	+1·2	+1·6	+ 2·4	+ 3·6	+ 5·2	+ 8·4
5	0·6 „ 0·99	−2·0	−1·5	−1·0	−0·7	−0·5	−0·2	0	+0·3	+0·5	+ 0·8	+ 1·5	+ 2·5	+ 4·0	+ 6·0	−2·5	−2·0	−1·5	−1·2	−1·0	−0·7	−0·5	−0·2	0	+0·3	+ 1·0	+ 2·0	+ 3·5	+ 5·5
		−0·5	0	+0·5	+0·8	+1·0	+1·3	+1·5	+1·8	+2·0	+ 2·5	+ 3·5	+ 5·0	+ 7·0	+11·0	−1·0	−0·5	0	+0·3	+0·5	+0·8	+1·0	+1·3	+1·5	+2·0	+ 3·0	+ 4·5	+ 6·5	+10·5
6	1·0 „ 1·49	−2·4	−1·8	−1·2	−0·9	−0·6	−0·3	0	+0·3	+0·6	+ 0·9	+ 1·8	+ 3·0	+ 4·8	+ 7·2	−3·0	−2·4	−1·8	−1·5	−1·2	−0·9	−0·6	−0·3	0	+0·3	+ 1·2	+ 2·4	+ 4·2	+ 6·6
		−0·6	0	+0·6	+0·9	+1·2	+1·5	+1·8	+2·1	+2·4	+ 3·0	+ 4·2	+ 6·0	+ 8·4	+13·2	−1·2	−0·6	0	+0·3	+0·6	+0·9	+1·2	+1·5	+1·8	+2·4	+ 3·6	+ 5·4	+ 7·8	+12·6
7	1·5 „ 2·09	−2·8	−2·1	−1·4	−1·0	−0·7	−0·3	0	+0·4	+0·7	+ 1·1	+ 2·1	+ 3·5	+ 5·6	+ 8·4	−3·5	−2·8	−2·1	−1·7	−1·4	−1·0	−0·7	−0·3	0	+0·4	+ 1·4	+ 2·8	+ 4·9	+ 7·7
		−0·7	0	+0·7	+1·1	+1·4	+1·8	+2·1	+2·5	+2·8	+ 3·5	+ 4·9	+ 7·0	+ 9·8	+15·4	−1·4	−0·7	0	+0·4	+0·7	+1·1	+1·4	+1·8	+2·1	+2·8	+ 4·2	+ 6·3	+ 9·1	+14·7
8	2·1 „ 2·79	−3·2	−2·4	−1·6	−1·2	−0·8	−0·4	0	+0·4	+0·8	+ 1·2	+ 2·4	+ 4·0	+ 6·4	+ 9·6	−4·0	−3·2	−2·4	−2·0	−1·6	−1·2	−0·8	−0·4	0	+0·4	+ 1·6	+ 3·2	+ 5·6	+ 8·8
		−0·8	0	+0·8	+1·2	+1·6	+2·0	+2·4	+2·8	+3·2	+ 4·0	+ 5·6	+ 8·0	+11·2	+17·6	−1·6	−0·8	0	+0·4	+0·8	+1·2	+1·6	+2·0	+2·4	+3·2	+ 4·8	+ 7·2	+10·4	+16·8
9	2·8 „ 3·59	−3·6	−2·7	−1·8	−1·3	−0·9	−0·4	0	+0·5	+0·9	+ 1·4	+ 2·7	+ 4·5	+ 7·2	+ 10·8	−4·5	−3·6	−2·7	−2·2	−1·8	−1·3	−0·9	−0·4	0	+0·5	+ 1·8	+ 3·6	+ 6·3	+ 9·9
		−0·9	0	+0·9	+1·4	+1·8	+2·3	+2·7	+3·2	+3·6	+ 4·5	+ 6·3	+ 9·0	+12·6	+19·8	−1·8	−0·9	0	+0·5	+0·9	+1·4	+1·8	+2·3	+2·7	+3·6	+ 5·4	+ 8·1	+11·7	+18·9
10	3·6 „ 4·49	−4·0	−3·0	−2·0	−1·5	−1·0	−0·5	0	+0·5	+1·0	+ 1·5	+ 3·0	+ 5·0	+ 8·0	+ 12·0	−5·0	−4·0	−3·0	−2·5	−2·0	−1·5	−1·0	−0·5	0	+0·5	+ 2·0	+ 4·0	+ 7·0	+ 11·0
		−1·0	0	+1·0	+1·5	+2·0	+2·5	+3·0	+3·5	+4·0	+ 5·0	+ 7·0	+10·0	+14·0	+22·0	−2·0	−1·0	0	+0·5	+1·0	+1·5	+2·0	+2·5	+3·0	+4·0	+ 6·0	+ 9·0	+13·0	+ 21·0
11	4·5 „ 5·49	−4·4	−3·3	−2·2	−1·6	−1·1	−0·5	0	+0·6	+1·1	+ 1·7	+ 3·3	+ 5·5	+ 8·8	+ 13·2	−5·5	−4·4	−3·3	−2·7	−2·2	−1·6	−1·1	−0·5	0	+0·6	+ 2·2	+ 4·4	+ 7·7	+ 12·1
		−1·1	0	+1·1	+1·7	+2·2	+2·8	+3·3	+3·9	+4·4	+ 5·5	+ 7·7	+11·0	+15·4	+ 24·2	−2·2	−1·1	0	+0·6	+1·1	+1·7	+2·2	+2·8	+3·3	+4·4	+ 6·6	+ 9·9	+14·3	+ 23·1
12	5·5 „ 6·50	−4·8	−3·6	−2·4	−1·8	−1·2	−0·6	0	+0·6	+1·2	+ 1·8	+ 3·6	+ 6·0	+ 9·6	+ 14·4	−6·0	−4·8	−3·6	−3·0	−2·4	−1·8	−1·2	−0·6	0	+0·6	+ 2·4	+ 4·8	+ 8·4	+ 13·2
		−1·2	0	+1·2	+1·8	+2·4	+3·0	+3·6	+4·2	+4·8	+ 6·0	+ 8·4	+12·0	+16·8	+ 26·4	−2·4	−1·2	0	+0·6	+1·2	+1·8	+2·4	+3·0	+3·6	+4·8	+ 7·2	+10·8	+15·6	+ 25·2
13	6·5 „ 7·79	−5·2	−3·9	−2·6	−1·9	−1·3	−0·6	0	+0·7	+1·3	+ 2·0	+ 3·9	+ 6·5	+ 10·4	+ 15·6	−6·5	−5·2	−3·9	−3·2	−2·6	−1·9	−1·3	−0·6	0	+0·7	+ 2·6	+ 5·2	+ 9·1	+ 14·3
		−1·3	0	+1·3	+2·0	+2·6	+3·3	+3·9	+4·6	+5·2	+ 6·5	+ 9·1	+13·0	+18·2	+ 28·6	−2·6	−1·3	0	+0·7	+1·3	+2·0	+2·6	+3·3	+3·9	+5·2	+ 7·8	+11·7	+16·9	+ 27·3
14	7·8 „ 9·09	−5·6	−4·2	−2·8	−2·1	−1·4	−0·7	0	+0·7	+1·4	+ 2·1	+ 4·2	+ 7·0	+ 11·2	+ 16·8	−7·0	−5·6	−4·2	−3·5	−2·8	−2·1	−1·4	−0·7	0	+0·7	+ 2·8	+ 5·6	+ 9·8	+ 15·4
		−1·4	0	+1·4	+2·1	+2·8	+3·5	+4·2	+4·9	+5·6	+ 7·0	+ 9·8	+14·0	+19·6	+ 30·8	−2·8	−1·4	0	+0·7	+1·4	+2·1	+2·8	+3·5	+4·2	+5·6	+ 8·4	+12·6	+18·2	+ 29·4
15	9·1 „ 10·49	−6·0	−4·5	−3·0	−2·2	−1·5	−0·7	0	+0·8	+1·5	+ 2·3	+ 4·5	+ 7·5	+ 12·0	+ 18·0	−7·5	−6·0	−4·5	−3·7	−3·0	−2·2	−1·5	−0·7	0	+0·8	+ 3·0	+ 6·0	+ 10·5	+ 16·5
		−1·5	0	+1·5	+2·3	+3·0	+3·8	+4·5	+5·3	+6·0	+ 7·5	+ 10·5	+ 15·0	+ 21·0	+ 33·0	−3·0	−1·5	0	+0·8	+1·5	+2·3	+3·0	+3·8	+4·5	+6·0	+ 9·0	+ 13·5	+ 19·5	+ 31·5
16	10·5 „ 11·99	−6·4	−4·8	−3·2	−2·4	−1·6	−0·8	0	+0·8	+1·6	+ 2·4	+ 4·8	+ 8·0	+ 12·8	+ 19·2	−8·0	−6·4	−4·8	−4·0	−3·2	−2·4	−1·6	−0·8	0	+0·8	+ 3·2	+ 6·4	+ 11·2	+ 17·8
		−1·6	0	+1·6	+2·4	+3·2	+4·0	+4·8	+5·6	+6·4	+ 8·0	+ 11·2	+ 16·0	+ 22·4	+ 35·2	−3·2	−1·6	0	+0·8	+1·6	+2·4	+3·2	+4·0	+4·8	+6·4	+ 9·6	+ 14·4	+ 20·8	+ 33·6
17	12·0 „ 13·59	−6·8	−5·1	−3·4	−2·5	−1·7	−0·8	0	+0·9	+1·7	+ 2·6	+ 5·1	+ 8·5	+ 13·6	+ 20·4	−8·5	−6·8	−5·1	−4·2	−3·4	−2·5	−1·7	−0·8	0	+0·9	+ 3·4	+ 6·8	+ 11·9	+ 18·7
		−1·7	0	+1·7	+2·6	+3·4	+4·3	+5·1	+6·0	+6·8	+ 8·5	+ 11·9	+ 17·0	+ 23·8	+ 37·4	−3·4	−1·7	0	+0·9	+1·7	+2·6	+3·4	+4·3	+5·1	+6·8	+ 10·2	+ 15·3	+ 22·1	+ 35·7
18	13·6 „ 15·29	−7·2	−5·4	−3·6	−2·7	−1·8	−0·9	0	+0·9	+1·8	+ 2·7	+ 5·4	+ 9·0	+ 14·4	+ 21·6	−9·0	−7·2	−5·4	−4·5	−3·6	−2·7	−1·8	−0·9	0	+0·9	+ 3·6	+ 7·2	+ 12·6	+ 19·8
		−1·8	0	+1·8	+2·7	+3·6	+4·5	+5·4	+6·3	+7·2	+ 9·0	+ 12·6	+ 18·0	+ 25·2	+ 39·6	−3·6	−1·8	0	+0·9	+1·8	+2·7	+3·6	+4·5	+5·4	+7·2	+ 10·8	+ 16·2	+ 23·4	+ 37·8
19	15·3 „ 17·09	−7·6	−5·7	−3·8	−2·8	−1·9	−0·9	0	+1·0	+1·9	+ 2·9	+ 5·7	+ 9·5	+ 15·2	+ 22·8	−9·5	−7·6	−5·7	−4·7	−3·8	−2·8	−1·9	−0·9	0	+1·0	+ 3·8	+ 7·6	+ 13·3	+ 20·9
		−1·9	0	+1·9	+2·9	+3·8	+4·8	+5·7	+6·7	+7·6	+ 9·5	+ 13·3	+ 19·0	+ 26·6	+ 41·8	−3·8	−1·9	0	+1·0	+1·9	+2·9	+3·8	+4·8	+5·7	+7·6	+ 11·4	+ 17·1	+ 24·7	+ 39·9
20	17·1 „ 18·99	−8·0	−6·0	−4·0	−3·0	−2·0	−1·0	0	+1·0	+2·0	+ 3·0	+ 6·0	+ 10·0	+ 16·0	+ 24·0	−10·0	−8·0	−6·0	−5·0	−4·0	−3·0	−2·0	−1·0	0	+1·0	+ 4·0	+ 8·0	+ 14·0	+ 22·0
		−2·0	0	+2·0	+3·0	+4·0	+5·0	+6·0	+7·0	+8·0	+ 10·0	+ 14·0	+ 20·0	+ 28·0	+ 44·0	−4·0	−2·0	0	+1·0	+2·0	+3·0	+4·0	+5·0	+6·0	+8·0	+ 12·0	+ 18·0	+ 26·0	+ 42·0
21	19·0 „ 20·99	−8·4	−6·3	−4·2	−3·1	−2·1	−1·0	0	+1·1	+2·1	+ 3·2	+ 6·3	+ 10·5	+ 16·8	+ 25·2	−10·5	−8·4	−6·3	−5·2	−4·2	−3·1	−2·1	−1·0	0	+1·1	+ 4·2	+ 8·4	+ 14·7	+ 23·1
		−2·1	0	+2·1	+3·2	+4·2	+5·3	+6·3	+7·4	+8·4	+ 10·5	+ 14·7	+ 21·0	+ 29·4	+ 46·2	−4·2	−2·1	0	+1·1	+2·1	+3·2	+4·2	+5·3	+6·3	+8·4	+ 12·6	+ 18·9	+ 27·3	+ 44·1
22	21·0 „ 23·09	−8·8	−6·6	−4·4	−3·3	−2·2	−1·1	0	+1·1	+2·2	+ 3·3	+ 6·6	+ 11·0	+ 17·6	+ 26·4	−11·0	−8·8	−6·6	−5·5	−4·4	−3·3	−2·2	−1·1	0	+1·1	+ 4·4	+ 8·8	+ 15·4	+ 24·2
		−2·2	0	+2·2	+3·3	+4·4	+5·5	+6·6	+7·7	+8·8	+ 11·0	+ 15·4	+ 22·0	+ 30·8	+ 48·4	−4·4	−2·2	0	+1·1	+2·2	+3·3	+4·4	+5·5	+6·6	+8·8	+ 13·2	+ 19·8	+ 28·6	+ 46·2
23	23·1 „ 25·29	−9·2	−6·9	−4·6	−3·4	−2·3	−1·1	0	+1·2	+2·3	+ 3·5	+ 6·9	+ 11·5	+ 18·4	+ 27·6	−11·5	−9·2	−6·9	−5·7	−4·6	−3·4	−2·3	−1·1	0	+1·2	+ 4·6	+ 9·2	+ 16·1	+ 25·3
		−2·3	0	+2·3	+3·5	+4·6	+5·8	+6·9	+8·1	+9·2	+ 11·5	+ 16·1	+ 23·0	+ 32·2	+ 50·6	−4·6	−2·3	0	+1·2	+2·3	+3·5	+4·6	+5·8	+6·9	+9·2	+ 13·8	+ 20·7	+ 29·9	+ 48·3

+ = Clearance; − = Interference; the upper figures in each square represent the Minimum, the lower figures the Maximum, Limit of Fit.

Two systems are in common use. In the unilateral system the tolerance is in one direction only from the nominal size and is usually positive: that is to say, every hole in the unilateral system is of the nominal size or larger. In the bilateral system the tolerance extends in both directions (but not always in equal amounts) from the nominal size: that is to say, holes in the bilateral system may be smaller or larger than the nominal size.

The Committee has unanimously agreed to recommend the use of the unilateral system as applied to cylindrical mating surfaces in cases where it does not conflict with predominating present practice.

In deciding on the system of gauging and the amount of the tolerances to be employed, it is important to distinguish between mating surfaces, in which the inter-relation between the surfaces in contact is the guiding feature, and non-mating surfaces in which only one surface has to be considered. The tables in the present report refer primarily to mating surfaces.

2. Table I gives both unilateral and bilateral limits of tolerance for holes. In the unilateral system, the nominal size is the low limit of the hole; in the bilateral system, the nominal size lies between the high and low limits of the hole.

3. Table II gives the recommended limits of tolerance for a standard series of graduated shafts suitable for pairing with either the unilateral or bilateral holes. Table III shows the numerical values of the limits of fit resulting from the assembly of any one of the standard shafts in a particular unilateral hole and a particular bilateral hole. The numerical values of the limits of fit resulting from the assembly of the standard shafts in any of the other holes specified in the tables can be arrived at by simple arithmetical calculation.

4. The effect of transferring any standard shaft from a bilateral hole to a unilateral hole tends either to increase the clearance or decrease the interference—i.e. it makes the fit easier.

6. Table I provides four standard grades of workmanship for holes, in which B (unilateral) and K (bilateral) holes represent the most accurate grade, while U (unilateral) and X (bilateral) holes represent those most commonly employed. Unilateral holes V, W, and bilateral holes Y, Z, have larger tolerances.

7. Table I also provides for the use of three holes A, G, H, each with two positive limits of tolerance. These are called oversize holes. They are common to both the unilateral and bilateral systems, and are included in the table to meet exceptional conditions.

The limit values in the tables are found by multiplying together the '*size multiplier*' '*m*' and the '*range factor*' '*r*,' odd half-ten-thousandths being eliminated by subtracting such odd amounts from the products in which they occur, negative products thus becoming more negative.

Example:

To find the limits for a $2\frac{1}{4}$ in. U hole.

The value of '*m*' corresponding with $2\frac{1}{4}$ ins. is 8; see Table I.

The value of '*r*' for the High Limit of U = 2 (in ten-thousandths of an inch[2]).

The value of '*r*' for the Low Limit of U = 0.

Hence, required limits are + 0·0016 in. and + 0.

Tables with metric values are also given in the Report.

Limit Arms for Micrometers.

A ready means of converting an ordinary micrometer to a limit gauge having a tolerance of from 0·0001 to 0·022 in. with variations of 0·0001 in. is provided by a pair of arms (fig. 13) manufactured by Messrs. Accles & Pollock, Oldbury, Birmingham. The micrometer having first been set to the low limit required, the pair of arms are cramped compass-friction-tight on to the micrometer barrel, the short arm just clearing the figures at the bevelled end. The short arm is now adjusted from its cramp end, so that it strikes the back of the frame at the

low limit. The screws are then tightened. The long arm is similarly adjusted so that it strikes the front of the frame at the high limit. When the change to new limits is greater than one

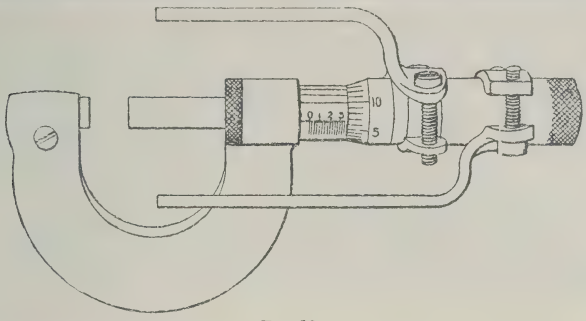


FIG. 13.

revolution of the thimble, the screws are slackened, and both arms brought to the top; the procedure is then as before. The ordinary use of the micrometer is not interfered with.
(' *Machinery.*')

INDICATOR FOR WIRE GAUGE.

A simply made indicator to enclose the notch of a wire gauge that is in use, and save time and trouble when making repeated gaugings, or searching amongst stock or scrap, can be made as

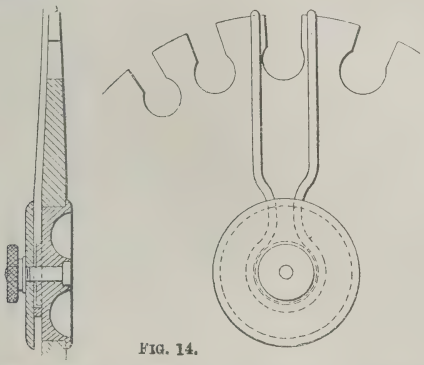


FIG. 14.

shown by fig. 14. The principle is to use springy wire, which may be altered to come just outside the edges of each opening. After holding it at this setting, clamping is done by tightening the knurled nut, which draws the large washer down on the wire and binds it. (' *Machinery.*')

Cost of Unnecessary Accuracy.

Very small limits are often indicated on drawings when it is not only extremely difficult to machine to them, but also when such accuracy is not only unnecessary but causes time and money to be wasted in the drawing office as well as by the man in the shop.

The following examples emphasize the need for sound judgment in deciding when close limits are and are not required.

A firm manufacturing small tools received an inquiry for a metal slitting cutter of the circular disc type. Tolerances were specified to $\pm$ 0·00005 in. The tool manufacturer quoted 20*l.* a piece for these cutters. On the customer replying that the price was excessive, the manufacturer asked him to specify the purpose for which the tools were to be used, and requested that he be

permitted to use his own judgment as to the accuracy required. The result was that perfectly satisfactory cutters were made with tolerances of ± 0·00025 in., which reduced the cost per cutter to 15*s.*

In another case a blue-print was submitted to a manufacturer calling for a screw over 20 ft. long, 3½ ins. in diameter, having four Acme threads to the inch. The tolerance on the pitch diameter and lead per inch was given as ± 0·0001 in. A quotation of over 2,000*l.* was made for the screw. It was estimated that, on account of the size of the screw and the difficulty of handling it, the setting up for making dependable measurements of a pitch diameter and lead error to within the limits required, would have cost about 4*l.* per measurement. In this case there was absolutely no need for accuracy. If there had been a $\frac{1}{10}$-in. tolerance on the diameter of the screw either way it would have made no difference in its application because the nut was to be fitted to the screw by the manufacturer who made the screw, and so long as the two fitted together, there was no question of interchangeability, because only one screw and nut were to be made. Finally, more reasonable tolerances were set on this screw, and it was made for 63*l.* instead of thirty or forty times that amount.

Frequently draughtsmen dimension fillets in decimals, indicating that great accuracy is required. The tool maker generally knows that no accuracy is really wanted, but if he were to follow the drawing exactly he would make the tools with such unnecessary accuracy as to increase their cost several times. If a fillet is marked $\frac{3}{16}$, it is clear that no great accuracy is required, but when it is dimensioned 0·1875 it would appear that an accuracy to the fourth decimal place is wanted.

(*'Machinery,'* August 19, 1926.)

Estimating Cost of Steel Bars.

Locate on axis OA (fig. 15) the position which represents the price in shillings per cwt. at which the particular steel can be bought. From that point travel to the right, horizontally, until the diagonal which represents the diameter of bar required is reached. From that point pass vertically down to axis OB and read off the price per foot for bar of that diameter in the material required.

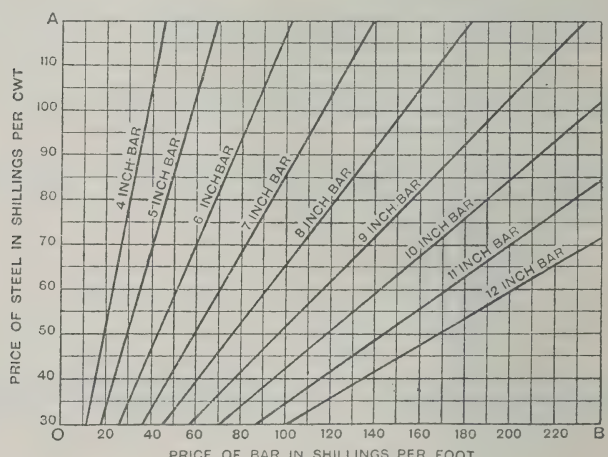


FIG. 15.

The chart covers steels from common mild steel to high grade nickel-chrome steels, and it can be used for determining the value of forgings if the price of plain forgings per cwt. is known.

(*Howard Ensaw,* '*Machinery.*')

SECTION XVII

FLOUR MILLING—COTTON MILLS—OIL MILLS—CANE-SUGAR MILLS AND MACHINERY

(pp. 783-797)

SECTION XVII

FLOUR MILLING — COTTON MILLS — OIL MILLS — CANE-SUGAR MILLS AND MACHINERY.

FLOUR MILLING.

(Contributed by E. R. & F. Turner, Ltd., Engineers, Ipswich.)

Fifty years ago the reducing of wheat to flour was done almost entirely by means of stones; now it is done by chilled iron rolls, except in a few countries where stones are still retained.

Stones are still used in England and elsewhere for making 'stone' flour, whole meal, farmers' grist, etc. They are usually from 36 to 48 ins. in diameter, top stone the runner, driven by gearing from a horizontal layshaft; speed about 150 revs. per minute; horse-power required from 8 to 12 b.h.p., but largely dependent on the condition of the grain and the sharpness of the dress in the stones.

ROLLER SYSTEM.

The very white, high-class flour of the present day is obtained by means of a very elaborate system. Every mill of any size is fitted with:—

1. Intake plant. 2. Wheat-cleaning plant. 3. Milling plant.

The intake plant for getting the wheat from trucks or barges into the mill may be either pneumatic (suction) or elevator, and is arranged so that the wheat may be deposited into any one of the storage bins or silos provided for the purpose.

The cleaning plant consists of scourers, aspirators, brush machines, etc., and also includes a plant for washing and drying such wheats as require this treatment.

The milling plant consists of roller mills, purifiers, and dressing machines connected by spouts, elevators, and conveyors. The actual roll contact surface is usually about 100 ins. per sack of 280 lbs. of flour per hour, which is divided up into approximately 40 ins. on the break rolls and 60 on the reductions. The break rolls are fluted from about 12 to 26 per inch, for opening the wheat berry, releasing the kernel and cleaning the bran, which is separated by means of sieves; the reduction rolls are smooth or very finely fluted or scratched, and reduce the kernel into fine flour, which passes into sifting machines and is dressed through fine silk. Sifting machines may be either of the reel type with internal beaters, commonly called 'centrifugals,' or the flat sieve, called 'plansifters,' both types being covered with silk containing from 100 to 150 threads per lineal inch for dressing out the finest flours, or coarser if required.

The semolina and middlings released by the break rolls are passed through purifiers in which strong currents of air pass through them, eliminating fine branny particles and other impurities before they are passed on to the reduction rolls.

Exhaust fans are connected to all machines for drawing away the hot air which is generated by the rolls and other frictional surfaces.

One sack of flour (280 lbs.) is made from 400 lbs. of wheat.

Angles of spouts for wheat should be not less than 25 degrees, and for flour and bran not less than 55 degrees.

Speed of elevators for wheat 300 to 350 feet per minute.
" " flour 180 feet per minute.
" " other mill stock 220 feet per minute.
Speed of spiral conveyors 150 feet per minute.
" band conveyors for wheat 9 feet per second.
" " " light stock 3 feet per second.
" break rolls 900 feet per minute.
" reduction rolls 630 feet per minute.

Power required about 10 to 12 b.h.p. per sack of flour per hour.

See also 'Modern Flour Milling Machinery,' by Lt.-Col. F. W. Turner (Paper read at Summer Meeting of the Institution of Mechanical Engineers, at Ipswich, June 15th, 1926). *Proceedings of the Institution of Mechanical Engineers*, No. 4, 1926, and *Engineering*, July 2nd, 1926.

COTTON MILLS.

(*BLACKBURN DISTRICT. YATES & THOM, ENGINEERS.*)

For 90,000 mule spindles and preparation.
Main building, 281 feet by 130 feet; 4 floors; 3 spinning rooms; 1 card room.
74 self-acting mules. Indicated horse-power, 1,380. Boiler-pressure, 160 lbs.
Engines.—Triple-expansion double-tandem condensing; one high-pressure cylinder, one intermediate cylinder, two low-pressure cylinders :—

Diameter of high-pressure cylinder, 21 inches; ratio, 1·0; stroke, 5 feet 6 inches.
Diameter of intermediate-pressure cylinder, 34½ inches; ratio, 2·7; speed, 60 per minute.
Diameter of each low-pressure cylinder, 39 inches; ratio, 6·9; piston speed, 660 feet per minute.

Rope pulley 30 feet diameter, grooved for 32 ropes 1⅝ inch diameter.
Velocity of ropes, 5,654 feet per minute.
Boilers.—Steel. 4 Lancashire type (one being spare); length, 30 feet; diameter, 8 feet. Furnaces, diameter 40 inches; working pressure, 160 lbs.
Economiser.—Green's type, from 278 to 384 pipes.
Mill Shafting.—Card shaft, 185 revolutions; 3⅜ inches to 2½ inches diameter.
9 drawing frames, 9 slubbing frames, 16 intermediate frames, and 44 roving frames of 176 spindles.
Roving shaft, 270 revolutions, 3½ inches to 2½ inches diameter.
Drawing, intermediate, and slubbing shaft, 225 revolutions, 3⅝ inches to 2½ inches diameter; drives also 27 cards.
Spinning-room shafts, 4¼ inches to 2¾ inches diameter.

For 71,000 ring spindles and preparation.
Boilers.—Four Lancashire, 30 feet by 8 feet, 160 lbs. per square inch, with Green's economiser of 384 pipes.
Engine.—Horizontal cross compound, 1,700 i.h.p. at 66 r.p.m.
Cylinders, 30 inches and 60 inches diameter by 5 feet stroke.
Jet condenser 50 cubic feet capacity. Two air pumps, each 31 inches diameter by 17-inch stroke at 66 r.p.m.
Flywheel, 24 feet diameter, 57 tons, 43 ropes 1⅜ inch diameter.
Rope-speed, 4,970 feet per minute; i.h.p. per rope, 39·5.
Piston speed, 660 feet per minute; air-pump speed, 187 feet per minute.
Cylinder ratio, nominal, 4 to 1; m.e.p., 30 lbs. per square inch.
Machinery.—Includes 15 opening and scutching machines, 132 cards, 18 drawing frames, 11 slubbing frames, 16 winding frames, 22 intermediate frames, 48 roving frames, 164 ring frames of 432 spindles each, 22 beaming machines, 6 warping mills, 100 reels, and 4 bundling presses.

The following are plants completed at Royton, Rochdale, Leigh, and Atherton, by Messrs. Yates & Thom, just before the war :—

ROYTON.

For 117,584 mule spindles, 10's to 24's counts.
Horizontal cross compound engine cylinders, 28 inches and 58 inches diameter, 5 feet stroke, 1,600 i.h.p.; 65 revolutions per minute.
Four boilers, 30 feet long, 8 feet 6 inches diameter, 180 lbs. pressure.
Economiser, with 480 pipes.
Fly rope pulley, 24 feet diameter; 40 1⅝-inch ropes.

For 132,000 mule spindles, 10's to 42's counts.
Horizontal cross compound engine cylinders, 28 inches and 58 inches diameter, 5 feet stroke, 1,600 i.h.p. ; 65 revolutions per minute.
Four boilers, 30 feet long, 8 feet 6 inches diameter, 180 lbs. pressure.
Superheaters.
Fly rope pulley, 24 feet diameter ; 40 1⅝-inch ropes.

ROCHDALE.

For 41,000 mule and ring spindles, 3's to 12's counts.
Horizontal cross compound engine cylinders, 19 inches and 40 inches diameter, 4 feet stroke, 600 i.h.p. ; 70 revolutions per minute.
Two boilers, 32 feet long, 8 feet 6 inches diameter, 180 lbs. pressure.
Economiser, with 192 pipes.
Fly rope pulley, 22 feet diameter ; 21 1⅝-inch ropes

LEIGH.

For 94,560 mule spindles, 40's to 80's counts.
Horizontal cross compound engine cylinders, 23 inches and 50 inches diameter, 5 feet stroke, 1,200 i.h.p. ; 65 revolutions per minute.
Three boilers, 30 feet long, 8 feet 6 inches diameter, 180 lbs. pressure.
Economiser, with 288 pipes.
Fly rope pulley, 24 feet diameter ; 33 1⅝-inch ropes.

For 85,000 mule spindles, 40's to 90's counts.
Horizontal cross compound engine cylinders, 23 inches to 50 inches diameter, 5 feet stroke, 1,200 i.h.p. ; 65 revolutions per minute.
Three boilers, 30 feet long, 9 feet diameter, 180 lbs. pressure.
Economiser, with 288 pipes.
Superheaters.
Fly rope pulley, 24 feet diameter ; 33 1⅝-inch ropes.

ATHERTON.

For 100,000 mule spindles, 60's to 90's counts.
Horizontal cross compound engine cylinders, 28 inches and 56 inches diameter, 5 feet stroke, 1,500 i.h.p. ; 66 revolutions per minute.
Four boilers, 30 feet long, 8 feet diameter ; 165 lbs. pressure.
Economiser, 384 pipes.
Fly rope pulley, 24 feet diameter ; 37 1¾-inch ropes.

(*STOCKPORT DISTRICT. J. & E. WOOD, ENGINEERS, BOLTON.*)

For 53,000 ring spindles and preparation.
Main building, 249 feet by 95 feet ; 5 floors and basement.
140 ring spinning frames of 376 to 396 spindles each, spinning 16's to 40's yarn.
Indicated horse-power, 1,500. Boiler-pressure, 165 lbs.
Engines.—Triple-expansion condensing double-tandem type : 1 high-pressure cylinder, 1 intermediate-pressure cylinder, 2 low-pressure cylinders :—

Diameter of high-pressure cylinder, 23 inches ; ratio, 1·00 ; stroke, 5 feet.
Diameter of intermediate-pressure cylinder, 37 inches ; ratio, 2·60 ; speed, 60 revolutions.
Diameter of low-pressure cylinder, 42 inches ; ratio, 6·66 ; piston speed, 600 feet.

Rope pulley 30 feet diameter, grooved for 52 ropes 1⅜ inch diameter.
Rope velocity, 5,655 feet per minute.
Boilers.—Steel. 4 Lancashire type, 30 feet by 8 feet (one being spare). Pressure, 165 lbs. Furnaces, 39 inches diameter.
Economiser.—Green's type, 448 pipes in 7 groups of 64 pipes each. Feed temperature, 75° F. Delivery to boilers, 320° F.
Chimney.—210 feet high, 8 feet diameter inside at top.
Machinery.—96 cards (flat-revolving type), 50-inch cylinder by 45 inches wide ; 4 drawing frames, each 3 heads, 4 deliveries ; 12 drawing frames, each 2 heads, 8 deliveries ; 12 slubbing frames, 104 spindles each ; 20 intermediate frames, 140 spindles each ; 52 roving frames, 180 spindles each ; 140 spinning frames, 376 to 396 spindles each.

(*HEYWOOD DISTRICT.*)

Engines.—The triple expansion condensing engines of the Mutual Spinning Company of Heywood were tested by Mr. J. F. L. Crosland with the following results :—

High-pressure cylinder, 21 inches diameter ; ratio, 1·00 ; stroke, 6 feet.
Intermediate-pressure cylinder, 33 inches diameter ; ratio, 2·50 ; speed, 53 revolutions.
Low-pressure cylinders, 35 inches diameter ; ratio, 5·61 ; piston speed, 636 feet per minute.
Total indicated horse-power, 1,090. Boiler-pressure, 156 lbs.
Cylinder vacuum, 11·2 lbs. ; condenser vacuum, 12·2 lbs. ; injection, 101° ; hot-well, 133° ; steam (2·5 per cent. wet) per horse-power hour, 12·51 lbs. ; dry, 12·2 lbs.
Dry coal per I.H.P. hour, 1·37 lb.

Economiser.—288 pipes ; mechanical stokers.

Boilers.—Two Lancashire boilers, 30 feet by 8 feet ; 66·5 square feet grate surface. Feed to economiser, 130° ; to boilers, 304°. Heating surface : boilers, 2,016 feet ; economiser, 2,880 feet.
Coal : Bituminous slack, 5·6 per cent. ash ; evaporating from and at 212°, 10·36 lbs., of which 16½ per cent. due to feed-heater. Coal per square foot of grate, 24·16 lbs. I.H.P. of engines, shafting and belts on loose pulleys, 242.

Egyptian Yarn (Fine).

For spinning fine yarns an indicated horse-power of 1,900 is found to drive 132,000 mule spindles and other preparation machinery at the Irwell Bank Mill, near Bolton.

Weaving Shed.

(*BLACKBURN DISTRICT. YATES & THOM, ENGINEERS.*)

172 feet 6 inches by 136 feet, to contain 512 looms.

Boiler.—One, 30 feet by 8 feet, 100 lbs. steam pressure.

Economiser.—96 pipes.

Engines.—Compound side-by-side type.
High-pressure cylinder, 16 inches diameter } Stroke, 4 feet ; revolutions, 55.
Low " " 28 " " }
Fly rope pulley, 20 feet diameter ; 7 grooves for 1⅝ rope.

The following are plants completed at Great Harwood and Leigh, by Messrs. Yates & Thom just before the war.

Great Harwood.

Three weaving sheds, each containing between 800 and 1,000 looms, 36 inches to 58 inches wide.
Horizontal cross compound engine cylinders, 17 inches and 34 inches diameter ; 3 feet 6 inches stroke, 450 i.h.p. ; 75 revolutions per minute.
Economiser, 128 pipes.
One boiler, 32 feet long ; 8 feet 6 inches diameter ; 160 lbs. pressure.
Fly rope pulley, 14 feet diameter, for 14 1¾-inch ropes.

Leigh.

Two weaving sheds, each containing 900 looms, 32 inches to 57 inches wide.
Horizontal cross compound engine cylinders, 16 inches and 32 inches diameter ; 3 feet 6 inches stroke ; 400 i.h.p. ; 80 revolutions per minute.
Boiler, 30 feet long ; 8 feet 6 inches diameter ; 180 lbs. pressure.
Economiser, 96 pipes.
Fly rope pulley, 15 feet ; 11 1¾-inch ropes.

Spinning and Weaving Mill.

Bury.

26,000 mule spindles, 32,000 ring spindles, 10's to 44's counts.
750 looms.
Horizontal cross compound engine cylinders, 33 inches and 66 inches diameter ; 5 feet stroke ; 2,000 i.h.p. ; 65 revolutions per minute.
Five boilers, 30 feet long ; 8 feet diameter ; 175 lbs. pressure.
Economiser, 480 pipes.
Fly rope pulley, 24 feet diameter for 46 $1\frac{3}{4}$-inch ropes.
Superheaters.

Mule Spinning Mill.

(OLDHAM DISTRICT. POLLIT & WIGZELL, ENGINEERS, SOWERBY BRIDGE.)

Mill 5 floors and cellar, 355 feet long, 129 feet wide inside ; 107,000 spindles, counts 8's to 40's ; one-third weft, two-thirds twist ; 1,650 indicated horse-power.

Engines.—Cross compound horizontal :—
High-pressure cylinder, 26 inches diameter
Low " " 52 " " } each 5 feet stroke, 75 revolutions per minute.
Flywheel 22 feet diameter, turned for 36 ropes $1\frac{5}{8}$ diameter. Weight of wheel-rim 36 tons.
Coal used for all purposes, 70 tons in summer, 84 tons in winter, per week.

Boilers.—6 Lancashire, 5 working, 1 cleaning ; 30 feet long, 8 feet diameter ; working pressure, 160 lbs. superheated.

Economiser.—Green's, 480 pipes.

Chimney.—7 feet diameter, 180 feet high ; main flue 8 feet high, 4 feet wide.

Mill Shafting.—Card-room shafts, 220 revolutions with 18-inch pulleys for drawing, roving and intermediate frames ; 170 revolutions for cards with 16-inch pulleys. Mule-room shafts 300 revolutions with 30-inch pulleys.

In the Yorkshire district it is not unusual to carry on cotton spinning in a shed of one floor as in India.

Ring Spinning Mill.

(POLLIT & WIGZELL, ENGINEERS, SOWERBY BRIDGE.)

One-floor shed, 203 feet wide, 346 feet 6 inches long ; in bays, lengthway, 10 feet 6 inches pitch ; 36,000 spindles, counts 0's to 32's ; average indicated horse-power 995.

Engine.—Tandem horizontal compound :—
High-pressure cylinder, 21 inches diameter
Low " " 42 " " } each 5 feet stroke, 75 revolutions per minute.
Flywheel 20 feet diameter, grooved for 26 ropes $1\frac{5}{8}$ diameter. Weight of rim, 30 tons.
Coal used for all purposes, 40 tons in summer, 58 tons in winter, per week.

Boilers.—3 Lancashire, 30 feet long, 8 feet diameter ; working pressure, 160 lbs., superheated

Economiser.—Green's, 272 pipes.

Chimney.—5 feet 6 inches diameter, 150 feet high ; main flue, 7 feet 6 inches high, 3 feet 3 inches wide.

Mill Shafting.—Card shaft, 175 revolutions with 17-inch pulleys ; drawing, roving, and intermediate frame shafts, 254 revs. with 27-inch pulleys ; spinning shaft, 254 revs. with 42-inch pulleys. The last seven bays of the shed are utilised for winding, reeling, and warping.

Boilers.

A Lancashire boiler to work at 160 lbs. pressure for a modern cotton-mill will usually be 30 feet long, 8 feet 6 ins. diameter, shell and end plates of $\frac{13}{16}$-inch steel with double butt strips, which in good work may be of equal breadth and treble riveted ; or the outer strap may be narrower than the inner strap and having only the double riveting. Furnaces and flues $\frac{9}{16}$ inch plate. The whole of the steel to be of 26 to 30 tons tensile strength, and to elongate not less than 20 per cent. in a length of 10 inches. For furnaces, steel with a tensile strength up to 32 tons is sometimes allowed.

Economisers.

THE 'GREEN' ECONOMISER.

(See Section XXX, Part II.)

Main Driving Ropes.

The main driving of cotton-mills is usually by cotton ropes of $1\frac{3}{8}$ inch to $1\frac{3}{4}$ inch diameter, the present tendency being towards the smaller diameter. The following tables are deduced from the weights of pure cotton ropes made by a Lancashire firm :—

Diameter of Rope in inches	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{7}{8}$	2
Weight per foot in pounds	·18	·25	·33	·40	·50	·60	·72	·84	·98	1·125	1·3
Diameter of smallest Pulley advisable	22″	25″	30″	34″	37″	42″	45″	48′	52″	57″	66″
Total stress in Rope in pounds	110	150	195	246	300	360	430	500	600	675	780
I.H.P. at 5,000 feet velocity	9	12	16	20	25	30	36	42	49	56	64

There is no practical difference in the power of ropes between 3,600 feet and 5,600 feet velocity. In Yorkshire, leather ropes $1\frac{1}{2}$ inch square are used, and will drive 80 i.h.p. per rope.

Cotton Yarn Measurement.

Cotton yarn is measured by the hank of 840 yards of single thread, and is said to be of 40's or 60's counts, when 40 hanks or 60 hanks weigh 1 lb. A lea is one-seventh of a hank or 120 yards. A wrap reel is 54 ins. circumference, so that 80 revolutions wind one lea and the skein thus wound is what is tested for strength. The softness of yarn varies with the number of turns per inch put into it. For equal quality in different counts the turns must vary as the square root of the counts. Thus turns per inch $= c\sqrt{\text{counts}}$.

For Indian and American cotton, $c = 3{\cdot}75$ for mule twist ; $3{\cdot}25$ for mule weft ; $4{\cdot}00$ for ring twist.

For Egyptian cotton the above values of c become $3{\cdot}606$, $3{\cdot}183$, and $3{\cdot}606$ respectively. Thus 50's American ring twist requires full 28 twists per inch.

List of Machinery in a Lancashire Mill, showing Average 30's Twist on Mules or Ring-frames, with approximate production at each process, speeds, etc.

77,000 Mule spindles for 28's to 32's (average 30's) twist spun counts at 31 hanks per spindle calculated to produce 79,566 lbs. of yarn per week of $55\frac{1}{2}$ hours from American cotton, $4\frac{1}{4}$ hanks single roving, or,

52,800 Ring spindles for 28's to 32's (average 30's) twist spun counts, at 45 hanks per spindle, calculated to produce 79,200 lbs. yarn per week of $55\frac{1}{2}$ hours from American cotton, $4\frac{1}{4}$ hanks single roving.

3 Hopper bale breakers, 36 ins. wide.

3 Automatic hopper feeders, 36 ins. wide, with filling apparatus.

3 Single Crighton cylinder parts, with valves and pipes to the exhaust opener, and lap machine.

3 Exhaust openers and lap machine, 48 ins. wide, for 46-in. laps, with lattice feeder, 4 ft. ntres, and dust trunks.

7 Single-finisher scutchers and lap machines, 46 ins. wide, for 45-in. laps, feeder for 4 laps. Lap 14 ozs. per yard.

1 Roving waste-opener and feeding-machine.

Details.	Hanks Roving.	Speed.	Prod. Lbs.	Hanks per Week.
			Each.	
108 Single cards, 45″ on wire, 50″ cyl., 106 s.s. flats, $1\frac{3}{8}$″ wide	0·15	Cyl. 170 r.p.m.	774	116
27 Drawing-frames, 2 heads of 4 deliveries = 27 finishing deliveries	0·13	350 r.p.m. of $1\frac{1}{2}$″ front roller	1150	149
9 Slubbing-frames, 96 spindles each 10″ × $5\frac{1}{2}$″ bobbins × 19″ staff	0·675	600 r.p.m. of spindle	94·8	59·2
17 Intermediate-frames, 140 spindles each 10″ × $4\frac{3}{4}$″ bobbins × $26\frac{3}{8}$″ staff	1·60	750 r.p.m. of spindle	34·1	54·5
42 Roving-frames, 180 spindles each 8″ × $3\frac{5}{8}$″ bobbins × $20\frac{1}{2}$″ staff	4·25	1,100 r.p.m. of spindle	10·6	45·1
74 S.A. mules, 1,100 spindles each 54″ stretch	30's, etc.	10,000 r.p.m.	1·03	31
120 Ring-frames, 440 spindles each $2\frac{1}{2}$″ distance, $1\frac{5}{8}$″ rings, $6\frac{1}{2}$″ lift	30's	9,000 r.p.m. of spindle	1·5	45

The above includes for a loss of 1 per cent. at each process, from the card-sliver to the spindle point.

In a modern mill the handling of cotton is reduced to a minimum. The bales of raw cotton are opened and slabs of the fibre flung into the bale-breaker, which delivers either to a mixing-bin or direct to a hopper-feeder, wherein the fibre is automatically regulated to weigh a given weight per yard. Thence it passes forward and is made into laps or rolls. These are removed by hand to the finisher scutcher whence the laps are run by a vertical tramway to the carding engines by hand. The cans of sliver made by the cards, and by the drawing frames and the bobbins, filled at all subsequent operations, must be moved and removed by hand.

The most modern mills are arranged, where possible, with all the card-room or preparation machinery on one floor. It is very usual to extend the ground floor by means of a shed extension along one side of the mill to enable this to be done. Usually the carding-engines are placed under this extension.

For fine counts, 120's to 140's, such as are now much spun in Oldham from Egyptian cotton the card sliver is combed before being drawn, and the modern tendency is to use combs for even coarser yarns. Generally, owing to Indian competition, the tendency in Lancashire is all in the direction of finer counts.

Horse-Power to Drive Cotton Machinery, with Speeds for Pulley.

(Bolton figures.)

	I.H.P.	R.P.M.
Hopper bale breaker	2 to $2\frac{1}{2}$	450 to 500
Porcupine opener	2 to $2\frac{1}{2}$	800
Hopper feeder	$1\frac{1}{2}$	225
Vertical opener, single	4	1,000
„ „ double	8	1,000
Exhaust opener	2	900
Opener with hop machine	12	1,200
Single scutcher	4	1,000 to 1,200
Double „	8	1,000 to 1,200
Carding engine	$\frac{3}{4}$	160 to 180
Sliver-lap machine	$\frac{1}{2}$	200 or 500
Ribbon „	1	262
Single-nip comb	$\frac{5}{8}$	305
Double-nip „	$\frac{3}{4}$	229
12-delivery drawing frame	1	150 to 300
Slubbing frame 90 spindles	2	200 to 300
Intermediate „ 120 „	2	300 to 385
Roving frame 150 „ / Jack „ 200 „	2	310 to 455
Self-acting mules	1 h.p. per 120 to 140 spindles	
Cop winding	1 h.p. „ 300 „	120
Ring frame	1 h.p. „ 90 to 100 „	
„ doubler	1 h.p. „ 55 to 65 „	
Drum winder	1 h.p. „ 100 drums	400 to 600
80-hank reel	$\frac{1}{2}$	250

The power required tends to be more or less according to the speeds of spindles, beaters, etc. For finer counts spindles run at higher speeds, but machinery generally is lighter.

Electrical Driving of Cotton Mills.

The load factor of a cotton mill is probably 90 to 95 per cent. This renders electrical driving of doubtful economy. It cannot be economically done from an outside power station, for this would be doing well if it had a 40 per cent. load factor. A cotton mill with its own generating plant can work more cheaply than by purchased current at a price that would pay an outside power station to charge. Then the transmission line is costly.

A group of mills within a small radius such as may be found in places might perhaps be economically driven from one central power house close to all the mills. But in electrical driving the generator is more costly than the rope rim, and the motors, if on line shafts, are more costly than the driven rope pulleys. The transmission losses and the generator and motor losses appear to be greater than the loss by rope driving, and where each machine has its own motor this and the transmission system seem to cost much more than shafting and pulleys. Where the motor stands on the floor at the end of a spinning or speed frame it compels the cutting out of several spindles, reducing the output capacity several per cent., or by fully as much as it is claimed each spindle will give an increased output as compared with an ordinary mill. But so far this increased output has been only demonstrated as between modern high-speed engines with electrical driving and old slow-speed engines in some out-of-date rope-driven mill. No comparisons are put forward with modern rope-driven mills with modern high-speed engines.

There are two systems of electric driving : (*a*) with the usual line shafts, each shaft with an electric motor taking the place of the usual rope pulley ; (*b*) with motors to each machine. And there may be (*c*) a mixed system on which there are line shafts for certain classes of work and separate motors for other machines. Rapid acceleration is claimed as an advantage of electrical driving. This, of course, demands motors much larger and more powerful than are merely necessary to drive the frame at its normal speed. The same acceleration may be secured with belt driving if the belts are made sufficiently larger than they need to be for the constant normal speed. Shafting does not require very much power to drive it, and where machines such as ring frames are closely spaced under a shaft, as are ring frames, it is difficult to see much benefit from electric driving of these machines, which do not stop until the bobbins are full. But with speed frames which must stop for every broken sliver it may be convenient that each machine should have its own motor. Mules which have an enormous and rapid variation of driving torque are badly suited for separate motors, and appear better driven from a line shaft. Given sufficient belt power any machine can be accelerated as rapidly as desired, and no single machine will be felt by the main engine any more than if accelerated electrically.

The abolition of belts is an advantage of some value in the separate motor system, but so far the experience of electrical driving has not been such as to warrant other than the most careful consideration of each case on its merits, and by an engineer who is a textile man first and an electrical engineer second. The following are estimates of cost of rope and electrical driving which may help the consideration of any particular case. The estimates are based on the driving of a 110,000 spindle mill (Bolton-fine-Counts) :

(Pre-war Prices.)

(*a*) *Rope Driving.*

Capital outlay—	£	*s.*	*d.*
Engine 1,500 i.h.p. and piping in engine-house	4,765	0	0
4 30′ × 8′ × 160 lb. boilers and superheaters	2,400	0	0
All piping in boiler-house (steel and cast iron)	600	0	0
Chimney and boiler seatings	1,000	0	0
Reservoir	1,500	0	0
Engine-house	600	0	0
„ bed	500	0	0
Rope race	500	0	0
Boiler-house	400	0	0
Economiser, dampers, engine, etc.	700	0	0
Lighting generator, 75 k.w., pilot-engine, and generator Switchboard and wiring, etc., 1,200 lights Foundations, etc.	1,450	0	0
Mill gearing, including dynamo driving	2,970	0	0
Cotton driving ropes	450	0	0
15-ton hand-power engine-house crane	300	0	0
Staging and stairs in rope race	300	0	0
	£18,435	0	0

(a) Running Costs.

	£	s.	d.
Coal, 3,850 tons per annum at 12*s.* 6*d.*	2,406	0	0
Wages: engineer, 55*s.*; oiler and greaser, 25*s.*; two stokers, 25*s.*	325	0	0
Oil, waste, stores, etc.	100	0	0
Insurance: engine, £80; boilers and pipes, £33; mill gearing, £60; buildings, £20	193	0	0
Rates, 7*s.* on 5 per cent. of £18,435	323	0	0
Interest and depreciation at 12 per cent. on £18,435	2,212	0	0
	£5,559	0	0

Equal to 0·477 pence per k.w. hour.

(b) Full Electrical Installation at Mill.

Capital outlay—	£	s.	d.
1,000 k.w. Turbo-generator, condensing plant, and piping	6,600	0	0
4 30′ × 8′ × 160 lb. boilers and superheaters	2,600	0	0
Boiler-house piping (steel and cast iron)	600	0	0
Chimney and boiler seatings	1,000	0	0
Reservoir	1,500	0	0
Generator house	400	0	0
,, bed	300	0	0
Motor tower and carrying for motors	250	0	0
Motors, cables, and switchboard	4,000	0	0
Boiler-house	400	0	0
Economiser, dampers, engine, &c.	700	0	0
Pilot engine and generator switchboard, &c.	1,150	0	0
Wiring for 1,200 lights, foundations, &c.			
Mill gearing	1,830	0	0
	£21,330	0	0

(b) Running Costs.

	£	s.	d.
Coal, 4,300 tons per annum at 12*s.* 6*d.*	2,690	0	0
Wages: engineer, 55*s.*; assistant, 25*s.*; 2 stokers, 25*s.*	325	0	0
Oil, waste, and general stores	75	0	0
Insurance: generator set, £110; boilers and pipes, £33; mill gearing, £40; buildings, £15; motors, £60	258	0	0
Rates, 7*s.* on 5 per cent. of £21,330	373	0	0
Interest and depreciation on £21,330 at 12 per cent.	2,560	0	0
	£6,281	0	0

Equal to 0·534 pence per k.w. hour.

To this ought to be added a 75 k.w. combined set for lighting, as it is hardly satisfactory to run lights and power off the same 3-phase generator. The cost would be £560.

(c) Electric Driving, but with Current from a Public Supply Station delivering at 10,000 *Volts.*

Capital outlay—	£	s.	d.
Boiler and piping, pump, &c., for heating	600	0	0
Chimney and boiler seatings	250	0	0
Transformer house	300	0	0
Motor tower and carrying for motors	250	0	0
Motors, cables, and switchboard	4,000	0	0
Transformers (spare set included)	1,000	0	0
Boiler-house	150	0	0
Mill gearing	1,830	0	0
Switchboard and wiring, 1,200 lights	900	0	0
	£9,280	0	0

(c) Running Costs.

	£	s.	d.
Coal for heating, 300 tons at 12*s.* 6*d.*	188	0	0
Wages: engineer, 55*s.*	137	10	0
Oil, waste, general stores	50	0	0
Insurance: motors, £60; transformers, £15; gearing, £40; buildings, £10.	125	0	0
Interest and depreciation on £9,280 at 12 per cent.	1,112	0	0

(*c*) *Running Costs*—continued.

	£	*s.*	*d.*
Current : average load 900 k.w., including lighting, based on uniform charge of 0·5625*d.* per hour	5,906	0	0
	£7,680	10	0

Equal to 0·657 pence per k.w. hour.

From the foregoing it would appear that if current can be bought for about 0·4*d.* per unit the driving by a public supply would cost the same as by ropes.

In the foregoing estimates of running costs, the boiler cleaning charges are omitted, the extra cost of this for (*a*) and (*b*) being about balanced by the omission of the wages of the stoker of the heating boiler in case (*c*). It is at present doubtful if current can be sold at a profit for 0·4*d.* per unit by any power station. The situation is made worse by the necessity for a heating boiler and chimney, &c., for case (*c*).

On the other hand driving electrically dispenses with the rope race which divides a mill halfway across into two sections, and it sets free the area of the race for machinery and increases the production within a given total area. But the real success of the system would come where a group of mills could be driven from a joint-power plant in the middle of the group, especially if the warming could be done from the same boilers. There is very little in the claim for steadier driving, this being merely a question of flywheel energy and cyclical regularity equally obtainable with ropes as electrically. No weight must be given to comparisons with old mills driven by old slow engines. Engineers should study the Brown Boveri ring frame, for which electricity does what is otherwise impracticable. In this machine the spindle speed is varied by moving the brushes of the motor so that the tension on the yarn is kept uniform at all stages of the bobbin. This cannot be done except by electrical means.

Machine Sizes in Plan—Pulley Size and Speed.

	Ft. Ins.	Ft. Ins.	Inches.				
Opener . . .	11 0	× 6	6	12 × 5	pulley	1,200 revolutions :	shaft, 400 revolutions
Double Scutcher .	18 8	× 6	6	9 × 4	„	1,500 „	„ „ „
Double Card . .	15 0	× 6	6	15 × 3	„	150 „	„ 150 „
Single Card . .	10 0	× 6	6				
Drawing Frame .	27 0	× 5	0	14 × 3	„	250 „	„ 120 „
Slubbing Frame .	27 4½	× 4	0	12 × 3	„	410 „	„ „ „
Intermediate . .	35 6	× 3	0	11 × 3	„	460 „	„ „ „
Roving . . .	35 6	× 3	0	10 × 3	„	480 „	„ „ „
Grinder . . .	6 0	× 2	10				

PAPER MILLS.

See 'Modern Paper Mill Plant,' *Proceedings of the Institution of Mechanical Engineers*, No. 3, 1926.

OIL MILLS.

The Manufacture of Vegetable Oils.

There are two methods of extracting oil from vegetable seeds and nuts. These are by means of hydraulic presses and by chemical solvents.

Two types of hydraulic presses are used. The Anglo-American, which has corrugated plates, between which the meal is confined in cloths, the sides being open ; and the 'cage' press, the box of which is built up of narrow steel bars held in a vertical position by weldless steel hoops, spaces being left between each bar for the escape of the oil, the solid matter, in the form of a round cake, being retained in the box.

The Anglo-American system is usually employed for seeds which do not contain more than 35 per cent. of oil; as the meal is confined between open plates it is difficult to prevent its escape, and the wrappers are soon destroyed when a large percentage of oil is present. For this reason a 'cage' press should be employed for all materials rich in oil.

The 'cage' press has the further advantage that a cold-drawn oil can be produced, and also double pressing can be resorted to, the second pressing being performed in either a 'cage' press or Anglo-American press.

In the expression of oil from seeds there are three main operations : crushing, heating, and pressing.

The crushing is of great importance; the oil cells must be thoroughly crushed and burst open. Unless this is effectively done it will be impossible to express the maximum amount of oil. The crushing rolls are usually five in number, superimposed, the seed being deflected by scraper plates, so that it is made to pass between the various rolls, and is thus subjected to four crushings, each receiving a heavier pressure than the preceding one. The heating is done in a mild steel kettle having a steam jacket, a mechanical stirrer, and a steam spray to give the necessary moisture to the meal.

The preliminary treatment of the seed is the same for Anglo-American and 'cage' presses.

For Anglo-American presses the heated meal has to be moulded, compressed, and wrapped in a cloth before it can be placed in the press, the usual number of cakes in each press being sixteen. The cakes leave the press with very oily edges; these have to be trimmed off by a paring machine, the oily edges being returned to the kettle for further treatment.

With 'cage' presses the meal passes directly into the box in measured quantities, a plate dividing each charge. With some seeds flat cloths are also placed between the meal and the plate to prevent cake from adhering to the plate and to assist the flow of the oil. The cakes leave the box in a finished state.

A small self-contained Anglo-American mill with two presses made by the Mirrlees Watson Co., Ltd., has a capacity of 7½ cwts. of seed per hour, and consists of the following parts:—

1 set of crushing rolls, five high, 12 ins. diameter, 18 ins. long.
1 meal-heating kettle, 42 ins. diameter, 27 ins. deep.
1 cake-moulding machine with a steam compressor.
2 hydraulic presses with rams 16 ins. diameter, working pressure 2 tons per sq. in., each press to make sixteen cakes per pressing.
1 cake-paring machine with fine reducing apparatus.
1 set of hydraulic pressure pumps, valves, and pipes.
1 oil collecting and settling tank.
1 rotary oil pump, and elevators to raise the seed to the rolls and meal to the kettle.

The parts are carried on a foundation plate, geared together and driven by a single belt. A smaller mill is also made with one press only. For larger capacities the machines are built separately and driven by belts from a shaft carried upon girders. They are fitted up in sets of four and eight presses, with capacities of 15 cwts. and 30 cwts. of seed per hour. The rolls are 16 ins. diameter, 48 ins. long.

For the eight-press and larger mills hydraulic accumulators are fitted, working at 500 to 600 lbs. and 2 tons per sq. in. The ballast is of sand contained in a mild steel case which is guided by mild steel guides. The sizes generally used are 9 ins. and 3 ins. diameter by 8 ft. stroke for the small plant, and 14 ins. and 4 ins. diameter by 10 ft. stroke for the larger plants.

Small self-contained mills with 'cage' presses are made by the Mirrlees Watson Co., Ltd., with capacities of 1½, 3, and 6 cwts., respectively. The rolls are of the Anglo-American type; the presses work at 2 tons per sq. in. and exert a pressure of over 35 cwts. per sq. inch on the cakes.

The 1½-cwt. mill has rolls 8 ins. diameter, 8 ins. long; kettle 21 ins. diameter, 18 ins. deep; press ram 10½ ins. diameter; box 11 ins. diameter, 54 ins. deep. The 3-cwt. mill has rolls 8 ins. diameter, 12 ins. long; kettle 27 ins. diameter, 20 ins. deep; press ram 13 ins. diameter; box 14 ins. diameter, 60 ins. deep. The 6-cwt. mill has rolls 8 ins. diameter, 18 ins. long; kettle 39 ins. diameter, 21 ins. deep; two presses with rams 13 ins. diameter; boxes 14 ins. diameter, 60 ins. deep.

The mills are complete with hydraulic pumps, valves, and pipes, the machines being geared together and driven by a single belt. The plants are also provided with reducing mills, so that large nuts, copra, etc., can be broken up previous to being passed to the crushing rolls. The cakes from the first pressing can also be broken up when double pressing is resorted to. Larger mills have independently driven machines and presses with boxes up to 19 ins. diameter by 9 ft. deep.

The expressed oil should be passed through a filter press to remove the sediment and produce a bright, clear oil. The filter press is composed of a number of plates with raised pyramid-shaped projections which support the cloth placed over each plate, the plates and cloth being securely clamped together. The oil is forced through the cloth and flows past the projections to the outlets at the bottom, and is collected in a trough fixed to the frames.

The oil passes through a further process of refining to remove the fatty acids and colouring matter, and when required for edible purposes is deodorised to remove the characteristic odour of the oil.

For the solvent process the seed must first be crushed, the meal is then placed in closed vessels, and the solvent, usually benzine, allowed to percolate through it in order to dissolve out the oil content. The mixture of oil and solvent is subjected to distillation—the solvent, being condensed, is used over again, and the oil is heated and steamed to remove the final traces of the solvent.

The meal after the solvent has been drained from it is also heated and steamed to remove all traces of solvent. Large plants are made with a series of vessels so that the solvent can be passed through a number and become heavily impregnated with oil before it is passed to the distiller. The residue from the process is in the form of meal, and should not contain more than about 2 per cent. of oil.

CANE-SUGAR MACHINERY.

For large sugar works, the canes are brought from the field to the factory by rail in trucks which are weighed and unloaded mechanically on to a carrier which continuously conveys them to the

Crushing Mills.

These usually comprise a pair of deeply serrated preparing rollers followed by several sets of three-roller crushing mills in tandem, and steam driven through heavy spur gearing.

The cane in transit through these successive mills is macerated with hot water, enabling ordinarily a juice extraction of over 80 per cent. on the weight of the cane to be obtained. This crushed cane (technically termed megass or bagasse), leaving the final mill with 45 to 50 per cent. of contained moisture, is continuously fed into special boiler furnaces and utilised for steam generation, supplying frequently the entire factory requirements.

MULTIPLE CRUSHING MILLS.

(The Mirrlees Watson Co. Ltd., Glasgow.)

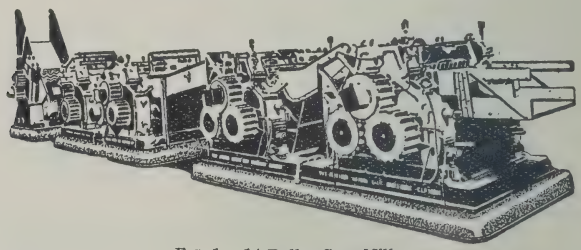


FIG. 1.—14-Roller Cane Mill.

Mill Rollers Dia. and Length.	Tons of Cane Crushed Hourly.		Engine Dia. and Stroke.	
	11-Roller Mills.	14-Roller Mills.	11-Roller.	(2 Engs.) 14-Roller.
Ins. Ins.			Ins. Ins.	Ins. Ins.
24×48	16	18	20×42	18×36
26×54	21	24	22×48	20×42
28×60	27	31	26×48	22×48
30×60	31	36	30×60	28×48
32×72	43	49	32×60	30×60
34×78	53	60	34×60	32×60
36×84	64	72	38×60	34×60

The mill capacity varies according to the class of cane crushed.

Vacuum Evaporators.

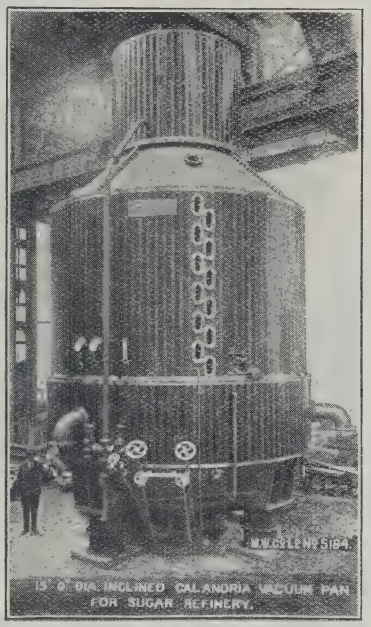
The cane juice expressed by the mills is clarified (by liming, heating, subsiding, and filtering) and then concentrated, in vacuo, to about one-fourth its original volume. The apparatus commonly used is a triple or quadruple effect, in which exhaust steam is used as the heating agent for the juice in the first vessel, whose vapour, under a reduced temperature, boils the partly concentrated in the second vessel, which again liberates vapour for boiling the still denser juice, under a still lower temperature in the succeeding vessel, which, in the case of a triple apparatus, is under a high vacuum and consequently low temperature. In a quadruple effect the principle is similar.

Mirrlees

Vacuum Pans

For the crystallization of liquids under high vacuum.

Our designs provide for rapid circulation with well distributed heating surface to ensure rapid boiling and economical steam consumption.



The MIRRLEES WATSON COMPANY LIMITED

ENGINEERS FOUNDERS

Head Office and Works: SCOTLAND ST., GLASGOW

London Office: 7 Grosvenor Gardens, S.W.1.

The Engineer.

ESTABLISHED 1856

1/- WEEKLY

The Metallurgist

IS A FREE MONTHLY SUPPLEMENT

to The Engineer.

EDITORIAL and PUBLICATION OFFICES:

28 ESSEX STREET STRAND LONDON W.C.2

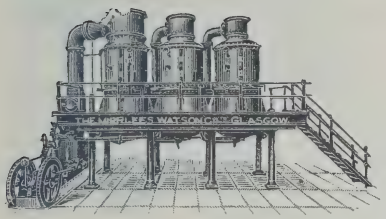
ANNUAL SUBSCRIPTION RATE £3 5 0

ECONOMY OF MULTIPLE-EFFECT EVAPORATION.

1 lb. of steam used		single	effect evaporates	$\frac{7}{8}$	lbs. water.
,,	,,	double	,, ,,	$1\frac{3}{4}$	,,
,,	,,	triple	,, ,,	$2\frac{5}{8}$	,,
,,	,,	quadruple	,, ,,	$3\frac{1}{2}$	,,

QUADRUPLE-EFFECT EVAPORATORS.

(*The Mirrlees Watson Co. Ltd.*)

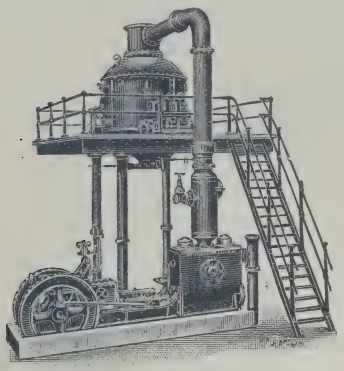


Triple-Effect Evaporator.

FIG. 2.

Diam. of Vessels.	Heating Surface.	Galls. treated per Hour from 9° to 30° Bme.
Ft. In.	Sq. Ft.	
6 0	4,440	2,960
7 0	6,248	4,170
8 0	8,168	5,450
9 0	10,500	7,000
10 0	13,120	8,750
11 0	15,800	10,530
12 0	18,800	12,530
13 0	21,940	14,610
14 0	25,270	16,830
15 0	28,860	19,220

The concentrated syrup is then granulated in vacuum pans.



Vacuum Pan and Air Pump.

FIG. 3.

VACUUM PAN DETAILS.

(*The Mirrlees Watson Co. Ltd.*)

Diam. of Pan.	Heating Surface.		Strike Capacity Gallons Massecuite.
	Coil Type.	Cal. Type.	
Ft. In.	Sq. Ft.	Sq. Ft.	
8 0	436	582	2,230
9 0	650	867	3,260
10 0	850	1,133	4,210
11 0	1,080	1,440	5,280
12 0	1,450	1,930	8,230
13 0	1,725	2,300	9,850
14 0	2,025	2,700	11,540
15 0	2,360	3,150	13,400

The resulting massecuite is discharged into crystallisers, thence to centrifugal sugar-curing machines, in which the molasses is separated from the crystals, which are thereupon dried and bagged for transport.

Vacuum Evaporators for Trade Liquids.

In the concentration of trade liquors by evaporation, it is generally important that this should be not only economically but also rapidly effected under low temperature, as many liquids exposed to heat deteriorate. Hence the extended adoption of multiple-effect evaporators on the

horizontal tube type of the Mirrlees Watson Co. Ltd., which, having a rapid flow of liquid through their evaporating tubes, meet these requirements with the minimum of deposition on the heating surfaces.

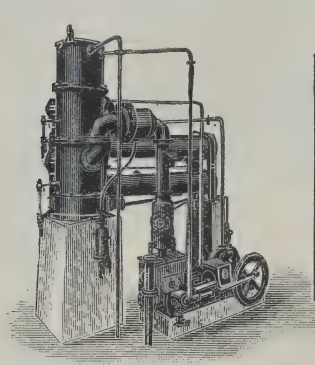


FIG. 4.—Double Effect.

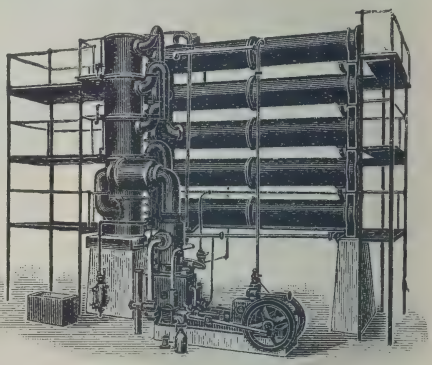


FIG. 5.—Quintuple Effect.

For bark and wood extracts, gelatine, glue, glucose, glycerine, and similar liquids of a viscid nature, a wide temperature difference between the vessels is desirable, hence the use of a double effect arrangement (fig. 4); their sizes range from 50 to 1,000 gallons hourly.

For soda ash recovery in paper mills, caustic soda, caustic potash, meat extract, distillery refuse, and other liquids readily amenable to concentration, triple, quadruple, quintuple (fig. 5), and sextuple (fig. 6) installations are in use; their sizes range from 100 to 3,000 gallons hourly.

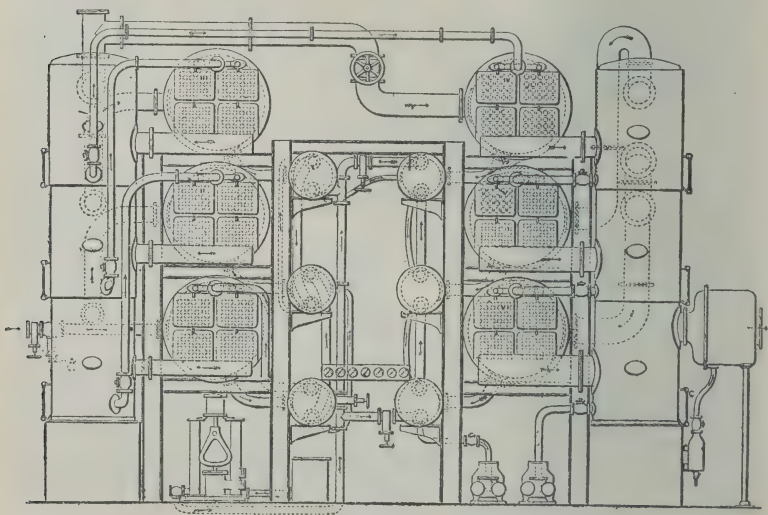


Fig. 6.—Sextuple-Effect Evaporator for Caustic Soda Lye.

Water Distillers.

To meet the demand for large quantities of distilled water at the lowest cost, the Multiple-Effect system of evaporation, extensively used in sugar manufacture, has also been applied largely, by the Mirrlees Watson Co. Ltd., to fresh-water distillers at home and abroad.

The principle on which multiple-effect evaporation is based is that the boiling point, and consequently the vapour temperature of liquids, varies with the surrounding pressure; the lower the pressure, the lower the boiling temperature; this principle is applied by working with diminishing pressures in the successive vessels of the multiple-effect evaporator.

Evaporators on the above principle are extensively used, but considerable experience is required so to proportion the various evaporating and condensing surfaces to secure the highest economy.

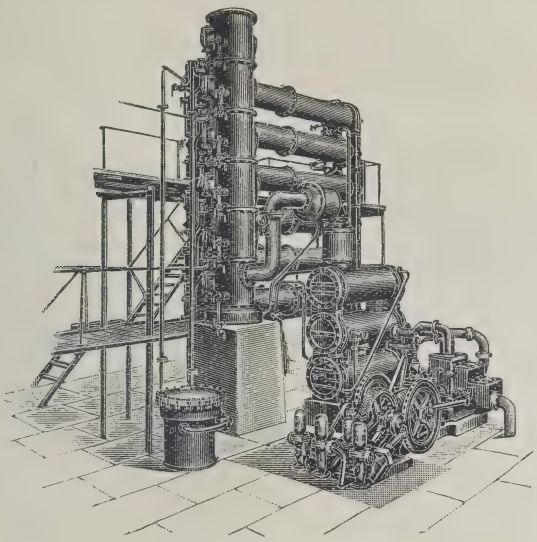


FIG. 7.—Sextuple Distiller.

Fig. 7 shows a sextuple-effect fresh-water distilling plant, which has been brought to a high state of efficiency, producing 44 lbs. of gained water (not including the drip from first vessel, which goes back direct to the boiler) per lb. of coal consumed.

Plants of this design are made to produce from 25 to 350 tons of distilled water per twenty-four hours. This type of distilling plant has been adopted by the British and other Governments.

For countries where earthquakes occur such apparatus is arranged with its evaporating vessels on a horizontal plane in place of the super-imposed type illustrated by fig. 7.

(*The Mirrlees Watson Co. Ltd.*)

NOTE ON CONDENSERS.

Great caution should be exercised in guaranteeing the efficiency of condensers which are used for other purposes than the condensation of exhaust steam from steam engines, as for example the exhausting of vacuum pans for sugar manufacture. In such cases it often happens that far more air is present than is the case with ordinary exhaust steam, and consequently the condenser may prove unequal to the task for which it was guaranteed. Those who quote prices or guarantee a particular efficiency in the case of a condenser should make it plain that the 'steam' to be condensed must only contain a certain (specific) quantity of air.

SECTION XVIII

CRANES AND CRANE APPLIANCES — ELECTRIC LIFTS — ELECTRIC WINDING ENGINES — AERIAL ROPEWAYS — CONVEYORS — MAGNETIC SEPARATORS

(pp. 801–851).

(With Contributions by John H. Huntley, M.I.E.S., and Professor John Goodman, M.Inst.C.E.)

Established since 1899. Making Lifts only.

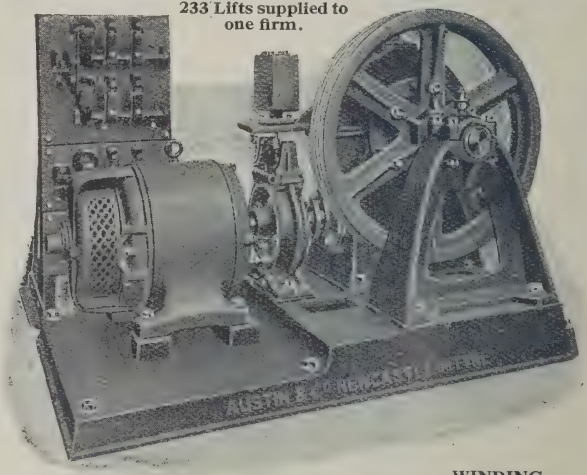
AUSTIN & CO

THE VERY HIGHEST QUALITY ALL-BRITISH

ELECTRIC LIFTS

WITH OUR PATENT "ACCRO" SELF-ADJUSTING BRAKE WHICH IS SILENT AND REQUIRES NO ATTENTION FOR YEARS.

233 Lifts supplied to one firm.



WINDING GEAR

Our Patent "Accro" Level-Stopping and Load-Indicating Invention is the latest important improvement in Lift Service. It is entirely mechanical, and of robust construction. It stops the lift smoothly and accurately level without the usual slowing-down, overtravelling and reversing. It indicates the weight of every load, guessing is unnecessary. It enables a lift to do three or four times as much work as formerly, with the same expense. Think of all the trouble and time it saves in many ways such as from fuse repairs, wear and tear.

No other Lift has these advantages

They are real money-saving inventions

DUNN STREET, NEWCASTLE-on-TYNE

TELEGRAMS: 'PERFECT' TELEPHONE: 34241

SECTION XVIII

CRANES AND CRANE APPLIANCES — ELECTRIC LIFTS — ELECTRIC WINDING ENGINES — AERIAL ROPEWAYS — CONVEYORS — MAGNETIC SEPARATORS

CRANES.

(Contributed by John H. Huntley, M.I.E.S.)

General Notes on Cranes.

The selection of a crane for any particular service is governed by several factors, such as load, space available, power at disposal of user, frequency and speed of operation, height of lift, duration of service and cost.

Cranes can be operated by manual, steam, hydraulic, pneumatic, or electric power, or by a combination of these powers.

In power stations, stores, and places where an occasional lift is all that is required, a hand crane is quite suitable, even for loads up to 20 or 30 tons, but for frequent operation, or where the load has to be moved a considerable distance, it is advisable to instal a power-driven crane, even for a load of a few hundredweights, as the saving in time and labour soon pays for the difference in initial cost.

The electric crane has the following points in its favour :—

1. Simplicity of operation.
2. No power consumed whilst crane is standing.
3. Power used varies in proportion to load.
4. Always ready for starting up.

The steam crane has the advantage that it can be used wherever a supply of fuel and water is available, but time is lost in raising steam, and fuel is consumed in keeping up steam between the lifts.

Hydraulic cranes are only economical in operation when constantly working under full load, as the water consumption is the same for light or heavy loads, although by means of double-power rams attempts have been made to overcome this defect. Trouble is caused by frost in exposed situations. Hydraulic cranes are still largely used in shipyards and boiler shops, for handling plates, portable riveters, etc., where a small lift at slow speed is required. The working hydraulic pressure is usually 1,500 lbs. per sq. in., but may be as high as 2 tons per sq. in.

Pneumatic power is applied only to small hoists and pulley blocks, the air pressure being usually about 100 lbs. per sq. in.

Petrol or Paraffin engines have been applied for driving small portable jib cranes, and experience has proved that the nominal horsepower of the engine should be at least 50 % more than the actual brake horsepower required in order to have a sufficient margin for acceleration.

Hand Cranes.

The ordinary height of handle above ground should be 3 ft.; radius of circle described by handle, not more than 18 ins.; 16 ins. is a usual length. Angle of jib, 45°.

Each man working at the handle of a crane will exert a force of 15 to 20 lbs., exclusive of friction.

If W = weight to be lifted by crane in lbs., P = force applied to handle in lbs., D = diameter described by handle in inches, n = number of revolutions of handle to one of barrel, B = diameter of barrel in inches, then

$$B = \frac{D \times P \times n}{W}. \qquad D = \frac{W \times B}{P \times n}. \qquad n = \frac{W \times B}{D \times P} \qquad P = \frac{W \times B}{n \times D} \qquad W = \frac{P \times D \times n}{B}$$

OVERHEAD TRAVELLING CRANES.

For service inside buildings or where the site conditions permit of the construction of a gantry the overhead travelling crane has proved to be the most efficient appliance. For outside work a Goliath crane is often equally serviceable, and has the advantage of a lower first cost than an overhead crane with gantry.

For hand cranes up to 40 ft. span and loads up to 5 tons, a single girder crane can be adopted, but for longer spans or heavier loads it is advisable to use the double girder type to obtain lateral stiffness.

Light cranes are usually operated from the floor level by hanging chains, but for cranes above 10 tons it is advisable to work them from the crane platform. The pull on a hand chain should not exceed 40 lbs. per man, and for crank handles not more than 25 to 30 lbs. per man. The handles should have a radius of 16 ins. to 18 ins. and the centre be at a level of from 2 ft. 6 ins. to 3 ft. above platform level.

Travelling wheels for both crab and crane are usually of cast iron, and should be fitted with ball or roller bearings.

The hoisting gear should be efficient, preferably with machine-cut teeth, pinions of mild steel, wheels of cast iron, and should be fitted with a self-sustaining brake which requires to be released by the operator before the load can be lowered, but which is inoperative in the hoisting direction. This condition can be obtained by an internal ratchet or a screw disc brake.

Worm gear should have a coarse pitch and the sustaining power of the crane should not be dependent upon the inefficiency of the gear.

OVERHEAD ELECTRIC CRANES.

For engineering shops, foundries, steelworks, etc., the overhead electric travelling crane is unequalled for general service.

The motions of hoisting, traversing and travelling are generally arranged, and a separate motor should be fitted for each motion. An auxiliary hoist having a lifting capacity of about one quarter the main hoist is desirable on cranes above 40 tons capacity, and should be operated by an independent motor.

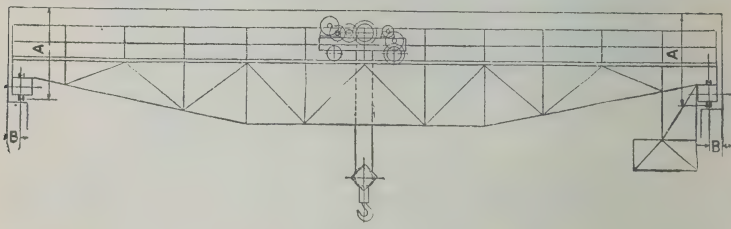


FIG. 1.

The following are particulars of standard 3-motor overhead travelling cranes (fig. 1), 30 ft. lift :—

TABLE OF STANDARD 3-MOTOR OVERHEAD ELECTRIC TRAVELLING CRANES.

Size.	Span.	Head-room. A.	End Space. B.	Wheel Base.	Max. Press. per wheel.	Wt. of Crane.	Speeds in feet per minute.			Falls of Rope.
							Hois	Traverse	Travel.	
Tons.	Ft.	Ft. Ins.	Ins.	Ft. Ins.	Tons.	Tons.				
2	30	5 0	7	6 0	2·7	5·8	20	100	300	2
2	40	5 6	7	8 6	3·0	7·3	20	100	300	2
2	50	5 6	7	8 6	3·3	8·5	20	100	300	2
5	40	6 0	8	8 6	5·3	9·8	20	90	300	4
5	50	6 0	8	8 6	5·8	11·5	20	90	300	4
5	60	6 0	8	9 6	6·1	13·0	20	90	300	4
10	40	6 6	8½	10 0	8·4	12·0	15	80	300	4
10	50	6 9	8½	10 0	9·1	14·5	15	80	300	4
10	60	6 9	8½	10 0	9·7	16·5	15	80	300	4

TABLE OF STANDARD 3-MOTOR OVERHEAD ELECTRIC TRAVELLING CRANES—*contd.*

Size.	Span.	Head-room. A.	End Space. B.	Wheel Base.	Max. Press. per wheel.	Wt. of Crane.	Speeds in feet per minute.			Falls of Rope.
Tons.	Ft.	Ft. Ins.	Ins.	Ft. Ins.	Tons.	Tons.	Hoist.	Traverse.	Travel.	
15	40	7 0	9	10 0	11·3	13·8	13	80	275	6
15	50	7 0	9	10 0	12·0	16·2	13	80	275	6
15	60	7 0	9	10 0	12·5	18·0	13	80	275	6
20	40	7 3	9½	10 6	14·5	16·5	10	80	275	8
20	50	7 3	9½	10 6	15·2	19·0	10	80	275	8
20	60	7 3	9½	10 6	15·8	21·0	10	80	275	8
25	40	7 6	10	11 6	17·5	19·0	10	80	250	8
25	50	7 6	10	11 6	18·5	22·0	10	80	250	8
25	60	7 9	10	11 6	19·2	24·2	10	80	250	8
30	40	8 0	10½	12 0	20·3	22·0	10	75	250	8
30	50	8 0	10½	12 0	21·5	25·0	10	75	250	8
30	60	8 3	10½	12 0	22·5	28·0	10	75	250	8
40	40	9 0	11	13 0	26·5	26·0	10	70	250	8
40	50	9 0	11	13 0	27·8	30·0	10	70	250	8
40	60	9 3	11	13 0	29·5	34·5	10	70	250	8
50	40	10 0	11	13 0	32·0	31·0	10	70	230	8
50	50	10 0	11	13 0	34·0	35·0	10	70	230	8
50	60	10 3	11	13 0	35·5	39·5	10	70	230	8
60	40	10 3	12	13 6	38·5	35·0	8	70	230	10
60	50	10 3	12	13 6	40·5	40·0	8	70	230	10
60	60	10 6	12	13 6	42·5	45·0	8	70	230	10

The above table is for cranes on four travelling wheels.

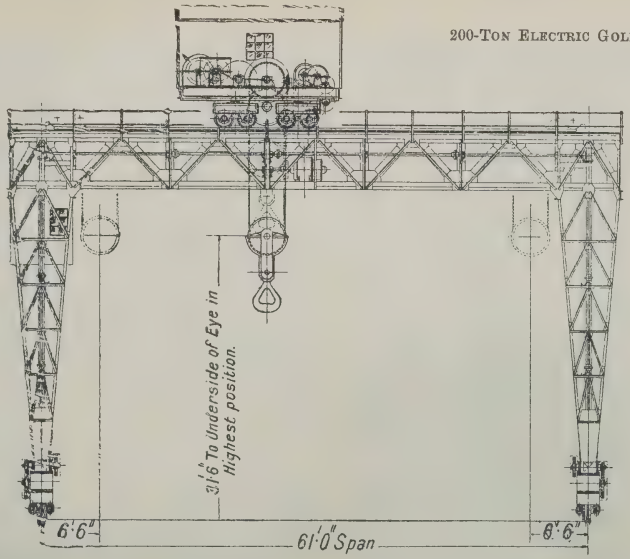
For 70-ton cranes and upwards the usual practice is to mount the crane on eight travelling wheels, and the wheel loads in the following table are based on this. The column headed 'Centres of Wheel Bogies' indicates the distance between the centre of each pair of wheels in each end carriage, and the following column gives the distance between the two wheels of each pair:—

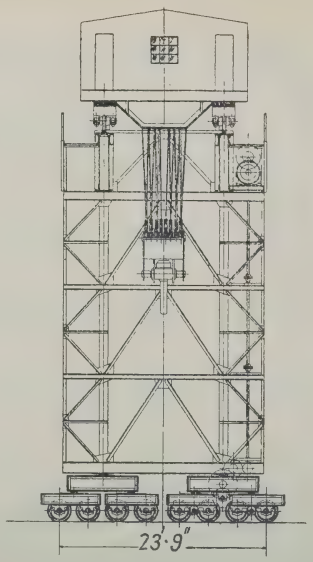
Size.	Span.	Head-room A.	End Space B.	Crs. of Wheel Bogies.	Bogie Wheel Centres.	Max. Press. per Wheel.	Wt. of Crane.	Speeds in F.P.M.			Falls of Rope.
								Hoist	Traverse.	Travel	
Tons.	Ft.	Ft. Ins.	Ft. Ins.	Ft. Ins.	Ft. Ins.	Tons.	Tons.	Ft.	Ft.	Ft.	
70	50	10 9	1 1	10 6	5 0	22·82	46	7	60	200	10
70	60	10 9	1 1	10 6	5	23·90	50½	7	60	180	10
70	70	11 0	1 1	10 6	5	24·87	56	7	60	180	10
80	50	11 0	1 2	11 0	5	25·40	51	6	60	180	12
80	60	11 0	1 2	11 0	5	26·75	56	6	60	150	12
80	70	11 3	1 2	11 0	5 0	28·00	62	6	60	150	12
100	50	11 3	1 3	11 6	5 3	32·12	62	5	60	150	12
100	60	11 3	1 3	11 6	5 3	32·62	68	5	60	150	12
100	70	11 6	1 3	11 6	5 3	35·00	75	5	60	150	12

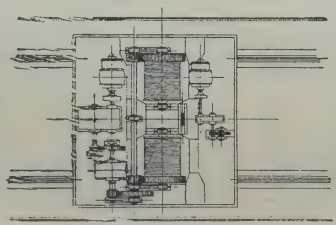
NOTE.—The headroom figures allow for about 3 ins. clearance and the end space for about 1½ in. clearance.

For short-span cranes of small size, a pair of 'H' beams make an economical set of girders, but the span should not exceed sixty times the width of the flange of one girder. For cranes up to 60 ft. span the most economical construction is the trapezoidal single web girder, with lattice auxiliary connected to the main girder by horizontal and diagonal bracing. This makes a very stiff construction, is easy to manufacture, and all surfaces are accessible for painting. The platform and travelling gear can be well supported by bearers between the main and auxiliary girders. For long spans and for outside service it is desirable to make the girder of the lattice

200-TON ELECTRIC GOLIATH CRANE.







SPEEDS AND MOTORS.

Hoisting	200	tons at	3	ft. per min. — Motor 65 B.H.P.
,,	75	,,	8	,, ,, — Change gear.
Traverse	200	,,	30	,, ,, — Motor 25 B.H.P.
Travel	200	,,	50	,, ,, — ,, 65 B.H.P.

FIG. 2.

(See page 807.)

type to reduce weight and wind surface. The depth of the girder is usually made one-twelfth the span. The maximum bending moment due to the crab, assuming each wheel equally loaded, is given by the formula :—

$$\frac{W}{28}\left(S - \frac{B}{2}\right)^2 \text{ ft. tons,}$$

where,

W = load per wheel in tons ; S = span in feet ; B = wheel base of crab in feet.

To this must be added the bending moment due to the deadweight of the girder and travelling gear. The modulus of the girder section should be such as to give a stress of between 5 and 6 tons per sq. in. The stresses in lattice girders are best determined by graphic methods.

The end carriages can be constructed of channels, and built up of plates and angles, when larger sections are required. They should be connected to the girders by large gusset plates, and fitted bolts to prevent cross winding. The wheel base should preferably be one-fifth to one-sixth of the span.

Rail wheels should be of cast steel, or of cast iron centres with steel tyres shrunk on. The width of the rolling surface of the rail is an important point, and for heavy loads a rounded surface should be avoided. The safe load on a wheel may be arrived at by the following empirical formula :—

For cast steel wheels, safe load in pounds = $800 \times W \times D$,

,, steel tyred ,, ,, ,, ,, = $1{,}200 \times W \times D$,

where,

W = width of rail face in inches ; D = dia. of wheel in inches.

Rails for heavy cranes are now rolled with a working face of 6 ins. and weight of 265 lbs. per yard.

To calculate the power required to travel a crane, a tractive force of from 60 to 70 lbs. per ton is usually taken, and for outside service the effect of wind pressure must be considered.

To prevent cross winding the cross shaft should be stiff enough to avoid undue torsional deflection when the driving wheels are unequally loaded due to the crab being at one end of the span. A foot brake should be fitted to the cross shaft on cage control cranes and a solenoid brake fitted to the motor shaft on all cranes working in the open.

The crab frame should be built up of steel sections well stiffened to form a rigid support for the motors and gearing. The traversing wheels should be of cast steel or have steel tyres, the latter only being necessary on crabs of the largest size.

It is unnecessary to fit a brake to the traversing motion unless the speed exceeds 100 ft. per minute, when a solenoid brake can be fitted.

The load is preferably lifted on wire ropes winding on to a cast iron drum rotated by gearing, the number of falls of rope depending upon the load. For cranes up to 3 tons capacity, the load may be lifted on two falls winding one fall on to the barrel. Where four or more falls of rope are used, two falls should wind on to the barrel, which should have right and left-hand spiral grooves and thus keep the load central between the girders.

The efficiency of the gearing varies from about 75 per cent. on small cranes to 65 per cent. on large cranes. A convenient rule to determine the horse-power of the hoisting motor is

$$\text{H.P.} = \frac{\text{foot-tons per minute}}{10}.$$

The speeds of working given in the table, pp. 802, 803, are suitable for general engineering shop work, but for steelworks very much higher speeds are used, and as continuous night and day service is the general practice in these works the design of the crane should be specially considered and made very robust.

All cranes are preferably fitted with two brakes, one of which should be an automatic solenoid brake; the type of the other brake will depend entirely on the service for which the crane is used.

Fig. 3 shows a small crab made by Messrs. Babcock & Wilcox. The framework consists of mild steel plates or channels connected by stretchers which support the motors and the

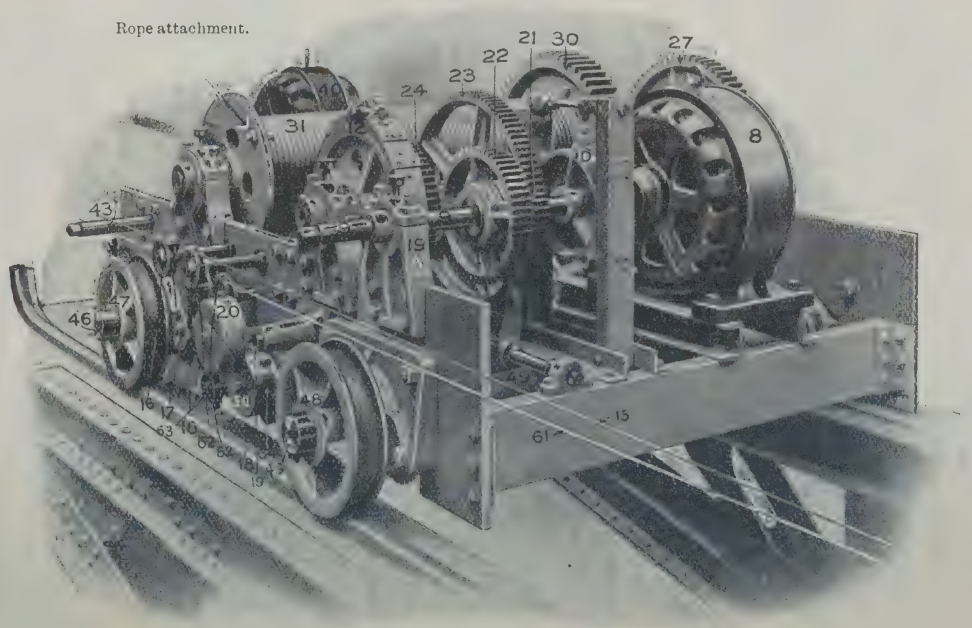


FIG. 3.—View of left-hand side of Babcock & Wilcox Crab, from hoisting motor end.

return sheaves of the hoisting ropes. The hoisting motor (8) is fitted with an extension shaft (9) on which is fixed the automatic magnetic brake (10). Two blocks press against the face of the brake pulley by means of a counterweight (50), when the crane is standing. When current is switched on, the weight is lifted by the action of the solenoid, and is automatically reapplied when current is cut off. There is also a mechanical brake (12), operated from the cage by a foot pedal attached to a lever. This lever is connected to the flexible wire (15) stretching from end to end of the cross-girders and passing over two fixed pulleys (16) and under a movable pulley (17) coupled to a counterweight (20) which serves to keep the brake strap clear of the brake-wheel. By operating the pedal the brake strap can be tightened round the brake-wheel.

The hoisting motor extension shaft (9) carries two sliding pinions, one for slow and heavy hoists (21), and the other for quick and light hoists (22), and these may be put into gear by means of a lever operated by hand from the driver's cage when the crab is opposite to it. These pinions gear with spur wheels (23) and (24) on the first motion shaft which carries the mechanical brake (12). At the farther end of this shaft, a pinion gears into the spur wheel (27) on the second motion shaft. On this latter shaft, another pinion gears with the main spur wheel (30) which is bolted to the hoisting barrel (31). The two ends of the hoisting rope are secured to the ends of this barrel, which is grooved right and left-handed, so that the rope on each side is wound uniformly towards the centre and a truly vertical movement given to the lifting hook. An overwinding gear is fitted consisting of a screw spindle and travelling nut which, when the hoisting hook has reached its highest position, strikes a lever connected to an emergency cut-out in series with the hoisting controller and motor. This action opens the switch, cuts the current off the motor and applies the magnetic brake, thus stopping any further winding. With D.C. motors the hoisting controller is fitted with rheostatic brake steps to control the load when lowering. With A.C. motors hand gear is used to release the magnetic brake mechanically. This gear consists of a lever attached to a flexible wire (61), stretching from one end to the other of the cross-girder, passing over two fixed pulleys (62) and under a movable pulley (63), fitted with the counterweight (50). By operating the lever, the movable pulley and weight may be raised, thereby freeing the brake to any desired extent.

On the spindle of the traversing motor (40) a pinion is fitted which gears with a spur wheel keyed on to the first motion traversing shaft (43). By means of another pinion and spur wheel this shaft is connected to the driving axle (46) to which are keyed two of the wheels (47) on which the crab runs. The remaining two wheels (48) are keyed to the trailing axle (49) which runs freely in its bearings.

For handling hot ingots in rolling mills, special cranes with semi-automatic tongs are used, and have been made to fulfil such varied functions as stripping the ingot from the mould, charging and withdrawing the ingot from the soaking pit, and lifting and replacing the soaking pit cover.

For handling molten metal, special cranes known as 'ladle cranes' have been designed, and are made up to 150 tons capacity, but cranes of 80 to 100 tons are more generally used. These cranes are fitted with two crabs, the main crab running on top of the girders and the auxiliary crab between the girders and on the lower flanges. The main crab has two barrels arranged so that the hoisting ropes pass outside the girders to sheaves attached to a lifting beam from which the ladle is suspended. The auxiliary crab has usually a lifting capacity of about 20 per cent. of the main crab and serves for general lifts and for tipping the ladle if required.

For handling forgings under hydraulic presses, special cranes have been developed with relieving gear to prevent overloads due to the action of the press in pulling down a forging which is suspended not quite level with the anvil. These cranes are made up to 200 tons capacity, and can be fitted with detachable gear for rotating the forgings.

For stores, warehouses, and machine shops an overhead travelling crane with underhung jib has many advantages owing to the increased area covered by the jib.

For outside service the Goliath crane fulfils the same function as an overhead traveller, and eliminates the gantry. Fig. 2 (p. 804) shows a crane of this type designed for lifting loads up to 200 tons.

The speeds of working and sizes of motors are:—

Hoisting, slow gear	200 tons,	3 ft. per min.	Motor 65 h.p.
,, fast ,,	75 ,,	8 ft. ,,	
Traversing	200 ,,	30 ft. ,,	,, 25 ,,
Travelling	200 ,,	50 ft. ,,	,, 65 ,,

The load is lifted on sixteen parts wire rope, winding one part on to each of two barrels rotated by the 65 h.p. motor through four reductions of gearing. Change-speed gear is provided for handling loads up to 75 tons at a quicker speed. An automatic solenoid and automatic mechanical brake are fitted, the latter working entirely enclosed in an oil box. The crab is mounted on four two-wheeled bogies and traversed by the 25 h.p. motor driving four rail wheels through four reductions of gearing. The crane travels on a twin rail track, there being eight centre-flanged steel-tyred wheels under each leg. Four wheels on each side of the crane are driven. The girders are of the lattice type with auxiliary girders and bracing to give lateral stiffness. All motions are operated from an enclosed cabin situated in an elevated position, the hoisting and travelling controllers are of the contactor type and the traversing of the drum type.

Overhead Runways.

Overhead runway systems, consisting of a track formed from a single rolled-steel joist which may be curved to suit local conditions, and have switches or turntables, are frequently installed in warehouses, steelworks, gas works, and other situations where an overhead crane is unsuitable or too expensive. Hand-pulley blocks and trolleys are used on the runways for occasional service and light loads, but for speed and loads up to five tons an electric trolley hoist should be installed. These may be either floor or cage controlled, and have usually separate motors for hoisting and travelling. The lifting speed may be as much as 100 ft. per minute, and the travelling speed from 200 to 800 ft. per minute, according to the duty required.

Grabs can be fitted to lift coal, ore, or similar material.

Derrick Cranes.

The Scotch derrick crane, largely used by contractors, has many advantages on account of its portability, simplicity of erection and long outreach. As commercially manufactured in stock patterns, these cranes have not usually a very large factor of safety, and if the full load is to be constantly handled it is advisable to specify a crane for about 25 per cent. greater load. The B.E.S.A. has drawn up Specification No. 327, 1928, as a guide on details of construction of these cranes with a view to securing a uniform design and factor of safety. Timber is prohibited as a stress-bearing material. Cranes of this type are specially made in large sizes for dock-side service in fitting out vessels under construction, but are giving way in favour of the more efficient cantilever crane.

British Standard Specification for Derrick Cranes.

(No. 327—1928.) (*Extract.*) *

Applies to the 'Scotch Derrick' or 'Stiff Legged,' 'Guy Derrick,' and 'Power Derrick,' types of cranes used in building operations.

Appendix III. of the Specification gives the following :

List of British Standard Specifications for Materials to be used in Manufacture of Crane Details.

Crane Detail.	British Standard Specification.	Crane Detail.	British Standard Specification.
Drums and iron castings . . .	No. 321.	Ropes . . .	No. 302.
Steel castings (both general and gearing)	No. 24, Specification No. 10, Class A.	Rivets . . .	No. 15, Grade A.
Hooks . . .	Best — Yorkshire Wrought Iron No. 51, or Mild Steel No. 24, Specification No. 8, Class B.	Rotating and fixed shafts . .	No. 24, Specification No. 8, Class B.
Shackles . .	Do. do. do.	Screw threads .	No. 92.
Turnbuckles . .	No. 24, Specification No. 8, Class B.	Ties and bridles .	Grade A Wrought Iron, No. 51. Mild Steel, No. 15, Grade A, Galvanised Steel Wire Rope with steel eyes.
Stays, sleepers, jibs and masts . .	Mild Steel No. 15, Class A.	Structural steel .	No. 15, Class A.
Keys . . .	No. 46.	Paint . . .	No. 262.

Locomotive Cranes.

For general service on railway sidings around works and shipyards the locomotive steam crane is unequalled. These cranes are made in standard sizes up to 10 tons capacity and lift the load free on track with the jib in any position.

* By permission of the British Engineering Standards Association.

The following table gives particulars of standard steam cranes :—

STANDARD LOCOMOTIVE STEAM CRANES, 4 FT. 8½ IN. GAUGE.

Size.	Radius.	Size of Engines. (Cylinders.)	Size of Boiler.	Steam Press.	Speeds.			Weight.
					Hoist, f.p.m.	Slew, r.p.m.	Travel, f.p.m.	
Tons.	Ft.	Ins. Ins.	Ft. Ins. Ft. Ins.	Lbs.				Tons.
3	16	7 by 10	7 9 by 3 6	100	80	2½	350	14½
5	16	8 by 10	8 3 by 4 0	100	80	2½	350	22½
7	16	8 by 12	8 3 by 4 0	100	50	2	300	32½
10	16	9 by 12	9 0 by 4 0	100	40	2	250	45
15	16	9 by 12	9 0 by 4 0	100	40	1½	200	54

For cranes of longer radius or greater lifting capacity than the above, blocking girders and rail clips are fitted to give the necessary stability, or the cranes may be made heavier.

Cranes that are to travel round sharp curves should have sliding axle gears or differential gears and the conditions pointed out to the crane makers at the time of ordering.

The travelling gear should be of steel, also the slewing ring and pinion.

The slewing spur-ring forms the roller path and should rest on a machined seating on the truck, the pressure of the slewing rollers being sufficient to hold it in position for normal working, but in the event of sudden loads it will slide sufficiently to absorb the shock and prevent damage to the gearing.

The engines are usually of the double cylinder non-condensing type, cranks at right angles and cut off at about three-quarter stroke. The boiler is of the vertical cross-tube type for a working pressure of 80 to 100 lbs. per sq. in. and should be lagged. Two means of feed should be fitted, and the incoming water should enter above the firebox and not impinge directly on the heated plates.

The mountings should preferably be flanged, and secured by studs to pads riveted to the shell of the boiler, rather than screwed directly to the shell. The safety valve should be separate and not a combined mounting with the stop valve.

A list of standard boiler mountings would consist of

1 steam-pressure gauge with syphon ;
1 water gauge with plate glass protection ;
1 spring loaded safety valve ;
1 steam stop valve ;
1 injector with steam cock and feed check valve ;
2 test cocks ;
1 steam jet cock and pipe to funnel ;
1 blow-off cock.

The feed pump may be either a steam donkey pump or be driven direct from the engine crosshead.

RUNABOUT CRANES.

For service in situations where overhead cranes or rail tracks are not available, a handy type of crane, known as the Runabout Crane, has been recently developed. It is operated by electric motors, taking current from storage batteries or a petrol-driven dynamo. The truck or chassis carries the source of power, the operating motors, and gearing. It is mounted on a fixed two-wheel axle, and at the rear on a two-wheel articulated castor which adapts itself to any unevenness of the ground and is used as the medium for steering and slewing. The jib is of the bascule type, pivoted at the apex of an A frame carried on the chassis. It is held and adjusted by a wire rope purchase acting on the back arm operated through derricking gear to vary the height and radius. The load is lifted on wire rope and held by an automatic solenoid brake. The cranes are made in sizes up to 2 tons at 6 ft. outreach and 25 cwts. at 10 ft. 6 ins. outreach, in front of the truck.

STEAM BREAKDOWN CRANES

for use on the permanent way, have been made up to 100 tons lifting capacity, but in Great Britain the usual size is from 10 to 35 tons. The truck of these cranes is provided with the usual permanent-way fittings and the axles are spring borne. The maximum load per axle with the crane in running order should not exceed 18 tons. When the crane is 'in train' the travelling gear should be disconnected and the superstructure locked to prevent it turning. Owing to the limited space due to the crane having to clear the loading gauge, a comparatively small size boiler must be fitted, and this should be of the multitubular type as the ordinary cross-tube boiler would not generate enough steam.

Wharf Cranes.

For dockside service in loading and unloading ships, the high gantry type of jib crane is principally used. The usual sizes are from 1½ tons to 5 tons and the radius from 40 to 60 ft. The gantry is made wide and high enough to admit of the passage of rolling stock underneath and to keep the jib clear of the sides of large vessels. A separate motor should be fitted for each motion.

Typical speeds of working, in feet per minute, are as follows :—

Description.	Size.		
	1½ tons.	3 tons.	5 tons.
Hoisting	200	150	120
Slewing	500	400	300
Travelling	100	100	80
Derricking	150	120	100

For high derricking speeds some form of compensating gear is essential to reduce the power consumption due to raising the load and jib each time the radius is reduced. With the ideal compensating gear the path of the load is kept horizontal as the jib is raised or lowered, and with the jib balanced the power required is only that necessary to overcome friction.

Numerous types of compensating gear have been invented during the past few years, some of which are very complicated, and the crane user should select those of the simplest construction with the minimum number of moving parts, and pay special attention to the lead of the hoisting rope, which often suffers considerably through being passed round numerous pulleys in order to give the desired level path to the hook as the jib is raised or lowered.

Luffing motion operated by a crank is often advocated, as the throw of the crank restricts the movement of the jib between the maximum and minimum radius; but in practice this is no real advantage as limit switches must be fitted as on rope-operated gear, otherwise the reversal of movement caused by the crank, irrespective of the direction of rotation of the motor, prevents the driver from knowing in what direction to start the motor to produce the required movement of the jib.

For rapid handling of coal, ore, or similar loose materials, the transporter grab crane is unequalled. The load is carried by a trolley which travels on a horizontal track, supported in a suitable manner by the crane structure, which latter may take the form of a tower with cantilevers on each side or a bridge, if it is desired, to convey the load over a considerable distance.

If working at the dockside the outer cantilever may be made to hinge up to clear the masts and funnels of ships when coming alongside.

The operating machinery can be fixed on the structure and the trolley carry only guide sheaves for the hoisting ropes and be hauled along by other ropes winding on to a separate winch, or the hoisting and transporting gear can be carried by the trolley. The latter system is usually adopted when the gross load exceeds 5 tons, as with the fixed machinery system the wear of the ropes becomes excessive in the larger sizes and off-sets the initial saving due to the lighter design of structure.

The advantage of this type of crane over the jib crane is that the load is conveyed in a direct line from point to point instead of through the arc of a circle. The transporting speed can be at the rate of 800 to 1,000 ft. per minute, whilst the maximum safe slewing speed is not more than 400 to 500 ft. per minute. With a 3-ton grab the transporter crane will usually handle 150 tons per hour against 100 to 120 tons on a jib crane.

On the larger types the driver travels with the trolley and is always immediately above the area of operations. In some cases the trolley is fitted with a revolving jib of about 10 ft. radius, from the point of which the grab is suspended. This enables the driver to cover the whole area of the ship's hatchway without moving the main structure, resulting in a speeding up of operations.

For handling coal, ore, and similar materials, much time and labour is saved by using grabs; these may be on the single or double-chain principle.

The single-chain grab may be fitted to any crane lifting on a single fall of rope, but grabs of the double-chain type require an extra drum to coil up the opening or closing rope. This should preferably be driven through a friction drive, although for short lifts a spring drum is quite suitable.

With the larger sizes two independent motor-driven winches, one for each rope, are sometimes used. The opening and closing of the grab is then performed by movement of the motor controller handles. For satisfactory operation automatic contactor controllers should be used with relays to prevent excessive overloading of the hoisting motor in the event of the driver being late in starting the holding rope winch when coming away from the material with a full grab.

Grabbing work is very severe on cranes, and the nominal size of the crane should be 50 per cent. in excess of the combined weight of grab and contents. The empty grab may be balanced by sliding counter weights to reduce the power consumption.

Double rope grabs have a tendency to spin and twist the ropes round each other. This may be overcome by fitting a large sheave at the grab head, and leading the opening and closing rope back to the jib head or winch, the grab being thus suspended on three ropes spread apart by the sheaves, which keeps it perfectly steady. On heavy grabs two hoisting ropes are used either on the single or double chain principle.

Automatic self-dumping single-rope grabs, which open when the weight of the grab is relieved from the hoisting rope, require only a single-drum winch for operation and give a higher duty than the single-rope grab with fixed discharge gear.

CAPACITY AND WEIGHTS OF GRABS.

For Coal.			For Iron Ore.		
Capacity.	Weight of Grab.	Approx. Weight of Coal.	Capacity.	Weight of Grab.	Approx. Weight of Ore.
Cub. ft.	Cwts.	Cwts.	Cub. ft.	Cwts.	Cwts.
45	30	15	21	38	20
56	36	20	31	49	30
63	42	25	42	60	40
82	48	30	65	90	60
112	68	40	150	220	200
180	112	70	—	—	—

SHIPBUILDING CRANES.

For service over shipbuilding berths, high gantries with overhead travellers and mono-rail cranes, gantries with cantilever cranes, steel derricks and tower cranes, are used, and experience favours the latter type.

Tower cranes may be either of the fixed or travelling type, practice in Great Britain being inclined to the fixed type, as they require less clearance between the berths, and use is made of valuable space which would otherwise have to be kept clear for the track of a travelling crane.

The crane structure consists of a horizontal cantilever supporting the track for the load trolley, and is carried on the top of a braced steel mast which is free to revolve inside a fixed or travelling tower. The cantilever is balanced by a tail carrying the machinery, which gives the crane a hammer-headed appearance.

The motions of hoisting, slewing, racking, and travelling are fitted, operated by independent motors. The cranes are made with a lifting height up to 120 ft., and a radius up to 140 ft. The usual capacity is 5 tons at 100 ft. radius and 10 tons at 50 ft. Travelling cranes of this type have been made to lift 30 tons at 100 ft. radius, 15 tons at 140 ft. radius, and with a height of lift of 120 ft., but these are exceptional sizes.

These cranes, if desired to lift varying loads at different radii, should be fitted with an automatic anti-overload device which will prevent a load above the safe limit for its radius being lifted, and will also prevent the trolley traversing beyond the safe radius for that particular load.

For fitting out, cranes of the wharf type with derricking jibs are used, and range in size from 10 to 40 tons, with radii up to 80 ft., according to requirements. Two speeds of hoisting are generally arranged, so that light loads may be lifted at a speed of 50 to 80 ft. per minute. A typical crane of this type lifts 30 tons at 55 ft., 20 tons at 72 ft., 10 tons at 80 ft. radius, and is mounted on a gantry to clear rolling stock, the centres of the gantry track being 22 ft.

Cranes of this type are installed for dry dock service, and it is occasionally necessary that they should work from both sides of the dock. This may be arranged by connecting the two sides of the dock by a track at the end and introducing turntables, or better by making the track sweep round on a curve and carrying the rail wheels on special swivelling bogies and inserting an epicyclic gear between the sections of the travelling motion cross shaft which connects the driving rail wheels on each side of the crane.

A large fitting-out crane is shown by fig. 4. This crane is designed to lift a load of 25 tons at 84 ft. radius, 5 tons at 112 ft. radius, and has a total range of lift above track level of 137 ft. The centres of the track rails are 40 ft.

The load of 25 tons is lifted at a speed of 20 ft. per minute, and 5 tons at 100 ft. per minute.

Floating cranes are occasionally used for fitting out and repair service, and have the advantage of being transferable from dock to dock, but with them it is not easy to deposit a heavy load with

precision, as the crane follows the movement of the water and its equilibrium is disturbed with every fluctuation of loading. These cranes are usually steam operated, but in recent years several large cranes have been electrically driven. This appears an unnecessary and expensive refinement, offering little advantage to compare with the additional cost of installing generators unless other use can be made of the power.

The pontoons are usually rectangular, flat-bottomed vessels, but if self-propelled it is advisable to give them a ship shape. The angle of heel under full load conditions and a 5 lb. wind should be not more than 5°. Travelling ballast is sometimes fitted to limit the heel under various loading conditions. A crane of this type, constructed by Sir Wm. Arrol & Co. Ltd., Glasgow, for the Brazilian Government, is designed to lift 30 tons at 10 metres outreach from the side of the vessel and is steam operated, the motions of hoisting, derricking, and slewing being fitted.

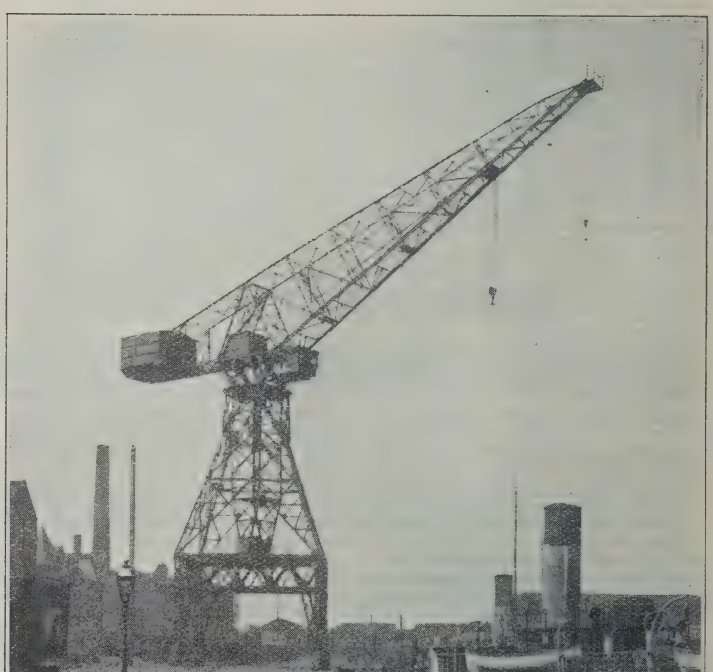


FIG. 4.—25-Ton Wharf Crane.

The pontoon, built by Messrs. Ritchie, Graham & Milne, of Glasgow, is of the self-propelled type, 150 ft. long by 50 ft. beam by 8 ft. deep, and is fitted with two sets of compound surface condensing engines having cylinders 13 ins. and 26 ins. by 18 ins. stroke, to give a speed of seven knots.

The outline of the largest floating crane so far constructed is shown in fig. 6, p. 814. This crane is designed to lift 350 tons at a radius of 100 ft., 300 tons at 121 ft., and 50 tons at about 160 ft. The range of lift with full load is 140 ft., and with 50-ton load 200 ft.

The crane, which was made by Messrs. Cowans, Sheldon & Co., Ltd., of Carlisle, for service in Japan, is operated by steam, there being nine sets of double-cylinder condensing engines, each motion being independently operated without the use of clutches. Control of the load for lowering is obtained by hydraulic brakes which enable very precise movements to be obtained.

The pontoon is 270 ft. long by 92 ft. wide, with draught of 10 ft., and is self-propelled by twin-screw compound engines supplied with steam at 150 lbs. pressure from two single-ended boilers.

The boilers are of the water-tube type and supply steam to the crane engines, auxiliary machinery and electric light set.

The Giant cantilever crane is adapted for loads from 75 to 250 tons and enables large portions of machinery, gun turrets, etc., to be lifted on board as a complete unit. A typical crane of this

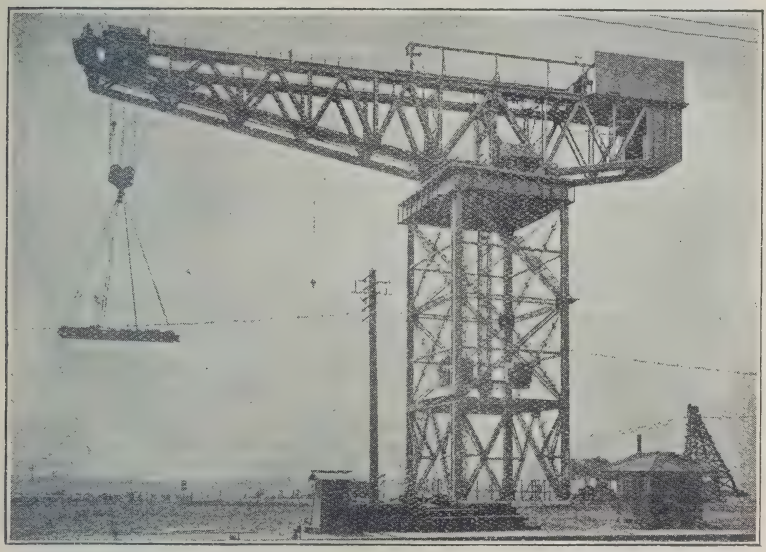


FIG. 5.—250-Ton Giant Crane.

type, manufactured by Sir Wm. Arrol & Co. Ltd., Glasgow, is shown in fig. 5. The leading dimensions are :—

Max. radius for 250 tons	105 ft.
,, ,, ,, 120 ,,	130 ft.
,, ,, ,, 80 ,,	160 ft.
Height to top of tower	140 ft.
Base of tower	50 ft. square.

The main hoisting machinery consists of independent hoists each of 125 tons capacity, which can be worked together when required. The auxiliary hoist is of 40 tons capacity and the jigger hoist of 10 tons, this latter having a separate track and trolley on the underside of one of the cantilever girders. The crane was satisfactorily tested with a load of 360 tons.

CAPSTANS.

For hauling trucks in goods stations or on the dock side a capstan designed to exert a pull of 1 ton at a speed of 150 to 200 ft. per minute is sufficient. The free bollard type with the rope attached to the bollard is more economical in use than the fixed bollard type which depends upon the friction of the coils of rope round the bollard for hauling power.

The warping of ships into position in the docks is done by capstans of from 10 to 20 tons pull. A constant torque at varying speeds is the desideratum for this duty, and the characteristics of the Austin constant-current motor make it eminently suitable for this work.

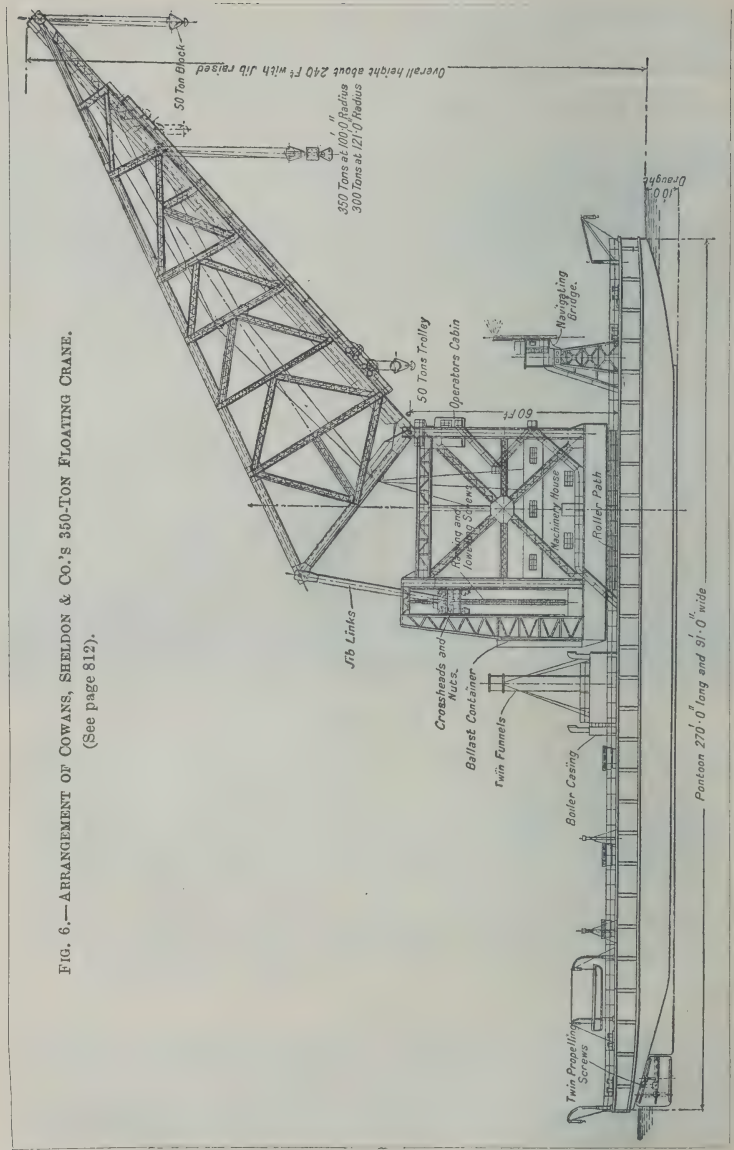


FIG. 6.—ARRANGEMENT OF COWANS, SHELDON & CO.'S 350-TON FLOATING CRANE.

(See page 812).

General Data.

Structures.

Crane structures should be made of open-hearth mild steel, 28 to 32 tons tensile strength, extension not less than 20 per cent. in eight inches. In determining the section of members, allowance should be made for the vibration due to the movement of the load and machinery, for the movement of the structure in the case of a travelling crane, and deflection should be considered where bearings for shafts are carried on structural members. The stresses should be such as to give a factor of 5 to 6 on the ultimate strength of the material. The ratio of length to radius of gyration should carefully be considered in struts, and charts plotted from Claxton Fidler's or Tetmeyer's formula form a reliable basis on which to determine the stresses. For wind bracings the factor of safety need not be more than three, as this is an exceptional loading infrequently applied.

Wind pressure on cranes in exposed positions in Great Britain may be taken as a maximum of 40 lbs. per sq. ft. It is inadvisable to work a crane in a wind pressure above 5 lbs. per sq. ft., and when designing the structure it is therefore unnecessary to allow for a higher pressure under full load conditions. The maximum wind pressure must of course be considered separately.

The revolving superstructure of jib or cantilever cranes may be carried on a ring of live rollers, or on a set of four or eight large diameter rollers designed as travelling wheels, but with conical treads and unflanged. The live rollers may be of either forged or cast steel and should run between two steel guard rings connected to the centre pin casting by radial rods. A gunmetal washer should be fitted between each roller and the outer guard ring, to take the outward thrust of the rollers. For unhardened rollers the pressure in lbs. should not exceed 400 by length in inches by diameter in inches. The roller-paths should be of steel, machined conical on the face to form a full bearing for the rollers.

On jib cranes up to about 5 tons capacity the roller-path frequently consists of a flat-bottom rail bent to a circle, and the superstructure is then mounted on four rollers 15 ins. to 18 ins. diameter. This is a cheap and serviceable construction.

A friction clutch should be incorporated in the slewing gear adjusted to slip and prevent damage in the event of excessive shock due to too sudden application of the brake or the fouling of jib with some stationary object.

Gearing.

On all high-class crane work the gearing should be machine cut. For hand and low-power cranes cast-iron wheels and steel pinions are satisfactory, but for all high-speed cranes the gearing should be of steel, with the possible exception of motor pinions, which, to ensure silent running, may be of rawhide, fabroil, or phosphor bronze.

For heavy loads and high speeds, machine-cut double helical gearing is very satisfactory, efficient, and silent.

The strength of spur gearing can be determined by the Lewis formula (page 915).

Gearing with stub teeth and 29° involute is advised for crane work, as the strength is greater than standard 14½° involute, by 30 per cent. with a rack, to 50 per cent. with a 12T pinion. The addendum is 0·27 of the circular pitch and the dedendum 0·32. With diametral pitch the outside diameter of the wheel is determined by adding 1·75 to the number of teeth and dividing by the pitch.

Worm wheels should consist of a phosphor bronze rim fixed on a cast-iron or steel centre, and have machine-cut teeth. The worm should be of high tensile forged steel and need not be hardened unless facilities exist for grinding the threads. In determining the size of worm gearing the heating effect must be considered as well as the strength.

For the highest efficiency a special globoidal gear, known as Enveloping Worm Gear, developed by Bostock & Bramley of Leeds, is recommended. This gear has a greater bearing surface than any other form and occupies less space than ordinary worm gear for the same power.

(See also 'Worm Gearing,' p. 937).

Shafts.

In designing shafts the resultant loads should be determined and the effect of combined bending and twisting moments considered. The equivalent twisting moment due to combined bending and twisting is equal to

$$BM + \sqrt{(BM)^2 + (TM)^2},$$

where,

BM = bending moment; TM = twisting moment.

Stresses should be low on account of the frequent reversal and impulsive loads common with crane work. Where keys are fitted, the shafts should be increased in diameter.

Bearings.

For cheap hand cranes the bearings are usually of cast-iron and not bushed. All power-driven cranes should have gunmetal bushed bearings. The bearing brackets should be fitted with caps, and the bushes be in halves. Studs for securing the caps are not desirable.

Tee-headed bolts should be used, and these should fit in a slot accessible from the outside of the pedestal, so that the bolts can be withdrawn after the cap is removed and without disturbing the steps. The pressure per sq. in. for shafts running at medium speeds should not exceed 600 lbs. For slow-moving pulleys and axles the pressure may be as high as 1,000 lbs. per sq. in. Fast-running shafts, such as motor spindles, should have ring lubrication. Axles are frequently fitted with float lubrication, which consists of a cylinder of cork floating in an oil-well formed in the inverted cap of the bearing and by contact with the axle supplying the necessary lubricant. Screw-down and spring-feed lubricators with grease are very satisfactory.

To reduce frictional losses, ball or roller bearings are sometimes fitted, and the makers should be consulted regarding the size of these. Ball bearings are very serviceable for worm thrust bearings, footstep bearings, and for under the hook.

Ropes.

Ropes for crane work should be of best plough steel quality, 6/37 stranding if above 1¾ in. circumference, below this size 6/24 or 6/19 stranding. For ordinary work a hemp core is suitable, but where exposed to heat a soft iron core is desirable. The diameter of the barrel and pulleys round which the rope passes should be between six to seven times the circumference of the rope. For high speeds it is advisable to use even larger pulleys than given by this rule. Sharp and reverse bends should be avoided, although with single ropes which may twist in winding the effect of a reverse bend is problematical. When lifting on a single fall, a so-called 'non-twisting' rope should be used. Ropes made from very high tensile steel wire are stiff and tend to 'bird-cage' or open out. The rope, particulars of which are given on page 828, is recommended for cranes. Ropes should be lubricated and examined in service for broken wires and corrosion. The barrels and pulleys should be machine-grooved to form a smooth and even surface for the rope to bed on. Hoisting ropes should have a factor of safety of at least seven, and even higher with small size ropes.

British Standard Specification for Round Strand Steel Wire Ropes for Cranes.*

(No. 302—1927.) (*Abstract.*)

(*This Specification is intended to provide for round strand ropes of Ordinary Lay having wires approximately uniform in diameter and of* 4 × 37, 6 × 19, 6 × 24, 6 × 37, *and* 6 × 61 *constructions, for the Lifting and Derricking, Ropes of Cranes and Coal Hoists, Transporters, and other similar lifting appliances.*)

3. The wire used in the manufacture of ropes to this specification shall be made from steel billets made by the acid open-hearth process and containing not more than 0·04 per cent. of sulphur and 0·04 per cent. of phosphorus.

The wire shall be of one of the following qualities as specified by the purchaser:

Grade A.—Special acid or 'Swedish' quality.

B.—Acid quality.

4. The wire used in the manufacture of ropes to this specification shall conform to one of the following descriptions, as specified by the purchaser:

Tensile Breaking Strength in Tons per sq. in.	Trade Description.
80– 90	Best Patent Steel.
90–100	Special Improved Patent Steel.
100–110	Best Plough Steel.
110–120	Special Improved Plough Steel.
115–125	Extra Special Improved Plough Steel.

14. The fibre used for the central or main core of the rope shall be new, and free, long fibre, good quality hemp, manila or jute.

(See also page 199.)

Hooks.

Fig. 7 gives proportions of hooks in terms of the diameter at the bottom of the thread. The size for any load can be arrived at by fixing the size of the shank to carry the load in tension. The area at the bottom of the thread must be taken. Hooks should be forged from mild steel.

Crane hooks depending from large travelling cranes in machine shops ought to be painted white, so as to be easily seen. With this improvement in visibility workmen will be less likely to run against a hook or to be struck by one which is moving.

Foundry ladle hooks fail from time to time without any previous indication of weakening, so that in many foundries it is usual to change them after a definite period of service. A German firm has brought out an improved hook made of a number of laminæ with an air space between.

* By permission of The British Engineering Standards Association.

This hook is said to be absolutely safe, for if a lamina should break, the remaining ones will be strong enough to hold the ladle. It is also necessary to replace only the broken strip. To ensure

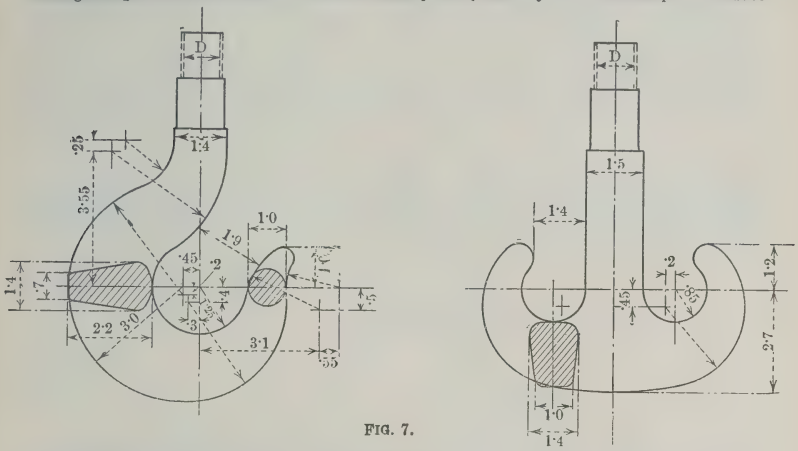


FIG. 7.

uniform wear, even when the load is at an angle, the jaws of the hook may be furnished with a two-piece bronze or steel roller bearing.

CHAINS.

Chains should only be used for grabs or situations where the fine strands of a rope would be damaged by abrasion. The working load for a welded iron chain is given by the rule.

$$W = \frac{(\text{dia. of iron in } \tfrac{1}{8} \text{ in.})^2}{9}.$$

Weldless steel chains are about 13 per cent. stronger, and make good slings on account of their lower weight. Steel plate driving chains are sometimes used for drives instead of gearing, and give good results if properly designed. The burden chain for large ingot rotating gears is best constructed of one inner and two outer links connected by steel studs.

(See also page 193.)

DESIGN OF LIFTING TACKLE.

Fig. 8 shows a faulty design of eye-bolt, likely to break without warning; and fig. 9 one which would stand rough usage, permit a pull at bad angles, and give notice before it broke, and at the same time possesses a useful strength (over 200 per cent.) beyond that of fig. 8.

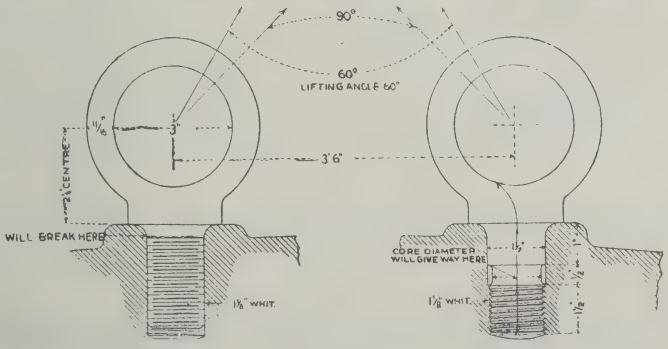


FIG. 8. FIG. 9.

These eye-bolts—hand-forged, of both types—have been tested to complete destruction at various 'punishing' angles, by Messrs. Kirkaldy, and the verdict has been from 50 per cent. to 250 per cent. in favour of the fig. 9 pattern.

The arrows depict the form and direction of the strains, and such were precisely followed in the tests by Messrs. Kirkaldy.

The increased strength, as a result of decreased material, refers to the reduced part A, for this reduction is an effective factor which contributes to the need that the strain of the load should be distributed over as large a volume of the composition of the bolt as is possible.

(*J. T. Towlson, 'The Electrical Review.'*)

Brakes.

One of the most important features on a crane is the brake, and this should have a large factor of safety. The brakes in common use are electric, automatic solenoid, centrifugal, automatic mechanical, and manual brakes. Hydraulic and pneumatic brakes are used for heavy loads, or where extremely fine control is desired.

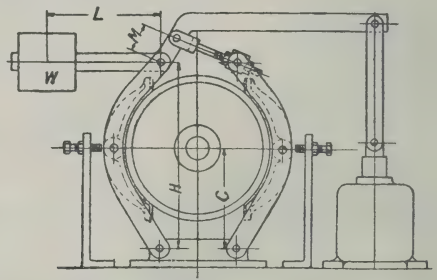


Fig. 10.

A typical arrangement of a solenoid brake is shown in fig. 10. The brake is applied by the weight and released by a solenoid magnet connected in the motor circuit so that it is excited immediately current is switched on to the motor.

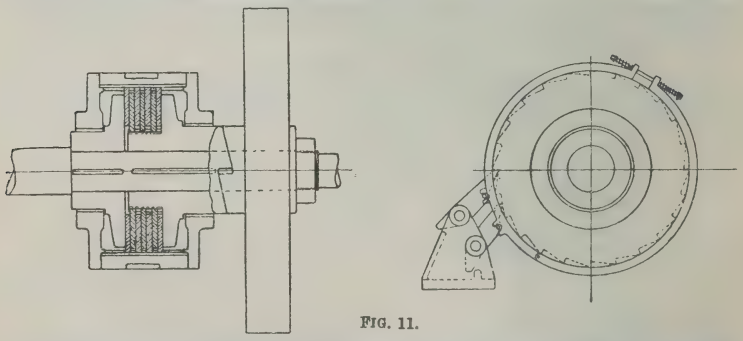


Fig. 11.

If F is the turning force on the rim and μ the coefficient of friction between the brake shoes and sheave, then the weight required to apply the brake equals

$$\frac{C \times M \times F}{H \times L \times 2\mu}$$

where C, M, H, and L refer to the dimensions shown in fig. 10.

The weight of the solenoid plunger and levers must be taken into account, and about 10 per cent. is usually added for friction in joints.

The centrifugal brake is designed to control speed, and does not serve to sustain the load at rest. In one form it consists of a series of weighted slippers which are pressed by centrifugal action on the inner rim of a fixed annulus, and thus have a retarding effect. The brake shoes should be slightly lubricated and ample radiating surface allowed to dissipate the heat generated.

The automatic mechanical brake, known also as the 'Weston washer brake,' depends for its action upon the friction between a set of fixed and rotating discs. The pressure on the discs is obtained by a screw and nut, or two helical surfaces which set up an axial thrust along the brake shaft. Fig. 11 shows one type of this brake in which the friction discs, alternately of bronze and steel, are enclosed in a cast-iron box, with ratchet teeth on the outside. One set of discs is connected by feathers to the box, and the other set to the thrust plate. The driving wheel has a helical face formed on the boss which engages with a corresponding face on the thrust plate. When hoisting, the discs are pressed up tight and the whole brake revolves, but when lowering, the pawls engage with the ratchet box, and hold it stationary. The pressure on the discs is relieved by the reversal of the motor, and the load is free to descend. As it tends to accelerate and gain on the motor, the discs are pressed together again and the friction between the surfaces of the fixed and revolving discs forms the braking effect. Ample lubrication and radiating surface are essential for the successful operation of this brake.

The pitch of the helical faces is determined by the following formula:

$$T = \frac{PL}{R \times \mu \times N'} \quad \text{and} \quad \frac{PL}{Tr} - f = \tan A;$$

therefore,

$$\text{lead of helix} = \tan A \times 2\pi r;$$

where,

R = average radius of disc;	μ = coefficient of friction = ·07;
PL = torque on driving wheel;	r = mean radius of helical faces;
A = angle of helix;	T = thrust or pressure on discs;
N = number of rubbing surfaces;	f = coefficient of friction of helical faces.

The brake should be proportioned so that an angle of helix of about 10° will give the requisite pressure. The brake should run at a speed not exceeding 150 revs. per minute, and the pressure on the discs should not be more than 200 lbs. per sq. in. and on the helical faces 500 lbs. per sq. in.

Manual brakes are usually of the band or post type. Band brakes should have a bar fitted round the outside of the strap with adjusting screws to ensure that when the brake is released the strap will come away evenly from the sheave.

Band brakes are generally only suitable for a torque in one direction, i.e. away from the fixed end of the strap.

To find the pull to be applied to the loose end of the strap, the following formula may be used:

$$P = \frac{F}{e^{\mu\theta} - 1},$$

where,

P = pull;	μ = coefficient of friction;
F = turning force on rim of brake sheave;	θ = angle of sheave embraced by strap in radians.
e = base of Napierian logs.	

Brake sheaves are usually made of cast iron, but steel should be used for severe duty. The brake shoes or strap may be lined with wood, leather, or one of the asbestos fabrics such as 'Chekko' or Ferodo lining; wood and leather are inclined to char and glaze and the coefficient of friction is reduced. Bonded asbestos (page 850) is not subject to these objections and maintains a constant coefficient of friction, which in practice may be taken as 0·3.

Electric cranes may be controlled by dynamic or rheostatic braking if direct current is used. Dynamic braking with potentiometer control is to be preferred to plain rheostatic braking, as practically automatic speed control is obtained in the lowering direction irrespective of the weight of the load. Heavy rushes of current and mechanical shock can be produced on the motor by either system of electric braking, and some form of current limit control should be provided.

Speeds up to 1½ times full load hoisting speed can now be obtained without difficulty in lowering the light hook under potentiometer control.

A shunt solenoid should be used with this system of braking, as the current may fall too low to excite a series coil sufficiently to hold off the brake.

Messrs. Laurence Scott have recently developed what is termed floating brake control for speed regulation, with the object of avoiding the disadvantages of dynamic braking. The brake consists of fixed and rotating discs mounted co-axial with the motor spindle. The discs are pressed together by springs and released by an electric magnet, excited when current is switched on to the motor. By arranging the starting resistance so as to divert current from the brake coils the magnetic flux is weakened to such a degree that the pressure of the springs begins partially to apply the brake and the speed is regulated according to the amount of current diverted from the series winding of the brake coils. The value of the resistance in circuit is adjusted through the medium of the controller used for starting the motor. This system has been used on a number of cranes with success.

To control the lowering of heavy loads at slow speeds, a hydraulic brake is the most efficient. In principle this consists of either a reciprocating or rotary pump with closed circuit and throttle valve.

The pump is in the train of gearing between the motor and the rope drum, and by means of a free wheel drive is inoperative in the hoisting direction. On lowering, the solenoid or other brake is released, and the descending load drives the gearing and pump. The speed is controlled by the throttle valve, and very fine regulation at constant speed can be obtained. The system is especially suitable for placing guns, turbines, and engines on board ship.

For gun tempering, armour treatment, and processes where immersion in a cooling bath is necessary, it is essential that the load should be rapidly lowered at speeds varying from 60 ft. to 150 ft. per min.

The lifting speed is about one-tenth of these speeds, and the usual practice is to disconnect the rope drum from the gearing and motor when lowering.

A coil clutch is used for the purpose, and a powerful brake must be fitted to control the lowering speed and sustain the load. Manual brakes have proved satisfactory for this purpose, particularly in conjunction with a centrifugal brake for speed control. On cranes of over 100 tons capacity elaborate electro-hydraulic systems have been used with great success, but they are very costly.

Electrical Equipment.

Cranes may be worked by either direct or alternating current, but the former is to be preferred for the following reasons :

1. Series-wound motors may be used with high starting torque and speed variation in proportion to load.

2. Very fine control can be obtained and dynamic braking used.

3. Less wiring than with slip-ring motors.

Alternating-current motors usually require less attention than direct-current motors, but the starting torque is limited and the running speed is practically the same under light and full load conditions. The starting torque for alternating-current motors should be not less than two-and-a-half times the full load torque, and in fixing the horse-power it is advisable to make it about 10 per cent. higher than for direct-current motors for the same duty.

With reference to paragraph 1, it should be noted that whilst with ordinary series motors the speed increases with a reduction in the load the variation is not very great between full and approximately half load, and as in many cases light loads are more frequently lifted, mechanical change gear is used to increase the speed and take advantage of the full power of the motor. This can, however, be eliminated by means of a special two-speed motor, designed to develop its full horse-power at two definite speeds besides having the usual series characteristic. The alteration in speed is obtained by diverting part of the field current, and with the Scott Bentley discriminator control this can be done automatically, thus relieving the operator from the trouble of changing gear and ensuring the correct speed for the weight of the load. The efficiency of the crane is increased without any added complications.

Two-speed squirrel-cage motors are now also manufactured, the change of speed being obtained by altering the number of poles.

The horse-power is in proportion to the speed, and the principal object of this design is to reduce the initial current peak by starting up at a slow speed and switching over to the higher when the gear is in motion.

Squirrel-cage motors are not in general suitable for crane work, although a number of cranes have been equipped with special squirrel-cage motors having high-resistance rotors and giving twice full load torque at starting. The starting current is about four to five times the full load current, and the switchgear should be of robust design. These motors are not suitable for a heavy duty or for where more than about 20 h.p. is required.

All motors should preferably be of the totally enclosed type. For ordinary engineering shop service crane motors rated to develop their full power for a period of half-an-hour with a tempera-

ture rise of 90° F. are quite suitable. For steel works and other situations where the duty is severe, the rating should be for one hour. It is desirable to have the yoke and bearings made in halves so that easy access may be had to the interior of the motor.

A special design of direct-current motor known as the Mill type is largely used on steelworks cranes. It has a steel carcase made to hinge open along the centre line to permit of easy removal of the armature. The armature is of small diameter to reduce inertia stresses on reversal, and the insulation and winding are such as to permit of stalling the motor without injury. It was originally developed for service in rolling mills, hence the name.

Controllers.

For ordinary crane service and motors up to about 60 h.p. the drum controller gives very good service. There should be ample room inside the controller for connections, all parts accessible, finger tips and drum contacts easily renewable, insulation of mica or micanite, and a definite step should be felt as the barrel is moved round to each contact. Magnetic blow-out should be fitted on direct-current controllers.

For frequent operation and large currents a solenoid operated or contactor controller should be fitted. The contactor switches handle the main current to the motor and are controlled by a small drum, or master controller, which deals only with the shunt current required to excite the contactor coils. A development of this system is the Holme Portable Remote Control, in which the master controller can be carried about by the operator and connection is made to the contactor panel by a multicore flexible cable.

Creeping control for foundry or riveting work is obtained by inserting resistance in parallel with the motor armature circuit, thus magnifying the strength of the field and enabling full-load torque to be obtained at a very slow speed.

Cam operated controllers have outwardly the appearance of drum controllers but the fingers consist of a series of simple contractors actuated by a number of cams mounted on a vertical spindle rotated by the controller handle. They are used where the current is considered rather heavy for an ordinary drum controller and yet not sufficiently heavy to justify an automatic controller.

On alternating current circuits speed regulation is obtained by resistance in the rotor circuit, but the same range of speeds cannot be obtained as with direct current, particularly on light loads.

When creeping speeds are required with alternating current the most practical method is to load up the motor with a manual brake, and then use the controller and resistance to adjust the speed.

Up to about five h.p. resistances are usually made of wire-wound porcelain bobbins, and above this power consist either of stamped steel grids or a drawn continuous strip bent into grid form. The grids should be well stiffened to eliminate vibration and sherardised to prevent rusting. Cast-iron grids should be avoided, as they give trouble due to breakage.

Resistances are usually rated to be in circuit two or five minutes out of every fifteen minutes, with a temperature rise not exceeding 350° to 400° F. For creeping control ten minutes or even continuous rating is desirable.

Switchboards.

Many cranes are fitted with a main switch and fuses for the protection of each motor circuit, but it is desirable that these should be replaced by a main automatic circuit breaker and relay switches for each branch circuit. The relay switches do away entirely with fuses and obviate the loss of valuable time at critical moments due to replacing them. Each switchboard should have a pilot lamp and socket for portable lamp connection. It is advisable, although not essential to fit an ammeter and a voltmeter.

Limit Switches.

Limit switches to prevent overwinding or overtravelling should be of the shunt type, and connected in the no-volt coil circuit of the main breaker on the switch-board, so that in the event of the switch coming into operation the main circuit is opened at the breaker.

Special contacts can be fitted on the controller to short-circuit the limit switch on bringing the controller on to the first notch in the reverse direction. The breaker can then be reset and the motor reversed. The limit switch should be of the totally enclosed self-resetting type.

Solenoids.

Solenoids for brake operation are preferably of the series type on direct-current circuits, but where rheostatic or dynamic braking is used they must be of the shunt type, in which case a discharge resistance should be fitted to take up the induced current on breaking circuit. For alternating circuits solenoids are preferably connected in all phases. The rating of a solenoid should be the same as the motor with which it is in circuit. The coils should be enclosed as a protection against damage, dirt, and damp.

The Lewis brake solenoid made by *Messrs. Ransomes & Rapier, Ltd.*, has two plungers, one sliding inside the other. The inner one is connected to the brake lever and is free to move independently of the outer plunger or armature. When the magnet is excited the two plungers are automatically and mechanically coupled and act as one in releasing the brake, but when the current is switched off they are disconnected and the inner plunger is free to follow the movement of the brake lever independently of the stroke of the magnet. As the brake surfaces wear the inner plunger is partially withdrawn from the outer, but owing to the automatic clutching arrangement, the full stroke of the magnet is always effective and releases the brake freely, obviating the rubbing and wear which take place with the ordinary brake if the shoes are not occasionally adjusted.

WIRING.

For crane work the current density may be in excess of ordinary practice for shop motors owing to the intermittent service, and the I.E.E. have drawn up a set of rules for this duty.

Cables, unless lead-covered and armoured, should be enclosed in metal conduits. Current is supplied to travelling cranes from bare conductors stretched along the track by means of slipper or trolley collectors. The bare conductors may be of copper wire, or, as is now common practice in steel works, steel tee or angle bars ground on the sliding face.

For jib cranes the conductors may be carried overhead or in a conduit. An alternative system is to run the cables in a closed conduit with junction boxes at intervals from which connection can be made by plug and flexible cable. A drum should be fitted on the crane to coil up the slack cable.

The drum may be hand operated if the crane is infrequently moved or if the length of travel is small, not exceeding 50 ft., and the cable light a spring can be used to automatically revolve the drum as the cable becomes slack and prevent loose coils lying on the ground. Where space will permit, it is better to have a weight-operated overhauling gear to exert a torque on the drum, and lengths of heavy cable up to 150 ft. long can be handled in this manner. Cable drums geared to the travelling motion are inadvisable owing to wheel slip on the rails, which may cause a considerable tension on the cable, with resulting damage.

Electro-Lifting Magnets.

Electro-lifting magnets are largely used in connection with cranes for handling all classes of iron and steel materials, and their application shows a considerable saving in time and cost compared with manual labour or even ordinary crane service with slings.

For lifting pig iron, small and heavy scrap, and large solid masses, a single circular magnet is adopted, and for lifting long and wide plates two or more of these magnets are used spaced along a beam which can be slung from the crane hook.

For rails and long bars or tubes a series of two or three rectangular or oval magnets on a beam are better, as greater lifting capacity is obtained with less dead weight.

The circular type of magnet consists of an outer casing or yoke of high permeability steel with a central boss which forms the inner pole of the magnet, the walls of the casing forming the outer pole. The annular space between the poles serves to contain a coil of high conductivity insulated copper wire or strip, secured in place and protected by a non-magnetic manganese steel bumping plate, the latter being held in position by the inner and outer pole shoes which are bolted to the casing and form the contact face of the magnet. In the 'Phœnix' magnet there are two annular recesses each with a coil; this construction increases the area of the central pole and adds to the lifting power of the magnet.

The outer casing is provided with lugs for attaching the sling chains, and a terminal box is fitted for making connection between the magnet coil and the supply cable.

Circular magnets are not suitable for stacking material in any systematic form, as the tendency with such magnets is to lift the material two or three deep only, where the centre pole gives contact, and a smaller quantity over the remaining surface of the magnet; consequently the quantity of material lifted is uncertain, and it is impossible to keep the stacks or banks in systematic heaps.

The Phœnix Patent Bi-Polar Magnet (fig. 12) has been specially designed for stacking systematically such material as rails, billets, ingots, bars, joists, etc. With this magnet the attractive force is distributed evenly across the pole faces, so that the magnet lifts and deposits the material in single layers without disturbing the underlying material. Further, it will lift single pieces out of a layer without disturbing the surrounding pieces. This systematic stacking is essential in works where large quantities of material are stored, and the use of the magnet must obviously effect great economies in time and labour.

The Phœnix bi-polar magnets can be used in several ways. The single magnet is used for lifting sections of comparatively short length—say up to 12 to 15 ft. To obtain greater lifting capacity with the same material than is given by a single magnet, two, three, or more magnets can be bolted together (see fig. 12, which shows three such magnets). When lifting material in long lengths, such as rails, a pair of single magnets are set at suitable centres and suspended from a lifting beam.

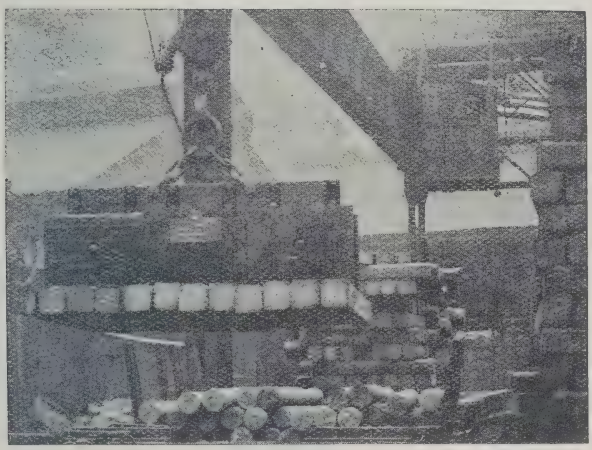


FIG. 12.

As regards the *lifting of hot material*, circular magnets are unsuitable, because they offer such large heating surfaces, and the windings are so close to the heat that there is bound to be overheating. The Phœnix bi-polar magnet offers the minimum surface to the heat, and when constructed for lifting hot material the pole pieces are extended about 8 ins. at the bottom to lift the coils farther away from the heat; the pole faces are then at a distance of 12 ins. from the coils.

The following typical cases are given to show the advantages accruing from the use of magnets:

1. 62-in. diameter circular Phœnix magnet, fitted on a 12-ton electric overhead crane, commanding a scrap storage ground. Weight of magnet about 65 cwts. Current consumption about 50 amperes at 220 volts.

2. A truck load of steel turnings unloaded in seven minutes, work which formerly took four men five hours.

3. Four large furnace boxes, each measuring 6 ft. by 2 ft. 7 ins. by 1 ft. 6 ins. deep, placed side by side on a rail lorry, filled in four lifts averaging 17½ cwts. each.

4. Four furnace boxes of the same size filled with 2½ tons of tube scrap in three lifts.

5. 52-in. diameter circular Phœnix magnet, handling steel turnings from stock heaps into railway trucks. Approximate weight of magnet 50 cwts. Approximate current consumption 40 amperes at 220 volts. The magnet made twelve lifts in seventeen minutes and handled in this time 95½ cwts.—an average of, say, 8 cwts. per lift. Steel turnings are difficult to handle, particularly if rusted together or with long turnings interlaced.

A 52-in. Phœnix magnet has also unloaded from trucks 600 tons of pig iron in 6¾ hours with the crane driver solely controlling the operations, no other labour being employed.

6. A pair of oval-shaped Phœnix magnets for handling combs of pigs. Without the magnets the time occupied to lift a bed of 22 combs was four hours; with one set of magnets the work is done in sixteen minutes.

With the magnets in use, only two or three ground men are employed, whereas hitherto 30 or 40 were required.

Steel may be handled by magnets up to a temperature of about 700° C.

The following table gives the approximate weights and lifting capacities with ingot and pig-iron for various sizes of circular lifting magnets:—

Diameter of Lifting Magnet.	Weight of Magnet.	Safe Load of Solid Steel Ingot.	Approximate Load of Pig Iron per Lift.	Current Consumption.
24 in.	5½ cwt.	5 tons	—	1·75 k.w.
36 ,,	16 ,,	7 ,,	9 cwt.	3·5 ,,
42 ,,	25 ,,	10 ,,	14 ,,	5 ,,
54 ,,	39 ,,	18 ,,	23 ,,	7·5 ,,
66 ,,	60 ,,	30 ,,	40 ,,	12 ,,

MOBILE FINGER MAGNET.

For lifting rails and bars of similar section, Messrs. Allen West & Co., Ltd., have developed a rectangular magnet (fig. 13) along the face of which are four rows of short steel pieces free

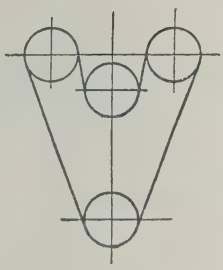
FIG. 13

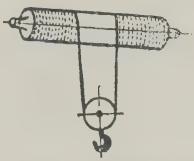
to move and adjust themselves to the contour of the bar to be lifted. Owing to these steel pieces forming a better path for the magnetic lines of force, the lifting capacity of the magnet is increased when dealing with irregular sections.

Vacuum Cup Lifting.

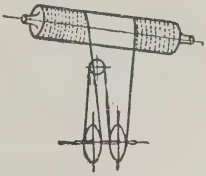
A German firm is using a system of vacuum cups, in place of slings, for handling non-magnetic plates with an overhead crane. A vacuum of 0·1 to 0·5 atmosphere is employed and is provided by a 5-h.p. rotary pump.

VARIOUS ARRANGEMENTS IN LIFTING APPLIANCES.

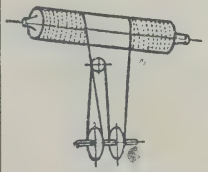




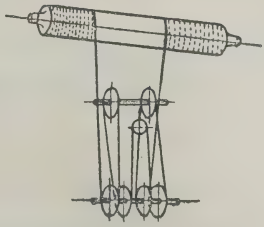
1.—One Bend.



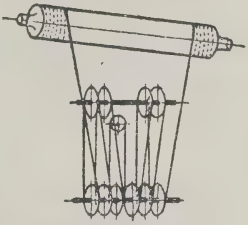
2.—Three Bends.



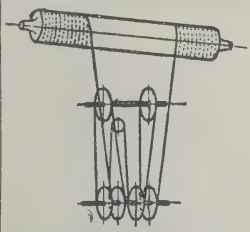
3.—Three Bends, One Reverse.



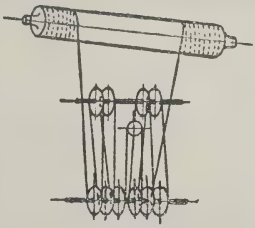
4.—Seven Bends.



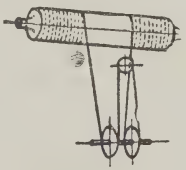
5.—Eleven Bends.



6.—Seven Bends, One Reverse.



7.—Eleven Bends, One Reverse.



8.—Three Bends, Large Bottom Pulleys.

Durability of Wire Ropes.

The two most important conditions appertaining to the manufacture and use of steel wire ropes that affect their durability are:

(a) Quality of material and size of wire.

(b) Diameter of pulleys and arrangement of ropes.

(a) Wire used for lifting ropes is of steel whose ultimate tensile strength varies from 80 to 130 tons per square inch. Ropes of material having a higher tensile strength are of smaller diameter for a given load and factor of safety, but since they are stiffer, larger drums are necessary to obtain the same durability.

The stress in a wire due to bending round a pulley is proportional to:

$$\frac{\text{(Modulus of elasticity)} \times \text{(Diameter of wire)}}{\text{(Diameter of pulley)}}$$

hence large wires are more quickly fatigued than small ones. On the other hand, small wires are more quickly worn through.

For a given ratio of pulley to rope diameter, given rope diameter and load, the life of the rope, if limited by abrasion and not by fatigue, increases as the square of the diameter of the wires of which it is made.

(b) For a given diameter of rope (d), diameter of wire (a) and load (P), the life of the rope depends on the square of the ratio of pulley to rope diameter (m) for large values of the latter.

The above statements have also been deduced from theoretical considerations by Prof. J. T. Nicolson, D.Sc., and may be represented by the formula given by him.

$$\text{Life of rope} = C\frac{m\,(m+1)\,d^2a^2}{P}$$

where the symbols m, d and a have the meanings given above, and C is a constant.

The practice of passing wire ropes on a crab as shown by fig. 14, entailing successively a 90° bend, a 180° reverse bend, and a 90° bend, is a bad one and deteriorates the strand wires;

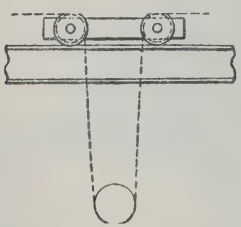


Fig. 14.

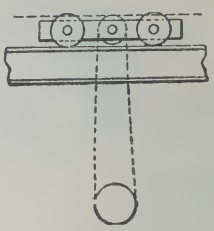


Fig. 15.

a better method is shown by fig. 15, the rope being taken over one upper pulley, down to the block, and up over the second pulley, the bend being always in the same direction, the reverse bend being thus abolished; this arrangement will increase the life of a rope as much as 50 per cent.

The effect of different arrangements of pulleys is shown by Nos. 1 to 7, page 825, and the accompanying tables, case No. 1 being taken as the standard.

COMPARISON OF ANTICIPATED LENGTH OF LIFE OF ROPES ARRANGED AS SHOWN IN NOS. 1 TO 7, PAGE 825.

No.	Number of Bends.	Relative Life of Rope.
1	1	300
2	3	100
3	3*	75
4	7	43
5	11	27
6	7*	37½
7	11*	25

REQUIRED INCREASE IN DIAMETERS OF ROPE DRUMS (MEASURED IN TERMS OF CIRCUMFERENCE OF ROPE) TO GIVE EQUAL DURABILITY.

No.	Increase of Diameter called for by No. 2.
3	1 circumference of rope
4	2½ circumferences of rope
5	4 ,, ,,
6	3 ,, ,,
7	4 ,, ,,

RATIO OF DIAMETER OF PULLEYS AND DRUMS TO CIRCUMFERENCE OF ROPE TO GIVE EQUAL DURABILITY.

No.	Ratio of Pulley and Drum Diameter to Rope Circumference.
1	4 to 1
2	7 to 1 (standard)
3	8 to 1
4	9·5 to 1
5	11 to 1
6	10 to 1
7	11 to 1

RELATIVE ROPE CIRCUMFERENCE ALLOWING FOR SMALLER ROPES DUE TO INCREASED NUMBER OF FALLS.

No.	Number of Falls.	Relative Rope Circumference.
1	2	140
2	4	100 (standard)
3	4	100
4	8	70
5	12	57
6	8	70
7	12	57

† Points to remember in connection with wire ropes :

1. In general, use the lowest grade of steel.
2. In general, use the middle flexibility.
3. Use Lang's lay, if possible.
4. Ratio of pulley to rope diameter should not be less than 26 ; 30 is better.
5. If possible, employ a ratio of pulley to rope diameter of 40, and then keep to the lowest instead of the middle flexibility.
6. In the case of peculiar double turns, use pulleys as large as possible.
7. Avoid unnecessary reverse bends.
8. Keep the ropes well oiled.

* Including one reverse bend which is twice as effective in wearing out the rope.

† Mr. George Hughes, M.I.Mech.E., in Proc. Inst. Mech. Engrs., 1912.

Extra Special Flexible Steel Wire Crane Ropes.

TABLE OF SIX-STRANDED ROPES (37 WIRES PER STRAND).

Circumference.		Diameter.		Approx Weight per Fathom.	Best Patent 80/90 Tons per square inch. Breaking Stress.		Special Improved Patent 90/100 Tons per square inch. Breaking Stress.		Best Plough 100/110 Tons per square inch. Breaking Stress.		Special Improved Plough 110/120 Tons per square inch. Breaking Stress.		Extra Special Improved Plough 120 Tons per square inch. Breaking Stress.	
Ins.	Mm.	Ins.	Mm.	Lbs.	Aggregate Tons.	Actual Tons.	Aggregate Tons.	Actual Tons.	Aggregate Tons.	Actual Tons.	Aggregate Tons.	Actual Tons.	Aggregate Tons.	Actual Tons.
1	25	$\frac{5}{16}$	8	1·08	3·1	2·6	3·5	2·9	3·9	3·2	4·3	3·5	4·4	3·6
1⅛	29	$\frac{3}{8}$	9	1·26	4·2	3·4	4·6	3·8	5·1	4·2	5·6	4·6	5·9	4·9
1¼	32		10	1·50	4·7	3·9	5·2	4·3	5·8	4·8	6·4	5·3	6·5	5·4
1⅜	35	$\frac{7}{16}$	11	1·80	5·7	4·7	6·4	5·3	7·1	5·8	7·8	6·4	8·1	6·7
1½	38		12	2·16	7·0	5·8	7·8	6·4	8·6	7·1	9·4	7·7	9·8	8·1
1⅝	41	$\frac{1}{2}$	13	2·58	8·3	6·8	9·3	7·7	10·2	8·4	11·2	9·2	11·7	9·6
1¾	44	$\frac{9}{16}$	14	3·00	9·7	8·0	10·9	9·0	12·0	9·9	13·2	10·9	13·7	11·3
2	51	$\frac{5}{8}$	16	3·96	13·0	10·7	14·5	11·9	16·0	13·2	17·5	14·4	18·3	15·1
2¼	54	$\frac{11}{16}$	17	4·44	14·7	12·1	16·5	13·6	18·2	15·0	20·0	16·5	20·8	17·1
2¼	57		18	5·04	16·7	13·8	18·6	15·3	20·6	17·0	22·5	18·6	23·5	19·4
2⅜	60	$\frac{3}{4}$	19	5·52	17·6	14·5	19·7	16·2	21·8	18·0	23·9	19·7	24·9	20·5
2½	63	$\frac{13}{16}$	20	6·12	19·7	16·2	22·0	18·1	24·4	20·1	26·7	22·0	27·8	22·9
2¾	70	$\frac{7}{8}$	22	7·38	24·2	19·9	27·1	22·3	29·9	24·7	32·8	27·0	34·2	28·2
3	76	$\frac{15}{16}$	24	8·70	29·2	24·1	32·6	26·9	36·1	29·8	39·5	32·6	41·2	34·0
3⅛	79	1	25	9·54	31·8	26·2	35·6	29·4	39·3	32·4	43·1	35·5	44·9	37·0
3¼	83		26	10·32	33·2	27·4	37·1	30·6	41·0	33·8	44·9	37·0	46·9	38·7
3⅜	86	$1\frac{1}{16}$	27	11·10	36·0	29·7	40·3	33·2	44·5	36·7	48·8	40·2	50·9	42·0
3½	89	$1\frac{1}{8}$	28	11·88	39·0	32·2	43·6	36·0	48·1	39·7	52·7	43·5	55·0	45·4
3¾	95		30	13·80	45·2	37·3	50·5	41·7	55·8	46·0	61·2	50·5	63·8	52·6
3¾	98	1¼	31	14·52	48·5	40·0	54·2	44·7	59·9	49·4	65·6	54·1	68·5	56·5
4	102		32	15·78	51·9	42·8	58·0	47·8	64·1	52·9	70·2	57·9	73·3	60·5
4¼	108		34	17·70	59·0	48·7	66·0	54·4	72·9	60·1	79·9	65·9	83·4	68·8
4⅜	111	1⅜	35	18·66	60·9	50·2	68·1	56·2	75·2	62·0	82·4	68·0	86·0	70·9
4½	114		36	19·98	64·7	53·4	72·3	59·6	79·9	65·9	87·6	72·3	91·4	75·4
4¾	121	1½	38	22·14	72·7	60·0	81·2	67·0	89·8	74·1	98·3	81·1	102·6	84·6
5	127		40	24·54	81·1	66·9	90·6	74·7	100·2	82·6	109·7	90·5	114·5	94·4
5⅛	130	1⅝	41	26·04	85·5	70·5	95·5	78·8	105·6	87·1	115·7	95·4	120·7	99·6
5¼	133		42	27·18	90·0	74·2	100·6	83·0	111·1	91·6	121·7	100·4	127·0	104·8
5½	140	1¾	44	29·58	96·9	79·9	108·3	89·3	119·8	98·8	131·2	108·2	136·9	112·9
5¾	146		46	32·28	106·6	87·9	119·2	98·3	131·7	108·6	144·3	119·0	150·5	124·2
6	152	1⅞	48	35·40	116·8	96·3	130·5	107·6	144·3	119·0	158·0	130·3	164·9	136·0
6¼	159	2	50	37·92	124·7	102·9	139·4	115·0	154·0	127·0	168·7	139·2	176·0	145·2
6½	165		52	40·92	135·7	111·9	151·6	125·1	167·6	138·3	183·5	151·4	191·5	158·0

LIFTS.

Notes on Lifts.

The usual height of a car from the floor to the top of the cross-head is 10 ft., and allowing the over-travel space or head room of 3 ft., this indicates that the minimum height to the underside of the floor of the engine or gear room from the top landing is 13 ft. The thickness of the floor depends upon the weight to be carried, but will not be less than 6 ins. When the engine is in the basement, 4 ft. is probably sufficient, but 6 ft. is desirable for the guide sheaves at the top of the shaft. Eight feet clear height should be regarded as the minimum for the engine room, whether it be situated in the basement or at the top of the shaft, and 9 ft. 6 ins. is desirable for the larger machines. Wherever possible a steel joist and lifting tackle should be installed in the engine-room to facilitate repairs when two or more lifts are provided.

The weight on the engine-room floor when the engine is located at the top of the shaft is, for a speed of 50 ft. per minute, approximately as follows, exclusive of the controller and cables :—

Car Capacity. *Passengers.*	Dead Load. *Tons.*
6	2·6
9	3·4
12	4·6
15	5·5
18	6·5

An allowance of 2 sq. ft. per passenger and 4 sq. ft. for the attendant (to enable him to handle the controller, car gate, and landing doors rapidly) is the usual basis for figuring car capacity ; for instance, a car having a floor area of 6 ft. by 5 ft. = 30 sq. ft. would have a passenger capacity of 30 less 4 = 26, which divided by 2 gives 13 passengers.

Similarly, the weight of the average individual has been ascertained to be 140 to 150 lbs. and, therefore, the assumed load is figured at 75 lbs. per sq. ft. of floor area.

Convenient car sizes are as follows :—

Depth.	Width.	Area.	Passengers.	Attendant.	Total.	Live Load.
ft. in.	ft.	sq. ft.				lbs.
6 0	7	42	19	1	20	3,150
5 0	6	30	13	1	14	2,250
4 4	6	26	11	1	12	1,950
4 0	5	20	8	1	9	1,500
3 0	4	12	4	1	5	900

In modern office buildings up to ten floors a very useful size for the car is 5 ft. wide by 4 ft. 4 ins. deep (*i.e.* 21·5 ft. super., carrying eight to nine passengers). Large departmental stores of the Oxford Street, Regent Street, and Kensington type generally require a larger car, and in this case the size is frequently 8 ft. wide by 6 ft. deep, giving a floor area of 48 sq. ft., and designed for a maximum load of 3,600 lbs. With these exceptions a 6 ft. by 6 ft. car is generally regarded as the largest standard car, rated for sixteen passengers and a gross load of 2,700 lbs.

Details of Tee Guide Rails.

Lbs. per foot run.	Width of Base (ins.).	Height of Web (ins.).	Width of Web (ins.).	Remarks.
$6\frac{1}{2}$	$2\frac{3}{4}$	$1\frac{5}{16}$	$\frac{21}{32}$	Counterweight Car
14	$4\frac{1}{2}$	$3\frac{7}{16}$	$\frac{21}{32}$	
21	$5\frac{1}{16}$	$3\frac{17}{32}$	$\frac{5}{8}$	
39·4	6	$5\frac{3}{16}$	1	

Guide rails, which are either cold-rolled or planed steel, are usually supplied in 14 ft. to 16 ft. lengths. Alignment is secured by groove-and-tongue joints and a splice plate bolted at the

back. Splice plates for car guides are frequently specified to be not less than 12 ins. long, four ⅝-in. bolts being required for each end of the guide. For counterweight guides, which are not called upon to stand heavy strains from the safety gear, 9-in. plates and ½-in. bolts are accepted. In cases where the supports on wall fixings are at a greater distance apart than 12 ft., angle iron or channel iron stiffeners are used, which run about 9 to 10 lb. per foot.

Guide grips should have a minimum clearance of ⅛ in. to avoid the risk of their being brought into action accidentally.

Usually the empty car is overbalanced to the extent of 40 per cent. of the maximum passenger capacity of the car, which has been shown to be the average load if the equipment is well designed. Careful balancing not only has a marked effect on the consumption of electrical energy, but also on the speed of an empty descending car and proper upward stopping distance.

As a very rough guide, the following figures are given for speeds :—

Travel of Car.	Load Speed.
30 to 40 ft. . . .	150 to 200 ft. per min.
45 to 70 ft. . . .	180 to 250 ft. per min.
75 to 100 ft. . . .	222 to 300 ft. per min.
100 ft. and over . .	300 to 400 ft. per min.

Due allowance must, of course, be made for car size, distance of average non-stop run, etc. In New York City, 500 ft. per min. is the legal limit of speed for local elevators and 700 ft. per min. for expresses, *i.e.* when there is a clear non-stop run of 80 ft.

Over-travel spaces at both the top and the bottom of the shaft are essential, as the brake is subject to wear and cannot be relied upon for accurate floor levelling, especially with high-speed traction machines.

The depth of the pit or sump—*i.e.* the over-travel at the bottom of the shaft—should be 3 ft. 6 ins., measured from the lowest landing. For higher speeds the space must be increased to 4 ft. 6 ins., and for traction-drive express lifts 7 ft. 6 ins. may be required. As an additional precaution against jar or shock, should the brake slip, spring or oil buffers are used under the car and also under the counterweight. The cost of a spring-loaded set, of a simple character, is small, and the installation is strongly recommended.

A clear space of 3 ft. is usually the minimum required between the top of the crosshead of the car and the underside of the shaft ceiling when the car is at its top landing. For speeds above 350 ft. per minute, 5 ft. is regarded as the minimum space.

All footings for columns and for foundations adjoining lift shafts should be carried below the pit in order to ensure that all sides of the pit are true and plumb with the finished sides of the shaft. Similarly, off-sets on the inner sides of the shaft walls should be avoided wherever possible, since the working forces of both the car and counter-balance weight guide rails must be truly vertical, and any inaccuracy in the builders' work will involve considerable additional expenditure for erection of the lifts.

A clear space of not less than 3 ft. should be provided between the bottom of the shaft and the lowest point of the underside of the car floor when the car is standing at its lowest landing, and between the top of the cross-head of the car and the underside of the ceiling of the shaft when the car is at the top landing. For express lifts these over-travel spaces must be increased up to a maximum of 7 ft. for the highest speeds, and even for slow-speed alternating-current lifts 4 ft. is desirable.

Collapsible gates are not desirable for brick shafts on account of the added fire risk, due to flue effect, and also because of draughts. The type of door should be that best adapted to the particular type of car to be installed, and the width of the opening should be such that ingress and egress will be as free as possible—usually 66 per cent. of the width of the car, in order to provide standing space for the attendant. Adequate allowance should be made for floor projections into the shaft, which vary with the type of door adopted, viz. combination slide and swing single slide; double doors sliding in the same direction; double doors sliding in opposite directions; double swing doors, etc. Wired glass panels should be provided in all landing doors to avoid accidents when opened quickly. The standard height of landing doors is 7 ft., and the minimum width should be 2 ft. 6 ins.

(*'The Builder,' Jan. 6, April 7, 21, 1922—Abstracts.*)

ELECTRIC LIFTS.

Typical Specification for Electric Lift.

The following are the main particulars of the lift :—

Extreme travel . . 38 ft. 9 ins. Speed . . . . 120 ft. per min.
Maximum load to be carried, 10 cwt. Inside dimensions of cage, 3 ft. 6 ins. × 3 ft. × 7 ft. high.

Position for Motor, etc.—The motor, etc., to be placed in the basement in the position shown in the plan.

If fixed overhead, the driving sheave or drum should be designed, if possible, to span correctly from the centre of cage to centre of weight, so as to save the use of diverting sheaves.

The overhead sheaves are to be supported on suitable rolled steel joists, to be supplied and put in position by the contractor.

Cage.—The cage walls are to be made of well-seasoned pine, with a substantial iron or steel framework, and adjustable guides at the top and bottom to work on the runners. No seat will be required. A Bostwick or other approved sliding gate to be provided at the cage entrance.

The construction of the cage may be either (*a*) framed with grooved and tongued panels, the woodwork forming the panels being put in diagonally, and the whole bolted to the iron framework of the cage ; or (*b*) of grooved and tongued planks on the flat bolted to the iron framework of the cage with an additional strengthening iron ; or (*c*) of an approved type. If method (*a*) is adopted, the framing must not be of less than 1½ in. stuff, and the panels not less than ⅞ in. ; if (*b*) the planks must be uniformly 1 in. thick. The floor of the cage must be of maple, 2 ins. thick. All the above dimensions are finished sizes.

An opening to be provided in the top to allow for the lighting of the cage by daylight. This opening is to be covered over and under with iron wire trellis of 1 in. mesh, of not less than No. 13 S.W.G. The wirework to be galvanised after it is made up.

The interior of the cage, with the exception of the floor, to be varnished with three coats of good oak varnish.

The cage to be fitted with an electric light fitting of the bulkhead or other approved type, with genuine enamel reflector. A bayonet collar lamp-holder, lamp, cut-out and tumbler switch for this fitting, and a length of flexible cable for making connection to a ceiling rose fixed about half-way down the lift-well, to be provided and fixed by the contractor.

Electric Bell.—Suitable bell pushes are to be provided and fixed on each landing, and connected to an electric bell, to be provided and fixed in and about midway up the lift-well, and to an indicator for each floor, to be provided and fixed in the lift cage. A battery for working the bell is to be provided.

The whole of the bell work, including the necessary wiring, is to be done in the most approved manner.

Gates for Cage and Openings in Lift Well.—Collapsible gates, with spring-catch lock of the Bostwick, B. and S., or other approved pattern, are to be provided and fixed to the lift-well openings on each floor, to the lift cage and motor enclosure. Six gates in all will be required. The locks and gates (except in the case where push buttons are required to be fitted) to be so constructed that they cannot be opened except by use of the authorised key.

The gates, which are to be carefully finished black, are required to open the width of the cage opening, and to reach to the top of the cage opening.

Lift-Well Enclosure and Wirework Guards.—A wirework guard is to be fixed to the gate uprights, and continued round the lift-well above and below the hand-rail from the basement to the second floor at a height of 7 ft. from the stair treads. Suitable supports for the attachment of the wire guard are to be provided, and care is to be taken to prevent the guard or its supports from interfering with the free use of the stairs hand-rail. The guard is to be over and under iron trellis of 1 in. mesh, not less than No. 13 S.W.G., and is to be finished black, and stiffened where necessary by angle iron or tubing.

A wirework screen with suitable channel iron supporting frame, at least 7 ft. high, with Bostwick gate, to be provided in the basement for the protection of the motor and gear.

*Carrying Capacity and Speed.**—The cage to carry a maximum load of 10 cwt., and to travel at a speed of 120 ft. per minute when fully loaded. The maximum variation of speed ascending or descending at any other load must not exceed 10 per cent.

* In the case of passenger lift with car switch operation, the speed is now from 200 to 300 feet per minute.

Safety Devices.—The most improved system of safety devices is required. A full description of all safety devices to be employed is to accompany tender. They must include the following :—(*a*) The locking of the cage on the guides or guide backings if one or more of the suspending ropes break or slacken ; (*b*) automatic stopping of the motor when the cage is brought to rest from any cause, unless the V wheel drive is used ; (*c*) automatic stopping of the cage before it reaches the top or bottom of the lift well.

Painting.—The whole of the materials fixed and supplied by the contractor, with the exception of the electric light and bell fittings, cables and resistances, bright metal work, folding gates, and the inside of the cage, which is to be varnished, is to be painted with three coats of oil paint, the finishing coat to be of a colour to be selected. The wood work, including all portions of the cage which are not visible, is to be knotted and primed before being painted.

Lubrication and Contact Cleaning.—The whole of the plant is required to work automatically and to require no attention beyond occasional lubrication, and occasional cleaning of the contacts.

Factor of Safety.—The factor of safety for the suspending ropes is to be not less than ten, and for all other parts a factor of safety of at least five is to be provided.

Precautions against Fire.—The contractor is required to make such arrangements as will prevent an outbreak of fire should the armature or any other part of the apparatus fuse or become excessively hot.

Quality of Material.—The material and workmanship to be of the best possible description throughout.

Tests.—Immediately the lift has been erected, the contractor will be required to make the following tests to the satisfaction of the inspecting officer :—

1. The cage to be loaded until the weight on the ropes is twice the combined weight of the cage and specified maximum load. The gear and suspending ropes must bear this weight for 30 minutes without showing any sign of weakness, temporary set, or permanent elongation of the rope strands.

2. Tests to determine if the specified speed is complied with.

3. Tests to determine if the various safety devices comply with the conditions of the contract. These are to include : (*a*) Starting the cage up and down and letting it run to the end of its journey without attention ; (*b*) slackening one or more as required of the suspending ropes.

4. Tests to determine the suitability of the switching gear.

The contractor is required to provide a man to start the lift and to attend to it for one week from the time when it is accepted as running satisfactorily.

Leading makers such as *Smith, Major & Stevens, Ltd.*, use the following speeds on their lift motors, for continuous current :—

Up to 3 h.p. . . .	1,000 revs.	Up to 7½–10 h.p. . .	650–750 revs.
„ 5 „ . . .	930 „	„ 15 „ . .	500–700 „

For single phase, 50 periods, 6 pole machines usually.

For 2 and 3 phase, 50 periods, 6 pole up to 10 h.p., and 8 pole above.

Ropes for Lifts.

See British Standard Specification for Round Strand Steel Wire Ropes for Lifts and Hoists No. 329, 1928.

ELECTRIC WINDING.

The Ward Leonard System.

(English Electric Co., Ltd.)

This system (fig. 16) is based on the fact that the speed of a separately excited D.C. motor is directly proportional to the voltage applied to the armature. If, therefore, the D.C. motor, 1, is used to drive the drum D, it can be made to run at any desired speed and in either direction of rotation, by varying the voltage and direction of the current impressed on the armature. This D.C. power is obtained from a separate generator, 2, known as the variable voltage generator, and driven at constant speed by a motor, 3, connected to the supply mains. The variable voltage generator and its driving motor are coupled together and mounted on a common bedplate, which also carries, on the same shaft as the other two machines, a small D.C. generator. This generator supplies the excitation current to the variable voltage generator and the winding motor, the triple combination being known as the converter set.

A regulator, 5, is used to vary the field of the V.V. generator and, consequently, its voltage between 0 and a ± maximum, and in this way the speed of the winding motor to which this voltage is applied can also be varied between zero and ± maximum.

The principal feature of the control is that the speed and direction of rotation of the winding motor are determined solely by the regulator or, in other words, by the position of the driver's control lever. For all practical purposes this holds true whatever the load in the cage may be, and whether the load is being raised or lowered. When in a running position—by moving the control lever towards the 'off' position—a strong electrical braking effect is obtained, as the motor then acts as a generator and returns energy through the converter set to the line. The same electrical braking effect can be obtained when lowering loads, and the low speeds required for inspection tests can be readily obtained by a slight movement of the lever from the 'off' position.

It should be noted that the controller only deals with the field of the V.V. generator (about 2 to 3 per cent. of the output), consequently, it is very small as compared with an A.C. controller, and can be readily operated even on the largest winders without the use of auxiliary gear.

As any given position of the control lever corresponds to one speed only, and one direction of rotation, it is possible to provide simple safety appliances of absolute reliability.

In Fig. 17 is shown diagrammatically the system used by the *English Electric Co.* for the automatic control of the acceleration and retardation.

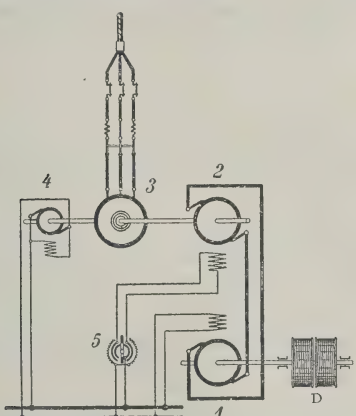


FIG. 16.

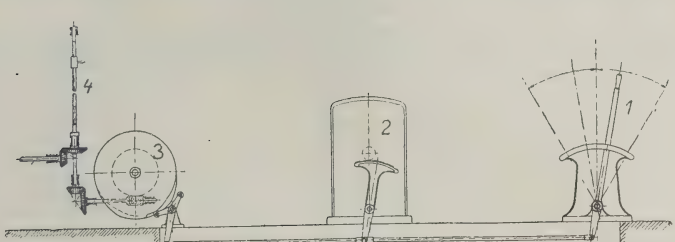


FIG. 17.—Diagram of Depth Indicator with Safety Device as used by the English Electric Co., Ltd.
(1) Operating Lever. (2) Controller. (3) Control Cam. (4) Depth Indicator.

The spindle of the depth indicator drives two cam wheels, one for each cage, through gearing proportioned so that the wheels make nearly one complete revolution. Each wheel has two cams, one for acceleration and one for deceleration. When starting, the control lever can only be moved from its mid position at a rate determined by the shape of the cams, which shape is designed to give the predetermined rate of acceleration. This rate cannot be exceeded. When retardation should begin, if the driver has not already started to bring back his lever, the deceleration cam gradually forces the lever back to the 'off' position. Acceleration, retardation, and speed of wind are thus within the driver's control, provided he does not attempt to exceed the limits for which the cams are designed.

In the event of an overspeed, electric braking automatically takes place. The winding motor will act as a generator and supply power to the V.V. generator, which will then act as a motor driving the set above synchronous speed, causing the induction motor to act as an asynchronous generator and pay back power to the line. Should the speed of the set exceed a predetermined maximum, a centrifugal switch driven from the shaft of the set will open the circuit of the brake magnet and apply the emergency brake.

AERIAL ROPEWAYS.

Aerial Ropeways may be divided into two fundamental types:—The SINGLE ROPE or MONOCABLE, and the DOUBLE ROPE or BI-CABLE. In the first, a single endless rope is used both to support and move the loads. In the second, there are one or more supporting ropes and an endless hauling rope. Adapted to these types are the 'Fixed or Non-detachable Clip System,' on the Single Rope Type, and the 'Jig-back' or 'To-and-fro System,' which is applicable to either type.

There is considerable divergence of opinion upon the merits and demerits of the two systems, which is due largely to the fact that a number of plants have been installed without giving due consideration to which type is most applicable to the nature of ground to be traversed, and the duty to be performed.

THE SINGLE ROPE OR MONO-CABLE SYSTEM.

Ropes.

By experience, it has been found the most suitable form of rope on an average length line is a 6·7 Lang's lay, with a specially hard hemp core. On short lines when the rope is subject to a great number of bends per hour around the terminal sheaves, compound lays are sometimes used. Wire cored ropes are also occasionally employed, owing to the fact that they tend to stretch less than those with hemp cores, but they have been found by experience not to have such a long life.

On a well-constructed line, depreciation on rope is covered by ·125 pence to ·3 pence per ton mile carried, on the basis of a line not less than 1,000 yards long, and a capacity of over 25 tons per hour; as the length and capacity increases, the depreciation per ton mile carried decreases. The fact that on a single rope system the entire length of the rope is constantly passing under observation at the terminals renders it very safe against failure, and it can be worked until a comparatively low factor of safety is reached. The diameter and breaking strain of ropes can be obtained from the formula given on page 839, under 'Diameter of Hauling Ropes.' No rule can be given, however, for finding initial tension, as this is dependent upon driving power required and spans to be negotiated, etc., apart from capacity of line and so forth.

Loads.

When possible, individual loads should be kept to about 1 per cent. of the hourly capacity of a line, for the purpose of distributing the weight on spans. Heavy individual loads tend to increase prime costs.

Carriers and Clips.

Among the best known are Roe's Patent Toggle and Saddle Clips.

The former, fig. 18, is capable of sustaining a load at an angle of over 45°, and in theory is the most efficient of all clips actuated by the load, as the whole of the tare weight also is used

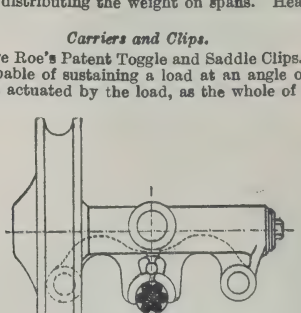


FIG. 18.

for gripping. In practice, however, the saddle clip, fig. 19, is found preferable and is usually adopted. The gripping in this type is performed by two cast chilled saddles, the sides of which are inclined, one side having a feather projection which engages into the lay of the rope, this projection being so formed that it disengages itself without damaging the strands when the carrier enters a station. The ruling grade on which this type of clip can be safely worked is 1 in 2¼. The tare weight of clip, carrier, and bucket is usually about one-third of the net load.

Fig. 20 shows the 'Roe Balance Beam.' The arrangement is used for distributing the pressure equally on a series of sheaves; these are arranged in groups of 2, 4, and 8. The introduction of a number of balanced sheaves in series greatly facilitates the working of long spans, and reduces wear and tear of rope where heavy pressure on any trestle has to be sustained.

A fair speed for the rope is 400 ft. per minute, though under certain conditions it can be run up to 550 ft. per minute, when no automatic return or dumping is required. The factor that rules the speed is the velocity at which the loads can be brought up to for engaging

with the rope, bearing in mind that the load and rope velocity should synchronise at the point of attachment.

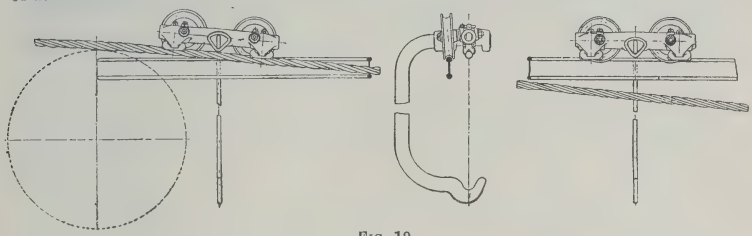


FIG. 19.

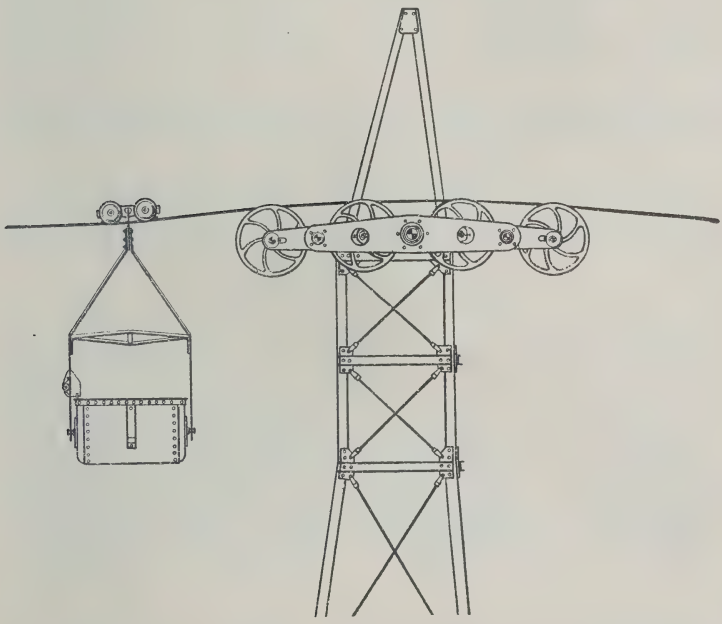


FIG. 20.

Power.

Any available power can be used as long as it is steady. Trouble, however, may be caused by using intermittent power, as this gives rise to oscillation of the rope in long spans. On automatic lines, the surplus power is preferably absorbed by means of a fan or turbine. When possible, the power should be placed at the upper end of a line, and the tension at the lower terminal. In long lines that have to be sectionalised, the driving or brake station, as the case may be, is sometimes placed between two sections, and the tension arrangements at the ends.

Tension.

To ensure a constant and even tension, it is essential that the counterweights should be always free and floating, and owing to the amount of stretch in hemp cored ropes, taking-up gear must also be allowed for. Fig. 21 shows an arrangement which combines a multiplying and taking-up device. The two sheaves at A are attached to the tension trolley, the winch and differential drum being mounted on an independent frame placed over the tension

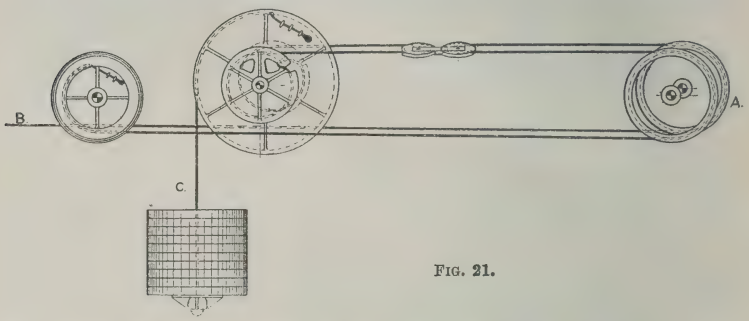


FIG. 21.

pit; one end of the rope B is anchored, the other being made fast to the tension winch drum. By this arrangement the resultant pull at A is four times the dead weight applied at C, and the weights can always be kept in suspension without the necessity of a deep pit, by taking in the slack with the winch.

Spans.

The distance between standards is dependent on the configuration of the ground, but over undulating country the average distance is approximately 330 ft. for lines of up to an hourly capacity of 60 tons. On a ropeway over the Andes, having a total length of 46 miles, several spans are being worked of 2,800 ft., though the longest recorded span on the single rope system is 3,600 ft.

DOUBLE ROPE OR BI-CABLE SYSTEM.*

Carrying Ropes.

Carrying ropes may be divided into two main classes, 'spiral,' fig. 22, and 'locked coil,' fig. 23. The main advantage of the spiral type is its cheapness, and in lines transporting a small tonnage

FIG. 22.

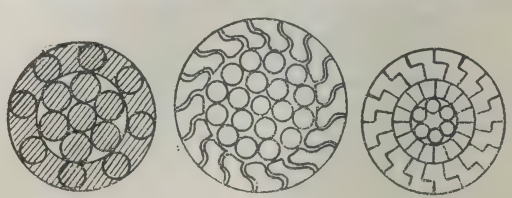


FIG. 23.

per hour it will generally answer its purpose. Ropes of this type should consist of 19 wires in ropes up to 1 in. diameter and of 37 wires in ropes up to 1½ in. diameter. With this latter the individual wires have a diameter of about ·205 in., which is about the limit to ensure perfect homogeneity of the material. Above 1½ in. diameter, locked coil ropes should invariably be used.

Locked coil ropes, fig. 23, are of two types, the special locked, *a*, and the full locked, *b* and *c*. The former consists of a spiral core surrounded by a layer of round wires kept apart by wires

* Abstract from 'The Engineer,' July 23, 1915

drawn to a special section. This rope is sold at a medium price between plain spiral rope and full locked coil. The surface of the special locked rope is not perfectly smooth, small interstices occurring between the wires. The full locked coil, on the other hand, presents a perfectly cylindrical surface to the passage of the cars. The second type of full locked construction—shown at *c*—is particularly useful for very heavy carriers, as, owing to the rope having an inner layer of tapered section wires, the crushing and distorting effect of the load when passing over the saddles on which the rope is supported is minimised. In addition to the great advantage of offering a smooth surface to the wheels of the cars (with the result that the wires may be of a lower breaking stress than with spiral ropes), the locked coil ropes possess the great advantage that, should by any chance one of the individual wires break, it is held in place by the neighbouring wires, and does not spring from the texture of the rope, as is the case with the spiral construction. This springing of the wires is often the cause of accidents, as the cars run against the outstanding wire, and are thrown off the line.

Fixture of Ropes.

The ropes are anchored at one end of their length, and some form of tensioning device applied at the other end. This generally takes the form of a hanging weight, which may generally be from one-quarter to one-fifth of the total breaking strength of the rope. A factor of safety of 4·5 is a good medium. It has been proved that for spiral ropes a breaking stress of 95 tons per square inch, and for the locked coil ropes 65 tons per square inch, give the best results. When calculating the tension weight required, the fact must be taken into account, that should the weight be situated on a lower level than the anchorage, the weight of the rope itself acts as a tensioning factor.

For fixing the ends of the rope a satisfactory arrangement is to employ the ring wedge coupling. In this anchorage—shown in section in fig. 24—the single wires are kept apart by small nail-like

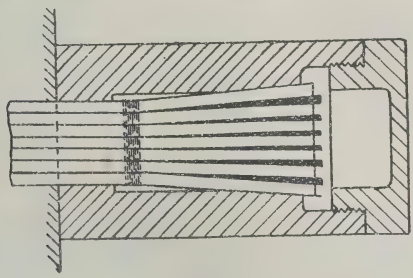


FIG. 24.

wedges, and the layers by means of ring wedges made in halves. These wedges are driven in and then a cap is screwed home, firmly locking them in place and entirely preventing any chance of the wires pulling out.

The same form of coupling, only made tapered at both ends, fig. 25, may be used for joining the various lengths of rope along the line.

FIG. 25.

The distance apart of the tensioning weights varies according to the carrying capacity of the line, weight of carriers, number of long spans occurring, etc. A good rule is not to exceed a length of 1½ mile between anchorage and tensioning device, although on level lines, with a small capacity, distances up to 2 miles may be taken.

Another method of tensioning is by means of buffer springs at either end of the rope, but this system is only applicable to short lines with light loads.

Diameter of Carrying Ropes.

It has been proved by experience that most trouble is caused by breaking due to bending near the saddles, and for this reason in order to obtain the diameter of the carrying rope

required, many firms employ formulæ based on stresses due to bending. A formula often used is :—

$$\text{Weight per metre of ropes} = \frac{0{\cdot}85}{100}\left\{(\text{weight of carrier}) + \Big((\text{distance between carriers in metres}) \times (\text{weight of traction rope per metre}).\Big)\right\}$$

This is for spiral construction ropes. With locked coil ropes the factor 0·85 becomes 1·05. This works well for small lines up to about 50 tons capacity hourly. For each additional 5 tons, 0·05 should be added to the factor 0·85. The following table shows the various diameters of a special locked rope for various loads and traction ropes :—

(Two-Wheeled Carriages.)

Dia. of Hauling Rope.	Weight of Loads in Cwts.									
	3	4	5	6	8	10	16	20	24	28
In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	Ins.
$\frac{1}{2}$	$1\frac{3}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1	$1\frac{3}{16}$	$1\frac{5}{16}$	—	—	—	—
$\frac{9}{16}$	$\frac{13}{16}$	$\frac{15}{16}$	$\frac{15}{16}$	$1\frac{1}{16}$	$1\frac{3}{16}$	$1\frac{1}{4}$	—	—	—	—
$\frac{5}{8}$	$\frac{7}{8}$	$\frac{15}{16}$	1	$1\frac{1}{16}$	$1\frac{3}{16}$	$1\frac{1}{2}$	$1\frac{1}{2}$	—	—	—
$\frac{11}{16}$	$\frac{15}{16}$	$\frac{15}{16}$	$1\frac{1}{16}$	$1\frac{1}{16}$	$1\frac{3}{16}$	$1\frac{1}{4}$	$1\frac{9}{16}$	—	—	—
$\frac{3}{4}$	$\frac{15}{16}$	$1\frac{1}{16}$	$1\frac{1}{16}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{5}{16}$	$1\frac{9}{16}$	$1\frac{11}{16}$	$1\frac{13}{16}$	—
$\frac{13}{16}$	—	$1\frac{1}{16}$	$1\frac{1}{8}$	$1\frac{3}{16}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{13}{16}$	$1\frac{3}{4}$	$1\frac{7}{8}$	2
$\frac{7}{8}$	—	—	$1\frac{3}{16}$	$1\frac{3}{16}$	$1\frac{5}{16}$	$1\frac{3}{8}$	$1\frac{3}{4}$	$1\frac{7}{8}$	$1\frac{7}{8}$	2

Design of Ropeway.

To lay out the profile for a ropeway, when the line is laid out on paper, the curve of the rope may be obtained and set out from the formula—

$$\text{Dip} = L^2W/8T,$$

where,

L = span ; W = weight per unit length ; 8 = constant ; T = tension,

and then the supporting trestles placed accordingly. It sometimes happens, however, in practice that when the rope is fully tensioned it 'floats' or rises up from the supporting tower, owing perhaps to a minute error in the drawn profile. To avoid this, some Continental makers employ a so-called factor of safety for the dip—that is, the denominator 8T is multiplied by a constant —the formula giving, as a result, a dip smaller than the actual. It will be obvious that if the curve with the dip so obtained is set out and the trestles drawn up to it, they will be slightly higher than need actually be the case, thus avoiding any possibility of 'floating' when the line is erected. This factor of safety may be 1·5 for spiral ropes and 1·1 for locked coil ropes.

There has been of late years an increasing tendency to employ saddles (upon which the rope is supported at the trestles) which are pivoted at the centre, and can consequently adapt themselves to the varying gradients of the rope, and there is little doubt that this type when properly designed, and well greased to allow of play in the rope, gives full satisfaction. In designing the shoes a good medium value for the pressure of the rope is 250 lbs. pressure per square inch of projected area of the rope. This is for locked coil ropes ; for spiral ropes a little less should be taken.

It sometimes happens that at a trestle, on either side of which are long free spans, there is a heavy downward reaction of the hauling rope, which, when the carrier passes over the saddle, produces a heavier pressure on the rope than that for which it was calculated. In this case the rope is protected by means of a steel plate cap over which the carrier runs, thereby removing the pressure from the rope. There is a difference of opinion as to when these caps should be employed, but it may safely be said that their use is to be recommended where the hauling rope reaction exceeds 450 lbs.

Hauling Ropes.

The *traction* or *hauling rope*, by means of which the carriers are propelled, is invariably a stranded rope with a hemp core, the flexibility depending on the size of the individual wires in the strands and the breaking stress of the material. This rope is endless and in constant motion. In the general case it passes round a driving sheave at one end of the line and round a surging or tension sheave at the other end. In some cases this plan may be varied, as, for instance, when the carriers are required to pass automatically round a large sheave at one end station. In this case no manual attention is required at this station and the driving and tensioning sheaves are combined in the station at the opposite end of the line. Whenever possible

the driving should take place at the upper and the tensioning at the lower end of the line. In this way the accumulative tension in the rope due to its own weight is utilised as a tensioning agent, whereas, were the tensioning to take place at the upper end, an additional amount of weight would be required to counterbalance the weight of the rope. The amount of the tensioning weight should be such as to give sufficient grip round the driving sheave to ensure perfect driving when the full load is on the line.

Diameter of Hauling Ropes.

The diameter of the hauling rope depends upon the highest tension occurring in it.

$$\text{Tension of rope in lbs.} = \left\{\text{Initial tension} + \left(\frac{W}{S} \times H\right) + (w \times H)\right\} + \left\{(W \times \mu_1 \times N) + (L \times \mu_2 \times w) + (R \times 2)\right\},$$

where,

W = weight of full carrier; S = spacing of carriers in feet; H = difference in level between stations in feet; w = weight per foot of traction rope; μ_1 = coefficient of friction of cars; μ_2 = coefficient of friction of traction rope; L = length of line in feet; N = number of carriers on one side of line; R = number of supporting rollers on one side of line.

This tension if multiplied by the factor of safety—6 to 8—and divided by the breaking stress, gives the area of the rope necessary. This is for the general case with the drive at the upper terminal and the loads ascending. When the loads are descending, the plus sign in the centre of the formula becomes minus. To obtain the tension when the ropeway is just put into work the second half of the formula should be doubled.

The usual breaking stresses of traction ropes are from 80 to 115 tons per square inch. The surge of the hauling rope tension sheave depends upon the distance apart of the carriers, whether long spans occur in the line, etc. As an average, 6 ft. play may be allowed for to commence with, and then an additional 12 ft. for each mile of length. In some cases in long lines it is necessary to split up the line into sections to avoid excessive tensions occurring.

Driving Power.

$$\text{Power in ft. lbs.} = \left\{\left(W_1 \times W_2 \times \mu_1 \times \frac{N}{2} \times V\right) + (w \times 2L \times \mu_2 \times V) + (2R \times V) + (F)\right\} + \text{or} - \text{theoretical power},$$

where,

W_1 = weight of full carrier; W_2 = weight of empty carrier; μ_1 = coefficient of friction for cars; N = total number of cars on line; V = velocity in feet per minute of cars; w = weight per foot of traction rope; L = length of line in feet; μ_2 = coefficient of friction for traction rope; R = number of supporting pulleys; F = power required for station friction.

The first four factors give the power necessary to overcome friction, and the theoretical power required or developed depends upon whether the loads are ascending or descending. The factor for station friction varies greatly for different lines, and is solely a matter of experience, although 0·02 of the total weight of the moving parts on the stations multiplied by the speed may be taken as a good mean.

Carriers.

There are two kinds of carriers in general use, those employing screw grips and those with lever grips. These may again be divided into sections, the overtype and the undertype grips. In the former, the hauling rope is attached to the carrier or running head at a point above, and in the latter at a point below the carrying rope.

The main advantage of the overtype grip is that the length of the stations is shorter than with the undertype grip, as the hauling rope may be led in and out of the jaws without interfering with the shunt rails. In the case of lever grips the overtype is generally the cheaper, and on lines containing no steep gradients can often be employed with advantage. Further, it may be more easily adapted for taking curves in both directions than the undertype. The overtype grip has, however, several great disadvantages. Chief amongst these is the fact that when ascending steep gradients the traction rope reaction falls outside the wheel base, tending to tip the whole running head, a tendency which may not be counterbalanced by the weight of the carrier, with the consequence that the whole apparatus leaves the line. Again, owing to the fact that the centre line of the hauling rope must fall outside the centre line of the carrying rope to avoid any entangling, there is a decided tendency for the carrier to tip sideways as soon as a heavy hauling rope reaction occurs, a tendency which is very detrimental to the perfect working of the line when it is required to discharge the contents of the carrier automatically whilst the carriage is on the open track. These defects may be remedied by having the hauling rope directly below the carrying rope, *i.e.*, using an undertype grip. Practically the only defect

of this type is that the stations are generally longer than with the overtype grip, thus involving extra structural work. Even this may be overcome by employing an undertype grip in which the jaws open downwards instead of upwards, as is usual.

The speed at which the carriers should be run depends upon whether it is necessary for them to be hauled automatically round curve or return sheaves by the traction rope. The following figures represent good practice :—

Straight lines	400–450 ft. p.m.
Curves up to about 45° change of direction	Undertype up to 400 ft. p.m. Overtype „ 300 ft. p.m.
With automatic haul-round—	
Sheave, 20 ft. dia.	Undertype, 350 ft. p.m. Overtype, 300 ft. p.m.
Sheave, 16 ft. dia.	Undertype, 300 ft. p.m. Overtype, 250 ft. p.m.
Sheave, 12 ft. dia.	Undertype, 200 ft. p.m. Overtype, 150 ft. p.m.

The running wheels should be of high quality cast steel, the diameter depending on the load and type of rope, larger diameter wheels being employed with spiral ropes than with locked coil. The runner pins should be self-oiling. The following table gives some diameters of wheels for varying loads, carriers with two wheels being employed :—

Weight of load and carrier, cwts.	Diameter of wheel, ins.
14	8
30	10
40	12

Screw Grips.

There are two distinct main classes of screw grip, in the first of which the grip is operated by a lever and strikers in the stations. In the second the grip is operated by the weight of the carrier and load. The best example of the former is shown in fig. 26. There are two jaws—(*a*) the movable and (*b*) the fixed. The movable jaw is actuated by a shaft carrying two screws, one a rapid thread and the other a fine thread. This shaft is turned by the lever C. The action of the grip is as follows :—On the lever C being turned over, the rapid thread operates and the movable jaw quickly approaches the fixed jaw until the rope is just gripped. The fine thread now comes into action and applies the necessary force to grip the rope firmly. To unlock the

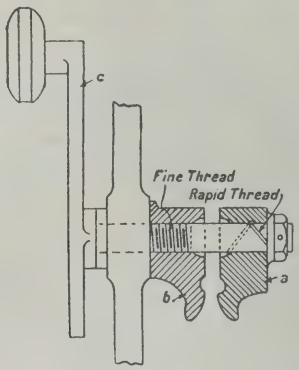


FIG. 26.

grip the reverse action takes place. The lever C is actuated at the stations by strikers constructed of angle irons.

In the other form of screw grip, in which the pressure is obtained from the weight of the carrier and load, the action is the same, the necessary turning moment being applied to the screw by hanging the carrier at the end of the lever arm. The grip is opened to allow of the rope entering or leaving, by means of rails, generally formed of angle irons, which depress small pulleys fixed at the end of the lever arm opposite to that on which the carrier is hung, thus raising the latter and thereby opening the jaws. It has been found in practice that the position of this lever arm should not be less than 35° to 40° to the vertical, to allow of easy depression of the rollers.

Lever Grips.

One of the best known Continental types is that shown in fig. 27. In this grip the pressure is applied by the weight of the carrier and load acting through a simple system of levers to give a multiplied pressure on the jaws. The grip is released by lifting the load, which is accomplished by means of small rollers running on to rails in the stations. Another type of grip—the

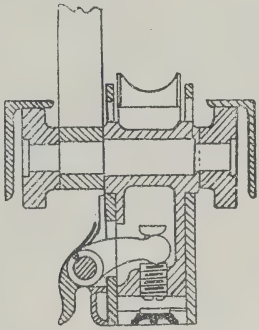


FIG. 27.

inclined plane type—is illustrated in fig. 28. The carrier and load are attached to a hanger having at its upper end a curved slot. This slot engages with a circular pin attached to the movable jaw. The downward pull of the hanger draws in this pin and with it the movable jaw, thus closing it upon the rope. When the pin is in the upper part of the slot the grip is absolutely locked, as no amount of upward pull of the rope can force the jaws apart. The grip is released

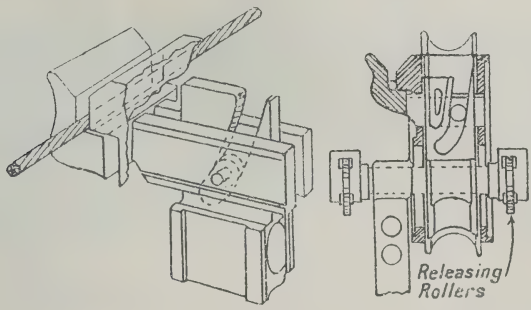


FIG. 28.

in the stations in the same manner as the type shown in fig. 27. In practice this form of grip has been found extremely simple and efficient.

In the matter of taking curves automatically in both directions, the advantage lies with the lever type of grip. In cost the latter generally has a great advantage. A ropeway with lever grips, carrying two-ton loads on an incline of 86 per cent., is being used for carrying timber in South-West Africa.

*Costs.**

The following tables of prime and working costs for ropeways represent the standard amounts for plants possessing no peculiar complications.

* The figures given are those current some time prior to the war.

Prime Cost of Plant with Structures in Steel.

Capacity in Tons per Hour.	Length of Line in Miles.			
	½	1	2	5
	£	£	£	£
5	1,900	2,000	3,300	6,900
10	2,100	2,300	3,650	7,400
20	2,350	2,600	3,900	8,000
40	2,500	2,870	5,000	10,500
60	2,850	3,200	5,700	11,900
80	3,100	3,500	6,300	13,100

The above prices are reckoned on the basis of employing spiral construction ropes up to 20 tons capacity and locked coil ropes for 40 to 80 tons capacity, and represent the total cost of the plant erected. For lengths above 5 miles a proportional cost may be taken.

Working Costs per Day.

Length in Miles.	Capacity per Day in Tons.					
	50	100	200	400	800	1,000
	s. d.	s. d.	s. d.	s. d.	s. d.	s. d.
½	17 6	21 0	25 0	32 0	42 4	48 0
1	19 5	24 0	28 10	37 7	52 0	58 5
2	26 6	31 6	36 4	49 7	68 10	76 10
5	45 7	51 0	61 0	83 3	126 0	156 10

The above costs are inclusive of depreciation, but exclusive of the cost of driving, this being so different in each case, depending on difference of height between stations, etc., that it is impossible to standardise, and it must therefore be reckoned as an extra to the above for each separate plant.

See also Cost of Aerial Ropeways—Section XLV.

Fixed Clip Single Rope System.

This type of ropeway is only applicable for light capacities, and is used largely on plantations and for the transport of valuable ores, where comparatively small quantities are handled.

The carrier head, fig. 29, is permanently attached to the rope by means of a spring steel strap, and is designed in such a way that it can pass freely over trestle sheaves, under depression wheels and round the terminals, without being detached from the rope. As both rope and carriers are in constant motion, loading has to be done by hand, or by means of a travelling hopper, which is usually hung from overhead rails at the terminal. The terminals are extremely simple and

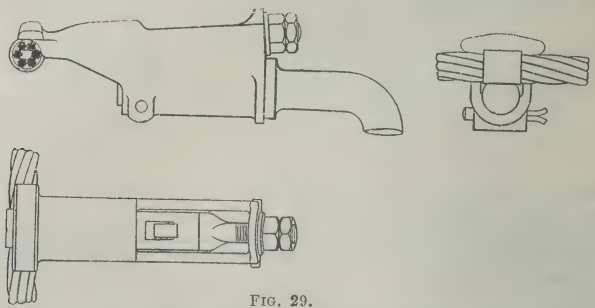


Fig. 29.

compact, as all shunt rails are done away with. The tension terminal is usually mounted on a trolley or on skids, the whole being movable by means of tension gear, to allow for stretch of rope, etc. Angles can be worked with the fixed clip system, but they should be preferably avoided, as they tend to produce excessive rope and gear wear at such places.

This type of line is run at a slow speed, namely, about 200 ft. per minute, and the only advantages it can claim are its simplicity, adaptability to steep grades, and its low prime cost when light capacities and moderate lengths have to be dealt with.

Jig-Back or To-and-Fro.

This type of ropeway is useful for short distances, and is operated with a pair of carriers that travel in reverse directions and are loaded or brought to rest alternatively at the opposite stations, but which do not pass round the terminals; it is, therefore, intermittent in its action, and the direction of travel is reversed after each journey. Where loads are to be carried which are heavy individually in relation to the capacity, and where the distance is short, the 'To-and-Fro System' is particularly applicable. A high carrier and rope velocity may be adopted during the run, especially where a clear span occurs between terminals, and where there are no intermediate trestles. These installations may be on the 'Monocable' or 'Bi-cable' principle, and in either case the moving rope is clamped to the carriers, and is not usually disconnected from them at the terminals.

It should be mentioned that, up to the present, the 'To-and-Fro' ropeway is the only system that is used for regular passenger traffic, as the carriers lend themselves to the adoption of a safety grip used in conjunction w th an auxiliary standing rope, but with such appliances these installations are very costly and present mechanical difficulties on lines of any length.

The following table, giving a rough idea of the capabilities of the various systems mentioned, shows the maximum lengths, capacities, and loads carried on various lines at present working or in course of construction:

System.	Present Maximum Lengths.	Present Maximum Hourly Capacity.	Present Maximum Individual Load.
	Yards.	Tons.	Tons.
Monocable .	80,196	200	$1\frac{1}{2}$
Bi-cable .	38,800	250	2
Fixed Clip .	2,330	8	4 cwts.
To-and-Fro .	1,850	Passengers' weight unknown.	17 tons.

Limitations of Types of Aerial Ropeways.

The *jig-back* type is the cheapest for short distances and a capacity of 8 to 12 tons per hour. Where the distance is considerable the loss of time in making the double journey between each load is too great, and where individual loads are heavy and the speed high too much time and energy is lost in starting and stopping.

The *monocable* type is suitable for a capacity up to 80 or 100 tons per hour where the grade of the rope is not steeper than 1 in $2\frac{1}{2}$ or 1 in $2\frac{3}{4}$. Where the rope grade would be steeper than this the boxhead clips would be liable to slip on the rope, and a bi-cable would have to be considered.

The *bi-cable* type is required for steep grades and large hourly capacity. It is evident from general principles that if the number and weight of the individual loads carried by a rope be gradually increased the tension and weight of the rope will have to be increased also, and at last a point will be reached when it is cheaper to fix the heavy main rope and use it for carrying purposes only and to provide an extra and lighter rope for hauling the loads. This point marks the natural transition from monocable to bi-cable.

(*A. J. Knight, B.Sc., M.I.Struct.E.*)

CONVEYORS.

(Contributed by Professor John Goodman, M.Inst.C.E.)

Belt Conveyors.

For conveying materials continuously over long distances in a horizontal or inclined direction band conveyors are usually employed.

Single conveyors are made up to 48 ins. in width and over 1,000 ft. in length. They can be employed for handling almost any class of material excepting those which are hot or sticky.

Solid woven cotton belts are lowest in first cost, but they are not suitable for continuous or heavy duties, because sharp substances soon cut the fabric, and, moreover, they are not waterproof and are very sensitive to climatic changes, which cause them to stretch and contract abnormally.

Belts treated with bituminous solutions are partially waterproof and resist wear much better, but have the disadvantage of becoming hard, and, moreover, they will not accommodate themselves to the troughing idlers.

Balata belts, when manufactured with the true vegetable gum, are satisfactory for work in the open, as they resist damp better than any other type. The best quality belts are durable and easily conform to the troughing idlers.

Vulcanised rubber belts of best quality, although high in first cost, are the best and ultimately the cheapest for conveying hard, heavy, or sharp materials. The cotton duck, which forms the basis, provides great tensile strength, whilst the rubber cover is the best-known material for resisting wear and abrasion. The cover on the carrying side should be at least $\frac{1}{16}$ in. thick.

Woven wire belts of various types are used for special purposes, as, for example, colliery picking belts, where it is desired to extract dust during conveyance.

The following table, taken from the *Engineering Magazine*, March 1909, shows the comparative values of different belts and materials to resist abrasion. Sample strips were submitted to equal sand-blast, and the volume of material worn away was measured.

Rubber belt	1·0	Woven cotton belt (high grade)	6·5
Rolled steel bar	1·5	Stitched duck (,, ,,)	8·0
Cast iron	3·5	Woven cotton belt (low grade)	9·0 to 15·0
Balata belt	5·0		

The life which may be expected from a best quality rubber belt when handling coal is given by the following expression:

$$N = 3W_b^3L$$

where,

N = the number of tons which a rubber belt may be expected to convey before it is worn out;
L = length of conveyor centres in feet;
W_b = width of belt in inches.

When estimating the life of belts made of other materials, the result obtained by the above expression must be divided by the corresponding figure in the table.

Belt Joints.—For short conveyors, endless belts may be used, but owing to difficulty of fixing they are not recommended for long conveyors. The best fastener for heavy duty is made of thin steel plates in the form of a hinge, with pins of hard copper, and used in combination with leather safes for the ends of the belt. The fasteners are secured by copper rivets.

For short and medium conveyors a satisfactory joint can be made with 'Alligator' fasteners.

Intermediate Idlers usually consist of C.I. pulleys, 5 ins. or 6 ins. diameter, mounted on hollow shafts fixed in C.I. brackets and provided with grease lubricators.

Table I. gives the thickness of belt and pitch of idlers recommended.

TABLE I.

Width of Band.	Effective Cross-section of Material carried in Sq. Ft.		Largest Pieces Belt will carry.	Ply.	Pitch of Idlers.
Inches.	Flat.	Troughed.	Inches.		Ft. Ins. Ft. Ins.
12	0·04	0·09	2½ cubes	3 to 4	5 6 to 6 0
16	0·07	0·15	3½ ,,	4 ,, 5	5 0 ,, 6 0
18	0·09	0·19	4 ,,	4 ,, 5	4 6 ,, 5 6
20	0·12	0·22	5 ,,	4 ,, 6	4 6 ,, 5 6
24	0·16	0·34	6½ ,,	5 ,, 7	4 6 ,, 5 6
30	0·27	0·55	10 ,,	6 ,, 8	4 0 ,, 5 0

Terminal drums should have a crowned face 3 ins. wider than the belt, and the diameter should be not less than 5 times the number of plies in the belt.

The driven terminal should be mounted on a carriage controlled by weights to automatically take up any belt stretch.

The usual speeds of conveyors when handling coal, ore, broken stone, etc., are from 250 to 350 ft. per minute.

Belts are usually carried on troughing idlers for the above-mentioned materials.

Speed for handling grain, 400 ft. to 600 ft. per minute. Grain containing chaff should run at the lower speed.

Grain bands usually run on flat idlers.

The capacity in cubic feet per minute may be taken from Table I. by multiplying the effective cross-section of material carried by the speed of the belt in feet per minute.

The pitch of the idlers for the return belt should be from 9 ft. to 12 ft., according to the weight of the belt.

The power required depends on workmanship, diameter of idlers, and other factors.

If fully loaded and the conditions are good, the following expression gives the horse-power required for band conveyors:

$$\text{H.P.} = \frac{(0{\cdot}15B + 0{\cdot}07M)S + HW}{33{,}000},$$

where,

B = weight of belt in pounds; M = weight of material on belt at any time in pounds; S = speed in feet per minute; H = height elevated in feet; W = weight of material delivered per minute in pounds.

Steel Belt Conveyors.

(Steel Belt Conveyors Ltd.)

Conveyors employing a flexible steel band in lieu of the textile fabric or rubber belt are in use in Scandinavia, on the Continent generally, in Great Britain and Colonies, and U.S.A., for handling a wide variety of materials, and have proved particularly successful in the stone, ore, cement, brick, confectionery, and sugar trades.

The band itself consists of finest quality cold-rolled, hardened and tempered Swedish charcoal steel of the "Sandvik" brand, is of very dense texture, highly resistant to rust and corrosion, extremely durable, and has a tensile strength of about 80 tons per sq. in., equal to about 3 tons per inch of width of belt. It is rolled at present in widths up to 24 ins., one-piece lengths up to 300–350 ft., and thicknesses up to 1 mm. (about 0·04 in.), the longitudinal edges being rounded and smooth. It is joined by means of simple lap riveted or butt strap riveted joints which can be made in less than an hour. Small conveyors can have brazed or welded joints. The flexibility of the material is such that a coil of 300 ft. of 16 in. belt can be sent out packed in a case 2 ft. square by 1 ft. 6 ins. deep; it weighs approximately 2 lbs. per foot run.

Consequent upon its transverse stiffness and ability to be loaded over its full width without the edges yielding downwards, steel belt has a high carrying capacity, and in addition for many materials it can be troughed by means of skirting boards (the belt forming the bottom of a rectangular trough) which increases the capacity three to four times. Thus, an open 16 ins. belt at 200 ft. per minute could handle about 35 tons of coal per hour, and if troughed from 100 to 120. Under similar conditions an open belt could handle 100 tons of broken granite per hour on the basis of 150 lbs. per cubic ft.

The usual end-drums and intermediate supports at suitable intervals for the top and bottom strands are used. The supports usually consist of simple Roller Idler Sets, but for light duty conveyors the strands can slide on timber runners that have a smooth and graphited face.

The belt does not stretch, and the only variation that has to be provided for is due to expansion and contraction; the tension devices are therefore of the simplest, and for a conveyor not exceeding 120 ft. in length working at normal temperature the sag in the bottom strand is sufficient to provide for all give and take required. Other than that a simple tension frame is employed, operated by either balance weight and rope, balance weight and levers, or springs.

Material up to a temperature of 200° F. may be conveyed without unduly affecting the belt.

Drag Bar or Scraper Conveyors.

These take the form of a wide chain travelling in a fixed trough, the material being dragged along by the chain bars. Considerable horse-power is required to drive this type, on account of the friction of the material and chain in the trough. Even with this fault the type is largely used where simplicity and few working parts are of premier importance. Examples are the 'Blackett' Conveyor, used at the coal face in collieries. The 'Bennis' Conveyor, much used in boiler stoking plants, and the 'Jenkins' 'D B' Patent Conveyor, which is principally used in gasworks for conveying hot or cold coke. It is also used for handling ashes, gravel, etc.

The conditions met with in handling coke are probably more severe than with any other material. First, because the coke reaches the trough at a temperature up to 1,700° F.; and, secondly, the coke is of a hard cutting nature, and contains a quantity of fine particles which work into the joints. Lubricants cannot be used, and finally the conveyor is in use night and day all the year round. Yet these chains frequently have a life of from three to four years, and convey up to 130,000 tons of material.

The 'DB' Conveyor consists of a special double chain connected together with cross-bars integral with special links at intervals of 2 ft. 1½ ins., and in widths from 12 ins. to 36 ins. or more. The detail of the chain is shown in fig. 30, and as used with a trough for hot coke, fig. 31.

It will be noticed that special short links are used alternately with long narrow links. Only the short links bear on the trough. The usual speed of such a conveyor is from 40 to 50 ft. per minute.

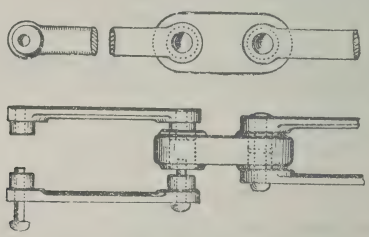


FIG. 30.

A trough 2 ft. 6 ins. wide, 7 ins. deep, will convey 20 tons of coke per hour and consume about 1 h.p. per 33 ft. of horizontal length.

The conveyor will work satisfactorily on inclines up to 33°, and with special cross-bars up to 70°.

The longest single conveyor installed is 350 ft. long, on the horizontal, with the addition of 60 ft. inclined at 30°.

The power required to drive this conveyor fully loaded is 14 b.h.p.

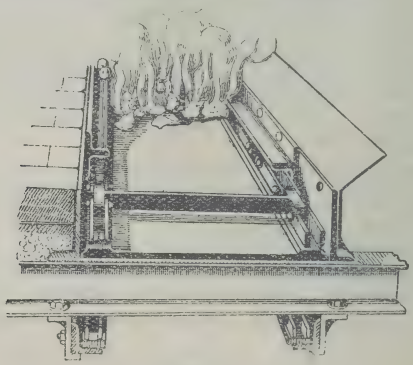


FIG. 31.

Scraper Conveyors are mostly used for handling coal when it is desired to discharge at short intervals and when dealing with quantities of from 20 to 40 tons per hour up to a maximum length of about 200 ft. For lengths beyond this it is better to use a Pushplate Conveyor, on account of the power required to drive being so much less.

The longest 'DB' Scraper Conveyor installed is 231 ft. long, dealing with 40 tons per hour, and requires 16 h.p. when fully loaded.

PUSHPLATE CONVEYORS.

These are in general use for conveying and delivering coal into storage hoppers. Conveyors of this type are superior to any other when it is desired to deliver coal at many intermediate points. For this purpose, outlets, with slide doors, are arranged in the trough where

desired. Generally the chain is provided with special attachment links at intervals, which carry M.S. plates and axles fitted with C.I. rollers, and grease lubricators. The plates are kept clear of the trough so as to avoid any noise when working.

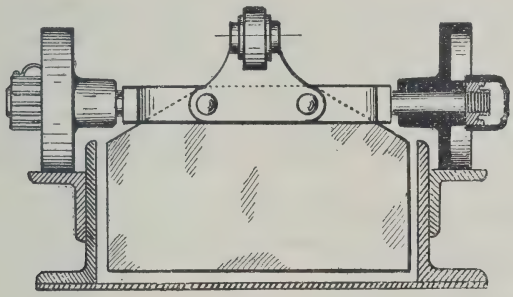


FIG. 32.

The usual speed is 100 to 120 ft. per minute, and as there is only the sliding friction between the coal and the trough the power taken is not excessive.

A typical Pushplate Conveyor is shown in section by fig. 32. For dealing with 25 tons of coal per hour the trough would be 15 ins. wide and 6 ins. deep. A single chain would be fitted with pushplates 17 ins. apart. The rollers would be grease lubricated, 6 ins. diameter and 4 ft. 3 ins. apart. A conveyor of this type, 250 ft. long, running at 120 ft. per minute, requires 5 b.h.p. to run empty, and 7 b.h.p. to run fully loaded.

TABLE II.—PUSHPLATE CONVEYOR FOR COAL.

Size and Capacity in Tons per Hour.			
Width of Trough.	Pitch of Pushplates.	Depth of Trough.	Tons of Coal per Hour.
Inches.	Feet.	Inches.	
12	1·5	6	20
15	1·5	6	28
18	1·5	6	35
24	1·5	6	45

The following gives the horse-power of scraper and pushplate conveyors:

$$\text{H.P.} = \frac{KS(M + G)}{33{,}000},$$

where,

K = constant, say 0·37 for scraper and 0·13 for pushplate conveyor with rollers; S = speed in feet per minute; M = weight of material on conveyor at any one time in pounds; G = weight of moving parts of chain in pounds.

The friction of the driving gear must be added to the above.

GRAVITY BUCKET CONVEYORS.

This type of conveyor is unique in the sense that one unit can transport vertically, horizontally, and at any intermediate angle, and for this reason they are much used in power-stations for elevating and storing coal in overhead bunkers.

The principle employed is a series of buckets mounted on steel spindles, which are connected to two endless chains at regular intervals, the ends of the spindles are provided with C.I. grease-lubricated rollers for the purpose of travelling on horizontal rails or between inclined guides.

The buckets are suspended above their centre of gravity in such a manner that they always remain in an upright position until they are tilted by cams or levers provided for the purpose.

The corner guides for changing the direction of the chain should preferably consist of two sprocket wheels large enough in diameter to allow the buckets to swing clear of the shaft; one wheel should be keyed in place, and the other allowed to turn freely on the shaft between collars, so as to rectify any trouble that may arise should one chain stretch more than the other.

As the buckets are mounted on the chains with a space of an inch or more between, it is necessary to provide a special 'feeding device' for filling the buckets. Several types are in use, but probably the most popular is in the form of a drum having a circumference equal to at least five pitches of buckets. Outlets are arranged corresponding with the pitch, with blank plate between.

The drum is rotated by direct gear with the chain, and the material is fed through the centre. The blank plates prevent the material spilling between the buckets.

It is sometimes necessary to feed buckets at several points; in this case the filler is provided with travelling wheels to run on separate rails, and is rotated by a large sprocket wheel which engages with one of the bucket chains.

TABLE III.—SIZE AND CAPACITY OF G.B. BUCKET CONVEYORS FOR COAL.

Tons per Hour.	Pitch of Buckets.	Buckets.			Speed.
		Length.	Width.	Depth	
	Inches.	Inches.	Inches.	Inches.	45 ft. per minute
30	21	22	16	9	
40	27	24	16	10	" "
50	27	24	20	11	" "
60	27	24	22	11	" "
100	27	26	30	12	" "

The following gives the horse-power required for gravity bucket conveyors:

$$\text{H.P.} = \frac{CH}{884} + \frac{S(W_m + W_c)}{180}$$

where,

C = capacity in tons per hour; H = height to be lifted in feet; S = speed in feet per minute; W_m = total weight of moving parts of conveyor in tons; W_c = total weight of coal in conveyor in tons.

Add for friction of gearing and motive power.

See also *Proceedings of the Inst. Mech. Engrs.*, No. 2, 1921.

PNEUMATIC CONVEYORS.

See "The Nuvako Pneumatic Conveying System"; *Engineering*, Dec. 9, 1927.

WATER CONVEYOR.

W. J. Jenkins & Co., Ltd., Retford.

In the Le Fleuve system of handling hot coke in Gas Works, the material, on being discharged from the retorts, is directed by means of a shoot to a concrete trough A, along which a stream of water is flowing (fig. 33.) When the red-hot coke comes in contact with the water, steam is given off in such quantities as to lift the coke and thereby to prevent it coming into contact with the bottom of the trough, thus the frictional resistance to motion is due to the flow of water, and not to dragging the coke along the trough. The coke and water are delivered to perforated trays on a short inclined conveyor, which allows the water to escape. Owing to the residual heat of the coke, the moisture carried over is evaporated, thus leaving the coke ideally quenched when delivered by the tray conveyor.

'LE FLEUVE' WATER CONVEYOR

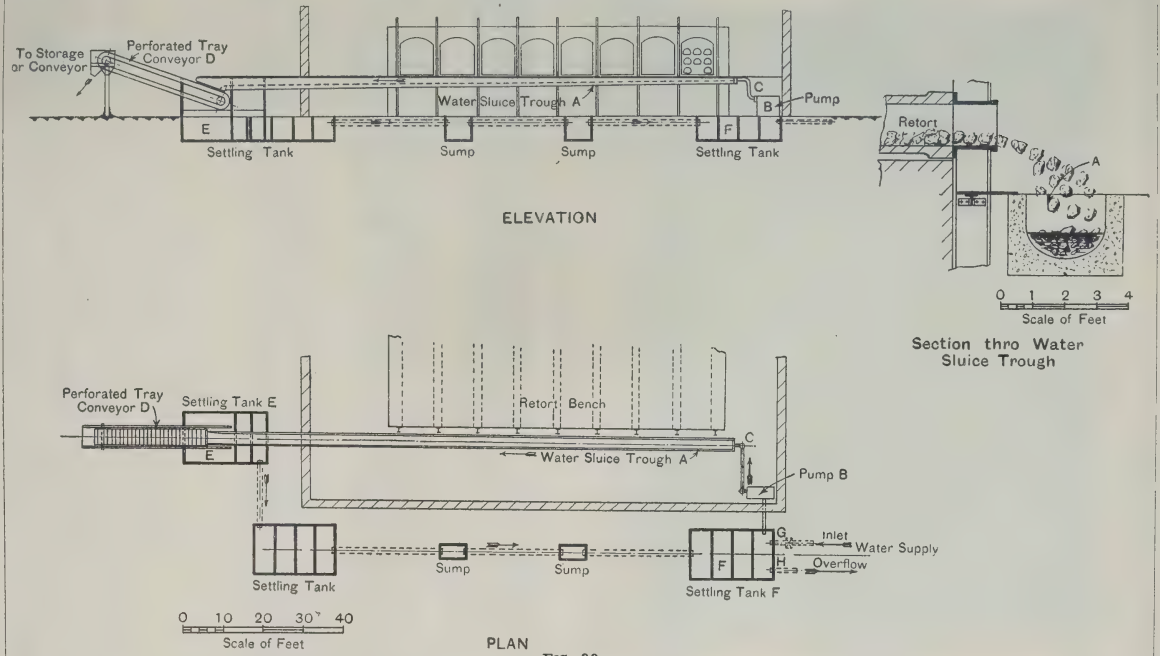


FIG. 33

The system is simplicity itself, and highly efficient. The coke is gently handled throughout consequently the breakage is reduced to a minimum.

When there is an ample supply of water it can, after use, be discharged into any convenient drain, after the solid matter has settled in suitable chambers E F, but when the supply of water is limited, it can be used over and over again by means of a circulating pump.

(*John Goodman.*)

Brake Linings.

Ferodo Applied to Brakes.

(*Ferodo, Ltd., Chapel-en-le-Frith.*)

Present-day high speeds and heavy loads require powerful and reliable brakes, these being especially necessary for modern lifting machinery. The friction coefficients of wood, leather, and metal are comparatively low, the values usually falling with heat and general working conditions; with the Ferodo lining on brakes (block, band, or disc type) the coefficient of friction increases as the temperature increases, the action is smooth, reliable and powerful, and reduces the wear and tear on the drums and gears. Ferodo fabric is readily fixed to a steel band, metal, or wood blocks.

For description of, and tests with, this material, see page 897.

Chekko Brake and Clutch Linings.

(*Cresswell's Asbestos Co., Ltd., Wellington Mills, Bradford.*)

These linings are made from 100 per cent. pure long asbestos fibre woven into brass wire in one solid piece; they are made in sheet from 2 in. to 12 in. wide, and in thickness varying from $\frac{3}{8}$ in. to $\frac{1}{4}$ in. The sheeting when cut to the required size is fastened to wood foundation blocks by means of counter-sunk brass screws, which screw into metal nuts, the latter lying in slots cut in the wood. In the case of a steel band being lined with the material, the screws pass through the band and screw into nuts on the outside of the band, or aluminium rivets may be used for the purpose. See also p. 897.

Notes on Fabric Linings.

A most important point to observe in regard to the use of fabric linings is that the blocks to be lined are first worn down so that they will make a true arc of contact, also that the fabric lining after being fitted is clear of the brake path at every point of its surface when not in action. In band-type brakes the same observations are essential, and an arrangement should be fitted whereby the top half of the band is kept well clear of the brake path by suitably arranged and powerful springs when idle.

Brake paths which are out of truth or have undue lateral play should be corrected, and the paths should be as smooth as possible. Given these conditions, experience proves that fabric linings are cheaper in the end as regards mere cost than materials which may cost little in themselves and are only costly in labour and renewal. In addition to the saving effected, there is improved efficiency, greater power, much sweeter action when engaging and disengaging, and greater durability.

In all brake paths the abutting edges should be tapered back a good 2 ins. to eliminate all possibility of any edge rising above the circumference of the path.

The manufacturers generally send out carefully printed instructions regarding fixing and applying, also as regards the correct material to use, and a little care and thought are all that are required to eliminate the possibility of failure.

Magnetic Separators.

(*The Rapid Magnetting Machine Co., Ltd., Birmingham.*)

Magnetic separators are made for removing iron as an impurity, recovering for its value, and preventing damage to crushing and other machinery.

One of the latest applications of magnetic separation, fig. 34, is for the recovery of iron (largely in the form of empty tins, used for canned food) from house refuse, *before passing to the destructor.*

The Rapid Magnetting Machine Co., Ltd., Birmingham, have taken out a patent for use in conjunction with the revolving screen used for removal of the dust as a preliminary operation.

The invention primarily consists of a very powerful electro-magnet, partly surrounding the ' delivery-off ' end of the screen. As the iron reaches the magnetic field the metal is drawn to

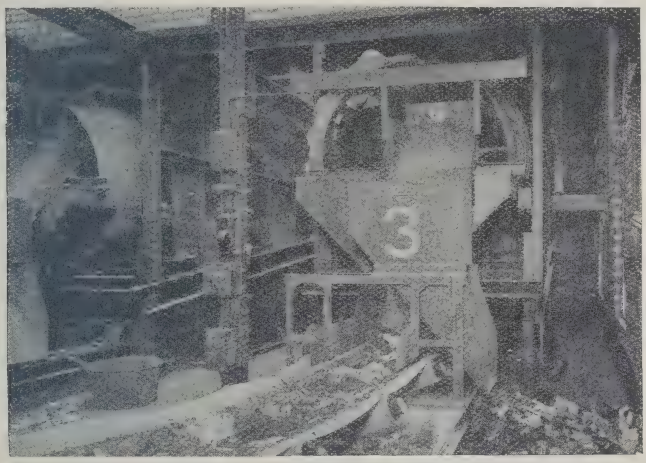


FIG. 34.

the inside surface of the screen and carried up to the vertical centre, where it is automatically released without the aid of mechanical cut-outs, falling into a shoot which diverts it to the side of the apparatus.

A common use of a magnetic separator is for the separation of iron from non-ferrous metals.

SECTION XIX

POWER TRANSMISSION

(Revised and amplified by Professor Dempster Smith, M.B.E., M.Sc.Tech., M.I.Mech.E.)

PART I

BELTING—ROPE DRIVING—CLUTCHES (pp. 855–897)

PART II

CHAIN GEARING (pp. 899–906)

PART III

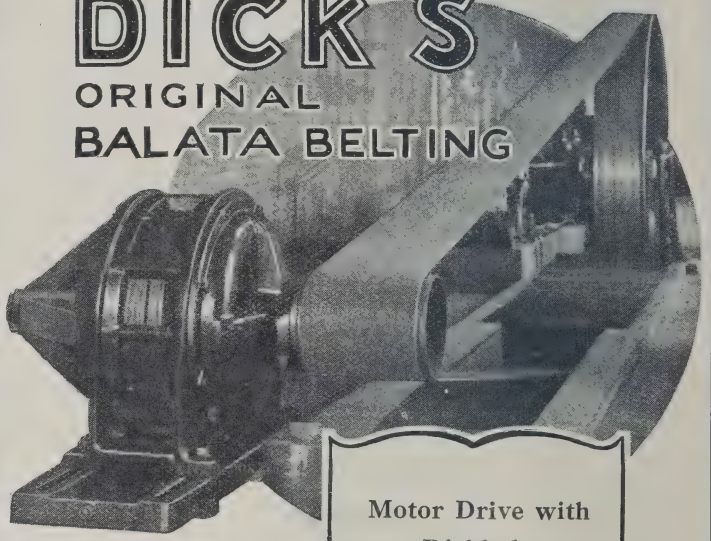
TOOTH GEARING — HELICAL GEARING — TURBINE GEARS—WORM GEARING—VARIABLE-SPEED GEAR (pp. 907–951)

(Revised by H. T. Davey, A.M.I.Mech.E.

PART IV

LINK MOTIONS (pp. 953–958)

DICK'S
ORIGINAL
BALATA BELTING



Motor Drive with
Dickbelt.
14 in. Face, 6 ply.

The superiority of the DICKBELT
is proved by its long life under
severe conditions

R. & J. Dick Limited

GREENHEAD WORKS, GLASGOW.

PATENTEES & ORIGINAL MAKERS OF

BALATA BELTING

AND DICK BALATA ELEMENT DRIVING ROPE

F. 855

SECTION XIX

(Revised and amplified by Professor Dempster Smith, M.B.E., M.Sc.Tech., M.I.Mech.E.)

PART I

BELTING—ROPE DRIVING—CLUTCHES.

BELTING.

Power Transmitted by Belting.

The following formula gives the power transmitted by any belt :—

$$H = \frac{(T - t)v}{33{,}000},$$

where,

H = actual horse-power transmitted ; T = tension in lbs. on the pulling side of the belt; t = tension in lbs. on the slack side of the belt ; v = speed of belt in feet per minute.

When a belt is doing its full work, T will be the greatest safe working tension that the belt will stand ; this is about 90 lbs. per inch of width for singles (more accurately 60 lbs. to 120 lbs., according to the thickness of the leather), and about 150 lbs. for doubles (135 lbs. for light and 160 lbs. for heavy doubles). In order that the belt shall do as much as possible, t must bear such a ratio to T that the belt just does not slip over the pulleys. In practice t varies from $\frac{1}{3}$ to $\frac{2}{3}$ of T, depending on the character of the surfaces of the belt and pulley, and on the fraction of the circumference of the pulley embraced by the belt ; $\frac{1}{2}$ of T will be a safe average value for t ; so that $T - t$ will be on the average 45 lbs. for singles and 75 lbs. for doubles for every inch of width ; hence the following rule for finding the power that any given belt should transmit :—

Multiply the width of the belt in inches by 45 if single, by 75 if double ; multiply this again by the speed in feet per minute, and divide by 33,000 ; the quotient will be the number of actual horse-power the belt will transmit.

EXAMPLE 1.—Required the horse-power which a 5-in. single belt, running at 1,000 feet per minute, will transmit.

$$5 \times 45 \times 1{,}000 \div 33{,}000 = 7 \text{ horse-power (nearly).}$$

EXAMPLE 2.—What power can be transmitted by a 12-in. double, running at 1,500 feet per minute ?

$$12 \times 75 \times 1{,}500 \div 33{,}000 = 41 \text{ horse-power (nearly).}$$

EXAMPLE 3.—An example of the reverse kind : What width must a belt be to transmit 20 horse-power, the belt running at a speed of 2,500 feet per minute ?

$$(20 \times 33{,}000) \div (2{,}500 \times 45) = 6 \text{ ins. nearly, if single.}$$
$$(20 \times 33{,}000) \div (2{,}500 \times 75) = 3\tfrac{1}{4} \text{ ins., about, if double.}$$

The values given in the table are more accurately calculated, and hence do not exactly agree with the above examples.

The size of the pulleys or the number of square inches of contact between belt and pulleys has no effect on the power transmitted ; that is to say, the same belt will have just as much grip, other things being equal, on 1-foot pulleys as on 4-foot pulleys. The fraction of the circumference of the pulley embraced by the belt, however, has an important effect on the friction between the two ; that is to say, if the belt is in contact with the pulley for more than half the circumference of the latter it will get a better grip, and will therefore transmit more power than if in contact for less than half the circumference ; hence the desirability, where possible, of making the pulling half of the belt the lower one, so that the slack upper side may sag down and lap further round the pulley.

The following table gives the number of horse-power which belts of various sizes should transmit. Allowance has been made for the loss of power at high speeds, due to the centrifugal force of the belt. This loss is negligible at low speeds, but increases with the velocity, amounting

to about 10 per cent. at a speed of 3,000 feet per minute, and about 30 per cent. at 5,000 feet per minute; no advantage is gained by running a belt faster than 4,000 or 5,000 feet per minute.

Under very good conditions of working, belting will transmit up to 1½ times the amounts given in the following table without being overstrained. (*Richardson.*)

TABLE SHOWING HORSE-POWER TRANSMITTED BY LEATHER BELTS UNDER AVERAGE WORKING CONDITIONS.

Speed of Belt. Ft. per Min.	Width of 'Single Thickness' Belts (ins.).									
	1	2	3	4	5	6	7	8	9	10
500	0·67	1·4	2·2	3	3·7	4·5	5·2	5·9	6·7	7·4
600	0·81	1·7	2·6	3·6	4·5	5·4	6·3	7·2	8·1	9
700	0·93	2	3	4·1	5·2	6·2	7·3	8·3	9·2	10·4
800	1·1	2·2	3·5	4·7	5·9	7·1	8·3	9·4	11·6	11·8
900	1·2	2·5	3·9	5·4	6·7	8	9·4	10·7	12	13·4
1,000	1·3	2·8	4·3	5·8	7·3	8·8	10·2	11·7	13·1	14·6
1,500	2·0	4·1	6·3	9·5	10·8	13	15	17·3	19·4	21·6
2,000	2·6	5·5	6·7	11·4	14·3	17·1	20	22·8	25·7	28·6
2,500	3·2	6·8	10·4	14·2	17·7	21·2	25	28·4	32	35·4
3,000	3·8	8	12·3	16·8	21	25	29·5	33·5	38	42
3,500	4·3	9·2	14	19·2	24	29	34	38·5	43	48
4,000	4·8	10·4	16	21·8	27·2	32·5	38	43·5	49	54·2
4,500	5·4	11·4	17·5	24	30	36	42	48	54	60
5,000	6·0	12·5	19·3	26·5	33	40	46	53	60	66

Light 'double thickness' belts will transmit 30 per cent. more power than that given in the above table.

For ordinary 'double thickness' belts the values given in the table may be increased 50 per cent., whilst for extra heavy 'double thickness' belts the values given in the table may be increased 70 per cent.

For intermediate speeds, and widths of belt, the h.p. transmitted is proportional to the speed or width of belt.

DICK'S ORIGINAL BALATA BELTING.

Table showing the horse-power which each inch of width of Dick belting, from 3-ply to 10-ply, will transmit at the speeds given.

Speed of Belt per minute.	3 Ply.	4 Ply.	5 Ply.	6 Ply.	7 Ply.	8 Ply.	9 Ply.	10 Ply.
Ft.	h.p.	h.p.	h.p.	h.p.	h.p.	h.p.	h.p.	h.p.
500	0·60	0·90	1·21	1·51	1·81	2·12	2·42	2·71
750	0·90	1·36	1·81	2·27	2·72	3·18	3·63	4·08
1,000	1·21	1·81	2·42	3·03	3·63	4·24	4·84	5·44
1,250	1·51	2·27	3·03	3·79	4·55	5·30	6·06	6·82
1,500	1·81	2·72	3·63	4·55	5·45	6·36	7·27	8·17
1,750	2·12	3·18	4·24	5·30	6·36	7·42	8·48	9·54
2,000	2·42	3·63	4·85	6·06	7·27	8·48	9·70	10·90
2,250	2·72	4·09	5·45	6·82	8·18	9·54	10·90	12·27
2,500	3·03	4·54	6·06	7·58	9·10	10·60	12·12	13·64
2,750	3·33	4·99	6·66	8·34	10·00	11·66	13·32	14·99
3,000	3·63	5·44	7·26	9·10	10·90	12·72	14·52	16·34
3,250	3·93	5·90	7·87	9·85	11·81	13·78	15·74	17·71
3,500	4·24	6·36	8·48	10·60	12·72	14·84	17·96	19·08
3,750	4·54	6·81	9·09	11·36	13·63	15·90	18·18	20·44
4,000	4·84	7·27	9·70	12·12	14·54	16·96	19·40	21·81
4,250	5·15	7·72	10·30	12·88	15·45	18·02	20·60	23·17
4,500	5·45	8·18	10·90	13·64	16·36	19·08	21·80	24·54
4,750	5·75	8·63	11·51	14·40	17·28	20·14	23·02	25·91
5,000	6·06	9·08	12·12	15·16	18·20	21·20	24·24	27·28

N.B.—These calculations, which are based on the most conservative lines, are given for ordinary loads over pulleys of proper diameters; anything abnormal in a drive should be taken into account in specifying the belt required.

The following table is the 'Dick Equivalent' of the best quality of leather belt :—

Dick belt 3-ply equals 14 ozs. single leather.
" 4 " " 18 ozs. "
" 5 " " 26–28 ozs. double leather (up to 12 ins. wide).
" 6 " " 30–32 ozs. double leather (over 12 ins. wide).

(*R. & J. Dick, Ltd.*)

German Practice.

In Germany it is usual to allow a higher effective tension per inch of belt width with increase in the diameter of pulley and speed. The table given below is compiled by Herr C. Otto Gehrckens and represents his practice for the past few years :—

Safe Effective Tension per Inch of Width (*Gehrckens*).

Diam. of Pulley.	Velocity Feet per Min.											
	600		1,000		2,000		3,000		4,000		5,000	
Ft. Ins.	A.	B.	A.	B.	A.	B.	A.	B.	A.	B.	A.	B.
0 4	11	—	13·5	—	16·5	—	16·5	—	19·0	—	19·0	—
0 8	16·5	—	22	—	27·5	—	30·0	—	33	—	36·0	—
1 8	27·5	44	38·5	49·5	44	55	49·5	60·5	55	66	60·5	71·5
3 3	33	55	46·5	66	55	77	60·5	88	66	93·5	71·5	99
6 6	38·5	66	55	82·5	66	110	71·5	121	77	132	82·5	137·5

A. Single thickness belts. B. Double thickness belts.

Reddaway Belting.

Table showing Horse-power Transmitted by the 'Camel Hair' Belting
(By permission of the Sole Manufacturers, *F. Reddaway & Co. Ltd., Manchester.*)

Vel. of Belt. Ft. per Min.	Width of Belt in Inches.												
	1	2	3	4	5	6	7	8	9	10	12	14	16
500	1·0	2·0	3·1	4·2	5·3	6·2	7·3	8·2	9·3	10·3	12·5	14·5	16·5
600	1·2	2·5	3·7	5·0	6·2	7·4	8·7	9·9	11·3	12·4	14·9	17·4	19·7
700	1·4	2·9	4·3	5·8	7·2	8·7	10·3	11·6	13·1	14·5	17·4	20·3	23·2
800	1·6	3·3	4·9	6·6	8·2	9·9	11·5	13·2	14·8	16·5	19·8	23·1	26·4
900	1·8	3·7	5·6	7·4	9·3	11·1	13·0	14·8	16·7	18·6	22·3	26·0	29·7
1,000	2·0	4·1	6·2	8·3	10·6	12·4	14·5	16·5	18·6	20·7	25·0	29·0	33·0
1,100	2·3	4·6	6·8	9·1	11·4	13·7	16·0	18·2	20·5	22·8	27·4	32·0	36·5
1,200	2·5	5·0	7·4	9·9	12·4	14·9	17·4	19·8	22·4	24·8	29·8	34·7	39·7
1,300	2·7	5·4	8·1	10·7	13·4	16·1	18·8	21·5	24·2	26·9	32·3	37·7	43·1
1,400	2·9	5·8	8·7	11·6	14·5	17·4	20·3	23·2	26·1	29·0	34·8	40·7	46·4
1,500	3·1	6·2	9·3	12·4	15·5	18·7	21·7	24·8	28·0	31·1	37·3	43·5	50·0
1,600	3·3	6·6	9·9	13·2	16·5	19·8	23·2	26·5	29·8	33·1	39·8	46·4	53·0
1,700	3·5	7·0	10·5	14·1	17·6	21·1	24·6	28·1	31·7	35·2	42·3	49·3	56·3
1,800	3·7	7·4	11·2	14·9	18·7	22·4	26·1	29·8	33·6	37·3	44·7	52·2	59·7
1,900	3·9	7·9	11·8	15·7	19·7	23·6	27·6	31·5	35·5	39·4	47·3	55·2	63·0
2,000	4·1	8·3	12·4	16·5	20·7	25·0	29·0	33·1	37·3	41·4	49·6	58·0	66·2
2,100	4·3	8·7	13·0	17·4	21·7	26·1	30·5	34·8	39·2	43·5	52·2	60·8	69·6
2,200	4·5	9·2	13·6	18·2	22·8	27·4	32·0	36·4	41·0	45·6	54·8	64·0	73·0
2,300	4·7	9·6	14·2	19·0	23·8	28·6	33·3	38·1	42·9	47·6	57·2	66·7	76·2
2,400	5·0	10·0	14·8	19·8	24·8	29·8	34·8	39·6	44·8	49·7	59·6	69·4	79·4
2,500	5·2	10·4	15·5	20·6	25·8	31·0	36·1	41·3	46·5	51·7	62·0	72·4	82·7
2,600	5·4	10·8	16·2	21·4	26·8	32·2	37·6	43·0	48·4	53·8	64·6	75·4	86·2
2,700	5·6	11·2	16·8	22·4	28·0	33·5	39·1	44·7	50·2	55·9	67·0	78·2	89·5
2,800	5·8	11·6	17·4	23·2	29·0	34·8	40·6	46·4	52·2	58·0	69·6	81·4	92·8
2,900	6·0	12·0	18·0	24·0	30·0	36·0	42·0	48·0	54·0	60·0	72·0	84·0	96·0
3,000	6·2	12·4	18·6	24·8	31·0	37·4	43·4	49·6	56·0	62·1	74·6	87·0	100·0

REDDAWAY BELTING.

TABLE SHOWING HORSE-POWER TRANSMITTED BY THE 'CAMEL HAIR' BELTING *(continued)*

(By permission of the Sole Manufacturers, *F. Reddaway & Co. Ltd., Manchester.*)

Vel. of Belt. Ft. per Min.	Width of Belt in Inches.												
	1	2	3	4	5	6	7	8	9	10	12	14	16
3,100	6·4	12·8	19·3	25·7	32·1	38·5	45·0	51·3	57·8	64·2	77·0	90·0	102·0
3,200	6·6	13·2	19·8	26·4	33·0	39·6	46·4	53·0	59·6	66·2	79·6	92·8	106·0
3,300	6·8	13·6	20·5	27·3	34·1	41·0	47·8	54·7	61·5	68·3	82·0	95·6	109·0
3,400	7·0	14·0	21·0	28·2	35·2	42·2	49·2	56·2	63·4	70·4	84·6	98·6	112·0
3,500	7·2	14·4	21·7	29·0	36·2	43·5	50·8	58·0	65·2	72·5	87·0	101·0	116·0
3,600	7·4	14·8	22·4	29·8	37·4	44·8	52·2	59·6	67·2	74·5	89·4	104·0	120·0
3,700	7·7	15·3	23·0	30·6	38·3	46·0	53·6	61·2	69·0	76·6	92·0	107·0	123·0
3,800	7·9	15·8	23·6	31·4	39·4	47·2	55·2	63·0	71·0	78·7	94·6	110·0	126·0
3,900	8·1	16·1	24·2	32·2	40·3	48·3	56·4	64·5	72·5	80·7	96·8	113·0	129·0
4,000	8·3	16·6	24·8	33·0	42·0	50·0	58·0	66·2	74·6	82·9	99·2	116·6	132·0
4,100	8·5	17·0	25·5	34·0	42·5	51·0	59·5	68·0	76·5	85·0	102·0	119·0	136·0
4,200	8·7	17·4	26·1	34·8	43·5	52·2	61·0	69·5	78·2	87·0	104·0	122·0	139·0
4,300	8·9	17·8	26·7	35·6	44·5	53·4	62·3	71·2	80·0	89·0	107·0	125·0	142·0
4,400	9·1	18·4	27·2	36·4	45·6	54·8	64·0	72·8	82·0	91·0	109·0	128·0	146·0
4,500	9·3	18·6	28·0	37·3	46·6	56·0	65·2	74·5	84·0	93·2	112·0	130·0	150·0

NOTE.—This table may be thoroughly relied upon for all ordinary conditions of working The results of tests made by Chas. Hopkinson, M.Inst.C.E., M.Inst.M.E., B.Sc., show that the 'Camel' Brand belt working under favourable conditions will safely transmit double the horse-power given in the above table.

Cross Section of Belts.

The following table shows what area of cross section of leather belt is needed to give various horse-powers, with various arcs of contact on cast-iron pulleys, at a belt speed of 3,000 feet per minute, the fastenings being single leather lacings. From it, having decided on the thickness. the width may be determined, or *vice versa*, by simple division:

Arc Contact.	Horse-powers.									
	5	10	15	20	25	30	35	40	45	50
Deg.	Sq. in.	Sq. in.	Sq. in.	Sq. in.	Sq. in.	Sq. in.	Sq. in.	Sq. in.	Sq. in.	Sq. in.
30	0·90	1·79	2·68	3·58	4·48	5·37	6·27	7·16	8·06	8·95
45	0·63	1·26	1·88	2·51	3·14	3·77	4·40	5·02	5·65	6·28
60	0·49	0·99	1·48	1·98	2·97	2·96	3·46	3·95	4·45	4·94
75	0·41	0·83	1·24	1·64	2·07	2·49	2·90	3·28	3·73	4·14
90	0·36	0·73	1·09	1·45	1·82	2·17	2·55	2·91	3·26	3·64
105	0·33	0·65	0·98	1·31	1·63	1·96	2·29	2·61	2·94	3·27
120	0·30	0·60	0·90	1·20	1·50	1·80	2·10	2·40	2·70	2·99
135	0·28	0·56	0·83	1·11	1·39	1·67	1·94	2·22	2·50	2·78
150	0·26	0·52	0·78	1·04	1·30	1·56	1·82	2·09	2·35	2·61
165	0·25	0·50	0·75	0·99	1·44	1·49	1·74	1·99	2·24	2·49
180	0·24	0·47	0·71	0·95	1·20	1·42	1·62	1·90	2·13	2·37
195	0·23	0·46	0·68	0·91	1·14	1·37	1·60	1·83	2·05	2·28
210	0·22	0·44	0·66	0·88	1·10	1·32	1·54	1·76	1·98	2·20
240	0·21	0·42	0·62	0·84	1·05	1·25	1·46	1·68	1·87	2·09
270	0·20	0·40	0·60	0·80	0·99	1·20	1·49	1·60	1·80	1·99
300	0·19	0·39	0·58	0·77	0·97	1·16	1·35	1·54	1·74	1·93

Belt Tension.

To find out whether the force carried by a belt is more than should be properly put upon it, or upon its fastening, we must know how much the arc of contact is, and what the tension on both the tight and the slack folds. The greatest tension upon it is that in the tight fold or side; and this is equal to that on the slack side plus the pull which the belt gives to the pulley or which the pulley gives to it. The greater the arc of contact, the greater the ratio between the tight and the slack side, and the less the tension will be upon the belt and its fastening to transmit a given power at a steady speed. Thus, if we have a pull of 10,000 lbs. necessary to carry a certain horse-power, when the arc of contact of the belt on the pulley is 90°, that means

CAMEL HAIR BELTING

The World's Greatest Power transmitter

CAMEL-HAIR BELTING
REGISTERED TRADE MARK

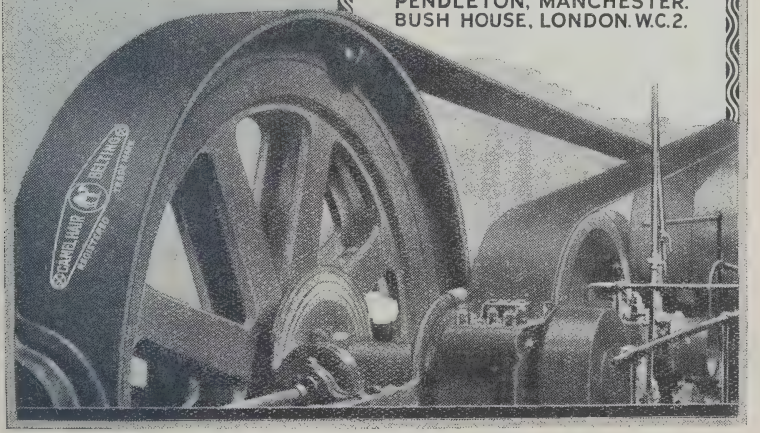
OVER 100,000,000 feet of Camel Hair Belting, varying in width from 2 to 60 inches, have been supplied to world-wide users since the date of its invention now **57 Years Ago.**

Camel Hair Belting gives to the user not only absolute efficiency in all attributes determining effective power transmission, but adds thereto the greatest economy in annual expenditure for upkeep.

INVENTORS & SOLE MAKERS

F. REDDAWAY & Co LTD

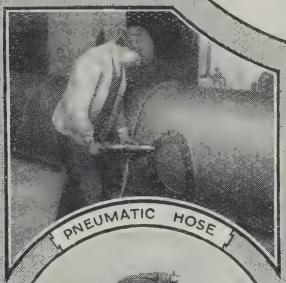
PENDLETON, MANCHESTER.
BUSH HOUSE, LONDON. W.C.2.



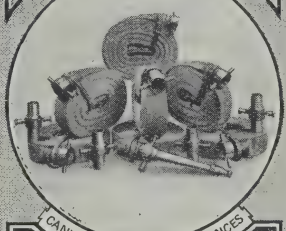
F. REDDAWAY & Co LTD

MANUFACTURERS and ENGINEERS

REGISTERED TRADE MARK



PNEUMATIC HOSE



CANVAS FIRE HOSE & APPLIANCES

RUBBER COVERED ROLLERS

Other Products of
F. Reddaway & Co.
Limited.

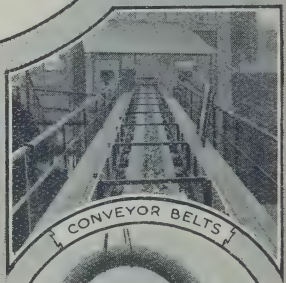
"SPHINCTER GRIP" HOSE for Rock Drill, Coal Cutter, Delivery, Steam, Gas, Oil, etc.

REDDAWAY'S PATENT HIGH-PRESSURE HOSE for pressures up to 10 Tons per square inch.

INDIA RUBBER GOODS for all Mechanical purposes

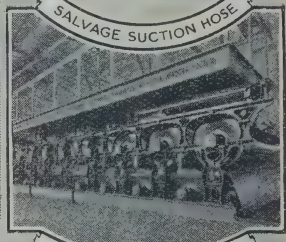
TEXTILE BELTING for Driving and Conveying

Sole Makers of
THE "CAMEL" PRINTING AND WASHING BLANKETS for Calico Printers, Etc.



CONVEYOR BELTS

SALVAGE SUCTION HOSE



PAPER MAKERS' DRY FELTS

Mills & Head Offices
PENDLETON, MANCHESTER.
BUSH HOUSE, ALDWYCH, LONDON W.C.2

that the ratio between the tension on the tight side and that on the slack side is 1·874; in other words, for every pound upon the slack side there will be 1·874 lbs. on the tight side; and if the difference between the tensions on the slack and the tight sides be 1,000 lbs., it will be necessary to have 1,000 ÷ 0·874 = 1,144 lbs. tension on the slack fold, and 2,144 on the tight one. (These figures apply to leather belts in good condition upon cast-iron pulleys, also in good condition.) The following table shows the ratio between the tension and transmitted pull for various arcs of contact from 30° to 300° :—

TABLE OF GREATEST TENSION ON BELTS.

Degrees of Arc . .	30	45	60	75	90	105	120	135
Ratio between tension and transmitted pull .	5·29	3·71	2·92	2·45	2·14	1·93	1·77	1·64
Degrees of Arc . .	150	165	180	195	210	240	270	300
Ratio between tension and transmitted pull .	1·54	1·47	1·40	1·35	1·30	1·23	1·18	1·14

EXAMPLE.—If it takes a pull of 300 lbs. to carry a given horse-power when the arc of contact is 195°, then the tension upon the tight side will be 1·35 × 300 lbs., or 405 lbs. Expressing results in horse-power instead of pounds pull, suppose that it is necessary to carry 175 horse-power with a belt running 2,000 feet per minute, and having 210° arc of contact; then the pull on the pulley will be 33,000 × 175 ÷ 2,000 = 2,887 lbs.; and the greatest tension upon the belt will be 2,887 × 1·3 = 3,753 lbs.

For double leather main belts,

$$\text{H.P.} = \frac{80\ W\ V}{33{,}000}$$

where

W = breadth in inches; V = velocity in feet per minute.

General Rules for Belts.

(1.)

A good all-round rule, and one easily remembered for belt calculations, is that 1 ft. per minute of belt speed per inch of width of belt is safely equal to the transmission of 1 watt of electrical energy. If we therefore multiply the belt speed in feet per minute by the width in inches, the result will be the total energy in watts; and if we divide the result by 1,000, the energy is obtained in kilo-watts, which energy the band may be relied upon to transmit without any undue tension being brought upon any part of the belt or excessive friction in any part of the machinery. These data, however, only apply to single leather belting or to woven belting of a similar thickness. If a light double band be employed, then 25 per cent. more energy may be calculated upon; or for heavy double leather belting as much as 60 per cent. more may be added. The great advantage for this method of calculation is that it fixes a constant for working tensions, so that the working tension becomes directly proportional to the width in inches, this constant being 44·24 lbs. pull per inch width of belt. If we wish, therefore, to know the difference in pounds pull on the tight and on the slack side of any belt, all that is necessary is to divide the total output of the machine in watts by the velocity of the belt in feet per minute, and multiply the result by 44·24, and the product will be the extra pull in pounds on the tight over the slack side. (*R. Berry.*)

(2.)

One horse-power will be transmitted by a single thickness belt 1 inch wide running at the rate of 1,000 feet per minute.

A belt $\frac{3}{10}$ths of a square inch section will transmit 1 horse-power at the foregoing rate.

Tension on belt by either of these rules will be 33 lbs. per inch of width and 165 lbs. per square inch of section.

With belts in good working order the frictional hold on the pulleys is about 58 per cent. of the pressure of the belt on the surface of the pulley as a maximum, and about 42 per cent. under ordinary atmospheric conditions.

It is safe to reckon that a double belt will do half as much work again as a single one.

Belting made of Helvetia leather is much stronger and will bear a heavier load than that made from ordinary tanned leather.

A double belt is more serviceable upon pulleys of large than upon those of small diameter, and should not be used upon pulleys whose diameters are less than about 3 times the width of the belt, unless the belt width exceeds about 4 ins.

Rubber belts will transmit from 25 to 40 per cent. more power per given arc of contact than leather belts. They should not be used in situations where they will be subject to a temperature of over about 90° F.

The average weight of a good four-ply rubber belt 7 ins. wide is 16 ozs. per linear foot. The best rubber belting is made with 32 ozs. duck, and then weighs 20 ozs. per linear foot.

It is a common error among mechanics and owners of factories to make the face of their pulleys narrow, in order to economise on the first cost of belting; but this false economy seldom decreases the cost of the machinery, and only saves a trifle in the first cost of belting. The small amount saved is soon lost by the stopping of machinery caused by the slipping of belts, strain on the shafting, increased friction (requiring additional driving power), and rapid destruction of the belts themselves. Were pulleys made of a proper size and width and face, and belts of proper width run with the *grain* side of leather to the pulley, a considerable saving of coal might be effected annually and an immense amount of trouble avoided.

Leather belts with grain side to pulley can drive 34 per cent. more than flesh side.

(3.)

To find the diameter of driving pulley.—Multiply the diameter of the driven by the number of revolutions it should make, and divide the product by the number of revolutions of the driver.

To find the diameter of driven.—Multiply the diameter of the driver by its number of revolutions, and divide the product by the number of revolutions of the driven.

To find the number of revolutions of the driven.—Multiply the diameter of the driver by its number of revolutions, and divide by the diameter of the driven.

To find the horse-power of a pulley.—Multiply the circumference of the pulley (feet) by the revolutions per minute, and the product thus obtained by the width of the belt, and divide the result by 600. The quotient will be the horse-power.

Length of Belts.

(1.)

(Not crossed.)

RULE.—Add the diameter of the two pulleys together, divide the result by 2, and multiply the quotient by $3\frac{1}{7}$; then add this product to twice the distance between the centres of the shafts; the result is approximately the length required.

(Crossed.)

RULE.—Mark off by lines the pulleys, set at their proper distance apar (either full size or to scale), and measure the length of the side of the belt, supposing the belt to envelop one-half the circumference only of each pulley, and add to this one-half the circumference of each pulley. If there is a great difference between the relative diameters of the pulleys, and the distance apart of the shafts is unusually small, the length of the straight sides of the belt should be measured, and the arcs of contact around the pulleys stepped around by compasses, the set of the latter being not more than about one-tenth the circumference of the pulley.

In the case of both running horizontally, the upper half of the belt should be the driving side.

(2.)

(Auguste Esnoul, A.C.G.I.)

Let X = distance between centres of pulleys; D and d = respective diameters * of pulleys; C and c = respective circumferences; P = a parameter equal to D − d for open, and to D + d for crossed belts, exclusive of lap for jointing; L = length of belt, then

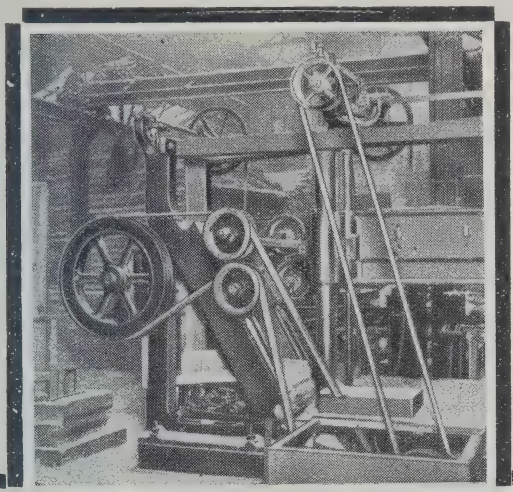
$$L = \tfrac{1}{2}(C + c) + 4X - \sqrt{4X^2 - P^2},$$

with an approximation such that the relative error is less than 1 in 1,000 for values of the ratio $\frac{X}{P}$ greater than or equal to 1·5; the relative error is about $\frac{1}{2}$ per cent. when $\frac{X}{P} = 1$.

For exact results, $L = \frac{1}{2}(C + c) + KX$, for both open and crossed belts; and $L = mX$ for crossed belts, where K and m are coefficients, tabulated herewith for the various values of the ratio $\frac{X}{P}$.

---

* Strictly speaking, D and d should be the theoretical diameters—that is to say, actual diameters plus thickness of belt.



HENDRYS'

LAMINATED BELTING

For Phenomenal Flexibility, Smooth, Vibrationless Running, Exceptional Durability and Maximum Power Transmission

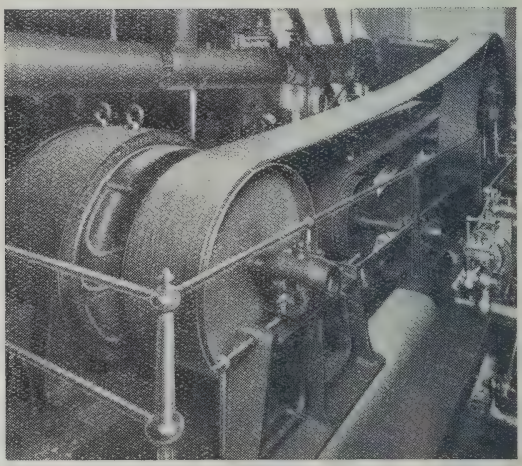
Sole Manufacturers

JAMES HENDRY LTD.

MAIN STREET, BRIDGETON, GLASGOW.

F. 860

ANY LENGTH—ANY WIDTH



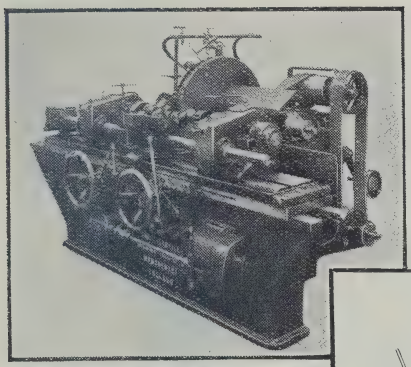
and up to 1" THICKNESS

Only the finest selected leather is used in the manufacture of Hendrys' Belting. Made under a special patent process, the leather is built up to the required size in a single ply from a single tested strand of solid leather. Efficient driving over pulleys is guaranteed with, in many cases, only half the usual width of belting. For smooth, vibrationless running always specify Hendrys'.

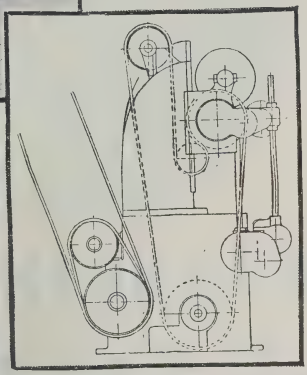
HENDRYS'

LAMINATED BELTING

JAMES HENDRY, LTD., BRIDGETON, GLASGOW



FOR EFFICIENT SHORT CENTRE TRANSMISSION

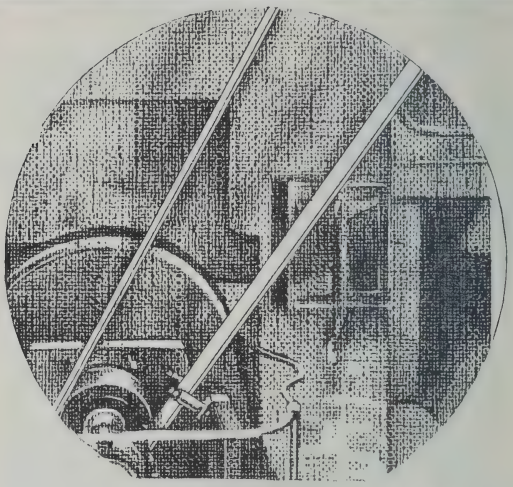


Maximum power transmission between long and short centres. No metal fasteners or cemented joints. The special construction of Hendry Belts ensures extraordinary flexibility and phenomenal grip

The illustration above is of Kearns' Horizontal Boring, Facing, Milling and Drilling Machine. Diagram below shows belt lay-out. All are Hendrys' No. 3×5″.

HENDRYS' LAMINATED BELTING

JAMES HENDRY, LTD., BRIDGETON, GLASGOW



ANY LENGTH ENDLESS

OF SOLID LEATHER IN A SINGLE PLY

Built under a patent process with no metal fasteners, Hendrys' Solid Leather Belting ensures efficient power transmission. Specify Hendrys', the Belting with a long life

HENDRYS'

LAMINATED BELTING

JAMES HENDRY, LTD., BRIDGETON, GLASGOW

$\frac{X}{P}$.	K.	m.	$\frac{X}{P}$.	K.	m.	$\frac{X}{P}$.	K.	m.	$\frac{X}{P}$.	K.	m.
0·700	2·537	4·781	0·81	2·394	4·333	1·02	2·246	3·786	1·7	2·082	3·006
0·705	2·528	4·756	0·82	2·384	4·299	1·03	2·241	3·766	1·8	2·078	2·951
0·710	2·520	4·732	0·83	2·374	4·266	1·04	2·237	3·747	1·9	2·070	2·897
0·715	2·512	4·709	0·84	2·365	4·235	1·05	2·232	3·728	2·0	2·063	2·848
0·720	2·505	4·687	0 85	2·357	4·205	1·06	2·228	3·710	2·2	2·051	2·765
0·725	2·498	4·665	0·86	2·349	4·175	1·07	2·224	3·692	2·4	2·043	2·697
0·730	2·491	4·643	0·87	2·341	4·146	1·08	2·219	3·674	2·6	2·037	2·641
0·735	2·484	4·621	0·88	2·333	4·118	1·09	2·215	3·656	2·8	2·032	2·593
0·740	2·477	4·600	0·89	2·325	4·090	1·10	2·211	3·639	3·0	2·028	2·552
0·745	2·470	4·578	0·90	2·318	4·063	1·12	2·203	3·605	3·5	2·021	2·470
0·750	2·463	4·557	0·91	2·311	4·037	1·14	2·196	3·574	4·0	2·016	2·408
0·755	2·457	4·537	0·92	2·304	4·011	1·16	2·189	3·543	4·5	2·012	2·361
0·760	2·451	4·518	0·93	2·297	3·986	1·18	2·182	3·513	5·0	2·010	2·324
0·765	2·445	4·498	0·94	2·290	3·961	1·20	2·176	3·485	5·5	2·008	2·293
0·770	2·439	4·479	0·95	2·284	3·937	1·25	2·162	3·419	6·0	2·007	2·269
0·775	2·433	4·460	0·96	2·278	3·914	1·30	2·150	3·358	6·5	2·006	2·248
0·780	2·427	4·441	0·97	2·272	3·891	1·35	2·138	3·302	7·0	2·005	2·229
0·785	2·421	4·422	0·98	2·266	3·869	1·40	2·128	3·250	8·0	2·004	2·200
0·790	2·416	4·404	0·99	2·261	3·848	1·45	2·119	3·202	9·0	2·003	2·178
0·795	2·410	4·386	1·00	2·256	3·827	1·50	2·112	3·159	10·0	2·002	2·159
0·800	2·405	4·368	1·01	2·251	3·806	1·60	2·099	3·081			

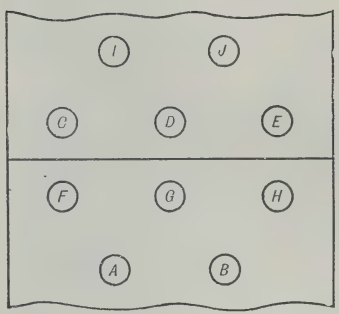


FIG. 1.

Jointing Belts.

LACED BUTT JOINT.

For belts about 3 ins. wide and over, the holes C, D, E, and F, G, H, should be about 1 in. apart and $\frac{5}{8}$ in. from the line of joint; A, B, and I, J should be about $\frac{1}{2}$ in. behind C, D, E, and F, G, H, respectively.

The sizes of the holes should be as follows:

Width of belt up to 4 ins.	Size of punched hole,	$\frac{1}{4}$ in.
„ „ from 4 to 8 ins.	„ „	$\frac{5}{16}$ in.
„ „ „ 8 ins. upwards	„ „	$\frac{3}{8}$ in.

WIDTH OF LACES.

Width of belt 24 ins. and over.	Width of lace,	$\frac{1}{2}$ in.
„ „ 6 to 24 ins.	„ „	$\frac{3}{8}$ in.
„ „ 2 to 4 ins.	„ „	$\frac{5}{16}$ in.
„ „ 2 ins. and less	„ „	$\frac{1}{4}$ in.

A square should always be used in cutting the ends of a belt preparatory to joining them, to ensure that they will be cut at a true right angle to the centre line of the belt. If a square is not employed, one or both ends are likely to be cut uneven or irregular, and these conditions cause the belt to slip from side to side of the pulley and impose irregular strains on the belt fibres, which no belt can endure indefinitely. Belts of widths up to about 18 ins. can be cut sufficiently true by holding one leg of the square against one side of the belt, having the other leg project across the belt, and then cutting along the edge of the latter with a sharp knife held perpendicular to the belt. The knife blade should be made wet occasionally to make it cut easily.

The ends of wide belts are difficult to cut square on account of slight variations in their widths and because the sides may not be exactly parallel; satisfactory results may be assured by the following method: At any point on the belt near where it is to be cut, measure across and find the centre, then at a point 2 or 3 ft. back from these dimensions, locate the centre again, and between these two points draw a clean sharp line. Next, by holding one leg of a square against the centre line just made, trim off one-half of the end of the belt by cutting along the other leg. By laying the square on the other side of the centre line, the other half of the belt can be cut. The square should be held firmly in position while cutting the belt, and if two small nails are driven in the belt on the centre line, they will help to keep the square from slipping.

Joining Thin Belts.

Place two ends of belt together, with edges fair one with the other, and make with an awl a row of holes, at *a*, through both ends. Then take about half a yard of strong twine, and draw half the length through the first hole; then pass each end of the twine through the second hole, one to the right and the other to the left, and draw both tight at the same time,

Fig. 2.

and so on until the last hole is reached, when one end only of the twine is passed through. The two ends of the twine are then knotted together, and the excess cut off.

Lap Joints.

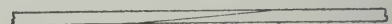


Fig. 3.

Cut down overlap to a plain bevel, cement surfaces together, and insert small taper wooden pegs, such as are used for boot and shoe manufacture. This is the strongest form of joint that can be made.

Length of Lap.

Width of single belt, in inches .	1 to 4½	5	6 to 8	9	10 to 14	15 to 24
Length of lap, in inches . . .	4½	5	6	6½	7	8

All double belts should have a 6-inch lap at least.

Cement for Belt Joints.

Equal parts of good glue and American isinglass; place the material in a glue pot and add sufficient water to cover the whole. Let the mixture soak ten hours, then bring the whole to a boiling heat, and add pure tannin until the whole appears like the white of an egg. Apply warm. Buff the grain of the leather where it is to be cemented, keep the joint surfaces pressed firmly together, let the joint dry for a few hours, and the belt will be ready for use.

(*Railway and Locomotive Engineer.*)

To Find Length in Feet of Belt in a Roll.

From outside diameter (D) deduct inside diameter (*d*), multiply by number of turns (*n*) and by 0·262.

Example.—D = 12 ins., *d* = 4 ins., *n* = 16 ins. Length = (12 − 4) × 16 × 0·262 = 33·5 ft.

Belt Dressings.

'Gripwell' Belting Syrup (*Asbestos Cement Building Products, Ld., Trafford Park, Manchester*) is particularly suited for fast and loose drives or shock drives, to prevent slip; the dressing is applied hot, so as to penetrate the belt freely.

Oiling and Greasing Leather Belts.

When the belt is pliable, and only dry and husky, apply blood-warm tallow, and dry in by heat of fire or sun. The addition of resin to the tallow for belts used in wet places is advantageous. Belts which have become hard and dry should have an application of neatsfoot or cod-liver oil mixed with a *small* quantity of resin.

Suggestions to Belt Users.

In order to obtain a given amount of power, it is better to use a narrow belt running at a high speed than a wide belt running at a low speed.

If possible, use a belt speed of about 3,000 feet per minute for smaller powers, and about 4,000 feet per minute for high powers. Thrashing machine belts run at about 2,200 feet per minute, and this is the usual speed for agricultural machines.

Use a pulley that is wide enough, so that a small amount of irregularity in the belt does not make it run off. Allow at least ½ in. on machine pulleys and 1 in. or more on countershaft and lineshaft pulleys. Use crowned pulleys wherever possible.

If possible, use a horizontal belt with large centre distances and with the pulling side at the bottom. In a horizontal drive, the slack side of the belt at the top of the drive increases the belt contact.

When using a vertical belt, do not have more centre distance than is absolutely necessary, because the weight of the belt reduces the efficiency of the drive.

Avoid crossed belts on short centre distances. Avoid the crossing of wide belts. If it is necessary to cross belts, see to it that the rubbing of the belt surfaces does not have a tendency to open up the laps.

Keep belts flexible and as clean as possible. Lubricating oil should not be allowed to remain on a belt, but should be removed by applying powdered chalk.

Do not use belt dressing containing mineral oils or resins.

Double belts should not be used on small pulleys.

Hints on Belt Driving.*

(John Tullis & Son, Ltd., St. Ann's Works, Glasgow.)

Pulleys having a high convexity prevent the belt from grasping its work. One-sixteenth of an inch is sufficient convexity for pulleys 6 ins. wide and under; $\frac{3}{32}$ of an inch is enough for pulleys over 12 ins. wide.

By compounding two single or double belts the gain in horse-power is close on 100 per cent.

By compounding three single or double belts the gain in horse-power will be close on 200 per cent.

Double belts should never be run on pulleys of small diameter. Compound single belts give much better results as regards power, longer life, and less cost.

Compound belts save the cost of wide pulleys as well as the room of wide pulley space.

Every inch in width of good double leather belting, travelling at 500 ft. per minute, while working on pulleys 24 ins. diameter, will transmit one horse-power, provided the arc of contact is 180 degrees.

Every inch in width of good single leather belting, travelling at 800 ft. per minute, while working on pulleys 24 ins. diameter, will transmit one horse-power, provided the arc of contact is 180 degrees.

A comfortable working strain for a single belt is about 50 lbs. for every inch of breadth, and for a double belt 80 lbs.

A belt running with the soft flesh side next the pulley will wear longest, because it is working according to the natural growth of the hide.

When driving on pulleys of 4 ins. diameter and under, the belt ought to be barely $\frac{1}{8}$ in. thick. On pulleys 4½ ins. to 8 ins. diameter the belt should be about $\frac{5}{32}$ in. thick. On pulleys 8½ ins. to 12 ins. diameter the belt should be about $\frac{3}{16}$ in. thick. On pulleys 12½ ins. to 18 ins. diameter the belt ought to be barely ¼ in. thick. On any diameter above 18 ins. heavy leather can be used with profit.

If the slack side of a main driving belt runs in wave form it is certain that the engine is getting more steam on one side of the piston than on the other. This unsteadiness is transmitted to all the working belts in the factory.

A badly balanced pulley, running at a high speed, will impart an unsteady motion to the belt running on it. The machine so driven makes bad work.

When a belt gets saturated with waste oil, an application of ground chalk will soon absorb the oil and make the belt workable.

Where possible, all belts should be run with the driving or tight side under the pulleys. The sag of the upper or slack side helps the belt to hug the pulley to the extent of 20 per cent. gain in driving power, as compared with a belt running with the tight side uppermost.

Belts running over pulleys of small diameter at high speeds ought to be as thin and as wide as possible. Orange-tan leather of uniform thickness answers remarkably well.

HIGH-SPEED BELTS.—A belt made from selected orange-tan leather, not too thick, softened, and made flexible, is the toughest and best leather belt for this class of drive. This belt will transmit great power when running at a belt speed of six to nine thousand feet per minute. The pulleys used for such drives must be strong and equally balanced, carefully turned to template, having a convexity no higher than one-sixteenth of an inch. Such pulleys must be hung perfectly true to each other upon the driving and driven shafts.

Thin belts, as wide as possible, give by far the best results working vertically. A thick vertical belt will not hug the pulleys.

In the use of vertical pulleys and guide pulleys, these pulleys ought to be twice the width of the belt, having no flange.

For machinery requiring the continuous shifting of the belt, single orange-tan leather is too thin, but double is highly recommended.

Leather link belting cut from the butt and made concave to suit the convexity of the pulleys will transmit from 25 to 30 per cent. more power than a flat belt.

Orange-tan leather rope, 1 in. square, running at 1,000 ft. per minute, on pulleys not less than 4 ft. in diameter, will transmit 10 horse-power.

In lacing hair belts the laces should run in parallel lines, each row laced separately, and no stitches placed across between the rows.

* Abstract from 'Guide to Belt Driving.'

A circular saw belt ought not to be narrower than one-fourth of the diameter of the saw. The spindle pulley should be as great in diameter as the width of the belt.

The following show tensile breaking stresses of leather belting :—

British Oak-tanned	.	.	.	4500–5000 lbs. per sq. in.
British Orange-tanned	.	.	.	6000–6500 „ „

According to *J. E. Rhoades and Sons*, the strongest part of belt leather is about one-third the way through from the flesh or rough side ; it is therefore desirable to run the grain or smooth side on the pulley so that the strongest part of the belt may be subject to the least wear. The flesh side is not as liable to crack as the grain side when the belt is old, hence it is better to crimp the grain than to stretch it. Experiments at Cornell University (U.S.A.) have established the fact that under reasonable shop tension the flesh side will transmit but 35 per cent. to 80 per cent. as much power as the grain side, with an average of 50 per cent. to 60 per cent., depending on the belt, the tension, etc.

A report issued by the Leather Belting Exchange Foundation, at Cornell University, on experiments to determine the relative power-transmitting capacity of belts on vertical, angular, and horizontal drives, gave the following conclusions :—

1. A vertical belt transmitted about 12 per cent. less power under average conditions than a horizontal belt, at 7 ft. 6 ins. centre distance. The percentage difference varies somewhat with the centre distance.

2. A given increase in tension increases the capacity of a vertical belt more than it does that of a horizontal belt, and *vice versa*. Therefore a vertical belt is more sensitive to tension, and ought to be retightened a little oftener to maintain an equally uniform transmitting capacity.

3. At the higher tensions, vertical, angle, and horizontal drives perform more nearly alike, while at low tensions they are widely different in transmitting capacity.

4. The power-transmitting capacities of four drives with the tight side below fall in the following order : First, horizontal belt ; second, 45° belt ; third, 67½° belt ; and fourth, vertical belt. There is practically no difference between the horizontal and 45° drives, or drives in between these two, except at low tension, when the horizontal is better.

5. Comparing the effect of using the tight side above and below with 45° and 67½° belts, it was found that when the tight side was below, leather belts gave about 12 per cent. more power on the 45° drive and about 6 per cent. more power on the 67½° drive, under average operating conditions.

Opinion differs somewhat as to which side of the belt should be run next to the pulley face —the grain or the flesh side,—but the balance appears to be in favour of the grain side. In the case of a good belt the grain side should transmit nearly twice the power conveyed by the flesh side. One disadvantage of using the flesh side next to the pulley is that whenever slip begins, the strength of the belt gets ground away by the quicker-running pulley acting like a grindstone. By running the grain side next to the pulley face, the essential sinews and muscles on the flesh side are preserved, while the less important and harder grain side takes the wear and tear caused by belt slip. (*Liverpool Journal of Commerce.*)

Belt Pulleys.

These should be at least an inch wider than the belt, and the rim should be convex to keep the belt in the centre ; about $\frac{1}{16}$ in. to $\frac{5}{32}$ in. of convexity is usually allowed in pulleys up to 12 ins. wide and over. The ratio of the diameters of two pulleys which work together should not exceed 6 to 1. The distance apart should, if possible, be such that the arc of contact on the smaller pulley is not less than 165°. If the arc of contact is less the width of the belt should be increased (see page 858). The diameter of pulleys is seldom less than 50 times the thickness of the belt in the case of ordinary belts, or 30 times the thickness of link belts ; that is, if any reasonable amount of power is being transmitted.

The slip in the case of a wood pulley with a single-ply oak-tanned leather belt is less than with one of cast-iron, if the belt tension is low.

Wood Pulleys.

In many respects wood pulleys possess considerable advantages as compared with pulleys made from cast iron and steel.

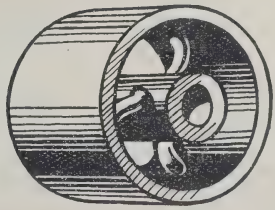
A wood pulley is some 60 per cent. lighter than a corresponding cast-iron pulley, while a comparison of the weights of wood and steel pulleys shows an advantage of some 30 per cent. in favour of the former. The light wood pulley is easily handled, and erection is considerably simplified, while the fact that a saving in power is effected must not be overlooked. These pulleys are particularly suitable for reversing drives where it is essential that the inertia of rotating parts should be reduced to a minimum.

Results of tests made in the United States with wood pulleys in regard to rim speed show that a wood pulley could be run up to a speed of 28,889 ft. per minute without destruction.

SOLVE YOUR DRIVING PROBLEMS *with* 'COLUMBIA' LEATHER BELTING

TULLIS

98% Stretchless and Straight Running



JOHN TULLIS & SON LTD., BRIDGETON, GLASGOW, S.E.

F. 864

The most efficient link between ***POWER*** *and* ***PRODUCTION!***

'Tullis'
Leather Belting

embodies vast experience, finest leather, and the most modern tanning and manufacturing process.

John Tullis & Son Ltd
LEATHER BELTING SPECIALISTS.

St. Ann's Works, Tullis St., GLASGOW, S.E.

Telephones: Bridgeton 1205-6-7-8.
(Private Branch Exchange.)
Telegrams: 'Tullis, Glasgow.'

AGENCIES:	Telephones:	Telegrams:
LONDON (Home)—116 Queen Victoria Street, E.C.4.	Central 9917/8.	Mekaniska, Cent, London.
LONDON (Export)—51/52 Bank Chambers, 329 High Holborn, W.C.1.	Holborn 8049	*Export:* Tullibody, Holb, London, W.C.1.
MANCHESTER—9 Blackfriars Road.	Central 2007	Tullis, Manchester.
BIRMINGHAM—Empire House, 161 Gt. Charles St.	Central 987	Tullis, Birmingham.
LEEDS—10 Aire Street.	Central 26095	Tullis, Leeds.
BELFAST—2 Linenhall Street, W.	Central 4292	Tullibody, Belfast.

With the same belt tension 33⅓ per cent. more power can be transmitted with a wood pulley than with either cast iron or steel, so that to transmit the same horse-power the belt on a wood pulley need not be so tight, and in consequence the strain on the bearings is relieved.

A belt obtains a better grip on a wood pulley than on an iron or steel pulley, so that there is much less liability of belt slip, and in consequence the life of the belt is enhanced.

Wood pulleys have been used to transmit up to 1,400 h.p.

A Simple Method of Crowning a Flat Pulley.

A belt-driven generator in a plant was originally fitted with a flat-faced pulley, and the belt would frequently run over to one side when an increase of load was thrown on the generator.

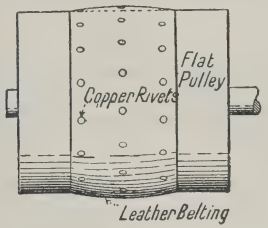


FIG. 4.

To overcome this, a crown was made on the pulley with leather lagging as shown by fig. 4. A piece of single-ply leather belting, 8 ins. wide, was procured and shaved down thin on the edges, and punched for three rows of ½-in. rivets. The leather was then fitted around the centre of the pulley, and the rivet holes were marked and drilled in the rim. The pulley was then warmed with a gasoline torch and a coat of belt cement applied and the leather pressed on smoothly, care being taken to draw it straight and to fit the ends together evenly. The rivets were then put in and headed up and the cement allowed to set twenty-four hours. This machine has run over four years since the job was done, and the driving belt has run perfectly under all conditions. (*Power.*)

Leather covers to pulleys are said to be conducive to excessive wear.

Belt Driving with Quarter or Right Angle Twist.

(1)

Fig. 5 shows the arrangement of pulleys for this purpose. In this case the centre of the bottom pulley should be in a dead line with the left-hand circumference of the top pulley, and the left-hand circumference of the bottom pulley should be in a dead line with the centre of the

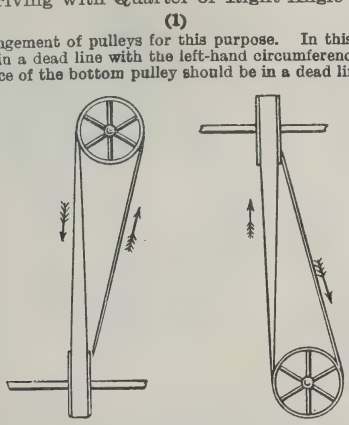


FIG. 5.

top. This is the correct position, but in actual practice the position of both pulleys has to be sometimes slightly altered to suit the running of the belt; this is a very simple matter, as it only means moving the pulley along the shaft one way or the other until it is seen that the belt

is in contact with the face of the pulley. No 'guide' pulleys are necessary. The pulleys should preferably be slightly crowned, *i.e.* a little larger in diameter at the middle than at the edges.

In practice, the centre plane of the lower pulley should be to the left of the edge of the upper pulley by an amount equal to half of the belt thickness.

With thick belts this point is sometimes of importance. No shifting of pulleys should usually be required if the drive is correctly set out.

With horizontal drives at long centres and pulleys with small crowning, a little adjustment is sometimes called for to compensate for dropping of belt due to its weight or perhaps due to the difficulty of getting a true horizontal line when 'setting out.' (*Geo. T. Pardoe.*)

(2)

The position for the machine pulley, with respect to the pulley on the shaft, depends on the relative direction of rotation of the pulleys. In any event the machine pulley must be in such a position with respect to the overhead pulley that when a plumb line is hung over the falling side of the overhead pulley at the middle of its face, the line must touch the rising side of the machine pulley at the middle of its face.

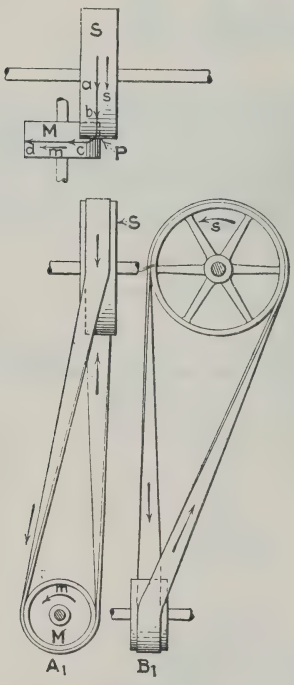


FIG. 6.

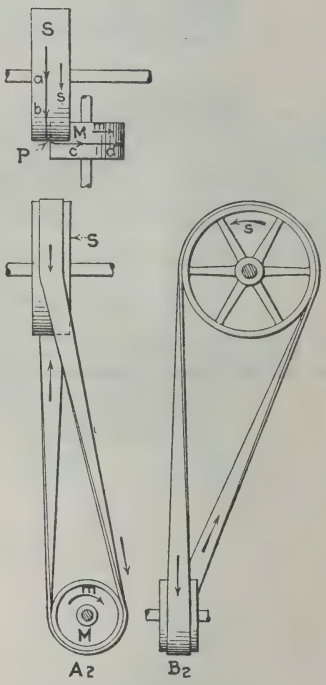


FIG. 7.

If S, fig. 6, is the overhead pulley whose direction of rotation is indicated by the arrow *s*, and it is desired to drive the machine pulley M in the direction indicated by the arrow *m*, then the middle of the face of the rising side of pulley M must be vertically under the middle of the face of the falling side of pulley S, that is, a plumb line hung over S at P, fig. 6 or fig. 7, should touch the rising side of pulley M at the middle of its face. It should be noted that where *s* and

m indicate the directions of rotation, whether in fig. 6 or fig. 7, the belt must pass over the top of the overhead pulley and thence to the far side of the machine pulley as indicated, in each plan, by the arrows *abcd*.

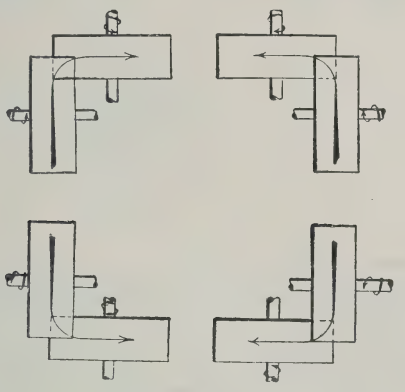


FIG. 8.

Mr. E. Bunzel points out that in laying out the drive, all that has to be seen to is that when looking down on the top of the two pulleys, the arrow of direction of the upper pulley follows the arrow of direction of the lower pulley. Fig. 8 shows the application of this rule for all four cases.

(*Power.*)

Link belts are not generally used for this kind of drive, but flat belts are made with a special curve to suit the drive in question; the curve of the belt varies according to the diameter of the pulleys, one to each other, and also to the length of the centres.

Putting on Heavy Belts.

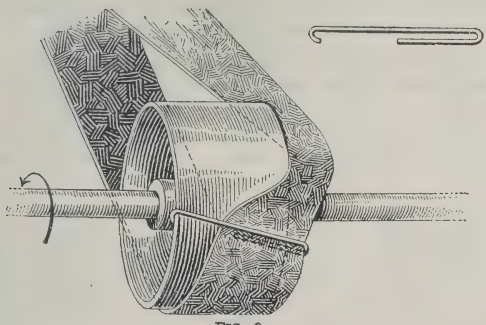


FIG. 9.

Fig. 9 shows a simple device for putting on heavy belts when it is necessary to shut down in order to do so, and where a rope is usually employed for the purpose.

Take a piece of $\frac{1}{2}$- or $\frac{5}{16}$-in. round iron and bend it, as shown in the small sketch. Hook the short end over the edge of the rim of the pulley and the long end under the belt as indicated. Start the engine slowly, and when the belt is on it is not necessary to stop, because the short end of the hook will straighten out and disengage itself. It is a good plan to use a rod of as light weight as will answer the purpose, since it will straighten out and free itself with less stress on the belt; the iron hook then falls to the floor. (*Power.*)

THE LINCONA ENDLESS BAND AND TENSION PULLEY DRIVE.

(James Dawson & Son., Ltd., Boultham Works, Lincoln.)

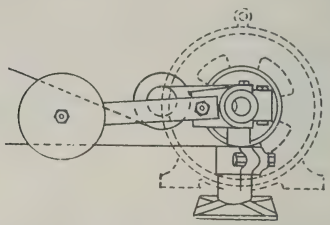


FIG. 10.

The degree of success obtained with the 'gravity idler' or 'jockey pulley' system of belt driving (fig. 10) has been governed by the difficulty of finding a fastener or joint which would stand up to the stresses produced by the severe bending resulting from the action of the idler pulley.

A further cause of failure has been the heavy stresses set up in the belt by the bending action round the small pulley usually found on such drives. The reverse bending, the high belt speed, and the short centres, all militate against the satisfactory operation of the ordinary type of belting. The 'Lincona Jointless Band' has been specially designed to meet the conditions, and is found to satisfactorily answer the required purpose. The joint is entirely eliminated, the basis of the belt being a very high tensile endless cotton band, which is impregnated with pure Balata gum, and folded, with the result that a very thin and strong belt has been produced, so that the bending stresses are reduced to a minimum, and frictional heat (which is rapidly destructive) is eliminated.

The advantages of a tension pulley drive are :—

1. A speed ratio of as much as 10 to 1 with a single drive.
2. The elimination of belt slip.
3. Saving in space, and, as a consequence of the shortness of the belt, the diminution of the collection of oil and dirt, and the avoidance of the inevitable 'taking up' for stretch.
4. A drive which is definitely over 98 per cent. efficient.
5. The efficient operation of vertical drives.
6. *A drive with all advantages of toothed gears and, in addition, the flexibility and 'safety-valve' qualities of a belt drive.*
7. The use of belt speeds up to 8,000 ft. per minute.
8. Much reduced belt tension, with the consequent drop in bearing pressure.

The belts are made to transmit powers as shown by the following table :—

'LINCONA' JOINTLESS DRIVING BANDS.

Speed of Belt in ft. per min.	Horse-Power Table per Inch of Width.							
	3-Ply.	4-Ply.	5 Ply.	6-Ply.	7-Ply.	8-Ply.	9-Ply.	10-Ply.
500	0·68	0·99	1·29	1·59	1·90	2·20	2·50	2·80
1,000	1·36	1·97	2·58	3·18	3·79	4·40	5·00	5·61
1,500	2·05	2·96	3·87	4·77	5·68	6·59	7·50	8·41
2,000	2·73	3·94	5·15	6·37	7·58	8·79	10·00	11·21
2,500	3·41	4·93	6·44	7·96	9·47	10·99	12·50	14·02
3,000	4·09	5·91	7·73	9·55	11·36	13·18	15·00	16·82
3,500	4·77	6·89	9·02	11·14	13·26	15·38	17·50	19·62
4,000	5·46	7·88	10·30	12·73	15·15	17·58	20·00	22·43
4,500	6·14	8·86	11·60	14·32	17·04	19·78	22·50	25·24
5,000	6·82	9·84	12·88	15·90	18·94	21·98	25·00	28·04
5,500	7·50	10·82	14·16	17·48	20·84	24·18	27·50	30·84
6,000	8·18	11·80	15·44	19·06	22·74	26·38	30·00	33·64
6,500	8·53	12·32	16·10	19·89	23·69	27·49	31·26	35·06
7,000	8·64	12·47	16·30	20·14	23·99	27·83	31·63	35·50
7,500	8·51	12·28	16·05	19·84	23·62	27·41	31·17	34·95
8,000	8·22	11·86	15·50	19·16	22·81	26·47	30·10	33·76

Link or Chain Belts.

HORSE-POWER TRANSMITTED BY LINK OR CHAIN BELTS PER INCH OF WIDTH.

Velocity of Belt in Feet per Minute.	Ins. $\frac{3}{8}$	Ins. $\frac{1}{2}$	Ins. $\frac{5}{8}$	Ins. $\frac{3}{4}$	Ins. $\frac{7}{8}$	Ins. 1
100	0·13	0·15	0·17	0·20	0·24	0·27
200	0·25	0·29	0·35	0·40	0·47	0·55
300	0·38	0·44	0·52	0·60	0·71	0·82
400	0·51	0·58	0·69	0·80	0·95	1·09
500	0·64	0·73	0·86	1·00	1·18	1·36
600	0·76	0·87	1·04	1·20	1·42	1·64
700	0·89	1·02	1·21	1·40	1·65	1·91
800	0·92	1·16	1·38	1·60	1·89	2·18
900	1·05	1·31	1·55	1·80	2·13	2·45
1,000	1·27	1·45	1·73	2·00	2·36	2·73
1,100	1·40	1·60	1·90	2·20	2·60	3·00
1,200	1·53	1·75	2·07	2·40	2·84	3·27
1,300	1·65	1·89	2·25	2·60	3·07	3·55
1,400	1·78	2·04	2·42	2·80	3·31	3·82
1,500	1·91	2·18	2·59	3·00	3·55	4·09
1,600	2·04	2·33	2·76	3·20	3·78	4·36
1,700	2·16	2·47	2·94	3·40	4·02	4·64
1,800	2·29	2·62	3·11	3·60	4·25	4·91
1,900	2·42	2·76	3·28	3·80	4·49	5·18
2,000	2·55	2·91	3·45	4·00	4·73	5·45
2,100	2·67	3·05	3·63	4·20	4·96	5·73
2,200	2·80	3·20	3·80	4·40	5·20	6·00
2,300	2·93	3·35	3·97	4·60	5·44	6·27
2,400	3·05	3·49	4·15	4·80	5·67	6·55
2,500	3·18	3·64	4·32	5·00	5·91	5·82
2,600	3·24	3·78	4·49	5·20	6·15	7·09
2,700	3·28	3·85	4·66	5·40	6·38	7·36
2,800	3·31	3·86	4·73	5·60	6·62	7·64
2,900	3·32	3·87	4·78	5·80	6·85	7·91
3,000	3·31	3·86	4·75	5·97	7·09	8·16
3,100	3·30	3·85	4·73	5·96	7·33	8·45
3,200	3·28	3·82	4·71	5·94	7·37	8·73
3,300	3·24	3·77	4·70	5·92	7·35	8·88
3,400	3·19	3·71	4·64	5·87	7·32	8·86
3,500	3·13	3·61	4·50	5·78	7·26	8·80
3,600	3·05	3·50	4·37	5·67	7·16	8·73
3,700	2·96	3·39	4·26	5·55	7·01	8·58
3,800	2·84	3·28	4·15	5·41	6·87	8·41
3,900	2·72	3·13	4·02	5·20	6·70	8·27
4,000	2·58	2·95	3·84	5·01	6·48	8·04
4,200	2·27	2·55	3·37	4·52	5·98	7·51
4,500	1·64	1·77	2·45	3·68	5·05	6·55
5,000	0·42	0·15	0·61	1·55	2·78	4·32

(*A. G. Brown.*)

Coefficients of Friction.

Dry new leather	·155 (average)
Old greased belts on well-polished pulleys .	·200 to ·220
,, ,, on pulleys not well-polished	·300 to ·330
Moist new leather	·120
Ropes, cords, twine, and sewing thread . .	·070 to ·075

Steel Belting.

Steel belts have been recently introduced for the purpose of transmitting power, and are in somewhat extensive use on the Continent. These belts are from $\frac{2}{10}$ to $\frac{9}{10}$ mm. thick, and from 22 mm. to 200 mm. wide, as required. The steel for such belts is of a very high temper and is manufactured specially for this purpose. The difficulty of jointing the belt is one of the great objections. At present the joint is generally made by the makers, and is usually effected

by soldering the ends to a keep plate on both sides of the band, and made doubly secure by screws. The length of the belt is obtained by stretching a special tape to a predetermined tension across the pulley, and both ends of the tape are cut at the overlap.

These steel bands run quite well on the bare pulley, but there is a tendency to polish the same, and it is therefore desirable to cover the pulley with a very thin cork covering, which is glued on to canvas which in turn is cemented to the pulley. With this arrangement there appears to be little or no slip whatever. This is proved in practice by the fact that the cork covering does not wear away.

With such a belt, 32 mm. wide by 0·3 mm. thick, and cork-faced pulley, Prof. Kammerer, of the Charlottenburg University, transmitted 26·3 h.p. at a speed of 15·7 metres per second. Both pulleys were 1,250 mm. diameter, and the slip was equal to 0·91 per cent.

ROPE DRIVING.

Material.

As a material for driving ropes hemp has been superseded by cotton; even though a good cotton rope costs roughly three times the price of a Manila rope its greater suitability has resulted in the almost complete disuse of Manila in the British Isles.

In the U.S.A. at the present time there are only one or two rolling-mills which employ Manila ropes, the remainder having gone over to the use of cotton, mainly of the three-strand variety.

Construction.

The objection raised to four- and seven-strand ropes that they must be subjected to unequal rates of stressing is not admitted by all makers. Both three- and four-strand ropes are used for the transmission of power. The three-strand rope is rather more pliable than the four-strand and is rather stronger. On the other hand the four-strand rope has, according to Messrs. Thomas Hart, Ltd., a lower rate of stretch and may have a larger contact surface with the pulley grooves. It is admitted that it is a little more difficult to splice, and that a three-strand rope well spliced is better than a four-strand rope badly spliced. There is, however, an increasing tendency to order endless ropes so that the problem of inexpert splicing is removed.

Pulleys.

Rope driving pulleys are usually made of cast iron. The metal should be of a good blend, which leaves a hard, bright finished surface after machining.

Grooves.

Both straight-sided and curved grooves are in use.

The best angle to employ between the faces varies with conditions. Some makers recommend 40° and some 45° for ropes over 1 in. in diameter, and 30° for smaller sizes. Sharper angles are employed in specal instances.

The grooves of a driving pulley should be polished, and the pulley as a whole should be accurately balanced. Pulley grooves are generally cored in the casting, but are sometimes milled from the solid for small diameters.

The old practice of leaving rope grooves rough 'to help the grip' is fortunately discredited. Groove surfaces left unpolished in the machine shop have to be finished at the cost of the rope, which becomes worn.

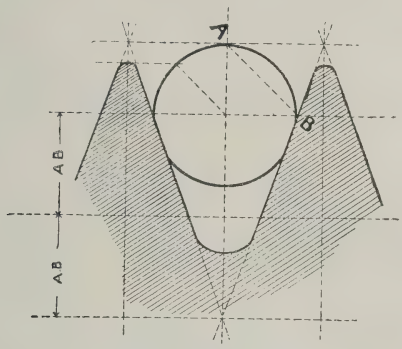


FIG. 11.—Flangeless groove with sides at 40°.

The method of setting out Hart's improved rope groove is shown by figs. 12 and 13. One great advantage of this groove is that it is made so deep as to prevent the rope ever reaching the bottom, a very important point.

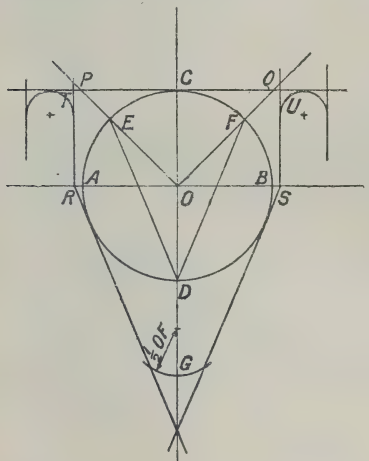


FIG. 12.

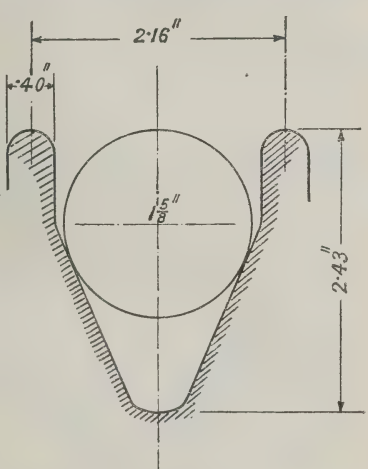


FIG. 13.

Section of Groove for Cotton Driving Ropes. (*Hart.*)

Table of Measurements in Inches.

Diameter of Rope	½	⅝	¾	⅞	1	1⅛	1¼	1⅜	1½	1⅝	1¾	1⅞	2
Pitch of Grooves .	·69	·84	1·02	1·18	1·35	1·51	1·675	1·82	2	2·16	2·35	2·51	2·72
Depth of Groove .	·75	·93	1·125	1·31	1·5	1·68	1.875	2·06	2·25	2·43	2·625	2·81	3
Thickness of Mid-feather . .	·125	·156	·187	·21	·25	·281	·31	·343	·375	·40	·43	·46	·5

CONSTRUCTION FOR A 45° ROPE GROOVE.

With centre O and radius OA, draw a circle equal in diameter to the rope required.

Draw horizontal and vertical diameters AOB and COD.

With A, B, and C as centres, and radius OA, draw arcs to intersect at P and Q.

Join P and Q to centre O, cutting circle at E and F.

Join DE and DF, and parallel to these two lines draw tangents to the circle cutting AB produced at R and S.

Through R and S draw RT and SU parallel to COD, cutting PCQ, produced in T and U.

Make DG = DO and draw arc through G with radius = $\frac{1}{2}$ OA.

Thickness of mid-feathers = $\frac{1}{2}$ OA.

Round off all corners of the groove TRGSU.

SPEED OF PULLEYS.

The limiting peripheral speed for solid c.i. pulleys is 6,000 f.p.m., and 5,3000 f.p.m. for split pulleys. Built-up pulleys with wrot arms, steel spokes, and c.i. hubs have been employed up to 8,750 f.p.m. Experiments in U.S.A. with wooden pulleys have gone up to a peripheral speed of nearly 29,000 f.p.m. without failure.

Porosity of rim at root of arms, cracked bosses, and transverse cracks at smallest sections of the arms in c.i. pulleys are the great dangers which have to be avoided by careful design and regional cooling-off in foundry.

Systems of Rope Driving.

There are two systems of rope driving—the 'multiple rope' system and the 'continuous rope' system. These are sometimes, and without real justification, called the 'English' and 'American' systems respectively. The former is the most commonly used because, consisting

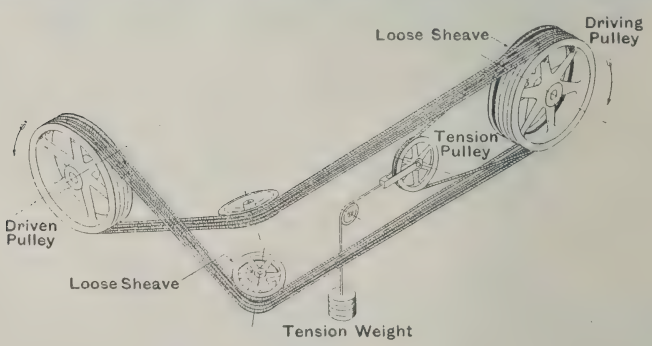


FIG. 14.

of a number of independent endless ropes running side by side, it has a great margin of reliability, which constitutes the unique advantage of rope driving. A crossed right-angle drive on the continuous system is shown diagrammatically by fig. 14.

By a reduction in margin of reliability the continuous drive can be usefully employed in some situations where a multiple drive would be expensive, inefficient, or inconvenient. Such cases are :

(1) Very short centre distances with large loads.
(2) Vertical drives.
(3) Limited widths with large loads.
(4) A few special angular drives.

A rope drive can be successfully arranged, by the aid of guide pulleys if necessary, for any possible relative position of the driving and driven shafts. This is the second great advantage of rope driving.

Design of Rope Drives.

The most important item in the design of a rope drive is to determine the correct ratio between pulley and rope diameters. The smallest pulley on the drive, *whether driving pulley or idler*, is the one to which this ratio applies. The old ' 30 times ' rule is obsolete, because the dimensional and speed limits have been widely extended during the past twenty years. Many rope drives are now running with highly satisfactory results, with a ratio of less than 10 in 1 between smallest pulley diameter and rope diameter.

Experimental work is still being carried out by the leading rope drive firms. The following is a table of rope pulley diameters extracted from a chart published by *Messrs. William Kenyon & Sons, Ltd.* This table is applicable to general heavy power conditions.

Minimum advisable Pulley Diameters in Inches.

Rope Sizes :	¾ In.	⅞ In.	1 In.	1⅛ In.	1¼ In.	1⅜ In.	1½ In.	1⅝ In.	1¾ In.	1⅞ In.	2 Ins.
Rope Speeds :	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins
1,000	10	12½	15	18½	20	24	28	31	35	40	45
1,500	11½	14	16½	20	22	26	29	33	38	42	47
2,000	12½	15	18	21½	24	28	31	35½	41	44½	50
2,500	13½	16½	19½	23	26	30	34	38½	44	48	53½
3,000	15	18	21	25	28½	32½	37	41½	47	51½	57
3,500	16½	19½	23	27	31	35	40	44	50	55	61
4,000	18½	21	25	29	33½	38	43	47½	53½	59	65
4,500	—	23	27	31	36	41	45½	51	57	63	69
5,000	—	25	29	33½	38½	44	49½	54½	60½	67	74
5,500	—	27½	31	35½	41	47	52	58	65	71½	79
6,000	—	30	33	38	44	50	55½	62½	69½	76	84
6,500	—	—	35	40½	47	54	59	67	74	82	90
7,000	—	—	37	43	50	58	63	71½	79	88	96

For high-speed light drives, such as from electric motors running at speeds above 700 r.p.m., it is best to consult a leading firm, as the fullest possibilities are not yet recognised or even definitely tabulated.

Wherever circumstances permit a margin above the lowest possible ratio of pulley to rope diameter should be allowed, as this will result in a direct increase of rope life. In special instances it is permissible to go below the minimum ratio if a shorter rope life is economically justified.

Ropes ½ in. diameter have given over four years' life running on 6 in. and smaller pulleys at speeds of up to 5,000 f.p.m.

Transmissible Power of Ropes.

The power transmissible by a cotton rope depends upon—

(*a*) Groove angle ;
(*b*) Coefficient of friction between rope and metal;
(*c*) Running back tension on drive ;
(*d*) Ratio of pulley diameter to rope diameter;
(*e*) Ratio between pulley diameters.

Obviously some of these factors are functions of several other terms, and any generalisation or table of figures must be handled with caution.

The following table of powers, however, has been found to answer well in general practice.

TABLE OF POWERS FOR COTTON ROPES.

(Messrs. Wm. Kenyon & Sons, Ltd., Dukinfield, near Manchester.)

Velocity in Ft. per Min.	Diameter of Ropes.										
	½″	⅝″	1″	1¼″	1½″	1⅝″	1¾″	1⅞″	2″	2¼″	2½″
2,000	3·6	5·0	6·5	8·2	10·1	12·2	14·9	17·2	20	22·9	26·1
2,200	3·9	5·5	7·1	9·0	11·1	13·4	16·3	18·9	22	25·3	28·7
2,400	4·3	6·0	7·8	9·9	12·1	14·7	17·8	20·7	24	27·6	31·3
2,600	4·6	6·5	8·4	10·7	13·1	15·9	19·2	22·4	26	29·9	33·9
2,800	5·0	7·0	9·1	11·5	14·1	17·1	20·8	24·1	28	32·2	36·5
3,000	5·4	7·5	9·7	12·3	15·1	18·3	22·3	25·8	30	34·5	39·1
3,200	5·7	8·0	10·4	13·2	16·2	19·6	23·8	27·6	32	36·8	41·8
3,400	6·1	8·5	11·0	14·0	17·2	20·8	25·3	29·3	34	39·1	44·4
3,600	6·4	9·0	11·7	14·8	18·2	22·0	26·7	31·0	36	41·4	47·0
3,800	6·8	9·5	12·3	15·6	19·2	23·2	28·2	32·7	38	43·7	49·6
4,000	7·2	10·0	13·0	16·4	20·2	24·5	29·7	34·5	40	46·0	52·2
4,200	7·5	10·5	13·6	17·3	21·2	25·7	31·2	36·2	42	48·3	54·8
4,400	7·9	11·0	14·3	18·1	22·2	26·9	32·7	37·9	44	50·6	57·4
4,600	8·2	11·5	14·9	18·9	23·2	28·1	34·2	39·6	46	52·9	60·0
4,800	8·6	12·0	15·6	19·8	24·2	29·4	35·7	41·4	48	55·2	62·7
5,000	9·0	12·5	16·2	20·6	25·3	30·6	37·1	43·1	50	57·5	65·3
5,200	9·3	13·0	16·9	21·4	26·3	31·8	38·6	44·8	52	59·8	67·9
5,400	9·7	13·5	17·5	22·2	27·3	33·0	40·1	46·5	54	62·1	70·5
5,600	10·0	14·0	18·2	23·1	28·3	34·3	41·6	48·3	56	64·4	73·1
5,800	10·4	14·5	18·8	23·9	29·3	35·5	43·1	50·0	58	66·7	75·7
6,000	10·8	15·0	19·5	24·7	30·3	36·7	44·6	51·7	60	69·0	78·3
6,200	11·1	15·5	20·1	25·5	31·3	37·9	46·1	53·4	62	71·3	80·9
6,400	11·5	16·0	20·8	26·4	32·4	39·2	47·6	55·2	64	73·6	83·6
6,600	11·8	16·5	21·4	27·2	33·4	40·4	49·0	56·9	66	75·9	86·2
6,800	12·2	17·0	22·1	28·0	34·4	41·6	50·5	58·5	68	78·2	88·8
7,000	12·6	17·5	22·7	28·8	35·4	42·8	52·0	60·3	70	80·5	91·4

At a rough generalisation these figures may be accepted under the following conditions:

(1) Pulley grooves not exceeding 45° in angle and straight sided.
(2) Minimum centre distance approximately equal to twice the mean pulley diameter.
(3) Inclination to horizontal not exceeding 30°.

For worn or wide angle grooves, additional ropes are necessary.

The permissible ratio of tight side to slack side tension for different groove angles are given below:

Groove Angle.	Groove Coefficient of Friction. μ_g.	Maximum Advisable Value of $\frac{T_1}{T_2}$.
30°	0·36	3·1
35°	0·298	2·5
40°	0·27	2·3
45°	0·24	2·125

Taking the coefficient of friction of lubricated cotton rope on polished cast iron as 0·0926, any value can be determined from the expressions:

$$\frac{T_1}{T_2} = e^{a\mu_g} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where μ_g = groove coefficient;

a = arc of contact of ropes on smallest pulley, in radians;

e = base of Naperian logs.

And

$$\mu_1 = \mu \operatorname{cosec} \frac{\theta}{2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where

μ = real coefficient of friction for flat surfaces.

θ = groove angle in degrees.

Centre distance is not limited except where the slack side of a drive is uppermost. Then the drive should be no longer than will permit a maximum sag of 10 per cent. before the two sides of the drive rub.

Rolling-mill drives have been installed with the pulley rims almost touching. If there is no limit to width, the multiple system may be used. If width is limited, the continuous system with a suitable tension weight will generally overcome the difficulty.

Rope Driving Speed.

At one time an arbitrary critical speed was fixed for rope driving at 4,800 f.p.m. This was determined by fixing a maximum permissible tension for each rope and subtracting the centrifugal

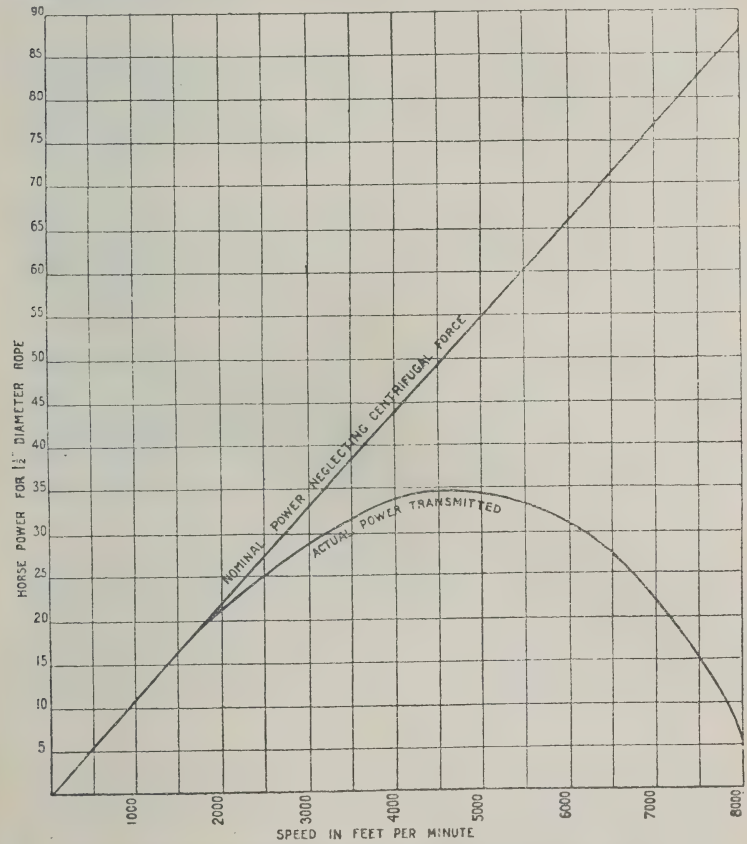


FIG. 15.—Diagram showing the effect of Centrifugal Force at different speeds of rope.

tension from this, the remainder being assumed to be the determining factor in power capacity. The result of these assumptions was a curve of horse-power capacity reading a maximum at 4,800 f.p.m. and then falling back to zero at about 8,300 f.p.m. as shown in fig. 15.

The total direct tension on a driving rope is, however, of small importance. The resistance to slip in the grooves is what determines maximum capacity, since a rope will slip along before the tension approaches the ultimate strength of the rope.

Furthermore, bending stresses are more important than direct stresses from the point of view of wear.

In practice the maximum direct stress in a driving rope at 4,500 f.p.m. does not exceed $\frac{1}{10}$th of its ultimate strength, and may, therefore, be increased appreciably without detriment, *provided the slipping point is not reached.*

Increase of speed actually raises the slipping point, because, if T_c is centrifugal tension; then the total running tension on the tight and slack sides respectively become $T_1 + T_c$ and $T_2 + T_c$, and it is obvious that

$$\frac{T_1}{T_2} > \frac{T_1 + T_c}{T_2 + T_c}$$

That is to say, for constant values of $e^{\alpha\mu}$ and T_2, as T_c increases, T_1 may be increased also. The power transmitted is a direct function of $(T_1 - T_2)$ so that power capacity actually increases with speed instead of decreasing. Allowance has to be made for bending stresses and increased rate of wear, however, so that the curve of capacity is actually below the straight line shown on fig. 15, but it continues to have a positive inclination up to a speed of 10,000 f.p.m. This explains why 2 in. diameter cotton ropes are actually transmitting 125–150 h.p. each at 8,750 f.p.m. on a Belgian rolling mill drive, with satisfactory results, when, according to the old theory, they ought not to be capable of transmitting power satisfactory at all.

Steadiness.

The use of cotton driving ropes for rolling mills in order to cushion heavy 'peak' loads is quite old; only recently, however, is the importance of resilience being appreciated in other directions. A book on 'The Driving of Spinning Frames,' published by Messrs. Wm. Kenyon & Sons, Ltd., gives a selection of tachograph records showing very strikingly how an irregular torque on a lineshaft can be damped out or considerably improved by the intervention of a rope drive.

Efficiency of Rope Driving.

Accurate and detailed data in respect of large drives is very scanty. Tests carried out by Prof. Bonte at Charlottenburg before 1914 on a drive of 200 h.p. capacity showed that rope efficiency increased with both speed and with loading. There is no simple law, however, and low loading it appears probable that efficiency will fall off beyond a speed of approximately 4,600 f.p.m.

For average rates of loading efficiencies of from 96 per cent. at 1,500 f.p.m. to 98·7 per cent. at 5,000 f.p.m. were obtained. This seems to indicate that the friction losses on ropes running under ordinary industrial conditions need not exceed 3 per cent.

Small drives usually have slightly lower efficiencies on account of the lower ratio of pulley to rope diameters, but at very high speeds efficiencies of 99 per cent. have been obtained.

Drives with Irregular Impulses or Loads.

When either the load or impulse is irregular, there are two main effects which must be provided against. The first is, if the ropes are fully loaded, or nearly so, for the general conditions of the drive, the irregular load or impulse has the effect of causing the ropes to slip, although under a load equal to the average load the drives might run perfectly well. In driving from single cylinder gas-engines (old type) as much as 50 per cent. extra ropes were sometimes found necessary. For a modern gas-engine 10 per cent. extra ropes are advisable.

The second is, that with either irregular impulses or loads, the ropes have a tendency to lash about, which, if not controlled, tends to excessive wear. The best thing to do is to regulate by flywheel effect and thus get rid of this action altogether. This, however, is not always possible. In that case the provision of a rope guard is advisable.

Rope Guards.

A design of the rope guard which has been used with great success is shown in fig. 16, and the method of installation in fig. 17.

To construct this, first determine the pitch, P, of the pulley grooves; let D = rope diameter, then thickness of blade $= P - \left(D + \frac{D}{8}\right)$.

The blades should be made of a good hard wood of even grain, without knots, with rounded edges, as shown on enlarged section. The ends are shaped to fit slots in cross-bars of frame where they are held in place by stretchers secured with coach-screws, thus facilitating replacement in case of breakage.

ROPE GUARD.

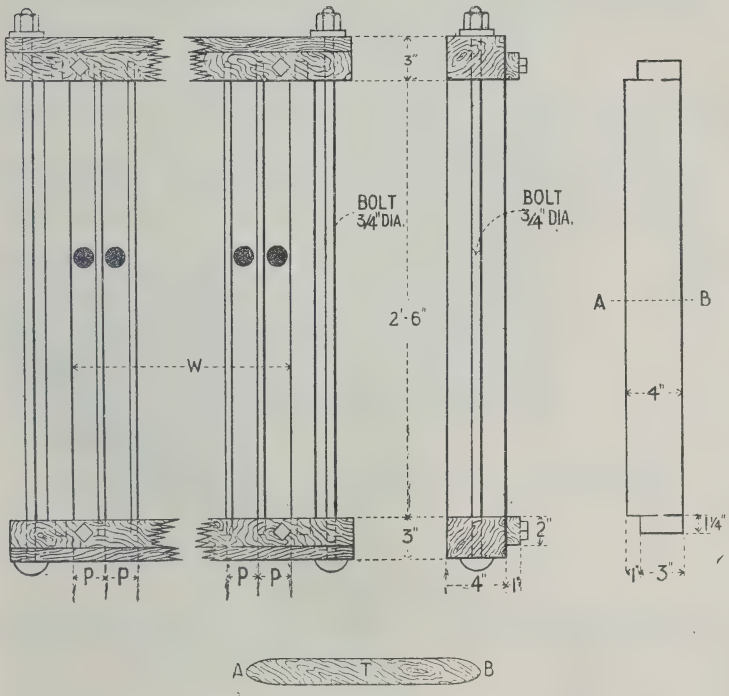


FIG. 16.

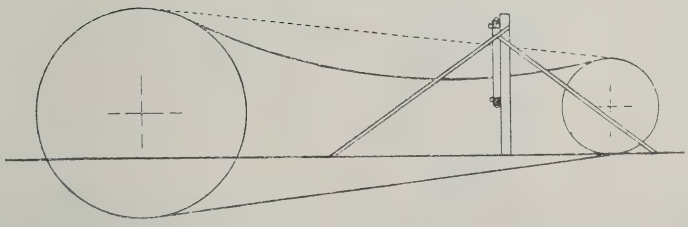


FIG. 17.

The method of applying this rope guard depends upon the relative positions of the two pulleys and direction of rotation. In oblique driving it is often attached to the roof or sides of a building, stayed by iron rods and tilted to meet the driving angle. These thin wooden blades are quite sufficient to stop the largest ropes from lashing; the action of lashing being accumulative, the blades steady each small impulse as it arises.

Long keyways are advantageous in fixing up angular drives, as they enable the pulley to be moved to the most desirable position in the circumstances.

Adaptability of Rope Drives.

In the case of a vertical half-cross drive, such as that shown in fig. 18, it is only necessary for success to ensure that the relations of the pulleys are such as to allow the slack to fall in a direct line, while the tight is deflected in the manner shown in the figure.

A right-angle drive is shown in fig. 19.

It will be seen that in this case the rope runs over four guide pulleys, so arranged in pairs of unequal size that the larger diameter leads to the smaller one, and *vice versa*. These guides run freely on studs which may be moved along the slotted sides of a rectangular bracket, thus serving as rope tighteners if required.

FIG. 18.

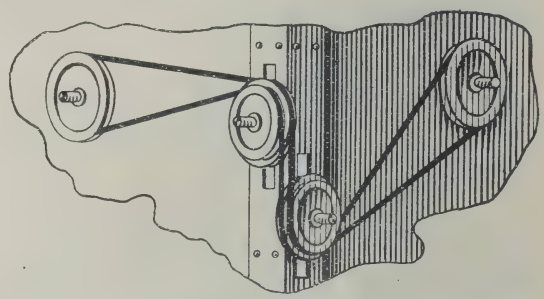


FIG. 19.

FAST AND LOOSE PULLEY.

Fig. 20 shows this combination in its most approved form. The working groove is set out at an angle of 30° on the lines previously described. A shallow intermediate groove provides for a gradual start and knock off, corresponding to the slip of a belt. The loose groove on the

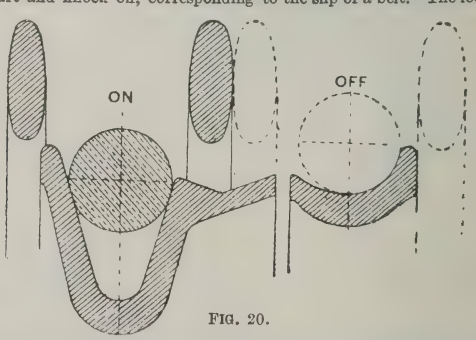
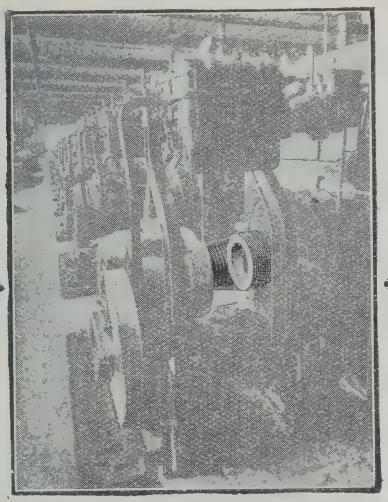


FIG. 20.

right is simply hollowed out to fit the rope, and is flanged on one side only. The rope shifter usually takes the curve of the pulleys, and is attached to a radius shaft and actuated by a small lever. A slight force on the slack side suffices to pass the ropes in or out of gear, and there is very little friction.



ROPE DRIVING

even under conditions
such as the above is

POSITIVE
RELIABLE
EFFICIENT
ECONOMICAL

For full information and advice
consult our Technical Dept.

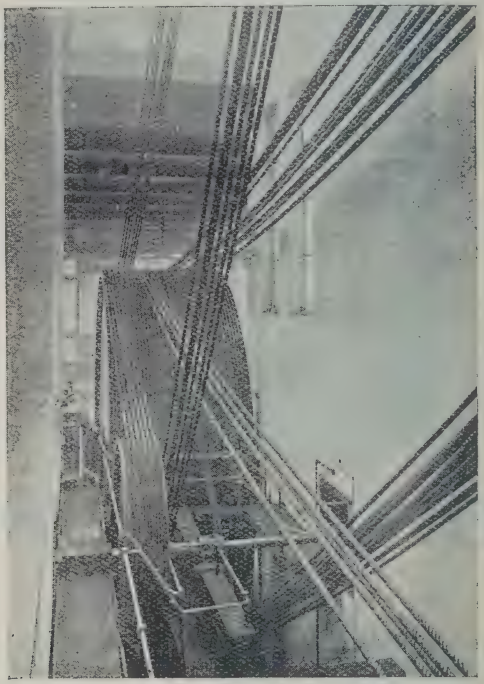
WM. KENYON & SONS, LTD.
Chapel Field Works, Dukinfield.

Rope Driving

affords the most advantageous System of Power Transmission. It is reliable, positive, shock absorbing, and offers sustained efficiency. Better than Gear, Belt or Chain Driving.

Lambeth Cotton Driving Ropes are in world wide use.

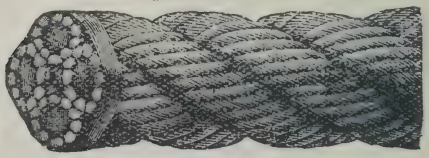
Everywhere recognised as the best.



Lambeth Cotton Driving Ropes have less stretch than any other make and will carry more power.

Illustration showing Rope Drives in Cotton Spinning Mill Engine 1750 I.H.P.

LAMBETH COTTON MAIN DRIVING ROPE, 4-STRAND.



(Also made 3-Strand)

Made by a firm with over 130 years' experience, and who were practically the pioneers of Cotton Rope Driving.

WRITE FOR 84-PAGE BOOK, 'ROPE DRIVING.'

THOMAS HART, LTD.

Rope Driving Specialists,
LAMBETH ROPE WORKS, BLACKBURN, ENGLAND.
Also makers of Hart's Special Mill Bandings and Hart's Anti-fraying Composition for Driving Ropes.

Telegraphic Address : 'Hart, Blackburn' (all Codes used).

Telephone : Blackburn 4264 (2 lines).

E 870

Notes on Rope Driving.

The following *Hints to users of ropes* are taken from Mr. Thomas Hart's book on 'Rope Driving':—

Take care that ropes do not get wet; water does them much harm.

Take care that ropes do not rub on anything in their flight, either when running at full speed or when starting and stopping the engine.

Take care that ropes do not slip; if they slip they are probably overloaded, or are too small for the grooves.

All ropes should be lubricated with a good anti-fraying and lubricating composition. Black-lead mixtures should not be used.

Instructions for Splicing Lambeth Cotton Driving Ropes (Four Strand).

(*Thomas Hart, Ltd., Blackburn.*)

Storing, Stretching, etc.

If it is necessary to store the ropes before fixing, they should be kept in a dry room, not on an earthen floor. When ready for fixing uncoil the ropes left-handed—the opposite to the way the hands of a clock move. Then stretch out with a pair of hand blocks, but do not allow the rope to revolve during this operation, or the 'turn' or 'twist' will be taken out of it. Everything depends upon the strength of shafting, etc., as to how tight the rope should be stretched, but under ordinary circumstances it is usual to have about four men pulling on a pair of hand blocks. Pass a string around the drive to get the exact running length of rope, and by tying two bands around the rope, mark off the exact running length whilst it is on the stretch, allowing ten feet (five feet each end) for the splice. For ropes 2 ins. diameter allow 12 feet of splice.

Splicing.

Unlay the five feet each end that has been allowed for splice, that is, as far back as the bands which mark the running length of rope; it is advisable to tie a string around the end of each of the loose strands to prevent the twist from coming out. The strands are now interlaced together, one from the right-hand side being next to one from the left-hand side, and so on, as shown in fig. 21. These strands should be pulled tight so that the bands butt together, and

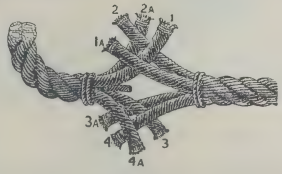


FIG. 21.

the loose strands on one side tied temporarily around the rope to keep the two ends of the rope in the position they now are.

Cut one of the bands, say the one on the right, and unlay strand No. 1A, at the same time laying in its place its fellow strand No. 1 from the opposite side of the rope. Do this for about four feet, and tie the two strands temporarily until all are ready for tucking in. The next strand to this, as will be seen in fig. 21, is numbered 2, but do not touch this now; take the next strand but one, numbered 3A, and unlay it, laying in its place strand No. 3, but only for about 1 ft. 6 ins.; tie these temporarily. It is very important in splicing four-strand ropes that two strands next to each other should not be laid up in the same direction.

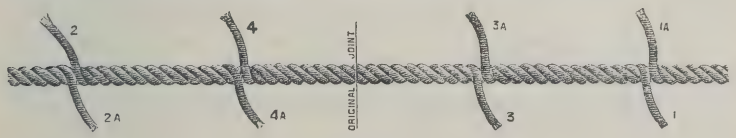


FIG. 22.

Next cut the other band, and proceed in the same manner with strands No. 2 and No. 4, but of course in the opposite direction. Care should be taken to keep the 'turn' or 'twist' in the strands, and they should be laid well and evenly down in their places. The splice should now be as fig. 22.

Begin now at joint No. 1 and 1A; untie the temporary fastening and shorten the strands to equal lengths of about 1 ft. 6 ins. An examination of the strands will show that the outside layer is composed of ten threads; for reference we will call these the friction bands; the internal portion of the strand we will call the tension strand. Take No. 1A and remove from it the ten friction bands and divide them into two lots of five, but do not cut them off. Now take strand No. 1 and unlay it two turns or laps, and whilst it is there remove from it the ten friction bands, divide them into two lots of five, but do not cut off. Now lay in again the tension strand only of No. 1 turn, and reduce it by leaving out about one-quarter, lay up the remaining three-quarters one turn, thus bringing it up to tension strand of No. 1A; tie the two with an overhand knot. At this knot the rope should be about its original diameter, and the joint should appear as shown in fig. 23.

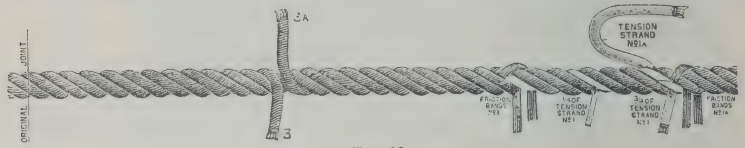


FIG. 23.

Take tension strand 1A and with the splicing pin work it under and over tension strand No. 1, as shown at fig. 24, passing it under and over spirally about five times; it will then have reached friction bands No. 1.

FIG. 24.

Now reduce it by cutting out a portion, and pass the remainder between the two lots of five friction bands, and lock by passing once or twice through the centre of the rope. This is clearly shown at fig. 25. Five friction bands are also locked through the rope as shown. This complete, turn to three-quarters of tension strand No. 1; this is locked exactly as tension strand No. 1A,

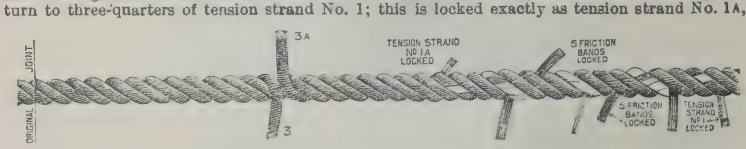


FIG. 25.

five of the friction bands are also locked as previously shown, and the splice should now be as fig. 25.

All loose ends are cut off and this portion is complete. Treat the two other joints in the same way and the splice is complete.

How to put a Rope on the Pulleys.

Put the rope on the small pulley, and as far around the large pulley as it will go; lash it there, but have the lashing so that the rope has freedom to slip through it. Then bar pulley slowly round by hand, or barring engine if there is one, and the rope will fall into its place. The groove of the small pulley and the face of the large pulley where the rope will touch in barring on should be well greased with tallow to prevent wedging. A piece of canvas may also be put on the rim of the large pulley to prevent it cutting the rope. (*Thomas Hart, Ltd.*)

Sizes of Pulleys and Drums.

The sizes of pulleys and drums should preferably be about ten times the circumference of the ropes to be used on the same, or, say, 1 foot in the diameter of the former for each pound of rope per fathom—*e.g.* a drum 3½ feet in diameter for a rope weighing 3½ lbs. per fathom.

V-Belting.

The capacity of a belt to transmit power depends upon the tension, the coefficient of friction, and the area of contact. By giving a belt the V-shape, the area of contact is greatly increased, for contact is along the sides. The area of contact also is increased by the wedge action of the belt in the groove, which forces the belt against the sides of the groove, and ensures that the entire surface of the belt passing over the pulley actually will be in contact with it. The V-belt gives maximum power transmission for a given width of pulley.

Whether a flat belt or a V-belt should be used depends largely upon the power to be transmitted and the speed at which the belt must be run. The V-belt will be found to operate satisfactorily at speeds as high as 3,500 feet a minute, while with flat belts on small pulleys, speeds up to 10,000 feet a minute are practicable. Single V-belts of $\frac{3}{4}$-in. width, running at 3,500 feet a minute on correctly designed pulleys, will transmit up to 3 h.p. If more power than this is to be transmitted, it will be found better practice to use twin belts, as shown at C, fig. 26, rather than to increase the size of the belts.

Pulley Grooves for V-Belts.

The pulley groove for a V-belt should be at least $\frac{3}{16}$ in. deeper than the belt thickness, to provide clearance, as indicated in fig. 26. A clearance of $\frac{1}{4}$ in. is better and usually will be found ample in any case. A V-belt must not bottom in the groove, as this would prevent the proper

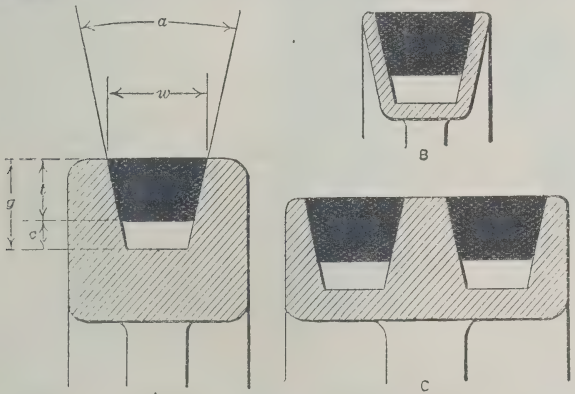


Fig. 26.

contact between the sides of the belt and the sides of the groove, and would seriously reduce the ability of the belt to transmit power. The width w of the pulley groove across the top of the V should be the same as the width of the belt, with a permissible tolerance of $\frac{1}{32}$ in. The sides of the groove, which form the contact surfaces, should be smooth, care being taken to avoid ridges, tool chatter marks, and other defects, as these cause rapid belt wear. The rim of the pulley should be as light as possible, as shown at B, except when a flywheel effect is desired, in which case the mass of metal in the rim should be increased, as shown at A.

Experiments have resulted in a recommended angle a of 38° for two-pulley drives. An included angle of 28° is recommended for the smaller pulleys of a three-pulley drive, due to the reduction of angular contact in this arrangement. The thickness t of a V-belt should be from 0·5 to 0·6 of the width w at the widest part of the V.

The 'Lincona' Balata 'V' Belt Rope.

(Patent.)

(*James Dawson & Son, Ltd., Boultham Works, Lincoln.*)

This driving rope, fig. 27, consists of a number of sections, each being built up of piles of cotton duck thoroughly impregnated with Balata gum (*Minusops Globosa*). The sections are held together with the Balata gum, and by a specially shaped rivet designed to prevent any creeping of the sections.

The belt rope is built up in such shape that the driving surfaces exactly fit the sides of the pulley grooves, fig. 28, and so provide a maximum of frictional surface in relation to the cubic measurements of the rope.

Advantages claimed for the 'Lincona' Balata 'V' belt rope are:—

1. It has a large frictional contact surface with the sides of the pulley groove, distributing the weight of the pull and protecting the internal fibres from side pressure, thereby ensuring long life.
2. The coefficient of friction is high, as no dressing or lubrication is necessary.
3. It can stand a heavy economical working tension on the pulling side. The breaking strain of a 'Lincona' Balata 'V' rope of 1 sq. in. section is 5,000 lbs.
4. As a result of the above it is capable of taking the heaviest loads at the highest speeds. A 'Lincona' Balata 'V' rope of 1 sq. in. section will transmit 60 b.h.p. at 4,000 ft. per minute, and give a satisfactory life and freedom from stretch.
5. Because of its high power transmission, a small number of ropes are required, therefore little power is absorbed in the transmission itself.
6. The joint is a simple splice, which any handy man can learn to make in a few minutes; thus the work of putting on the ropes can be done on the spot and at once.
7. No dressing and no attention needed once the ropes have been fitted.
8. No ill effects from damp or exposed conditions.

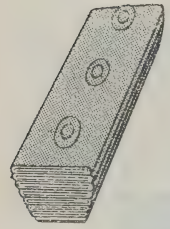


FIG. 27.

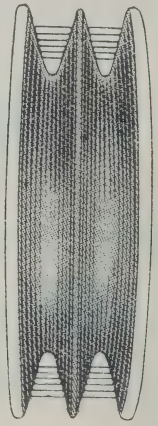
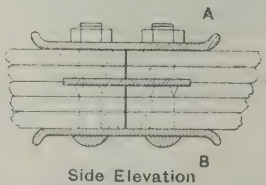


FIG. 28.



Side Elevation

Section through A.B.

FIG. 29.

This type of rope is being used extensively on steel rolling mill drives, main drives, pumps of all descriptions, fans, D.C. generators, and haulage gearing, and is giving improved service and life.

It can be satisfactorily fitted to grooves already in existence for interstranded cotton ropes, some slight alteration in design from the standard rope being necessary to effect a perfect fit in the groove. New installations should be fitted with pulleys having grooves of an angle of 28°.

When *Jointing*, the belt is drawn together by means of screw clamps, the several sections are cemented and riveted, the butted joints of each section being staggered in such a manner as to ensure the rivets being kept upright.

When fitting or taking up it is important that only the sections of the butted joints on one side are cut. The sections of the other half of the joint then act as a guide to correct cutting.

A fastener capable of holding together the butt ends of a Lincona belt rope has recently been devised; it forms a compact and strong joint which will stand up continuously to the heavy strains which it is required to bear. The arrangement, fig. 29, consists of three strong metal plates, A, B, and C, provided with spikes (shown dotted), the centre plate having these spikes on both sides. Bolts pass through the plates and bind the latter and the ends of the belt firmly together.

The following table represents length of life and freedom from bearing trouble rather than a high power figure; it will be found to be reliable in practice for speeds up to 6,000 ft. per minute.

'Lincona' Balata 'V' ropes have been tested up to 7,000 ft. per minute and found to transmit the power given in the table.

From the table it will be seen that the horse-power is increasingly transmitted up to 7,000 feet per minute rope-speed, with a falling off noticeable after 4,000 feet per minute due to centrifugal force.

In calculating the number of ropes required for a drive, it is important that allowance should be made if the arc of contact of the rope on the smaller pulley embraces less than 180° of the circumference.

TABLE SHOWING HORSE-POWER TRANSMITTED BY 'LINCONA' BALATA 'V' BELT ROPE.
(Cross-section area of belt = 1 sq. in. Speed in ft. per minute.)

Speed	1000	2000	2500	2600	2700	2800	2900	3000	3100	3200	3300	3400	3500	3600
H.P. .	12·5	24·7	30·6	31·5	32·7	33·7	34·9	35·9	37·0	37·9	39·0	40·0	40·9	41·8
Speed	3700	3800	3900	4000	4100	4200	4300	4400	4500	4600	4700	4800	4900	5000
H.P. .	42·8	43·7	44·5	45·0	46·3	47·0	48·0	48·7	49·4	50·2	50·8	51·4	52·3	52·5
Speed	5100	5200	5300	5400	5500	5600	5700	5800	5900	6000	6500	7000	7500	8000
H.P. .	53·2	54·1	54·5	55·0	55·7	56·0	56·3	57·0	57·2	57·4	58·8	59·2	58·5	57·0

For any other cross-section area, multiply horse-power by this area.

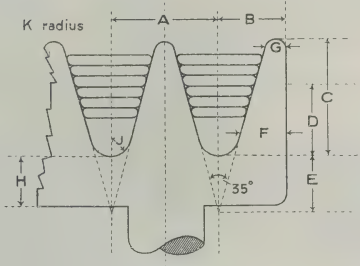
The following allowance should be made:—

Arc	180°	170°	150°	140°	130°	120°
Percentage of power transmitted .	100	97	95	90	85	80

It must be borne in mind that the arc of contact is lessened if the slack is on the bottom of the drive.

An allowance, depending on the tension that can be put on the ropes, must be made if the drive is vertical or at an acute angle.

'LINCONA' 'V' BELT ROPES, STANDARD BELTS, AND PULLEYS.



Area in Sq. Ins.	Min. Pulley Diam.	Dimensions in Ins.								
		A	B	C	D	E	F	H	J	K
	Ft. Ins.									
0·25	1 4	1 7/16	7/8	1 1/4	2 7/32	7/16	1 1/16	1/2	3/16	3/16
0·38	2 0	1 7/32	7/8	1 1/4	3/4	7/16	1 1/16	1/2	3/16	3/16
0·5	2 0	1 9/16	1	1 9/16	1	19/32	3/4	5/8	1/4	1/4
0·65	2 8	1 9/16	1	1 1/2	29/32	19/32	3/4	5/8	1/4	5/16
0·8	2 8	1 7/8	1 1/4	1 3/4	1 3/16	3/4	7/8	3/4	5/16	5/16
1·0	3 4	1 7/8	1 1/4	1 3/4	1 3/32	3/4	7/8	3/4	5/16	5/16
1·35	4 0	2 1/16	1 3/8	2 1/16	1 7/16	7/8	1	7/8	3/8	9/32
1·5	4 2	2 1/8	1 3/8	2 1/16	1 3/32	1	1 1/16	1	7/16	3/16
1·75	4 8	2 5/16	1 1/2	2 1/4	1 13/32	27/32	1 1/8	1 1/16	7/16	5/16
2·0	4 10	2 7/16	1 9/16	2 1/2	1 7/16	1 1/32	1 1/16	1 1/4	7/16	11/32
2·25	5 4	2 5/8	1 5/8	2 1/2	1 1/2	1	1 5/16	1 5/16	7/16	11/32
2·5	5 6	2 21/32	1 1/4	2 5/8	1 9/16	3/16	1 3/16	1 3/8	1/2	13/32

HORSE-POWER TABLE FOR DICKBALATA ELEMENT DRIVING ROPES.

Speed of Rope in ft. per min.	¾-in.	⅞-in.	1-in.	1⅛-in.	1¼-in.	1⅜-in.	1½-in.	1⅝-in.	1¾-in.	1⅞-in.	2-in.	2⅛-in.	2¼-in.
1,000	3·5	5·5	8·3	10·4	12·5	14·5	17·0	20·1	22·8	25·1	27·8	30·8	33·8
2,000	6·9	11·7	16·3	20·5	24·6	28·5	33·5	39·6	44·9	49·4	54·8	60·7	66·6
2,500	8·5	14·4	20·0	25·2	30·3	35·1	41·2	48·7	55·3	60·8	67·5	74·7	82·0
2,600	8·8	14·9	20·7	26·9	31·4	36·3	42·6	50·4	57·2	62·9	69·9	77·3	83·7
2,800	9·4	15·9	22·1	28·7	33·4	38·7	45·5	53·8	61·1	67·1	74·6	81·4	89·4
3,000	10·0	17·0	23·5	30·5	35·5	41·1	48·4	56·1	65·0	71·3	79·3	86·6	95·1
3,200	10·6	18·0	24·9	32·3	37·6	43·5	51·3	59·4	68·9	75·5	84·0	91·7	100·8
3,400	11·1	18·8	26·0	33·8	39·4	45·6	53·7	62·2	72·1	79·1	88·0	96·1	105·5
3,600	11·6	19·7	27·2	35·3	41·1	47·6	56·1	65·0	75·4	82·7	91·9	100·4	110·3
3,800	12·0	20·4	28·2	36·7	42·7	49·4	57·2	67·6	78·5	87·0	95·9	104·5	115·0
4,000	12·5	21·2	29·3	38·1	44·3	51·3	60·5	70·1	81·2	89·2	99·2	108·2	119·0
4,200	13·1	22·2	30·7	39·9	46·5	53·8	63·4	73·4	85·1	93·4	103·8	113·4	124·5
4,400	13·6	23·1	31·9	41·4	48·3	55·8	65·8	76·2	88·4	96·9	107·8	117·7	129·3
4,600	14·0	23·8	32·9	42·7	49·7	57·5	67·7	78·5	91·0	99·8	111·0	121·2	133·1
4,800	14·3	24·3	33·6	43·6	50·7	58·7	69·2	80·2	92·9	101·9	113·3	123·8	135·8
5,000	14·6	24·8	34·3	44·5	51·8	60·0	70·6	81·9	94·9	104·1	115·7	126·4	138·8
5,200	15·0	25·5	35·2	45·7	53·2	61·6	72·6	84·1	97·5	106·9	118·9	129·9	142·6
5,400	15·3	26·0	35·9	46·6	54·3	42·8	74·0	85·8	99·4	109·0	121·3	132·4	145·5
5,500	15·5	26·3	36·4	47·2	55·0	63·6	75·0	86·9	100·7	110·5	122·9	134·2	147·4

Branches :

LONDON : LEEDS : MANCHESTER :
BIRMINGHAM : NEWCASTLE-ON-TYNE :
BRISTOL : DUNDEE : BELFAST.

DICKBALATA
ELEMENT DRIVING
ROPES

A super driving rope of exceptional flexibility and durability, built for maximum service and guaranteed to give it.

◆ ◆ ◆

Easy to splice

Has Long Life

Price List on application

6 Ropes 2″×2¼″ transmitting 700 BHP. 3 1½″ Ropes driving Dynamo.

Positive grip owing to the large surface contact of the 'V' section

R. & J. Dick Limited

GREENHEAD WORKS, GLASGOW.

PATENTEES & ORIGINAL MAKERS OF

BALATA BELTING

AND DICK BALATA ELEMENT DRIVING ROPE

The Dickbalata Element Driving Rope.

(R. & J. Dick, Ltd., Greenhead Works, Glasgow.)

This rope is constructed V shape, as shown by fig. 30. The binder or top portion is made like ordinary Dickbelt (p. 856), with Dickbalata blocks solutioned and riveted to it.

The rope is of very great strength, as can be seen from the horse-power table (page 884); it is claimed to give practically double the power of cotton ropes. It does not need a special expert to fit it; any engineer or belt fitter can make the splice without difficulty and in quick time; the splice is as strong as the main portion of rope.

Fig. 30.

The Dickbalata Rope has positive grip, no slip, runs in steamy or damp atmospheres without being affected, and without noise; no dressing is needed. Owing to its flexible construction, the rope can be used at short centres if necessary and over small pulleys.

The rope can be used for cross-driving.

WIRE ROPES.

Lang's-lay.

In the 'Lang's-lay' all the wires forming the strands, and the strands comprising the rope are laid in the same direction. These ropes have been used for from 3 to 6 years hauling some 3,500,000 tons of coal, at speeds varying from 5 to 15 miles per hour, over distances up to 220,000 miles.

Haulage Wire Ropes, Useful Sizes (Lang's-lay).

Approximate Diameter.	Circumference of Rope.	Number of Strands.	Number of Wires in Strands.	Number of Wires in Core	Gauge of Wires.	Lay of Strands.	Lay of Rope.
Ins.	Ins.				In.	Ins.	Ins.
$\frac{5}{8}$	2	6	6	1	·072	$2\frac{1}{4}$	6
1	3	6	6	1	·105	3	$7\frac{1}{2}$
$1\frac{1}{4}$	4	6	8	7	·116	$3\frac{3}{4}$	9
2	6·2	6	10	7	·146	$4\frac{3}{4}$	13

The Third Report of the Wire Ropes Research Committee of the Institution of Mechanical Engineers gives in the summary the following:

Twelve wires to a strand with a fibre core was the best construction when associated with Lang's-lay.

Ordinary lay was best for nineteen-wire strands, 12 on 6 on 1, except on the pulley of 850 wire diameters.

A soft core wire in a seven-wire strand proved to be undesirable for a rope of high-tensile wire which was used on a small pulley.

A Lang's-lay rope with strands of six 100- to 110-ton wires laid on a soft core wire was best on a large pulley of 500 wire diameters, but even on this pulley the soft core wire was a weakness in a 80- to 90-ton rope.

6 × 6 and 6 × 7 ropes compared favourably with 6 × 12 and 6 × 19 on the same pulley when its diameter equalled 500 diameters of the larger wire, or 18 circumferences of either rope.

'Trulay' ropes were superior to those made by the usual method of manufacture, particularly on small pulleys and in 6 × 7 construction.

The Lang's-lay ropes with 12 wires to the strand on a fibre core gave particularly good results on pulleys of 500 wire diameters and larger, and under these conditions the alleged deformation of this form of rope will not occur.

WIRE ROPE SPLICING.

FIG. 31.

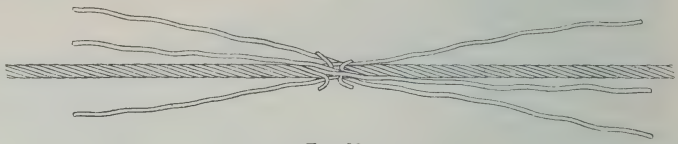


FIG. 32.

FIG. 33.

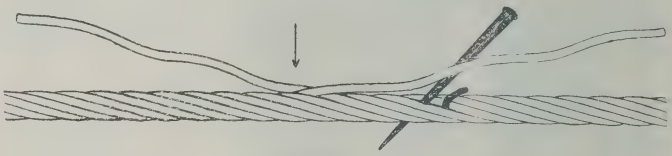


FIG. 34.

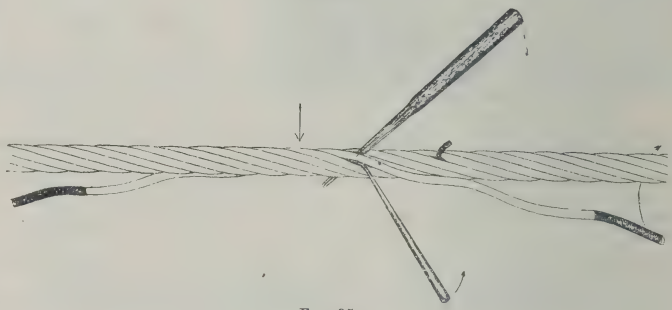


FIG. 35.

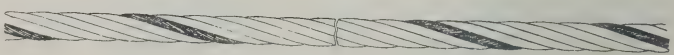


FIG. 36.

The tensile strength of a wire was reduced 18 per cent. by severe lateral pressure.

For 'Horse Power required for Haulage,' see p. 2598.

Wire Rope Splicing.

LONG SPLICE.

This splice is used for haulage and where endless ropes are required.

Mark off both ends the length required for splicing, which depends on the size of rope, say 48 ft., that is 24 ft. each end. Having marked each end, unlay three alternate strands from each end to within 6 ins. of mark, as shown in fig. 31, L, p. 886.

Cut off the three remaining strands with the core and bend back at mark, as shown in fig. 31, R.

Lock both ends together, close up, so that the long strands pass alternately right and left, as shown in fig. 32.

Grip the rope on one side of the lock with tongs, unlay one short strand and lay up in its place the corresponding long strand until within 4 ft. of the end. To facilitate operations, the short strand may now be cut 4 ft. from the cross, the two ends measuring 8 ft. Lay up the other two long strands, the crosses being 8 ft. apart; next repeat the operations with other side, cutting the ends at 4 ft. and the crosses being 8 ft. apart, as shown in fig. 33.

The next operation is to run the ends in and substitute the core. It is advisable to serve the ends, this gives the outer strands a firm grip on them and prevents loose wires protruding through interstices of the strands. Grip the rope with tongs where indicated by arrow, drive the spike through centre of rope three strands in advance of the strand to be run in, and with the aid of short spike pick the core out, as shown in fig. 34.

Run spike back towards the tongs, insert needle over the end in place of the spike, hook spoon under the end, and in same place as the needle, as shown in fig. 35.

Take two or three turns out of the strand and twist it round the rope as indicated by arrow, at the same time bring the handle of the needle and spoon together, as indicated. This forces the strand into centre of rope, and by twisting the needle round the lay the strand will enter the rope one side and force the core out on the other. Cut the core close up to the end of strand, and turn and run the opposite end in exactly the same way.

Run in the other five pairs of ends in exactly the same way as the first pair.

Care should be taken not to run the core too far back, otherwise flat places will occur.

This completes the splice, and if correctly executed there should be very little difference between the splice and any other portion of the rope.

With a Lang's-lay rope the ends of the strands are run into the right of each other, as shown in fig. 36, p. 886.

With the ordinary lay rope the ends of the strands are run into the left of each other.

Wire Rope Socketing.

Wire rope sockets should be made from carefully selected and tested steel of high strength, and be either drop forged or of well-annealed cast steel, depending upon the diameter of the rope to which the socket is attached.

When the work is done in the ordinary way the wires are bent back on each other and the basket of the socket filled in with metal, pins being used to lessen the quantity of metal required. In following this method it is important to have all the wires engaged, otherwise some of them creep back into the rope, thus preventing a perfect contact between the rope and the socket, consequently developing a weakness in the connection due to the disproportionate strain obtained on the strands.

When sockets are attached in the following manner, the full strength of the rope or the socket is obtained:—

The rope is first served securely with soft wire, fig. 37 (p. 888), and the strands opened, as shown for a distance equal to the length of the basket of the socket, and the hemp core cut out.

The wires are then separated in each strand, as shown, fig. 38, and well cleaned from all grease by means of petrol.

The wires are next dipped for a few minutes (not more than five) into a solution of equal parts of hydrochloric acid and water, then washed thoroughly in water, and dried. A stronger acid solution than that indicated should not be used.

The wires are then bound together by means of soft wire, as shown by fig. 39.

The socket being warmed, the wires are pushed through the basket of the socket till they are even with, or a little above, the top. The binding then being pulled off, the wires spring out.

Clay is put round the rope at the bottom of the socket to prevent metal running through, and then melted white metal is poured into the socket at A, fig. 40, until the socket is filled up, care being taken that the metal is not too hot to take the temper out of the wire.

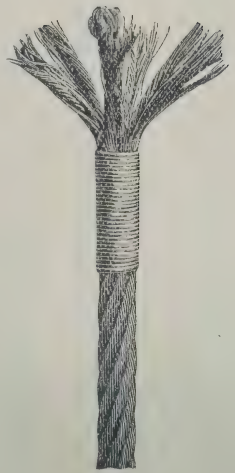


FIG. 37.

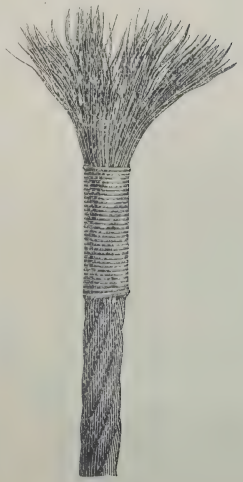


FIG. 38.

FIG. 39.

FIG. 40.

Ordinary virgin spelter is probably the best metal for socketing wire ropes, but this has to be used with caution, as it can be heated up too high unless the workman is very careful. As soon as it begins to char a pine stick (a match will do) should be used.

Another mixture that is recommended is 80 per cent. zinc and 20 per cent. tin; this has a low melting point and consequently is not apt to affect the wires. (*Brunlons, Musselburgh.*)

The Coal Mines Act of 1911 requires that the filling metal should not have a melting point above 750° F. Cast zinc is not good as a filler, as its melting point is high—786·9° F.—and it is liable to be heated considerably above this temperature.

Blackett's Wire-Rope Socket.

Fig. 41 represents a section of Blackett's wire-rope socket. The cap or socket, A, made of iron or steel, and fitted with a lining, B, of some soft metal, is placed round the rope-end, C, which is brought into proper position, and then forcibly driven outwards against the lining, B, within

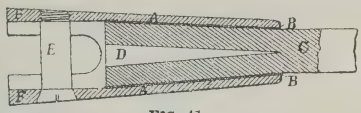


Fig. 41.

the socket, the wire ends being held asunder by a tapered plug, D, made of soft metal similar to the lining, B. A bolt, E, which serves to carry the load, or another socket, passes through a double eye. This arrangement forms a strong and compact job, and the process of socketing can be easily and rapidly performed.

Attaching ropes to joints or sockets by means of pins or rivets driven through the former is a detrimental, if not a dangerous, procedure.

Wire Rope Lubrication.

According to the operating temperatures that may be involved, and the possibility of the presence of an excess of water, the viscosity of a wire rope lubricant should be 1,000 seconds Saybolt at 210° F., or somewhat lower. In warm climates, or when the ropes are adjacent to ovens or furnaces, where there is danger of the lubricant becoming thinned down to such an extent that it will drip off, it is advisable to use a lubricant of about 1,000 seconds viscosity at the temperature prevalent under operating conditions.

On the other hand, in relatively cold districts, it is advisable to use a somewhat thinner lubricant selected in accordance with the range of the operating temperatures involved.

Wire rope lubricants should, in general, be straight mineral petroleum products, devoid of fillers or thickening mediums, i.e. whatever the viscosity, it should be an inherent property of the lubricant, not an artificial characteristic which cannot be depended upon. Greases or soap-thickened mineral oils are quite unsuited to wire lubrication.

Application of Wire Rope Lubricants.

As a general rule, wire rope lubricants must be applied in a heated condition.

The best method of treating wire ropes is to use a form of split box for the lubricant which can be parted to permit it to be placed around the rope, the hole through which the rope passes being on the parting line. The lubricating box may be made from wood or metal. The shape of the box is determined somewhat by the angle or position of the rope. For horizontal ropes, a V-shaped box or trough with holes at each end near the bottom is generally used. Conical shaped boxes or containers are used for vertical ropes, the rope passing through a hole at the small end of the cone. Boxes of the kind described can be readily built in the average plant, with suitable provision for rendering it sufficiently tight to prevent the lubricant from leaking out even when reduced in viscosity by heating. The slow passage of the rope through such a bath of heated compound will not only ensure thorough coating of the surface but also will cause the lubricant to penetrate to the inner strands. Further working of the rope over the sheaves before the lubricant has time to cool entirely will aid in bringing about maximum penetration.

It is good practice to apply lubrication at least every six months, and every three months under some conditions.

CLUTCHES.

Friction Clutches.

Fig. 42 is one of the simplest forms of friction clutch. Part A clutches with part B (which is keyed to the shaft) by advancing the latter so that the conical surfaces are pressed together: these surfaces are sometimes covered with leather or fibre in order to increase the frictional resistance. When the members are mounted on separate shafts, it is generally advisable to mount

the sliding member on the driven shafts so that it may be at rest when not engaged. Where oil is likely to lodge between the two parts, small escape holes should be drilled parallel to the shaft axis through one of the members. These holes also allow of the escape of any air which may be trapped between the faces.

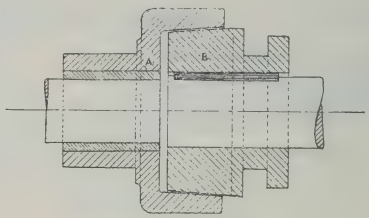


FIG. 42.

THE WESTON FRICTION COUPLING.

This device is probably more frequently used as a brake than as a clutch, but is equally suitable for both purposes. In the design shown in fig. 43, the muff A is keyed to the shaft S, but can be moved along the same by means of a fork which fits into the groove at the right-hand end. This muff A is provided with several feathers F around its inside surface (or the inside surface

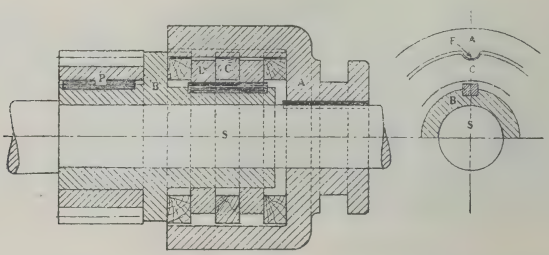


FIG. 43.

may be hexagonal), and into these are fitted the annular discs C. These discs are sometimes made of hard wood, but may be of iron. The member B, riding freely upon the shaft S, has a pinion P (or pulley) keyed to the outer end, and the metal rings D keyed to the sleeve. The rings D are free to move longitudinally along the sleeve, and are clear of the feathers F. Similarly the discs C are clear of the key in the sleeve B. On pressing A forward the axial force P is exerted on each of the rubbing faces and the frictional or turning effort increases with the number of rings.

The principle of the Benn and Hele-Shaw clutches is identical with the above.

THE HELE-SHAW INDUSTRIAL CLUTCH.

(*The British Hele-Shaw Patent Clutch Co., Ltd., Oldham.*)

In this form of clutch (fig. 44) all the plates have driving notches either on their outer or inner circumference. The plates with driving notches on their outer circumference are made from high-grade phosphor-bronze, the plates with inner driving notches being of high quality mild steel. The illustration shows a pulley type of clutch; in this case the drive may be transmitted either from the pulley D to the shaft by means of the core A, or *vice versa*. The plates are housed in an oil-tight casing, which revolves with the pulley. The cover B contains the springs L, which

maintain the pressure on the discs sufficient to transmit the drive, the end O is the mechanism for operating the clutch and also provides for adjustment when wear occurs.

The method of operation is simple. By means of a gun-metal strap ring with two pegs, which fits in the actuator H, the toggles, I, are moved. These move the presser J, which in turn moves the pins K, compressing the plates together.; this compression of the plates usually being sufficient to start the transmission. As the presser J moves over further, the springs L are compressed until sufficient pressure is put on the discs to ensure transmission without slip. When wear takes place the clutch is maintained in efficient condition by putting into the driving position whilst stopped, the pin M is removed and the ring N is screwed up one or more turns until the springs are again fully compressed.

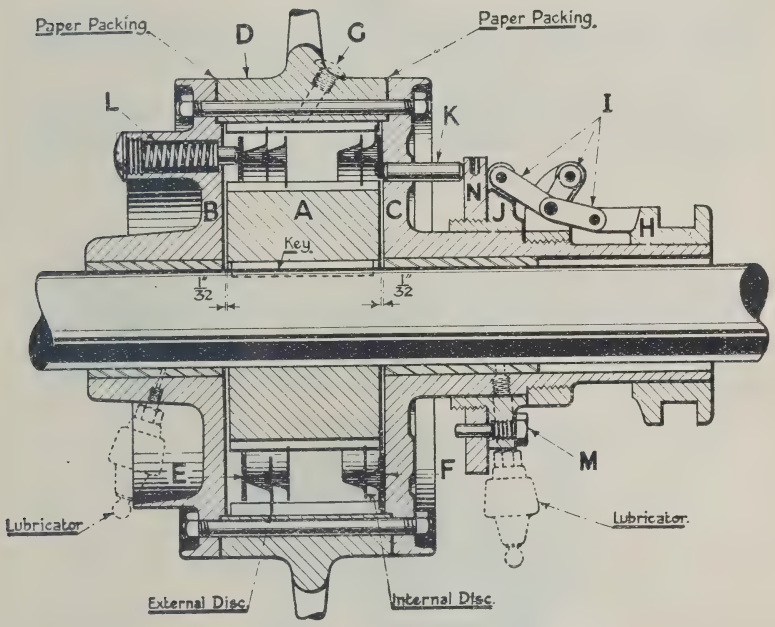


FIG. 44.

The ease with which a Hele-Shaw Clutch is operated is due in part to the mechanical advantage obtained from the toggles and from the slow compression of the discs. It should be noted that the discs, running in a bath of oil, engage without any jar, and that acceleration is gradual through out. The clutch is self-locking both in and out of gear, and that there is no thrust on the shaft when in gear or out of gear.

The Hele-Shaw Clutch can be arranged in various forms: pulley, coupling, gear clutch, a pair of clutches combined for a reversing motion or with an extended boss to take small gear or pulley.

In practice it is found that the V-grooved twin plates are approximately three times as powerful as the same number of flat plates under the same pressure, and hence for the same work require one-third the spring pressure or operating pressure to engage. The plates being well separated right up to the point of work, and their only touching at the side of the V-grooves, free circulation of oil and extreme rapid dissipation of wear are ensured, so that plates never burn the oil, and may be engaged indefinitely without causing undue heat, whilst their working condition remains constant, also long periods of slipping can be endured as part of their regular work.

The V-groove in the twin plates adds so greatly to their traverse strength that buckling and distortion are impossible. Even under the most unfavourable working conditions, such as through accidental loss of oil and extreme heat being set up, the V-grooved plates still retain their shape.

The latest development of the Hele-Shaw Clutch is a Safety Coupling. It is in appearance similar to a Hele-Shaw Clutch without the operating end. This consists of a coupling which is permanently in gear, but which is capable of transmitting power without slip up to a predetermined maximum. When the maximum power is exceeded, slip takes place, and shows the operator that an abnormal condition exists. Slipping may be maintained for more than five minutes without damaging the coupling. These couplings are usually applied between an electric motor and a worm-reduction gear.

The advantages of the Hele-Shaw Safety Coupling are these :—The power at which slip occurs is adjustable and is also more constant than other types, such as are used when a pin is sheared. The internal parts are fully protected, and when the load returns to normal the coupling picks up the load smoothly and then acts as a solid coupling until further overload develops.

The Hele-Shaw Clutch is ideal for use in exposed positions, such as out of doors, in coal-mines, etc., as the friction surfaces are fully protected, and, furthermore, since it is a clutch running in oil it is especially suited for use with chain drives, as the driving chain requires lubrication, which is detrimental to Ferodo type clutches. The applications are universal, such as :—electric winches for ships, machine tools, paper-making machinery, cable-making machinery, etc., etc.

The Benn Friction Clutch.

In this clutch (Type H), fig. 45, the friction plates move axially, practically bear against the entire faces of the drum, and are continuous. As a result of this construction the diameter, for a given power, is relatively small. As the movements of the plates are axial and not radial, they are unaffected by centrifugal force and the drum being relieved from bursting strains the clutch is suitable for high speeds. The lever stops H, as they approach, show at a glance the amount of wear and compensation, for this can be made by screwing in the plate M one or two notches. By screwing out the ring M and slacking back two set screws, the interior parts can be withdrawn without disturbing the shafts or parts keyed thereto.

The construction and action of the clutch is as follows :—

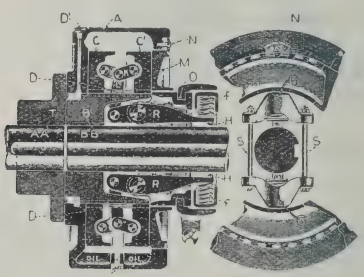


Fig. 45.

The boss T of drum A is keyed to the shaft AA, and the driving boss B to the shaft BB. On the driving boss B, the friction plates CC′ are free to slide on feathers DD. These plates CC are connected by toggles KK, which, in turn, are connected by radial links LL to the levers RR. The ends of these levers RR carry compression springs FF, having their outer ends seated in the yokes GG. These yokes are connected to each other by the bolts SS, thus forming an equalising member, which absolutely *ensures uniform pressure* between all friction surfaces throughout the clutch. The cover M, forming one end of the drum, is accurately adjustable, and is notched to receive the lock plate N. The lever stops HH are to enable the position of the toggles KK to be ascertained from outside the drum.

The figure shows the condition of the clutch when engaged, i.e., the plates bearing against M and A, the muff O in its inmost position, springs F in compression, the toggles KK in their most advantageous position (nearly straight), and the clutch locked by the radial link L being beyond the centre. On withdrawing the muff O the link L is pulled towards the right, the toggles KK are drawn towards the centre, and the plates C withdrawn from the drum faces.

The Johnson Friction Clutch.

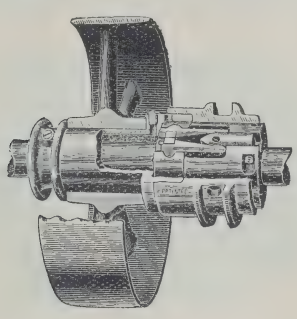


Fig. 46.—Clutch with Pulley Mounted on Hub.

This clutch, figs. 46, 47, consists of a body fastened to the shaft and carrying a split ring into which are inserted a pair of levers. A curved shaped wedge, which is made part of a sliding sleeve, forces the levers apart, expands the ring, bringing its outer surface into fractional contact with the inner surface of the friction cup, the hub of which is made to suit requirements. One screw which moves two taper blocks, set into the levers, adjusts the contact of the ring and cup to any degree.

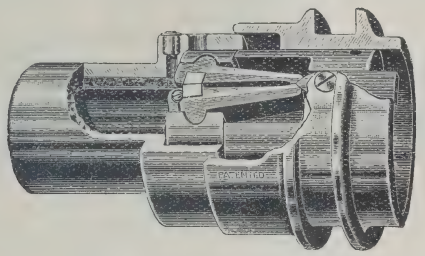


Fig. 47.—Section broken away, showing Clutch disengaged.

Patent Automatic Trip Centrifugal Clutch.

This clutch, fig. 48, has been designed for use with electric motors, driving any class of machinery, *to enable the motor to start without load and to automatically pick up the load* when a certain speed has been attained.

For three-phase and two-phase motors of the squirrel-cage type, as well as for all single-phase induction motors which possess a low-starting torque, a clutch of this description is most essential.

The clutch consists of a cast-iron body (B), keyed to motor shaft and provided with machined slides to receive the slippers (S), and with fulcrum pins (F) to carry the trip levers (L).

The slippers (S) are of cast iron, machined to suit the slides in body (B), and are shod with 'Perodo' friction surface linings and fitted with pins (P) to carry rollers (A).

The trip levers (L) are cam-shaped to bear against the rollers (A) and hold the slippers (S) away from the friction rim (R). They are held in position by the springs shown, the tension on which may be adjusted by means of the adjusting bolts (T).

The action of the clutch is as follows:—When the motor has reached a certain speed, the centrifugal force of the levers (L) overcomes the tension of the springs, causing the levers to open outwards, and thereby releasing the slippers (S), which move outwards against the friction rim (R) and automatically pick up the load.

When the current is cut off, the levers are gradually returned to their former position by means of the springs, and withdraw the slippers from contact with the friction rim.

The clutches are *designed to engage at 75 per cent. of normal speed*, but the springs are provided with adjusting bolts to allow the slippers to come into operation at any desired speed between 50 per cent. and 90 per cent. of normal speed.

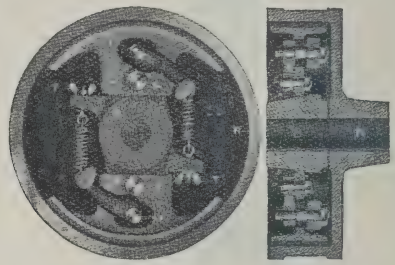


FIG. 48.

By this design of clutch *full advantage of the weight of the slippers* for the required frictional contact is secured, *thereby enabling the various parts to be as light as possible, and reducing the time of rubbing of the friction surfaces to a minimum.*

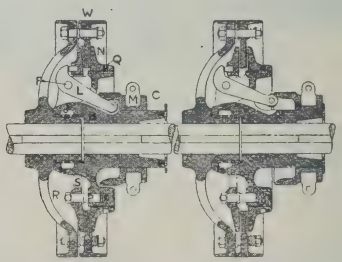
These clutches are made by Frank Wigglesworth & Co., Ltd., in sizes from 6 ins. to 30 ins. and capable of transmitting from 0·7 to 500 horse-power at a speed of 500 revolutions per minute. The power increases almost in proportion to the cube of the speed and fourth power of the size.

HIGH-SPEED DISC FRICTION CLUTCH.

The general design allows this type of clutch, fig. 49, to be run at very high speeds, and forms an ideal clutch in connection with electrical machines.

End pressure is not required to keep the clutch in or out of gear, and all projecting parts are shrouded.

Its construction is as follows:—



In Gear Position. Out of Gear Position.

FIG. 49.

The body (B) is a cast-iron circular disc, with boss bored and keywayed to suit shaft, and having lugs cast on the inside, forming fulcrums for levers (L).

Driving pins (*p*) are fitted into the body (B), which carry the inside annular friction ring (S).

The levers (L) are of steel, and having friction rollers fitted to one end to allow of easy movement of sliding collar over them.

The sliding collar is of cast iron, having extended circular flange, the inside being tapered in order to depress the ends of levers (L). It is complete with cast-iron clams (M), which receive the momentary pressure from operating lever for putting clutch in and out of gear. A spring grease lubricator is fitted to clams (M).

The shell (R) is a cast-iron circular bell-shaped disc, with boss bored and keywayed to suit shaft, and to its extreme diameter the annular disc is fixed.

The internal boss (J) is bushed with gun-metal, and forms a bearing for the extended boss of body (B), thereby ensuring perfect alignment of clutch. Grease lubricators are fitted to this bearing.

Its action is as follows:—

When the clutch is put into gear, the sliding collar (C) is moved (by means of operating lever) from the out-of-gear position (shown by dotted lines) to position shown on drawing, and depresses the ends of levers (L), their opposite ends pressing hard on the annular ring (S) gripping the annular disc against the friction face of adjusting ring (N).

When the clutch is put out of gear the springs shown force the rings clear of the ring (W), which also moves clear of the friction face on ring (N), as shown by out-of-gear position.

Adjustment is made by rotating the ring (N), and afterwards locked by locking screw (Q).

These clutches are made by Frank Wigglesworth & Co., Ltd., in sizes from 12 ins. to 46 ins., which are capable of transmitting from 3 to 1,200 horse-power at speeds from 10 to 1,500 revolutions per minute. The power transmitted varies directly as the revolutions.

Coil Clutch.

The clutch shown in fig. 50 consists of a bushed driving plate (*c*) mounted on a shaft and having lugs cast on its face to engage the transmitting coil (*a*). A drum (*b*) of chilled material is keyed to the shaft adjacent to the driving plate. The transmitting coil (*a*), of special steel, having a highly-finished internal surface, is mounted on the drum and has one end engaged with the driving plate lugs, whilst the other is free to be operated. A sliding member (*d*) is mounted on the shaft beyond the drum, its function being to pull the free end of the coil through a steel lever (*e*), which is pivoted on an extension of the driving plate (*c*), and engages the free end of the coil.

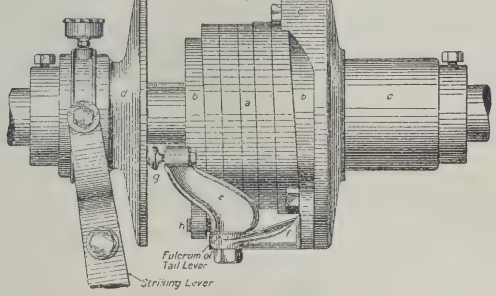


Fig. 50.

In engaging the clutch, the plate (*d*) is moved towards the drum, thereby tilting the lever (*e*) and tightening the free end of the coil, which is contracted and gradually tightens on to the drum (*b*), causing the different members of the clutch to be locked together. The action is similar to that of a man tightening the free end of a rope on a revolving capstan.

The clutch, made by the Coil Clutch Co. Ltd., Johnstone, is simple in construction and compact, and is used in colliery haulage gear, heavy cranes, and rolling mills.

The Wallace Patent Centrifugal Multi-groove Clutch.

(British Hele-Shaw Patent Clutch Company Ltd., Oldham.)

This clutch (fig. 51) transmits power by the action of slipper blocks acted upon at speed by centrifugal force, thus creating frictional contact with the interior face of the clutch, and developing driving power rapidly as the speed increases. The patent lies in the use of grooves on the friction surfaces, giving superior grip over flat faces, and also in the fact that lubricant is introduced which preserves the wearing surfaces, and ensures smooth action and long service.

The clutch consists of a spider—mounted on the motor as the driver portion—which carries on radial guides four movable slippers, which, by centrifugal force, transmit the drive to the case with its dust- and oil-tight cover. The clutch is spring-controlled.

When the clutch begins to drive at just over three-quarter speed the oil collects in special groove extensions, leaving the friction surfaces practically dry at full speed, thus preventing further slippage. For quick pick-up little oil is used, for later pick-up more oil.

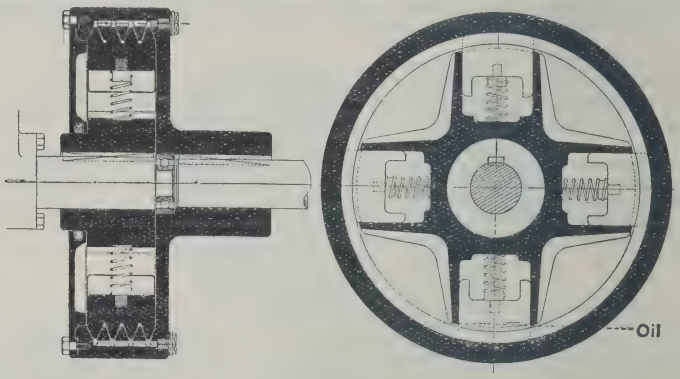


FIG. 51.

The great advantage of the clutch lies in the fact that the motor can start under no load, and as speed is developed the load is applied gradually and automatically, thus reducing strain and wear and tear. The use of the clutch on three-phase motors of the squirrel-cage type cuts down the initial heavy draught on the line and so facilitates starting.

These clutches are particularly adaptable and valuable in starting up individually driven machines under full load. In these cases one part of a centrifugal coupling clutch is mounted on the armature shaft and the other part is mounted on the machine shaft, thus giving perfect connection, reduced starting strain, and driving at full speed like a solid coupling.

For group driving, clutches built within a pulley and mounted on the motor shaft give perfect connection where belt or rope pulley is required. In these cases the clutch case generally forms the pulley, the whole running on ball bearings.

The clutches are made in sizes from 4 ins. to 36 ins., or as required.

Magnetic Clutches.

These consist of two members—the field or driving member, which carries the magnetising winding, and the armature, or driven member. In the design, by the Igranic Electric Co., shown in fig. 52, these members are carried on the face of annular spring steel plates attached to the

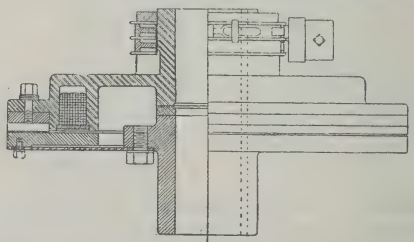


FIG. 52.

driving and driven hubs. Normally the members are held apart, but when the current is supplied so the windings of the field member they are magnetically drawn together, the flexibility of the spring steel plates allowing the necessary movement. The power is then transmitted entirely by

the friction. The engaging surface, fitted externally to the poles of the field member, is composed of woven asbestos and brass wire, and stands proud of the pole faces so as to prevent these coming into contact with the armature. This type of clutch can pick up the load gradually, and can be controlled from remote positions.

Note on Clutches.

In selecting a clutch, attention must be given to the starting torque required in overcoming the inertia of the part, means of stopping quickly in case of accident, and the effect of centrifugal forces on engaging parts in retarding their release or in throwing the members out of action.

Friction Surfaces.

In friction clutches the mating surfaces are sometimes of special materials in order to increase the grip. These materials are usually secured to the body of the clutch by pins, and trouble occasionally arises through the pins shearing. Preferably the fabric should be secured to the body with brass bolts. For low speeds and light pressures wooden surfaces are very suitable, but under opposite conditions the wood becomes charred. When the mating surfaces are of steel one of these should be casehardened. Steel should not be used with cast iron unless the latter has been chilled. In general, close-grained cast iron on cast iron gives the best results. A soft material should not be used with a metal which tends to crumble, as hard particles of the latter get imbedded in the former, and severe scoring results.

(*Professor Dempster Smith.*)

Ferodo Friction Linings.

(*Ferodo Ltd., Chapel-en-le-Frith.*)

Ferodo Brake and Clutch Linings are fabrics of distinct forms, i.e. fibre cotton base, and bonded asbestos, the object and use of each being to provide a surface which in rubbing contact with metal gives a high coefficient of friction which rises with the temperature and is not physically affected by the presence of oil or water—in fact, a damp or greasy surface improves the durability of Ferodo friction linings, but decreases their frictional value in proportion to the quantity and nature of the lubricant present.

Tests on Ferodo fibre prove the average coefficient of friction to be 0·5; the maximum obtained being 0·7—the temperature limit for the fabric being about 230° F.; the material has great mechanical strength and wearing qualities.

Tests on Ferodo bonded asbestos prove the average coefficient of friction to be 0·3 and that the efficiency of the fabric is maintained at any temperature up to 500° F. (260° C.).

Ferodo Fabrics form linings which do not score or seize under any speed or pressure and allow a smooth gradual engagement against dry metal surfaces. The high coefficient of these materials renders very fine angle cones unnecessary. The angles recommended range between 24° and 34° (included).

For flat-plate clutches, Ferodo Friction Linings are made up in the form of circular discs, either jointless or with a close butt joint stretched with fine soft metal wire.

Chekko Friction Linings.

(*Cresswell's Asbestos Co., Ltd., Wellington Mills, Bradford.*)

These linings are made from 100 per cent. pure long asbestos fibre woven into brass wire in one solid piece. They are made in sheet from 2 in. to 12 in. wide, and in thickness varying from $\frac{3}{32}$ in. to $\frac{3}{4}$ in. The sheeting when cut to the required size is fastened to wood foundation blocks by means of counter-sunk brass screws, which screw into metal nuts, the latter lying in slots cut in the wood. In the case of a steel band being lined with the material, screws pass through the band and screw into nuts on the outside of the band, or aluminium rivets may be used for the purpose.

The average coefficient of friction of Chekko is given by the makers as 0·35 against cast iron or steel.

Flat-plate Chekko clutch rings are made up in the stitched joint circular rings or discs hydraulically moulded, and also supplied cut from sheets.

SECTION XIX

PART II

CHAIN GEARING.

(Association of British Driving Chain Manufacturers (A.B.D.C.M.).)

[NOTE.—The British Standard Dimensions for Steel Roller Chains and chain wheels and the clauses relating thereto, contained in Specification No. 228—1925, were adopted by the British Engineering Standards Association from a report presented by the Association of British Driving Chain Manufacturers, which co-operated throughout in the preparation of the Specification.]

The adaptability and high efficiency of chain gearing for the transmission of power is now extensively recognised. Although the advantages of chain driving are not yet fully appreciated, the attention of power users and engineers when considering methods of transmission should be directed towards embodying chain drives of suitable type.

Although belts, ropes, and the several forms of toothed wheel gearing will always have their place, chain drives can in many cases be employed with greater advantage. Chain gearing ensures a positive transmission, at the same time affording a beneficial measure of elasticity. The average efficiency of a well-designed chain drive when new is 99 per cent., and the loss of power will not exceed about 2 per cent. at the end of its useful life. Thus, chain gearing has a higher maintained efficiency than any other form of transmission.

The initial tension required with a belt is unnecessary with a chain, and the bearing friction due to it is therefore eliminated.

Chain drives can be used under very varying conditions in respect of:—(*a*) Speed ratios, which can be readily modified; (*b*) centres of shafts (chain gearing can be used at centres which are too long for spur gear and too short for belts or ropes); (*c*) nature of the load; (*d*) temperature; (*e*) humidity; (*f*) power to be transmitted.

TYPES OF CHAINS.

The two general typeg of power transmission chains are shown by figs. 1 and 2. Speaking generally, the Inverted Tooth type is employed where silence is a consideration, and the Bush Roller type for general applications.

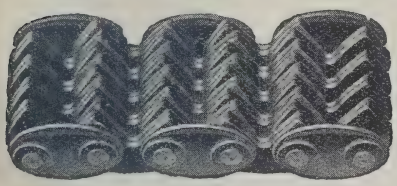


FIG. 1.—Inverted Tooth Type.

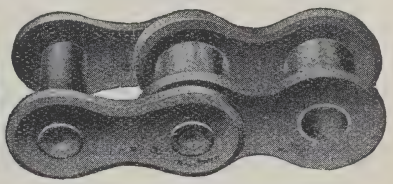


FIG. 2.—Roller Type.

Desirable Conditions for Application of Chain Gearing.

The following statement, prepared by the Association of British Driving Chain Manufacturers, shows the approximate limits that usually obtain in the economical application of chain gearing; where proposed drives are not within these limits, the chain manufacturer should be consulted.

Gear Ratio (maximum): Roller Type, 7 : 1; Inverted Tooth Type, 7 : 1.

Chain Speed.

Normal maximum, assuming the number of teeth is 19.

Pitch.	Roller type. Feet per min.	Inverted tooth type. Feet per min.
Up to ½ in. pitch	1,000	1,500
⅝ in. and ¾ in. ,,	1,000	1,400
1 in. and 1¼ in. ,,	900	1,350
1½ in. and 1⅝ in. ,,	900	1,300
2 in. and 2½ in. ,,	800	1,250
3 in. ,,	700	1,200

Number of Teeth in Wheels.

Minimum	Roller type 17	Inverted tooth type 19
Maximum	,, 120	,, ,, ,, 134

This applies only to inverted tooth chains with a profile based on a 60° link face angle.

In cases where the chain speed is substantially less than shown above, for any given pitch, the number of teeth in the pinion may be reduced, but it must be borne in mind that within limits the greater the number of teeth in pinion the longer will be the life, and the higher the efficiency, of the drive.

Centre Distance between Shafts.

The following expression gives limiting values of centre distance for drives running under ordinary conditions with steady loads, at chain speeds between 1,500 and 300 feet per minute:

$$\text{Maximum centres (in feet)} = 20 \sqrt{\frac{\text{teeth in wheel}}{\text{r.p.m. of pinion}}}$$

Where the chain speed is less than 300 feet per minute, the centre distance should not exceed:

$$\tfrac{1}{2}\sqrt{\text{Pitch} \times \text{teeth in wheel} \times \text{teeth in pinion}}.$$

The values will hold up to inclinations of 60° when the wheel is above the pinion, and up to inclinations of 45° when the pinion is above the wheel; for greater inclinations than these, for impulsive or irregular loading, or where any special considerations enter into the installation or running of a drive, the chain manufacturer should be consulted.

(*a*) *Desirable Minimum.*—The centre distance should be such that the arc of contact between the chain and any wheel is not less than 120°, and with at least 7 teeth in engagement.

(*b*) *Adjustment.*—It is advisable to provide suitable means for the adjustment of the centre distance, and the normal minimum amount of this adjustment should be 1½ times the pitch of the chain.

In some cases adjustment can only be obtained by the use of a jockey wheel, which must engage with the slack side of the chain and have machine-cut teeth.

(*c*) *Chain Length.*—It is desirable that the number of pitches in the chain should not be a multiple of the number of the teeth in either the pinion or the wheel.

Disposition of Drives and Direction of Rotation.

It is preferable that the positions of the driving and driven shafts shall be such that the slack or non-driving free length of the chain will not be inclined at a greater angle than 60° to the horizontal, otherwise adjustment will have to be made frequently.

Drives should be so arranged in relation to the direction of rotation of the shafts that the tight or driving free length of the chain will be uppermost.

Subject to the foregoing, it is preferable when chain cases are fitted that the pinion should be above the wheel.

Clearance.

The normal allowance for clearance over the back of a chain mounted upon a wheel is an amount equal to half the pitch of the chain. When, however, a chain case is fitted an extra amount of clearance above this should be allowed.

The clearance for sag should be greatest at a point somewhere midway along the free portions of the chain, and this clearance should be allowed for both above and below each free portion of chain.

The minimum amount of clearance is

$$\frac{\text{Centre distance between wheels (in inches)}}{12} + \tfrac{1}{4} \text{ in.}$$

The width inside a chain case, and therefore the amount of clearance at the sides of the chain, will frequently be determined by the width of the wheel boss; where this condition does not obtain, and for drives where no case is fitted, a minimum clearance of $\frac{3}{8}$ in. plus $\frac{1}{8}$ in. per foot of centre distance should be allowed.

The clearance between the rivets of inverted tooth chains running in multiple should be $\frac{3}{8}$ in. for pitches up to and including $\frac{3}{4}$ in.; for 1 in. pitch and upwards allow $\frac{1}{2}$ in.

Lubrication.

The provision of chain cases is advisable for all drives where the chain speed exceeds 1,000 ft. per minute, and for drives below this speed when the surrounding air is charged with particles which may adhere to the chain.

Efficient lubrication of chains is of the utmost importance and is, needless to say, more easily obtained where drives are encased.

Drives with a chain speed in excess of 1,350 ft. per minute, those employing a wide chain, or more than one chain, should be fitted with the sump type of chain case, and a pump to deliver oil to the working face of the chain. For lower speeds and narrow chains it will be sufficient to provide an oil-bath case and allow the chain to dip into the oil whilst running.

A drip feed lubricator delivering oil to the inner face of inverted tooth chains, or between the inner and outer plates of roller chains, may be used when the chain speed is not greater than 1,000 ft. per minute. A splash guard should then be fitted to prevent the oil being thrown on to adjacent objects. One drip should be provided for each 3 in. width of inverted tooth chain, and 2, 3, or 4 drips for roller chains, depending upon whether a single, a duplex, or a triplex chain is used.

Erection.

Chain wheels must be rigidly mounted on shafts which are parallel and truly horizontal, and the teeth of the wheels must be in correct alignment.

Where possible, wheels should be placed close to bearings, and must run true on their shafts.

Care should be taken that the chain connection is properly secured.

Before starting up the drive the chain should be lubricated and centres adjusted to take up any undue slack. No tension should be put upon the chain.

Maintenance.

Periodic examination is advisable to ascertain:

(*a*) If correct alignment of the wheels has been maintained; (*b*) if undue wear is taking place; (*c*) if adjustment of chain is needed; (*d*) if the chain requires cleaning; (*e*) if lubrication is being effectively carried out.

Improved service usually results when chains are removed from time to time for cleaning. They should be washed with paraffin and, after draining, thoroughly lubricated before being again used.

When replacing worn chains by new ones always ascertain that undue wear has not taken place on the teeth of the wheels.

A.B.D.C.M. STANDARDS FOR ROLLER CHAINS.
'SHORT PITCH' SERIES. (0·375 – 0·75 in. PITCHES.)

Association Chain No.		Pitch.	Max. Roller Diam.	Min. Width between Plates.		Tooth Width.			
						Narrow.		Wide.	
Narrow.	Wide.			Narrow.	Wide.	Max.	Min.	Max.	Min.
3 N	3 W	0·375($\frac{3}{8}$)	0·250	0·155	0·230	0·141	0·134	0·213	0·204
Cycle Sizes.		0·5($\frac{1}{2}$)	0·305	0·130	0·192	0·117	0·110	0·176	0·168
4 N	4 W	0·5($\frac{1}{2}$)	0·335	0·205	0·305	0·189	0·181	0·285	0·275
5 N	5 W	0·625($\frac{5}{8}$)	0·400	0·255	0·380	0·237	0·228	0·356	0·345
6 N	6 W	0·75($\frac{3}{4}$)	0·475	0·310	0·460	0·290	0·279	0·433	0·420

All dimensions are in inches.

1·0–3·0 PITCHES. 'SHORT PITCH' SERIES. (1·0 in. to 3·0 in. PITCHES.)

Chain No.	Pitch.	Maximum Roller Diam.	Minimum Width between Plates.		Tooth Width.			
					Narrow.		Wide.	
			Narrow.	Wide.	Max.	Min.	Max.	Min.
Refer to individual members' catalogues.	1·0	0·625	0·500	0·670	0·472	0·458	0·635	0·618
	1·25	0·750	0·560	0·770	0·530	0·514	0·731	0·712
	1·5	0·850	0·625	0·875	0·592	0·575	0·830	0·810
	—	1·000	—	1·000	—	—	0·952	0·928
	1·75	1·000	0·750	1·000	0·712	0·693	0·952	0·928
	—	1·100	0·875	1·220	0·832	0·810	1·163	1·135
	2·0	1·150	0·875	1·220	0·832	0·810	1·163	1·135
	2·25	1·350	—	1·350	—	—	1·288	1·257
	2·5	1·550	—	1·500	—	—	1·432	1·398
	3·0	1·900	—	1·800	—	—	1·720	1·680
	—	2·000	—	1·500	—	—	1·432	1·398

All dimensions are in inches.

NOTE.—The narrow series is not recommended for general use.

TOOTH CUTTER FORMS.

For the convenience of chain users and to ensure complete interchangeability, the Association has standardised tooth forms which, while not being identical with any of the existing forms,

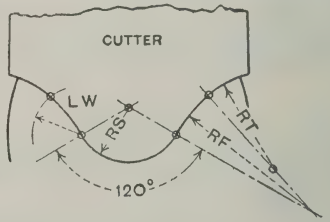


FIG. 3.

incorporate the essential features of each. The general construction is shown in fig. 3. It consists of a *roller seating*, *working faces*, and *topping curves*, all of which are arcs of circles.

The angle of 120° subtending the roller seating has its origin at the centre of the diameter of the roller seating. The working faces are comparatively flat curves tangential to the roller seating, and the topping curves are tangential to the working faces and of such a radius as to give a suitable height of tooth. The dimensions of tooth forms for the Association Standard Chains are given in the following table:

Roller Diameter.	Cutter No.	No. of Teeth to Cut.	R.S. Radius of Roller Seating.	R.F. Radius of Working Face.	L.W. Length of Working Face.	R.T. Radius at Top of Tooth.
0·250	1	9–12	0·126	0·45	—	0·45
	2	13–19	0·126	1·125	0·12	0·18
	3	20 and over	0·126	3·825 min.	0·154 min.	0·052 max.
0·335	1	9–12	0·169	0·6	—	0·6
	2	13–19	0·169	1·5	0·16	0·24
	3	20 and over	0·169	5·1 min.	0·205 min.	0·07 max.
0·400	1	9–12	0·202	0·75	—	0·75
	2	13–19	0·202	1·875	0·20	0·3
	3	20 and over	0·202	6·375 min.	0·256 min.	0·087 max.
0·475	1	9–12	0·240	0·9	—	0·9
	2	13–19	0·240	2·25	0·24	0·36
	3	20 and over	0·240	7·65 min.	0·307 min.	0·105 max.
0·625	1	9–12	0·316	1·2	—	1·2
	2	13–19	0·316	3·0	0·32	0·48
	3	20 and over	0·316	10 2 min.	0·41 min.	0·14 max.
0·750	1	9–12	0·379	1·44	—	1·44
	2	13–19	0·379	3·6	0·384	0·576
	3	20 and over	0·379	12·24 min.	0·492 min.	0·168 max.
0·850	1	9–12	0·429	1·632	—	1·632
	2	13–19	0·429	4·08	0·435	0·653
	3	20 and over	0·429	13·872 min.	0·558 min.	0·190 max.
1·000	1	9–12	0·505	1·8	—	1·8
	2	13–19	0·505	4·5	0·48	0·72
	3	20 and over	0·505	15·3 min.	0·615 min.	0·21 max.
1·100	1	9–12	0·556	2·1	—	2·1
	2	13–19	0·556	5·25	0·56	0·84
	3	20 and over	0·556	17·85 min.	0·717 min.	0·245 max.
1·150	1	9–12	0·581	2·208	—	2·208
	2	13–19	0·581	5·52	0·589	0·883
	3	20 and over	0·581	18·768 min.	0·754 min.	0·258 max.
1·350	1	9–12	0·682	2·592	—	2·592
	2	13–19	0·682	6·48	0·691	1·037
	3	20 and over	0·682	22·032 min.	0·886 min.	0·302 max.
1·550	1	9–12	0·783	3·0	—	3·0
	2	13–19	0·783	7·5	0·80	1·2
	3	20 and over	0·783	25·5 min.	1·025 min.	0·35 max.
1·900	1	9–12	0·960	3·6	—	3·6
	2	13–19	0·960	9·0	0·96	1·44
	3	20 and over	0·960	30·6 min.	1·23 min.	0·42 max.
2·000	1	9–12	1·010	3·6	—	3·6
	2	13–19	1·010	9·0	0·96	1·44
	3	20 and over	1·010	30·6 min.	1·23 min.	0·42 max.

All dimensions are in inches.

Wheels.

It is of the utmost importance that chain wheels should have their teeth accurately machined with cutters of correct form. Pinions should preferably have an odd number of teeth to equalise wear on the chain.

(*a*) *Section Profile of Teeth for Roller Chains.*—Chain wheel blanks should be machined as indicated in fig. 4.

Roller Chain Wheels Tooth Profile.

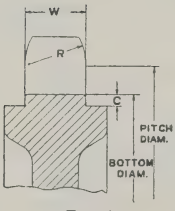


FIG. 4.

Pitch diam. . . .	= Chain pitch cosec $\frac{180°}{\text{Number of teeth.}}$
Bottom diam. max.. .	= Pitch diam. minus max. roller diam.
Tolerance on bottom diam.	= Minus 0·001 in. per 1 in. pitch diam., but in any case to be within the following limits: min., 0·005 in.; max., 0·020 in.
W (mean width of tooth)	= (95 % of width between chain inner plates), minus 0·01 in.
Tolerance on W . .	. ± (1 % of width between chain inner plates), plus 0·002 in.
R (Tooth side radius)	. = Roller diameter.
C (minimum shroud depth)	= $\frac{\text{Chain pitch} - \text{R}}{2} + \frac{0{\cdot}8 \text{ pitch}}{\text{Number of teeth.}}$

Measuring Loads.

To ensure interchangeability of chains it is necessary that measurements be taken under similar conditions.

It is, therefore, necessary that chains be measured before oiling, but after the test load of one-third the breaking strength has been applied, and that at the time of measurement the chain should be supported horizontally throughout its entire length.

As the necessary load should bear a definite relation to the breaking load of the chain, and inasmuch as actual breaking loads, in practice, are approximately in the ratio of pitch^2, the measuring load is taken as equal to (Pitch^2 × 112) lbs. For the ' long pitch ' series the measuring loads are the same as for the ' short pitch ' series of corresponding strength.

Measuring Length and Tolerance.

The actual length of test chains for measuring purposes is taken as 50 pitches for all pitches of chain, and a tolerance of 0·1 per cent. per pitch per length of 50 pitches is allowed in excess of the theoretical length.

'Long Pitch' Series.

A standard ' long pitch ' series of chains is formed from the standard series by using rollers, bushes, and rivets of a given pitch, but with inner and outer plates having twice the pitch.

Profiles of Teeth.

As for standard series of roller chains, but based on half the nominal pitch of a long pitch chain.

Cutters.

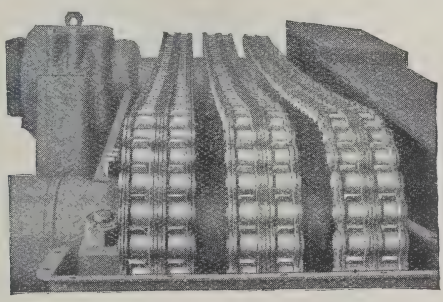
As for standard series of roller chains. (That is, for a given roller diameter a standard cutter may be employed for both standard or long pitch chains.)

RENOLD
CHAIN DRIVES

COMPLETE Chain Drives from 1 to 100 h.p. are available from stock, and quick despatch can be given on drives up to 1,000 h.p.

Send for "Renold Stock Chain Drives" Booklet Ref. No. 016/03 and "Renold Standard Design Drives" Booklet Ref. No. 016/16.

Three strands of 3" pitch Renold Duplex Roller Chain transmitting 1,000 h.p. from Steam Engine to Generator.



HANS RENOLD LIMITED
MANCHESTER ENGLAND

The Engineer.

Established 1856. Registered as a Newspaper

PROVIDES THE LATEST AND THE MOST RELIABLE INFORMATION FOR ENGINEERS

IN ALL BRANCHES OF THE PROFESSION.

The last issue in each month contains a Supplement, **The Metallurgist,** *a journal devoted to the science of metals, ferrous and non-ferrous.*

Subscriptions

		Per annum
INLAND, YEARLY		£3 5 0
CANADIAN SUBSCRIPTIONS.	Thin paper edition ...	£2 18 6
	Thick paper edition ...	£3 3 0
FOREIGN SUBSCRIPTIONS.	Thin paper edition ...	£3 3 0
	Thick paper edition ...	£3 7 6

'THE ENGINEER' DIRECTORY AND BUYER'S GUIDE

SENT FREE ON APPLICATION.

Chain Wheels.

pitch diameters.

The following table gives correct pitch diameters (P.D.) for all types of chain having a pitch of 1 inch. The pitch diameters for wheels to suit other pitches of chain are directly proportional to the pitch of the chain.

No. of Teeth.	P.D.	No. of Teeth.	P.D.	No. of Teeth.	P.D.	No. of Teeth.	P.D.
9	2·924	32	10·202	55	17·517	78	24·835
10	3·236	33	10·520	56	17·835	79	25·153
11	3·549	34	10·838	57	18·153	80	25·471
12	3·864	35	11·156	58	18·471	81	25·790
13	4·179	36	11·474	59	18·789	82	26·108
14	4·494	37	11·792	60	19·107	83	26·426
15	4·810	38	12·110	61	19·425	84	26·744
16	5·126	39	12·428	62	19·744	85	27·063
17	5·442	40	12·746	63	20·062	86	27·381
18	5·759	41	13·063	64	20·380	87	27·699
19	6·076	42	13·381	65	20·698	88	28·017
20	6·392	43	13·700	66	21·016	89	28·335
21	6·710	44	14·018	67	21·335	90	28·654
22	7·027	45	14·336	68	21·653	91	28·972
23	7·344	46	14·654	69	21·971	92	29·290
24	7·661	47	14·972	70	22·289	93	29·608
25	7·979	48	15·290	71	22·607	94	29·927
26	8·296	49	15·608	72	22·926	95	30·245
27	8·614	50	15·926	73	23·244	96	30·563
28	8·931	51	16·244	74	23·562	97	30·881
29	9·249	52	16·562	75	23·880	98	31·200
30	9·567	53	16·880	76	24·198	99	31·518
31	9·885	54	17·198	77	24·517	100	31·836

The Morse Rocker-Joint Chains.

(The Morse Chain Co. Ltd., Letchworth, Herts.)

These chains, fig. 5, form an important departure in silent chain driving. Until their introduction the principal objection to chains was found in the rapid wear and consequent stretch. This is necessarily present with all chains having round joint-pins, friction due to sliding surfaces

Fig. 5.

Fig. 6.

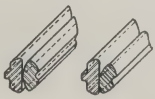


Fig. 7.

being unavoidable. A very ingenious feature is the joint, which in its latest development, as shown in fig 7, consists of two pieces, i.e. the rocker and seatpin, the former being provided with a knife edge which rocks on the face of the latter when the joint passes on or off the wheel. The bearing surface extends throughout the width of the chain. The pins for all practical purposes

are an integral part of the link, fig. 6, hence the only motion which takes place at the joint is a rocking movement. This ensures absence of sliding friction, which means the reduction of wear at the joint to the minimum, with consequent maximum life and efficiency. The shape of the tooth permits silent running, even under high speed.

Another important departure in the Morse chain drive is that represented by the No. 25 reinforced type chain—shown in fig. 8. In this construction the outer links are all knit together by short rivets and spacing pieces, adding very considerably to the material strength of the chain; also distributing the load more equally, and enabling the chain to run at higher speeds than has hitherto been possible.

The greatest ratio of gearing under favourable conditions with the Morse chains is 10 : 1. The following table gives the various pitches manufactured and the maximum permissible R.P.M. (revolutions per minute) for each:

Max. R.P.M. .	4,000	2,000	1,600	1,400	1,100	1,000	800	700	600	500	400	250
Pitch in ins. .	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	·9	1	1·2	1·25	1·5	1·75	2	3

The chain width depends on the speed, determined as follows:—

$$\text{Speed} = \frac{N \times P \times \text{R.P.M.}}{12} = \text{F.P.M.} \qquad \text{Load} = \frac{\text{H.P.} \times 33{,}000}{\text{F.P.M.}} = \text{total load on chain in lbs.}$$

where,

H.P. = Maximum horse-power; P = Circular pitch in inches;
N = No. of teeth; F.P.M. = Feet per minute.

This total load divided by the load per inch width advised gives the width of the chain.

Average permissible loads per inch in width for general machine-shop drives are as follows:—

Lbs. per inch wide .	65	85	105	125	160	200	220	240	300	400	500	800
Pitch in ins. . .	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	·9	1	1·2	1·25	1·5	1·75	2	3

Good chain speeds or average machine-shop drives are from 1,200 to 1,500 ft. per min.

As no friction due to sliding surfaces takes place, however, the chains can be run at speeds which would, owing to centrifugal force, throw off all lubricant, and drives of this make are now transmitting light loads satisfactorily at 3,000 ft. per min. chain speed.

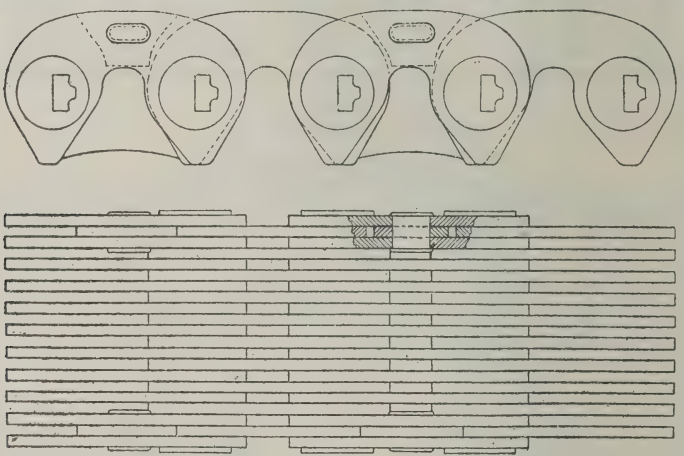


FIG. 8.

In view of its special joint construction, the Morse chain is claimed to surpass all others in point of size and strength, and the fact of being able to manufacture so large a size of silent chain as the 3-in. pitch in widths up to 24 ins. has resulted in such chain drives being applied for transmissions up to even 5,000 h.p. in a single drive.

The Morse chains are made in nineteen different types and pitches and single drives can be furnished for transmitting from $\frac{1}{4}$ h.p. up to 1,000 h.p.

SECTION XIX

PART III

TOOTH GEARING—HELICAL GEARING—TURBINE GEARS—WORM GEARING—VARIABLE-SPEED GEAR.

(Revised by H. T. Davey, A.M.I.Mech.E.)

Definitions.

The '*pitch circle*' is a virtual circle representing the periphery of a disc that would transmit by friction, when in contact with a similar disc, the required relative motion.

'*Base Circle.*'—The generating circle, or circle from which the involute curve is developed, in the case of the involute profile.

'*Diameter of a gear*' is the diameter of the pitch circle, and is often referred to as the pitch diameter.

'*Circular pitch*' is the length of the arc of the pitch circle measured from a point on one tooth to the corresponding point on the next tooth. It is equivalent to the circumference of the pitch circle divided by the number of teeth in the gear.

'*Diametral pitch*' is the number of teeth for each inch of pitch diameter, and is equivalent to the number of teeth in the gear divided by the diameter. It is a ratio and not a dimension. If a gear has a pitch diameter of 4 ins. and has 40 teeth, then it has a diametral pitch of 10.

'*Module*' is the reciprocal of the diametral pitch, and hence represents the pitch diameter per tooth. The module multiplied by the number of teeth gives the pitch diameter. The module is commonly used in Europe where the metric system is universal.

'*Chordal pitch*' is the length of a straight line drawn from the pitch line centres of two contiguous teeth.

'*Addendum*' is the height of a gear-tooth outside the pitch circle or pitch line.

'*Dedendum*' is the depth of a tooth space below the pitch circle or pitch line.

'*Clearance*' is the distance allowed between the top of the tooth of one gear and the bottom of its mating tooth space. It is equal to the difference between the addendum and the dedendum.

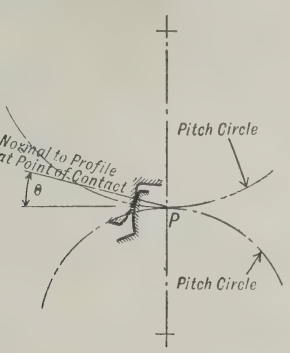


FIG. 1.

'*Whole depth*' is the total distance measured radially between circles containing the periphery of the teeth and the bottom of the tooth spaces.

'*Working depth*' is the distance which the teeth of one gear extend into the spaces of its mating gear. It is equal to the whole depth minus the clearance, and is also equal to the sum of the addenda of two mating gears.

'*Active profile*' is that part of the tooth which actually comes into contact with the profile of its mating tooth.

'*Line of action*' (see fig. 1).—In properly formed gears a line normal to the point of contact of two engaging teeth passes through the pitch point P. All points of contact of engaging teeth should fall on this line in order to secure uniform transmission of motion, for involute gears.

'*Angle of obliquity of action*,' θ (fig. 1), is the angle made by the intersection of the line of action with the tangent at P.

'*Interference*' occurs when points of contact of mating teeth do not fall on the line of action. Motion cannot be uniform under those conditions.

'*Width of tooth face*' is the length of the surface of contact of two mating gears. It is usually equal to the full width of the gear-wheel.

'*Centre distance*' is the linear measurement between the centres of two mating gears.

'*Line of centres*' is that line containing the centres of two mating gears.

A '*trundle*,' '*lantern pinion*,' or '*wallover*,' is a pinion the teeth of which are formed by round bars or rods of equal length, the ends of which are inserted in two discs, mounted centrally on a spindle. Such a pinion will operate wheels having, within limits, any number of teeth.

A '*hunting tooth*' or '*cog*' is an odd tooth provided in a wheel in order to secure uniformity in the wear of the teeth. If the number of teeth in the wheel is exactly divisible by the number of teeth in the pinion, then the same teeth are continually coming into engagement. This tends to produce uneven wearing, and hence, in mill gearing particularly, it is usual to introduce the hunting tooth. In accurately cut steel gears the same difficulties do not so readily arise.

'*Minimum number of teeth*' in a *pinion*.—Pinions having less than 10 teeth should not be used generally for power transmission. In cases where size is a very important controlling factor, then pinions having as few as 5 teeth can be made to engage sufficiently well. In such cases, however, the involute tooth profile must be continued practically to the root of the tooth, the standard tooth form being quite unsuitable. Such conditions imply a high-pressure angle.

FORMS OF GEAR TEETH.

Involute Teeth.

The involute tooth profile is now widely used, since it presents several important advantages compared to the older cycloidal form.

Three involute systems are in vogue: the standard 14½° involute system, the 20° full depth tooth system, and the 20° stub tooth system. The former system is a compromise between the cycloidal and involute tooth forms. Actually very little of the profile is involute. The need for smaller numbers of teeth and greater accuracy in the gears led to the introduction of the 20° full depth system. Such gears have stronger teeth, less undercutting being required in gears with few teeth.

In these two forms gears are interchangeable, and any two wheels of the same pitch will run together, provided the centre distance is suitable, irrespective of the number of teeth.

20° stub teeth have reduced tooth height, and the general proportions are different. The teeth are stiffer than those of the former types.

Gears having involute tooth profiles have the advantage that should the wheel centres be slightly inaccurate the teeth will still mesh properly. This occurs due to the fact that the path of contact is a straight line tangential to the base circle. The effect of excessive wheel centre distance increases the angle of obliquity producing increased back lash and pressure on the bearings.

Since the two engaging surfaces in contact are convex the surface is small, and hence the wear on involute teeth is rather greater than is the case with cycloidal teeth.

With a 14½° angle of obliquity and the standard addendum of $0{\cdot}318p$, excessive relative sliding occurs at the tooth points and local wear becomes very pronounced. Further, a 12-toothed pinion gearing with a rack having straight teeth, with a 29° included angle between the working faces, produces severe interference between the pinion flanks and rack tooth points. The cutting away of the teeth to obviate this defect reduces the path of contact, which is a further disadvantage. If the two angles are adhered to, then the shape of the engaging teeth needs some modification in the case of pinions having a small number of teeth. Actually the flanks of the pinion teeth are usually made radial under such conditions, and the rack tooth faces are rounded off.

The trouble referred to can be obviated by using 20° or 22½° angles of obliquity, when under such conditions the true tooth profile can be used without fear of interference.

The effect of increasing the angle of obliquity to 20° or 22½° is to increase the pressure on the bearings by 4 to 6 per cent.

With a pinion of 28 teeth and 14½° pressure angle interference is avoided, as is also the case with a 20-tooth pinion having a 20° angle of obliquity. With small numbers of teeth interference can be avoided if the stub teeth proportions are adopted for the addenda. With such teeth the efficiency of transmission is increased, due to a better rolling action in preference to the sliding action, which occurs when teeth have too great an addendum.

Setting out Involute Teeth.

Let ABCD (fig. 2) be the pitch circles of two mating gears, and EF their line of centres. Through the intersection of the two pitch circles with EF, draw MN such that angle θ is equal to the angle of obliquity of action desired. MN is the line of action of the two gears,

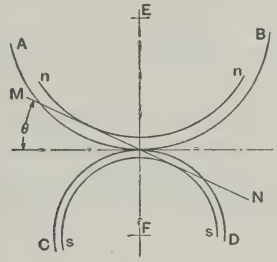


FIG. 2.

and is therefore the common tangent to the two base circles from which the teeth profiles are developed. Hence, with centres on EF draw the base circles *nn* and *ss*, just touching the line MN. The teeth profiles can then be set out as follows: Let A be the base circle of the gear and K the generator, which in effect is a straight edge (see fig. 3). Cut a notch in the edge of K and fix a

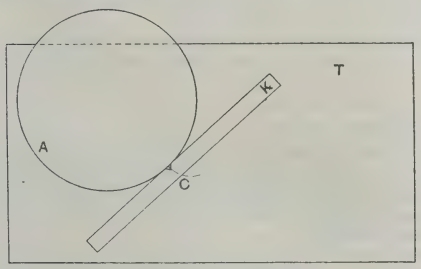


FIG. 3.

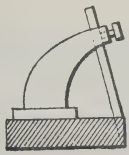


FIG. 4.

steel point in a clamp, as shown in fig. 4, so that the point touches the circle A through the notch. Press the straight edge up to circle A and steadily roll straight edge round circle. If a piece of finely sand-papered tin T is placed beneath as shown, a curve will be traced on it, as K rolls, shown at C. This is the tooth profile.

Proportions of Involute Gear-Teeth for Spur Gearing.

(Machine-cut gears of 14½° and 20° obliquity.)

(*Messrs. Brown and Sharpe.*)

p = circular pitch; $d.p.$ = diametral pitch.

Thickness of tooth and width of tooth space	$0{\cdot}5p$ or $1{\cdot}571/d.p.$	Dedendum	$0{\cdot}368p$ or $1{\cdot}157/d.p.$
Addendum	$0{\cdot}318p$,, $1/d.p.$	Bottom clearance	$0{\cdot}05p$,, $0{\cdot}157/d.p.$
		Total height of tooth	$0{\cdot}686p$,, $2{\cdot}157/d.p.$

Proportions of Short Involute Teeth.

(Stub teeth, 20° obliquity.)

Thickness of teeth and width of space	$0{\cdot}5p$ or $1{\cdot}571/d.p.$	Working height	$0{\cdot}5p$ or $1{\cdot}571/d.p.$
Addendum	$0{\cdot}25p$,, $0{\cdot}785/d.p.$	Total height of tooth	$0{\cdot}55p$,, $1{\cdot}728/d.p.$
Dedendum	$0{\cdot}3p$,, $0{\cdot}942/d.p.$	Clearance	$0{\cdot}05p$,, $0{\cdot}157/d.p.$

Rolling-Mill Gears.

The shape of the teeth for this purpose varies somewhat from the standard involute form, although the involute profile is closely followed. Good radii are provided at the root of the teeth, and a $22\frac{1}{2}°$ pressure angle is used. Usually the tooth face is exceptionally wide, often being made equal to the pitch diameter of the gear. This makes provision for the heavy wear to which these gears are subjected.

Typical tooth proportions are as follows:—

Addendum, $0 \cdot 275p$; dedendum, $0 \cdot 325p$; width of tooth at pitch circle, $0 \cdot 42p$.

Machine-Moulded Teeth.

Proportions for these vary. The following are typical values:—

Thickness of tooth, $0 \cdot 485p$. Width of tooth space, $0 \cdot 515p$. Addendum, $0 \cdot 25p$ to $0 \cdot 35p$. Dedendum, $0 \cdot 35p + 0 \cdot 08$ in. to $0 \cdot 4p + 0 \cdot 08$ in. Total height of tooth, $0 \cdot 6p + 0 \cdot 08$ in. to $0 \cdot 75p + 0 \cdot 08$ in.

Cycloidal Teeth.

Small amount of wear occurs with this form of tooth profile due to large surface of contact. To ensure quiet running and no back lash it is essential that the wheel centres be accurately located, since the path of contact consists of two curves which change direction at the pitch point.

These gears are difficult to cut accurately in a machine, and if the profiles of pinions of 12 teeth to a rack are generated with rolling circles greater in diameter than half the diameter of the smallest pinion, then the flanks cannot be cut by the ordinary rotating cutter.

Gears of this form have good wearing qualities, but the advent of higher speeds of rotation have made the defects of these teeth very pronounced, and involute gears have superseded them.

Epicyclic Gears.

In epicyclic gearin the arm rotates, and hence its motion must be added to that of the wheels. In general, the action of epicyclic gear trains can be summarised as follows:—the number of revolutions of any gear-wheel in the train for each single revolution of the arm is equivalent to the number of revolutions that the gear-wheel would make if the arm was fixed and the first gear-wheel turned through 1 revolution + 1 for wheels rotating in the same direction, and − 1 for wheels rotating in the reverse direction to that of the arm.

If 'n' is the number of gears in a train, then the nth gear-wheel will rotate in the *same direction* as the arm if 'n' is an *even number*, and in the reverse direction if 'n' is odd and the velocity ratio greater than 1. If the velocity ratio is less than 1, then direction of rotation is the same as that of the arm, 'n' being odd.

With epicyclic gears high velocity ratios can be obtained in a convenient form.

Involute Gear Cutters.

Since the dimensions of gear teeth are proportional to the circular or diametral pitch in the standard tooth forms, it follows that, theoretically, a different cutter is required for gears having different numbers of teeth.

In actual practice, however, a set of 8 cutters is employed for the $14\frac{1}{2}°$ system, and another set of 8 for the 20° system.

The cutters are employed as follows and are numbered from 1 to 8, each one being known by its particular number:—

No. 1	for cutting gears having	135	teeth	to a rack			
No. 2	,,	,,	,,	,,	55	,,	134 teeth
No. 3	,,	,,	,,	,,	35	,,	54 ,,
No. 4	,,	,,	,,	,,	26	,,	34 ,,
No. 5	,,	,,	,,	,,	21	,,	25 ,,
No. 6	,,	,,	,,	,,	17	,,	20 ,,
No. 7	,,	,,	,,	,,	14	,,	16 ,,
No. 8	,,	,,	,,	,,	12	,,	13 ,,

The shape of the cutter conforms to that of the gear having the smallest number of teeth.

The above cutters are satisfactory for gears working under general conditions where special attainments are not needed, and for gears working under steady loading conditions.

For more accurate work and more precise conditions, cutters bearing half numbers are made as follows:—

No. 1½	for cutting gears having		80	teeth to	134	teeth
No. 2½	,,	,,	,,	,,	42	,,	54 ,,
No. 3½	,,	,,	,,	,,	30	,,	34 ,,
No. 4½	,,	,,	,,	,,	23	,,	25 ,,
No. 5½	,,	,,	,,	,,	19	,,	20 ,,
No. 6½	,,	,,	,,	,,	15	,,	16 ,,
No. 7½	,,	,,	,,	,,	13 teeth only		

Spur Gear Calculations.

Relation between circular pitch and diametral pitch.

c.p. = circular pitch; *d.p.* = diametral pitch; *p.c.d.* = pitch circle diameter. N = number of teeth; $\pi = 3{\cdot}1416$.

$$c.p. = \frac{\pi \times p.c.d.}{N}; \quad d.p. = \frac{N}{p.c.d.}$$

Hence,

$$c.p. \times d.p. = \pi; \quad c.p. = \frac{\pi}{d.p.}; \quad d.p. = \frac{\pi}{c.p.}$$

Example: A wheel has 60 teeth of 10*d.p.* What is its *p.c.d.* and *c.p.*?

$$p.c.d. = \frac{60}{10} = 6 \text{ ins.}; \quad c.p. = \frac{\pi}{10} = 0{\cdot}314 \text{ ins.}$$

The Number of Teeth being given and the Diametral Pitch, to find the Outside Diameter of the Blank required for 14½° standard Involute Gear.

RULE.—Add 2 to the number of teeth and divide by the diametral pitch.

EXAMPLE.—No. of teeth 40, *d.p.* 4. Add 2 to the 40, making 42, and divide by 4; the quotient is 10½ ins., which is the required outside diameter of the blank.

The distance between centres of two mating gears is found by adding the number of teeth together and dividing half the sum by the diametral pitch.

If two mating gears have respectively 60 and 40 teeth, add 60 and 40 together, making 100, which divided by 2 gives 50. Assuming the diametral pitch of the gears is 4, 50 divided by 4 gives 12½ ins., the required centre distance.

To determine the Pitch Line Velocity of a Gear.

RULE.—Multiply the pitch circle diameter by π and the product by the number of revolutions per second. Divide the final product by 12, and the result is the pitch line velocity in feet per second.

EXAMPLE.—A wheel of 6 ins. *p.c.d.* runs at 320 revs. per min. What is its pitch line velocity?

$$\text{Pitch line velocity} = \frac{3{\cdot}1416 \times 6 \times 320}{12 \times 60} = 8{\cdot}4 \text{ ft. per sec.}$$

or 504 ft. per minute.

To compute the Circular Pitch of a Wheel.

RULE.—Divide circumference at the pitch line by the number of teeth.

EXAMPLE.—A wheel 40 ins. in diameter requires 75 teeth; what is its circular pitch?

$$\frac{3{\cdot}1416 \times 40}{75} = 1{\cdot}6755 \text{ ins.}$$

To compute the Chordal Pitch.

RULE.—Divide 180° by the number of teeth, ascertain the sine of the quotient, and multiply it by the diameter of the wheel.

EXAMPLE.—The number of teeth is 75, and the diameter 40 ins.; what is the chordal pitch?

$$\frac{180}{75} = 2° 24' \text{ and sine of } 2° 24' = \cdot04188, \text{ which} \times 40 = 1\cdot6752 \text{ in.}$$

To compute the Diameter of a Wheel.

RULE.—Multiply the number of teeth by the circular pitch, and divide the product by 3·1416.

EXAMPLE.—The number of teeth in a wheel is 75, and the circular pitch 1·675 ins.; what is the diameter of the wheel?

$$\frac{75 \times 1\cdot675}{3\cdot1416} = 40 \text{ ins.}$$

To compute the Number of Teeth in a Wheel.

RULE.—Divide the pitch circumference by the circular pitch.

To compute the Number of Teeth in a Pinion or Follower to obtain a given Speed Ratio.

RULE.—Multiply the number of teeth in the driving gear by the speed ratio of the two gears.

EXAMPLE.—A driving gear makes 40 revs. per min., and has 54 teeth. The pinion is to run at 120 revs. per min.; what number of teeth is required?

$$\text{Speed ratio} = \frac{40}{120} = \frac{1}{3}.$$

$$\text{No. of teeth required in pinion} = \frac{54 \times 1}{3} = 18.$$

To compute the Diameter of a Pinion when the Diameter of the Wheel and Number of Teeth in the Wheel and Pinion are given.

RULE.—Multiply the diameter of the wheel by the number of teeth in the pinion, and divide the product by the number of teeth in the wheel.

EXAMPLE.—The diameter of a wheel is 25 inches, the number of its teeth 210, and the number of teeth is the pinion 30; what is the diameter of the pinion?

$$\frac{25 \times 30}{210} = 3\cdot57 \text{ inches.}$$

To compute the Number of Teeth required in a Train of Wheels to produce a given Velocity Ratio.

RULE.—Multiply the number of teeth in the driver by its number of revolutions, and divide the product by the number of revolutions of each pinion, for each driver and pinion.

EXAMPLE.—If a driver in a train of 3 wheels has 90 teeth, and makes 2 revolutions, and the velocity ratios required are 2, 10, and 18, what are the number of teeth in each of the other two?

$$10 : 90 :: 2 : 18 \quad \therefore \text{ teeth in 2nd wheel} = 18$$
$$18 : 90 :: 2 : 10 \quad \therefore \text{ teeth in 3rd wheel} = 10$$

To compute the Circumference of a Wheel.

RULE.—Multiply the number of teeth by their circular pitch.

To compute the Revolutions of a Wheel or Pinion.

RULE.—Multiply the diameter or circumference of the wheel, or the number of its teeth, as the case may be, by the number of its revolutions, and divide the product by the diameter or circumference of the wheel, or number of teeth in the pinion.

EXAMPLE.—A pinion 10 inches in diameter is driven by a wheel, 2 feet (24 inches) in diameter, making 46 revolutions per minute; what is the number of revolutions of the pinion?

$$\frac{24 \times 46}{10} = 110\cdot4 \text{ revolutions.}$$

To compute the Speed of a Pinion.

RULE.—Divide the diameter, circumference, or number of teeth in the driver, as the case may be, by the diameter, etc., of the pinion.

When there are a Series or Train of Wheels and Pinions.

RULE.—Divide the continued product of the diameter, circumference, or number of teeth in the wheels, by the continued product of the diameter, &c., of the pinions.

EXAMPLE 1.—If a wheel of 32 teeth drives a pinion of 10, upon the axis of which there is one of 30 teeth driving a pinion of 8, what are the revolutions of the last?

$$\frac{32}{10} \times \frac{30}{8} = \frac{960}{80} = 12 \text{ revolutions.}$$

EXAMPLE 2.—The diameters of a train of wheels are 6, 9, 9, 10, and 12 inches; of the pinions, 6, 6, 6, 6, and 6 inches; and the number of revolutions of the driving-shaft or prime mover is 10; what are the revolutions of the last pinion?

$$\frac{6 \times 9 \times 9 \times 10 \times 12 \times 10}{6 \times 6 \times 6 \times 6 \times 6} = \frac{583200}{7776} = 75 \text{ revolutions.}$$

To compute the Number of Teeth in each Wheel of a Train for a given Speed Ratio.

The numbers of teeth in the wheels are inversely proportional to the speeds.

EXAMPLE.—What should be the number of teeth in each of three wheels so that the speeds of each are in the ratio of 3 : 2 : 1, the pinion having 18 teeth?

$$\text{No. of teeth in second wheel} = 18 \times \frac{3}{2} = 27.$$

$$\text{,, ,, third ,,} = 27 \times \frac{2}{1} = 54.$$

FORMULÆ FOR DETERMINING THE DIMENSIONS OF GEARS BY THE METRIC PITCH.

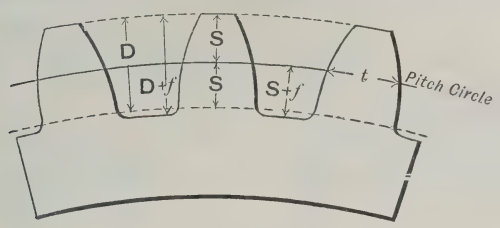


FIG. 5.

M = module; D′ = pitch diameter of gear; D = whole diameter of gear; N = number of teeth in gear; D″ = working depth of teeth; t = thickness of teeth on pitch line; f = amount added to depth for clearance.

Then

$$M = \frac{D'}{N} \text{ or } \frac{D}{N+2}. \qquad D' = NM. \qquad D = (N+2)\,M.$$

$$N = \frac{D'}{M} \text{ or } \frac{D}{M} - 2. \qquad D'' = 2M. \qquad t = M\,1{\cdot}5708. \qquad f = \frac{M\,1{\cdot}5708}{10}.$$

The module is equal to the part marked 'S' in fig. 5, measured in mm. and parts of mm.

RELATION BETWEEN MODULE AND DIAMETRAL PITCH.

Module.	Diametral Pitch.	Module.	Diametral Pitch.	Module.	Diametral Pitch.	Module.	Diametral Pitch.
0·5	50·800	2·5	10·160	6	4·233	14	1·814
0·75	33·867	3	8·466	7	3·628	15	1·693
1	25·400	3·5	7·257	8	3·175	16	1·587
1·25	20·320	4	6·350	9	2·822	17	1·494
1·5	16·933	4·5	5·644	10	2·540	18	1·411
1·75	14·514	5	5·080	11	2·309	19	1·336
2	12·700	5·5	4·618	12	2·117	20	1·270
2·25	11·288			13	1·954		

NOTE.—In the above table the module is worked out in millimetres. By reducing the dimensions to inches the equivalent diametral pitch is obtained.

EXAMPLE.—Module = 0·5, determine equivalent diametral pitch.

$\frac{1}{0·5} = 2$. Since 25·4 mm. = 1 in., ∴ equivalent diametral pitch = 50·800.

Mill Gearing.

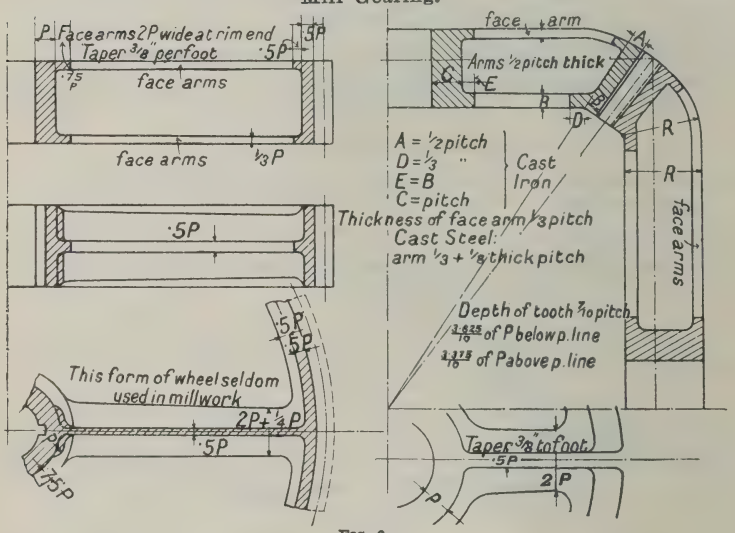


FIG. 6.

Fig. 6 shows the proportions of tooth, arm, faces, bosses, etc., commonly used in mill gearing. (*J. Nuttall.*)

General Method of Determining Dimensions of Gear Teeth to carry a Given Load.

Horse-power (H.P.) to be transmitted by gear $= \frac{WV}{33{,}000}$; W = tangential load at pitch line in lbs.; V = pitch line velocity in feet per min.

$$W = \frac{33{,}000 \times H.P.}{V} \text{ lbs.}$$

In general, it is assumed that the load W is concentrated at the point of the tooth, and the tooth to act as a cantilever.

Hence, if h = height of tooth in ins., f = face width in ins., b = breadth of tooth in ins. at the *weakest section*, s = safe stress for the material used in lbs. per sq. in.

$$Wh = \tfrac{1}{6}fb^2s; \quad f = \frac{6Wh}{b^2s}; \quad b = \sqrt{\frac{6Wh}{fs}}.$$

The breadth of a tooth and its height can be fixed initially, according to general proportions. Usually it is convenient to calculate the required face width to give the required strength.

The safe stress can be calculated from the formula following in connection with the Lewis formula. It will depend upon loading conditions and material used.

Experience has proved that a cast-iron wheel having teeth 3 ins. pitch and a face width of 7½ ins. will transmit at 6·66 ft. per sec. 60 horse-power.

To compute the Horse-Power transmitted by a Gear.

RULE.—Multiply the tangential load in lbs. at the pitch line by the pitch line velocity in ft. per min. and divide the product by 33,000.

EXAMPLE.—A gear 8 ins. *p.c.d.* runs at 400 r.p.m. and carries a pitch line load of 800 lbs. What horse-power is transmitted?

$$\text{Pitch line velocity} = \frac{\pi \times p.c.d. \times N}{12} = \frac{3{\cdot}1916 \times 8 \times 400}{12} = 837 \textit{ ft. per min.}$$

$$\text{H.P.} = \frac{800 \times 837}{33{,}000} = 20{\cdot}3.$$

The Lewis Formula.

(1.)

This formula is in general use for determining gear strength, and is used extensively in design. It was intended for use on machine-cut gears, but with suitable modification can be applied to machine moulded teeth and even ordinary cast gears when suitably modified.

W = safe tangential load at pitch line in lbs.; y = Lewis factor, which depends upon the tooth form; s = safe stress for the material used in lbs. per sq. in.; f = width of tooth face in ins.; p = circular pitch of teeth in ins.

Then

$$W = ysfp.$$

The values of s and y are given in the following tables:—

LEWIS'S VALUES OF s.

Speed in Feet per Minute.	100 or less.	200	300	600	900	1,200	1,800	2,400
Cast iron .	8,000	6,000	4,800	4,000	3,000	2,400	2,000	1,700
Steel . .	20,000	15,000	12,000	10,000	7,500	6,000	5,000	4,300

LEWIS'S VALUES OF y.

Number of Teeth.	Involute (20 deg. obliquity).	Involute (15 deg. obliquity) and Cycloidal.	Radial Flanks.	Number of Teeth.	Involute (20 deg. obliquity).	Involute (15 deg. obliquity) and Cycloidal.	Radial Flanks.
12	0·078	0·067	0·052	27	0·111	0·100	0·064
13	0·083	0·070	0·053	30	0·114	0·102	0·065
14	0·088	0·072	0·054	34	0·118	0·104	0·066
15	0·092	0·075	0·055	38	0·122	0·107	0·067
16	0·094	0·077	0·056	43	0·126	0·110	0·068
17	0·096	0·080	0·057	50	0·130	0·112	0·069
18	0·098	0·083	0·058	60	0·134	0·114	0·070
19	0·100	0·087	0·059	75	0·138	0·116	0·071
20	0·102	0·090	0·060	100	0·142	0·118	0·072
21	0·104	0·092	0·061	150	0·146	0·120	0·073
23	0·106	0·094	0·062	300	0·150	0·122	0·074
25	0·108	0·097	0·063	Rack	0·154	0·124	0·075

The values for s will depend upon pitch line velocity, and can be selected from the above table or can be determined from the following formula :—

$$S_v = S_o \left(\frac{600}{600 + V}\right) \quad \text{NOTE:—SEE ADDENDUM.}$$

where,

S_v = Safe stress at given pitch line velocity in lbs. per sq. in.; S_o = Safe stress at basic velocity in lbs. per sq. in.; V = Actual pitch line velocity in ft. per min.

Suitable values for S_o.—Nickel-chrome gears, 100,000 lbs. per sq. in.; steel-cut gears, 20,000 lbs. per sq. in.; cast iron, 8,000 lbs. per sq. in.

When serious shock loads are likely to occur, or when alternations in load occur, the above values must be modified accordingly.

EXAMPLE.—A mild steel gear having a pitch circle 6 ins. diameter is to transmit 20 horse-power at 400 revs. per min. Determine the required pitch and width of face.

$$\text{Velocity of pitch line} = \frac{\pi \times 6 \times 400}{12} = 628 \text{ ft. per min.}$$

$$S_v = S_o \left(\frac{600}{600 + V}\right) = \frac{20{,}000 \times 600}{1228} = 10{,}000 \text{ lbs. per sq. in. approx.}$$

$$W = \frac{33{,}000 \times \text{H.P.}}{V} = \frac{33{,}000 \times 20}{628} = 1{,}052 \text{ lbs.}$$

Assume value of y for 20 teeth = 0·09 and width of face = $3p$.

$$W = ysfp\,; \quad 1{,}052 = 0{\cdot}09 \times 10{,}000 \times 3p^2.$$

$$p = \sqrt{\frac{1{,}052}{0{\cdot}09 \times 10{,}000 \times 3}} = \tfrac{5}{8} \text{ in. circular pitch } (0{\cdot}625).$$

Check number of teeth since originally number was assumed.

$$\text{No. of teeth in gear} = \frac{\pi \times p.c.d.}{c.p.} = \frac{3{\cdot}1416 \times 6}{0{\cdot}625} = 30$$

Actual pitch circle diameter would be $\dfrac{30 \times 0{\cdot}625}{\pi} = 5{\cdot}968$ ins.

Width of face = $3p = 1\tfrac{7}{8}$ in.

From the table of values of y its value for 30 teeth is greater than that for 20 teeth, the assumed value. Hence the actual safe load for the gear is a little greater than is required. In cases of considerable differences fresh values of y must be chosen, and the calculation for pitch repeated.

MODIFICATION OF LEWIS FORMULA.

By H. T. Davey, A.M.I.Mech.E.

(2.)

For the application to gearing problems where it is intended to use circular pitches,

$$W = ysfp.$$

For similar problems where it is intended to use diametral pitches,

$$W = \frac{zsF}{P}$$

where,

W = Safe tangential load at pitch line in lbs.; y = Lewis factor for circular pitch gears; s = Safe stress at given pitch line velocity in lbs. per sq. in.; f = Width of face in ins.; p = Circular pitch of teeth in ins.; z = Lewis factor for diametral pitch gears; F = Width of face in ins.; P = Diametral pitch.

Most practical problems consist of determining a suitable pitch of teeth to transmit a given horse-power, together with the width of face, which in order to solve the equation for any particular case, is given some value in terms of the pitch. The safe stress at the given pitch line velocity is determined either from a table, or can be calculated from the equation—

$$S_v = S_o \left(\frac{600}{600 + V}\right) \quad \text{NOTE :—SEE ADDENDUM.}$$

where,

S_v = Safe stress at pitch line velocity V ft. per min. in lbs. per sq. in.; S_o = Safe stress at zero speed in lbs. per sq. in.

The following values of S_v may be taken :—

Nickel-chrome gears, 100,000 lbs. per sq. in. ; steel castings and ordinary steel cut gears, 20,000 lbs. per sq. in. ; cast iron, 8,000 lbs. per sq. in.

If the gearing is liable to serious shock or alternations in the loading, then the values of S_v calculated from the above should be reduced accordingly.

In a practical problem the approximate pitch circle diameter is usually known, or can be fixed according to the conditions ruling.

In applying the Lewis formula to the problem of finding the pitch, we have two unknowns—(*a*) Lewis factor ; (*b*) pitch of teeth. The Lewis factor depends upon the number of teeth, as also does the pitch for a given pitch circle diameter. It is common to *assume* a number of teeth and select from a table the corresponding Lewis factor. If the number of teeth assumed is incorrect as shown by the application of the formula, then fresh values for the number of teeth have to be selected until a suitable pitch is obtained consistent with strength. The method is, therefore, one of trial and error, and requires considerable judgment and experience in order to obtain suitable results.

In order to obviate this, the formula can be written as follows :—

Let,

$$\text{Width of face } f = 3p \text{ (which agrees with average practice)};$$

then, for circular pitch gears,

$$W = 3ysp^2.$$

Also,

$$p = \frac{\pi D}{N};$$

where,

$$N = \text{number of teeth}; \quad D = \text{pitch circle diameter};$$

and,

$$W = \frac{3ys\pi^2D^2}{N^2}; \quad \frac{y}{N^2} = \frac{W}{3\pi^2D^2s}.$$

Multiplying the right-hand side of equation by 10,000, adjusting values of $\frac{y}{N^2}$ accordingly, we get

$$\frac{y}{N^2} = \frac{337{\cdot}7W}{D^2s},$$

which permits easier tabulation and facilitates reading.

For diametral pitch gears,

$$W = \frac{S.F.Z.}{P} = \frac{S.F.Z.D}{N}; \quad W = \frac{3\pi^2D^2SZ}{N^2}; \quad \frac{Z}{N^2} = \frac{W}{3\pi D^2S}.$$

Or, multiplying the right-hand side by 10,000,

$$\frac{Z}{N^2} = \frac{1061W}{D^2S}.$$

If other values for the width of face are required the formulæ become

$$\frac{y}{N^2} = \frac{1013{\cdot}2W}{KD^2S} \text{ for circular pitches,}$$

and

$$\frac{Z}{N^2} = \frac{3183W}{KD^2S} \text{ for diametral pitches.}$$

$$K = \frac{\text{width of face}}{\text{circular pitch}}, \text{ a value to be decided upon initially.}$$

If values of $\frac{y}{N^2}$ and $\frac{Z}{N^2}$ are tabulated, then it becomes unnecessary to assume the number of teeth, since this can be seen at a glance from a table when the above calculation has been made, and the required pitch of teeth can then be found for the given pitch circle diameter.

Some slight adjustments may be necessary in order to make the pitch such as to enable the gear to be cut by the usual standard cutters.

No. of Teeth.	Involute 20° Obliquity.				Involute 15° Obliquity.			
	y	z	$\frac{y}{N^2}$	$\frac{z}{N^2}$	y	z	$\frac{y}{N^2}$	$\frac{z}{N^2}$
12	0·078	0·245	5·42	17·0	0·067	0·211	4·65	14·6
13	0·083	0·261	4·91	15·4	0·070	0·220	4·14	13·0
14	0·088	0·276	4·49	14·1	0·072	0·227	3·67	11·6
15	0·092	0·289	4·41	12·8	0·075	0·236	3·33	10·5
16	0·094	0·296	3·67	11·5	0·077	0·242	3·00	9·45
17	0·096	0·302	3·32	10·5	0·080	0·252	2·77	8·69
18	0·098	0·307	3·02	9·47	0·083	0·261	2·56	8·05
19	0·100	0·314	2·77	8·69	0·087	0·274	2·41	7·59
20	0·102	0·321	2·55	8·02	0·090	0·283	2·25	7·07
21	0·104	0·326	2·35	7·39	0·092	0·289	2·09	6·55
23	0·106	0·333	2·00	6·29	0·094	0·296	1·77	5·59
25	0·108	0·340	1·72	5·44	0·097	0·305	1·55	4·88
27	0·111	0·349	1·52	4·78	0·100	0·314	1·37	4·31
30	0·114	0·358	1·26	3·98	0·102	0·321	1·13	3·56
34	0·118	0·371	1·02	3·21	0·104	0·326	0·90	2·82
38	0·122	0·384	0·845	2·66	0·107	0·336	0·741	2·32
43	0·126	0·396	0·681	2·14	0·110	0·346	0·595	1·87
50	0·130	0·408	0·520	1·63	0·112	0·352	0·448	1·40
60	0·134	0·421	0·372	1·17	0·114	0·358	0·316	0·994
75	0.138	0·434	0·245	0·771	0·116	0·365	0·206	0·648
100	0·142	0·446	0·142	0·446	0·118	0·371	0·118	0·371
150	0·146	0·459	0·0644	0·204	0·120	0·377	0·0533	0·168
300	0·150	0·472	0·0166	0·0524	0·122	0·384	0·0135	0·0426

Example.—A spur gear is required to transmit 60 h.p. from a motor shaft running at 400 revs. per min.; the gear is to be of steel, having an approximate pitch circle diameter of 8 ins. What will be the pitch of the teeth, and width of face, for 20° involute gear?

$$\text{Horse-power transmitted} = \frac{WV}{33{,}000}; \quad V = \frac{\pi \times 8 \times 400}{12};$$

therefore,

$$W = \frac{60 \times 33{,}000 \times 12}{8\pi \times 400} = 2{,}362 \text{ lbs.}; \quad S_v = 20{,}000 \left(\frac{600}{600 + 837{\cdot}5}\right) = 8{,}350 \text{ lbs. per sq. in.};$$

$$\frac{y}{N^2} = \frac{337{\cdot}7 \times 2362}{64 \times 8350} = 1{\cdot}49.$$

From the table, the value of 1·49 corresponds most nearly to 27 teeth; the face width will therefore be 2·8 ins.

The Lewis formula is based upon the assumption that the entire loading comes upon one pair of mating teeth, and that the load is concentrated at the tooth point. The compressive and shear components are ignored. The result is that the teeth so designed are somewhat on the heavy side. It should, however, be realised that teeth wear away from their original true shape, and hence the point of application of the load may reach the tooth point, and it may possibly occur that the load is borne by one pair of mating teeth at some particular instant.

Gearing on Armature Shafts.

In gearing, large revolving masses—as, for example, armatures of motors—should be separated from adjoining masses, such as large gear-wheels of the second motion, by some elastic medium, such as a length of shafting or a flexible coupling. This is intended to allow for slight variations in the angular velocity, due to unavoidable errors in cutting the teeth, to take place without giving rise to destructive loads between the teeth. The stresses due to inaccuracies in the spacing of the teeth may exceed the working stress. With an error of only 0·2 per cent. in the pitch the excess load due to acceleration and retardation may amount to as much as 28 times the load transmitted. Such loads increase as the square of the mean pitch-line velocity and as the mass of the wheel considered.

Non-Resonant Pinions.

Raw-hide, fibre, fabroil, bakelite, and other similar materials, are often used for non-resonant pinions. The pinions driving from the shafts of electric motors are frequently made of one of these materials.

The stipulation that the number of teeth in the wheel should not be an even multiple of the number of teeth in the pinion is now obsolete, if modern methods of gear cutting are employed.

For bakelite pinions the safe stress at the basic speed $S_o = 6{,}000$ lbs. per sq. in. In order to allow for pitch line velocity, the American Gear Manufacturers' Association recommend the use of the following formula, which can also be applied when either of the other materials mentioned are used, provided suitable values of S_o are adopted.

$$S_v = S_o\left(\frac{150}{200 + V} + 0{\cdot}25\right).$$

The general procedure for calculating gear strength is then as for metal gears.

Raw-hide Pinions.

Often made to replace metal pinions running at high speed. 1,600–1,700 ft. per min. is generally considered maximum pitch line speed. When power to be transmitted is high, metal reinforcing plates on the sides are required. For this purpose bronze plates bolted or riveted are common. The maximum tooth pressure per in. width of tooth face is about 250 lbs., the pressure being decreased as the pitch line velocity increases. For 1-in. pitch gears about 140–150 lbs. is usual.

It is important to keep oil and grease away from raw-hide pinions, as mineral oil proves most destructive. A dressing of raw linseed-oil occasionally keeps the material in good condition and preserves it.

Fabroil Gears.

(By permission of The British Thomson-Houston Co., Ltd.)

Fabroil as used by the B.T.H. for non-resonant gears has a distinct advantage, that it is not affected by the action of water, acid fumes, or hot or cold oil.

Gears made from this material have been used extensively for quayside and wharf machinery, on donkey-engines and various purposes aboard ship, and also in chemical works.

The material is elastic, and is thus capable of absorbing shock. For delicate machinery, where vibration would be detrimental, these gears are most useful.

Fabroil gears are made in three forms, each of which can be supplied with spur, bevel, or spiral teeth : (*a*) shrouded ; (*b*) moulded ; (*c*) without shrouds, made from sheet material or board.

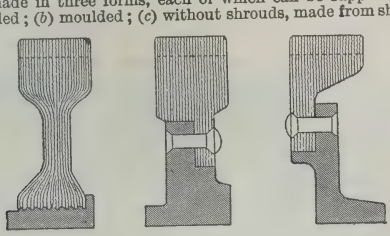


FIG. 7.

Fig. 7 shows how the fabric elements follow the contour lines of the gears, the web being obtained without machining.

The gears are cut to Brown and Sharpe's standard 14½° involute section, but other forms can be made if desired.

The wearing qualities of this material have proved excellent, and besides extending the life of the mating gear, will often outlast a metal gear working under the same conditions.

Before running these gears a light dressing of vaseline and graphite should be given, and periodically similar treatment applied.

Fabroil gears must not be used in conjunction with uncut cast-iron gear-teeth or with other teeth which are badly worn.

The strength of the teeth may be calculated by using the Lewis formula for the particular case under consideration, provided suitable constants are employed. These are as follows :—

Values of 'S' * (Unit Stress) for Fabroil Gears at Different Speeds.

The table below has been compiled from the results of recent intensive tests. The values given for 'S' at the stated pitch line velocities allow greater tooth loads to be carried at high pitchline

* These values of 'S' apply only to shrouded gears. Multiply by 0·8 for other types of gear.

velocities than is the case with cast-iron gears. This is permissible in view of the fact that the Fabroil material has much greater shock-absorbing capacity than cast iron.

Pitch Line Velocity in ft. per min.	100 or Less.	200	300	600	900	1,200	1,800	2,400
Values of 'S'	5,500	3,440	3,025	2,405	2,120	1,960	1,785	1,695

Size of Tooth.

The following table gives the diametral pitch of gearing recommended for the maximum H.P. transmitted. While smaller teeth may afford sufficient strength in some cases, these standards should be adhered to as far as possible :—

Max. H.P. Transmitted.	Diametral Pitch.	Max. H.P. Transmitted.	Diametral Pitch.	Max. H.P. Transmitted.	Diametral Pitch.
$\frac{1}{6}$	12	5	6	50	$2\frac{1}{2}$
$\frac{1}{3}$	10	$7\frac{1}{2}$	5	60	$2\frac{1}{2}$
$\frac{1}{2}$	10	10	4	75	2
$\frac{3}{4}$	10	15	3	100	2
1	8	20	3	125	$1\frac{1}{2}$
2	8	25	3	150	$1\frac{1}{2}$
3	6	—	—	—	—

The minimum number of teeth should be 18 to 20, if possible.
Gears with less than 16 teeth should never be used if it can be avoided.

Width of Face of Fabroil Gears.—The width of the non-metallic face in the case of shrouded gears should be equal to the face of the mating gear plus the aggregate end play of both shafts. The metal mating gear must not be allowed to mesh with the shrouds.

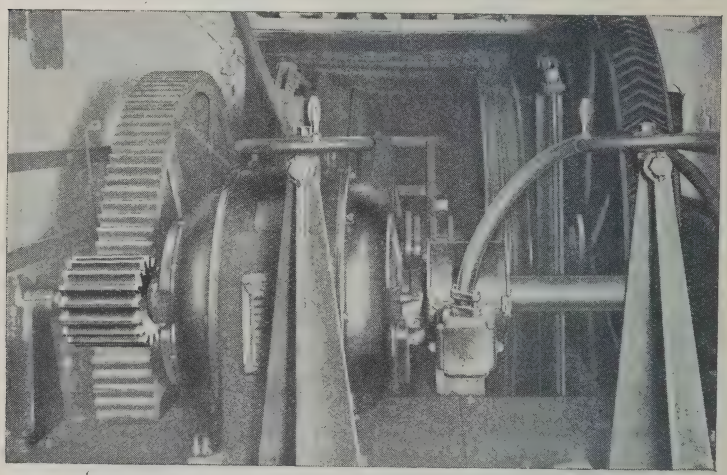


Fig. 8.—Colliery Haulage Gear with Fabroil Gear Drive.

Resilient Steel Gears.

(Seymour's Patent.)*

These gears are designed and constructed specially for drives where the strength and durability of steel gears is necessary, but where the noise accruing from their use would not be permissible.

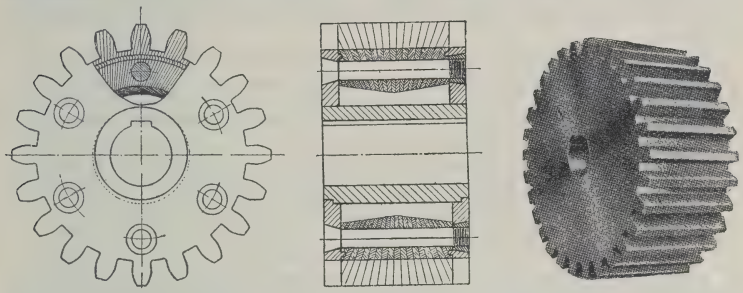


Fig. 9.—Section of Pinion with a Portion of End-plate cut away.

The portion of the gear-wheel from which the teeth are formed is made up of steel laminæ, having an ultimate tensile strength of 45 tons per sq. in., the boss and shroud plates being constructed of cast or forged steel.

The resilience of the working surfaces of the teeth afforded by this method of construction is sufficient to render the running of the gears as noiseless (even when the same are considerably overloaded) as gears constructed of raw hide, paper, and woven fabrics.

On account of the superior strength and durability of steel over the other substances named, the resilient steel gears transmit many times the load for any given size of tooth, and their life is longer proportionally.

Owing to the internal system of automatic lubrication, only the necessary amount of lubricant is supplied to the surfaces of the teeth, thus preventing waste and effecting a saving of $3\frac{1}{2}$ to 4 per cent. of the power transmitted.

Important improvements in the construction of these gears have been made, notably the connecting together of the lateral extremities of the laminæ by welding, thus the laminæ are firmly secured together independently of the support supplied by the shroud plates and bolts.

These gears have been successfully applied to almost every class of drive, including mine and quarry machinery and haulage, electric motor drives, pumps, air compressors, machine tools, cranes, etc., etc.

Fig. 9 shows laminæ and shroud plates held together with bolts.

The cavity between the boss and the laminæ is the receptacle for the lubricant; centrifugal force propels the lubricant radially outwards between the laminæ, thus efficiently lubricating the entire working surface of the teeth.

The power transmitted by such gears can be calculated from the following formula :—

$$F = \frac{HP \times 1000}{K \times N \times R'}$$

where,

F = Face width of pinion in inches; N = No. of teeth; K = Factor for each pitch (see table); R = Revolutions of pinion per minute; HP = Horse-power to be transmitted.

Circular Pitch . .	$\frac{1}{2}''$	$\frac{5}{8}''$	$\frac{3}{4}''$	$\frac{7}{8}''$	$1''$	$1\frac{1}{4}''$	$1\frac{1}{2}''$	$1\frac{3}{4}''$	$2''$	$2\frac{1}{4}''$	$2\frac{1}{2}''$	$3''$
Factor K . . .	·225	·35	·505	·689	·9	1·41	2·03	2·75	3·6	4·55	5·63	8·1
Diametral Pitch . .	6	5	4	$3\frac{1}{2}$	3	$2\frac{1}{2}$	$2\frac{1}{4}$	2	$1\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{1}{4}$	1
Factor K . . .	·246	·355	·555	·723	·975	1·41	1·74	2·22	2·92	3·93	5·67	8·9

* The Skerne Works, Ltd., Darlington, informs us that they no longer make this gear but produce the Skerne Resilient Gear instead.

Gearing Noises.

During the last few years considerable attention has been given to noise produced by gears. The kind of noise produced by a pair of gears will often give some indication of defects present.

High-class gearing running in perfect order produces a low-pitched humming sound with absence of beats, the sound being quite 'smooth' and continuous.

The greater the accuracy in machining the teeth and the better-class finish put upon the tooth faces, all tend to reduce noise. Irregular spacing of teeth is a common defect, and can be recognised by an intermittent clicking or a pulsating howl. Similar sounds occur when teeth are mis-shaped.

A pulsating howl may also arise due to improper setting of the gears upon their spindles. Eccentricity is a common cause.

High-pitched screeching often occurs, due to rough tooth faces. A common defect, particularly in heavy gearing, is to use either too little support for the shafts or too small a shaft diameter. Such errors set up spring in the shafts, resulting in pulsating sounds of varying pitches according to the pitch-line velocities.

A good condition of a set of gears, particularly those running at high speed, is indicated by a more or less musical sound, or sounds of constant pitch.

In the case of gears which are required to run in either direction, the fact that noise prevails more in one direction than another indicates that the teeth are not radial.

Generally, fine-pitched teeth and wide face widths tend toward noise reduction.

Critical Speeds* of Gearing.

Certain conditions of gear-teeth surfaces after running for some time cannot be directly traceable to general causes. Results of investigations go to show that the running of gear shafts at speeds too near that of the critical speeds results in the development of rough tooth profiles and general deformation of the surfaces in contact. Vibrations are often manifest in gearing, particularly in the case of high speeds. These will be considerably increased if they harmonise with those of the shaft at its critical speed, and such conditions may produce fracture.

In cases where gears are running at or near the critical speeds of their shafts, unsatisfactory running will invariably result. Hence dimensions of shafts must be arranged so that this defect is avoided.

Approximate rule for determining critical speed :—

$$S_c = \frac{188}{\sqrt{\delta}}$$

where,

S_c = critical speed of shaft in r.p.m.; δ = max. deflection of shaft in ins.

If a gear is required to run above the first critical speed, then when starting up the gear must be allowed to pass quickly over this critical speed period. In general, the working speed of a gear should not be within 28 per cent. or 30 per cent. of the critical speed.

Grinding of Gear Teeth.

The necessity of running gears at high pitch line velocities up to 16,000 or 17,000 ft. per minute has given rise to the need for finer and more accurate finishing of tooth profiles. Hardened steel gears used under severe conditions are ground. At high pitch line velocities small defects in the tooth faces are very detrimental to efficient and continuous working.

Defects in tooth faces produce noise, and may also reduce the safe load that can be transmitted by a tooth.

As compared with cut gears, those which are ground produce a characteristic high-pitched note when running under load and at high pitch line velocity. Even fine cut grinding-wheels do not produce a sufficiently smooth surface to entirely eliminate noise, but after ground gears have been in service for a short time they should, if accurately cut, run almost noiselessly.

When gears are to be ground, errors due to the wearing of the grinding-wheel must be guarded against. The material from which the abrasive wheel is made must be carefully selected. No specific recommendation can be given as details affect each particular case, as does the material to be ground. Neither very hard nor very soft wheels should be employed. To secure most accurate results and hence guard against wear of wheel, precaution should be taken when grinding gears. A suitable dividing head should be employed, and arrangements made so that the first and last teeth to be ground are not adjacent. Such an arrangement reduces error due to wear of abrasive wheel. Care should be taken to grind teeth in such a way that the error is evenly distributed.

Lapping Gear Teeth.

The idea of running hardened gears together with lubricant and abrasive material has been tried. For producing a highly finished surface to the tooth faces the method is not satisfactory,

* See Morley's 'Strength of Materials' for complete treatment.

but it serves to correct slight errors in tooth profiles. From a production point of view the method is not satisfactory.

Good results as regards finish and accuracy can be obtained by running a hard steel gear in mesh with one made of softer material and having a wide face. Both gears should be run at high speed and with load on while abrasive material and lubricant are applied to the tooth faces. The hard gear should be made to traverse the face of the wider soft gear.

In some cases cut gears are run with load on with two or three hardened and ground gears. This process tends to produce a well-burnished tooth surface.

Hardening of Gears.

Appreciable wear on gear teeth rapidly causes deterioration of the tooth faces, with consequent serious results. Hence some method of hardening is required, particularly in cases where the working conditions are severe, as is the case of those for automobile drives. Such gears are invariably hardened.

Processes vary greatly, and great care is needed to prevent serious warping. To preserve accuracy slow and uniform heating is essential. Such treatment also tends to produce a homogeneous metal as a result. After preheating the chemical treatment is applied to the skin until a condition is secured suitable for the final quenching. The quench, which is usually made in an oil bath, should be so carried out as to ensure rapid and even cooling.

A successful process of hardening automobile gears is given by the *American Machinist*. It is claimed that a minimum amount of warping occurs, and the shrinkage of the shaft holes is inappreciable. The average case thickness is approximately 25 thousandths of an inch.

The gears are preheated in a large gas-oven or forge until a dull red heat is obtained; temp. about 1300°–1900° F. The gears are then boiled in a solution of cyanide of potassium for about 1 hr. After this treatment the gears are plunged into a bath of fish oil which is kept constantly in circulation to ensure uniform temperature throughout.

Heat Treatment of Gears for Machine Tools.

The use of case-hardened gears for this, and many other purposes, is to a large extent being superseded by the use of high carbon steel treated so as to give uniform texture throughout the gear tooth section. Although gears so treated do not appear to be so hard as when case-hardened, yet when put into service they wear well and retain their shape and size for a very long time.

Straight carbon, nickel, chrome-vanadium, and chrome-nickel steels are used for gears which are intended for case carbonising. Other similar alloys are also obtainable for the same purpose.

Generally speaking, a chrome alloy steel is to be recommended in preference to a nickel one for gear manufacture, since lamination is less likely than with nickel steel, particularly the 3½ per cent. variety.

For case-hardening purposes the carbon should be about 0·2 per cent. The carbon in case should be changed by the use of carbonising material, and working at correct temperature, to about 0·9 per cent. Temperature of carbonisation for the steels mentioned should be about 1600°–1650° F.

The final treatment by heat after the casing is particularly important, as both the core and the case need bringing to the proper state in order to secure full advantage of the hardening process.

Treatment.

The gears should be reheated, after having been cooled off when carbonised, to about 1600° F. and quenched. This produces the required core texture. To harden the case again reheat to about 1400°–1425° F. and quench. To provide additional toughness the gear may then be drawn through an oil bath at a temperature of 425° F.

High Carbon Gears.

The advantages of alloys in this category have been referred to. In addition gears made from these alloys, and heat-treated in proper manner, are less liable to serious distortion by warping, and are stronger. The process of hardening is generally of a simpler character and probably cheaper, although material may be slightly more expensive.

Chrome-nickel alloy is the chief material used for gears intended to be hardened throughout. Chrome-vanadium steel is also used.

Before heat treating, the correct temperatures to be used in the processes should be obtained from the steel makers. Generally the temperature should be raised gradually to about 1500°–1550° F., and then quenched in an oil bath. Further treatment consists of drawing through an oil bath after the first quenching. Such treatment, as given above, is applied to automobile gears and to other gearing working under equally severe conditions.

In all cases where gears have to be hardened the chief difficulty to contend with is warping, and hence in gearing design, sudden changes of section should be avoided; under-cutting at the tooth roots is also liable to give trouble and should not be resorted to when gears have to be hardened.

Bevel Gears.

In the case of bevel gears the diameter understood is the diameter of the base of the pitch cone D, fig. 10. The pitch is measured on this circle. In fig. 10, A and B are a bevel pair revolving on shafts OM and MN. AM represents the pitch cone surface of A and B. OA and EK are drawn at right angles to AM to cut the axes at O, H, and KL respectively. AH and AO are then

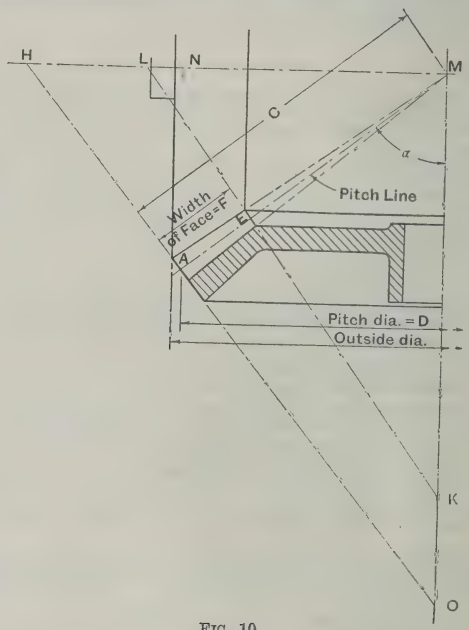


FIG. 10.

the virtual radii of the wheels A and B, and represent the radii of two spur gears which will gear similarly to the bevel pair. These are used as base-lines for setting out the teeth of the bevels A and B. Similarly, EL and EK are the radii for setting out the inner ends of the bevel teeth. The complete tooth lies evenly between these two shapes thus produced.

N' = no. of teeth in the equivalent spur gear; α = pitch cone angle.

$$N' = \frac{\text{No. of teeth in bevel gear.}}{\cos \alpha}.$$

Mitre Wheels.

If two shafts are at right angles and are connected by a bevel pair whose diameters are equal, giving equal speed of rotation, then such wheels are called *mitre wheels*.

Strength of Bevel Gears.

With slight modification the Lewis formula can be applied to bevel gears.

The modified form for bevel gears is—

$$W = \frac{Y s F(C - F)}{PC},$$

where,

P = diametral pitch; Y = Lewis factor selected from the table (p. 925); C and F are the dimensions given in fig. 10.

The factor $\frac{(C - F)}{C}$ approximately expresses the ratio of strength of a bevel gear to that of

a spur gear having the same number of teeth and pitch. As the pitch changes in a bevel gear the strength varies and hence the use of the modifying factor.

EXAMPLE.—Determine the maximum safe load that can be carried by a cast-iron bevel gear 3*d.p.* 20 ins. diameter. Pitch cone angle 76°, speed of wheel, 75 r.p.m.; angle of obliquity, 20°; width of face, 4 ins. Calculate also the safe horse-power that could be transmitted.

$$\text{No. of teeth in bevel gear} = 3 \times 20 = 60.$$

$$\text{Pitch-line velocity} = \frac{3{\cdot}1416 \times 20 \times 75}{12} = 400 \text{ ft. per min.}$$

$$N' = \frac{60}{\cos 76^\circ} = 250 \text{ teeth}; \quad S_v = S_o\left(\frac{600}{600 + V}\right) = \frac{8{,}000 \times 600}{1{,}000} = 4{,}800 \text{ lbs. per sq. in.}$$

$$C = \frac{10}{\sin. 76^\circ} = 10{\cdot}3 \text{ ins.};$$

$$W = \frac{Y S F(C - F)}{PC} = \frac{0{\cdot}467 \times 4{,}800 \times 4(10{\cdot}3 - 4)}{3 \times 10{\cdot}3} = 1{,}830 \text{ lbs.}$$

$$\text{Max. safe horse-power} = \frac{1{,}830 \times 400}{33{,}000} = 22.$$

VALUES OF 'Y' FOR BEVEL GEARS.

N'.	Y. 14½° Involute Teeth.	Y. 20° Involute Teeth.	N'.	Y. 14½° Involute Teeth.	Y. 20° Involute Teeth.
12	0·210	0·245	27	0·314	0·349
13	0·220	0·261	30	0·320	0·358
14	0·226	0·276	34	0·327	0·371
15	0·236	0·289	38	0·336	0·383
16	0·242	0·295	43	0·346	0·396
17	0·251	0·302	50	0·352	0·408
18	0·261	0·308	60	0·358	0·421
19	0·273	0·314	75	0·364	0·434
20	0·283	0·320	100	0·371	0·446
21	0·289	0·327	150	0·377	0·459
23	0·295	0·333	300	0·383	0·471
25	0·305	0·339	Rack	0·390	0·484

Maag Gearing.

This type of gearing introduces certain novel features both in production and shape of tooth profile. It is claimed that the full benefit of the involute tooth profile can be obtained with this

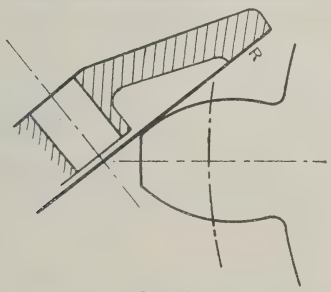


FIG. 11.

form of gearing. Standard tooth form is not adhered to, the teeth being cut on the generating principle and a true involute profile for each gear tooth obtained. Any pitch can be cut, using the same standard tool, which is in the form of a straight-sided rack cutter; i.e. the teeth are formed by cutters whose shapes correspond to the shape of the rack belonging to the gear wheel.

One of the essential features of this type of gearing is in the hardening and grinding of the teeth. The case is hardened to a depth of approximately ·04 in., after which it is drawn to a temperature of 200°–300° F. and ground.

The highest possible degree of precision is obtained in this grinding process, carried out on special machines having specially shaped abrasive wheels, which are accurately mounted and tested for wear by special means. The generating principle is used in this operation as in the milling operation.

Two abrasive wheels are inclined to each other so as to form by their lateral surfaces the tooth profile of the rack to which the gear belongs. The surfaces of the abrasive wheels are made hollow, so that the part marked R (fig. 11) only, forms the working plane.

Despite wear, this projecting part describes a working plane at right angles to the axis of rotation of the shaft.

Against this grinding form the gear is rotated in the correct way to give the proper shape of tooth—that is, it rolls with its pitch circle on a line corresponding to that of the rack.

The grinding wheels are moved backward and forward, so as to completely traverse the whole face width of the gear. The gear is also rolled until grinding is complete across the whole tooth face.

To allow for the wear of the abrasive wheels, compensation is provided by way of an automatic electrical device which keeps the planes of the grinding wheels in their exact position.

It is claimed that much higher tooth load can be carried by Maag gears at higher pitch-line velocities than with other types of gearing. It is further claimed that pitch-line speeds of 10,000 ft. per minute to 15,000 ft. per minute are possible, while the load per inch width can be double that allowed with double helical gears.

In the Swiss 2,500 h.p. locomotives, Maag gearing was used in place of the double helical gearing.

The increase in load-carrying capacity of Maag teeth is accounted for by the fact that the teeth are thicker at the root than ordinarily cut gears.

DEVICE FOR ADJUSTING ABRASIVE WHEELS FOR WEAR.

(German Patent No. 303014.)

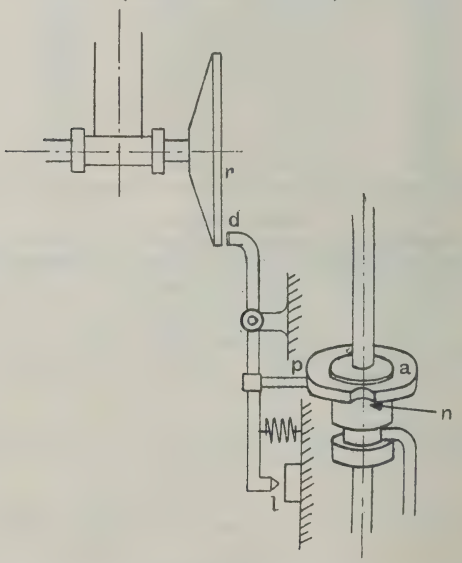


FIG. 12.

Fig. 12 shows the mechanism. *d* is a diamond contact piece which at uniform time intervals, controlled by the rotating disc *a*, is brought into contact very gently with the rim *r* of the abrasive wheel. The disc *a* has a notch *n* in its rim, into which the projecting piece *p* drops; thus the contact of *d* with *r* is at equal time intervals as *a* rotates uniformly.

If the abrasive wheel has worn, then the diamond piece follows it and brings an electrical contact l into action, which operates a mechanism to bring the abrasive wheels into their correct position. The electric contact breaks as soon as the adjustment is complete.

In order to check the accuracy of tooth profile a special machine is used. (*Saurer Arbur.*)

HELICAL GEARING.

Spiral or helical gears are used to transmit motion between two non-intersecting shafts, the axes of which may be inclined to one another at angles varying between 0° and 90°. When the shafts are at 90° to one another the drive is similar to that of a worm and worm wheel. Although the term 'spiral' is often used to indicate this type of gearing, actually each tooth is part of a helix, and the correct description of such gearing is helical rather than spiral.

Definitions.

'*Normal circular pitch*' is the distance between the centres of two consecutive teeth measured round the pitch cylinder and at right angles to the tooth.

'*Circumferential or real circular pitch*' is the distance between the centres of two consecutive teeth measured round the pitch cylinder and in a plane perpendicular to the axis of rotation. See fig. 13.

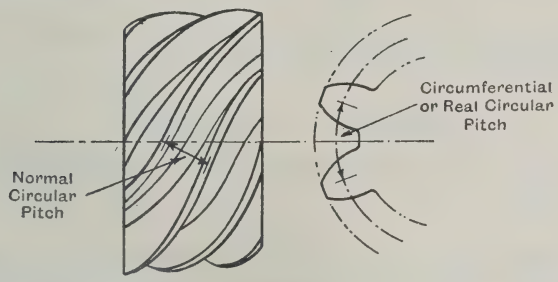


FIG. 13.

'*Spiral angle* (θ).'—The angle which the tooth helix makes with the axis of rotation. Fig. 14.

'*Lead angle* (λ).'—The complement of the spiral angle, or the angle between the tooth helix and a plane perpendicular to the axis of rotation. Fig. 14.

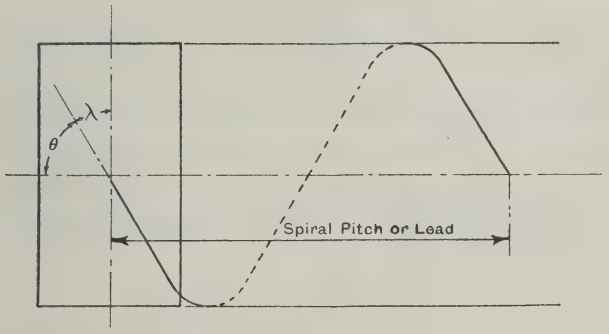


FIG. 14.

'*Spiral pitch or lead.*'—A spiral gear can be considered as a portion of a multiple-threaded screw. If the gear was made wide enough each tooth would make a complete turn on the blank.

The angle of the teeth controls the pitch of this complete turn, known as the *spiral pitch or lead.* In order to cut the teeth at any desired angle the lead is required. See fig. 14.

$$\text{Lead} = \frac{\text{Pitch circumference}}{\text{Tan } \theta}, \text{ where } \theta = \text{spiral angle.}$$

Increase of spiral angle increases the circular pitch, and hence the pitch diameter for the same number of teeth.

'*Normal pressure angle*' is the angle between a tangent to the tooth profile at the pitch cylinder and a radial, the angle being measured normal to the spiral. Definitions given previously for pitch cylinder, pitch diameter, etc., are similarly applied to spiral gears.

θ = spiral angle *r.c.p.* = real circular pitch ; *n.c.p.* = normal circular pitch ;

N = no. of teeth ; *r.d.p.* = real diametral pitch ; *n.d.p.* = normal diametral pitch ;

$$\text{Normal circular pitch} = \text{real circular pitch} \times \cos \theta.$$

$$n.c.p. = \frac{\pi}{n.d.p.}; \quad r.c.p. = \frac{n.c.p.}{\cos \theta}$$

$$\text{Pitch dia.} = \frac{r.c.p. \times N}{\pi} = \frac{N}{r.d.p.}; \quad \text{Outside dia.} = \text{Pitch dia.} + 2\,(\cdot 3183\ n.c.p.);$$

$$r.d.p. = n.d.p. \times \cos \theta.$$

The most common example of a spiral drive is that in which two spirals are used to transmit motion between two shafts at right angles to each other.

Consider two gears A and B meshing under the conditions above.

Let

D = pitch dia. of A ; d = pitch dia. of B.
N = no. of revs. of A ; n = no. of revs. of B.
α = spiral angle of A ; β = spiral angle of B.

Velocity ratio $= \frac{N}{n}$. The number of revolutions of the wheels is inversely proportional to the numbers of teeth.

$$\text{Number of teeth in A} = r.d.p._{A} \times D; \quad \text{Number of teeth in B} = r.d.p._{B} \times d.$$

Also,

$$\text{Number of teeth in A} = n.d.p._{A} \times \cos \alpha \times D; \quad \text{Number of teeth in B} = n.d.p._{B} \times \cos \beta \times d.$$

Since the wheels are in mesh the normal pitches must be the same, hence $n.d.p._{A} = n.d.p._{B}$.

Therefore,

$$\frac{\text{No. of teeth in A}}{\text{No. of teeth in B}} = \frac{D \cos \alpha}{d \cos \beta}$$

$$\frac{N}{n} = \frac{d \cdot \cos \beta}{D \cdot \cos \alpha} \quad . \quad . \quad . \quad . \quad . \quad (1)$$

and since $\alpha = 90 - \beta$, and $\cos \beta = \sin \alpha$, therefore

$$\frac{N}{n} = \frac{d}{D} \tan \alpha \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The above relations are true for right- or left-handed spirals, and hold good for any two wheels in mesh, whether axes are at right angles or not. The only difference will be in the relation between α and β.

From equation (2) the required spiral angles can be obtained for a given velocity ratio and given diameters.

EXAMPLE.—The camshaft for a 4-stroke cycle gas-engine is to be driven by a right-angled spiral drive. Determine the spiral angles of the two gears.

The gear diameters are usually equal in cases of this sort. The speed reduction is 2 to 1, the camshaft running at half engine speed.

$$\frac{N}{n} = 2 = \frac{d}{D} \tan \alpha. \quad \frac{d}{D} = 1,$$

therefore,

$$\tan \alpha = 2.$$

Hence,

$$\alpha = 63° 26', \text{ and } \beta = 26° 34'.$$

When the spiral angles have been calculated, the actual diameters must be obtained, so that with a standard normal pitch an integral number of teeth is given.

In the case of single helical drives the provision of some means to take up end-thrust is essential. Ball thrust bearings are commonly used, while for light duty a plain collar on the shaft, bearing against a bronze bush or end-pad is satisfactory.

Let

H.P. = horse-power transmitted by a pair of gears; V = pitch line velocity in ft. per min.; P = tangential load on the teeth; $\text{H.P.} = \frac{\text{PV}}{33{,}000}$; $\text{P} = \frac{33{,}000\ \text{H.P.}}{\text{V}}$

If P_a = axial thrust, then $\text{P}_a = \text{P} \tan \theta$.

'Tooth face width.'—This should not be less than about $2\frac{1}{2}$ to 3 times the normal circular pitch, to ensure proper continuity of action. The normal tooth profile is involute.

The cutter for milling spiral gears is not selected according to the number of teeth as is the case with spur gears. The number of teeth in the spiral gear is divided by the cosine of the spiral angle cubed, i.e. $\frac{\text{N}}{\cos^3 \theta}$. When this calculation has been made, the resulting number is the number of teeth for which the cutter is selected.

Addendum $= \frac{1}{n.d.p.}$; Whole depth of tooth $= \frac{2 \cdot 157}{n.d.p.}$; Normal tooth thickness at pitch line $= \frac{1 \cdot 571}{n.d.p.}$.

Double Helical Gears, or Herring-bone Gears.

When accurately designed and well made this type of gearing has many advantages compared with other forms of gearing. The chief advantages are:—

(*a*) The engagement of the teeth is continuous when proper face width is used, and hence greater loads can be transmitted with high degree of efficiency.

(*b*) Vibration is practically eliminated.

(*c*) Shock is eliminated as the teeth are in continuous engagement, hence the load is transferred from tooth to tooth smoothly. Wear is thereby reduced, and also noise.

Tooth Profiles.

The normal tooth profile used is the involute. Pressure angles of 20° are used by most manufacturers. In some cases the $14\frac{1}{2}$° angle has been adopted, while for special cases other angles have been employed.

Spiral angles.—In the case of staggered teeth the angle adopted is usually 23°. For fine pitched turbine gears 30° is common, while for winding machinery, haulage purposes, etc., 45° is often used.

Face Width.

The total face width should not be less than 5 times the normal circular pitch. High ratio gears have face widths up to 12 times the normal circular pitch.

Tooth Proportions.

There does not appear to be any standard tooth proportions. The following are, however, values conforming to average practice:—

Staggered teeth, 23° pressure angle, addendum $= \frac{\cdot 8}{dp}$; dedendum $= \frac{\cdot 96}{dp}$.

Continuous teeth, 45° angle. Addendum $= \frac{\cdot 7}{dp}$; dedendum $= \frac{\cdot 85}{dp}$.

Strength of Double Helical Gearing.

Proportions of Teeth.

Thickness of tooth $= \frac{cp}{2}$; Addendum $= \frac{\cdot 89 \times cp}{\pi}$; Dedendum $= \frac{1 \cdot 05 \times cp}{\pi}$.

cp = circular pitch.

In the case of helical gears with continuous teeth, the formulæ vary according to the kind of machinery for which the gears are intended, and they are very numerous.

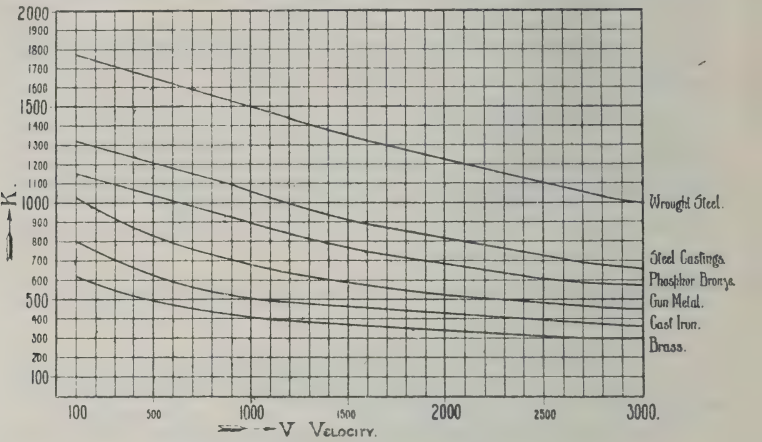


FIG. 15.

MOUNTING OF DOUBLE HELICAL GEARS.

It is of first importance that the axes of rotation of a double helical gear reduction should be *parallel and in the same plane.* Uneven load distribution will result if this is not so. It is important that the centre distance be such that the requisite amount of back-lash is obtained. Back-lash must not be excessive, but must be sufficient to ensure proper lubrication. In special cases, such as in gun-sights, back-lash is undesirable. Excessive back-lash is liable to create hammer-blow. Back-lash may be from 5 to 10 thousandths of an inch per inch of circular pitch. The tooth thickness of both gears is reduced by half of the amount given. In high-speed gears lubricated by spraying, the gears having fine pitches, greater back-lash is often desirable.

In order to ensure that there shall be no end thrust, it is necessary for the gears to take up a position of equilibrium, and hence flexible couplings between shafts should be used.

LUBRICATION.

Low-speed gears are often lubricated by allowing the wheel to dip in an oil bath. Violent churning of the oil should be avoided, and hence this form of lubrication must not be adopted for high-speed gears. For this latter purpose sprayers are used, the oil being supplied by a pump.

Triple Helical Gears.

A gear having two changes in the hand of the helices is called a triple helical gear. Such gears are sometimes used in mining machinery. The idea of the triple helical gear apparently evolved from the notion that helical gears gave best results when the apex of the teeth pointed in the direction of rotation, and hence the triple gear provided the best form of gear for reversing. Such a contention does not, however, appear to have much to support it in the light of modern gear technology. They are, however, still in some demand, and engineers occasionally specify them.

TURBINE GEARS.

The fact that large horse-powers are transmitted at high pitch-line velocities calls for special consideration in manufacture. Pitch-line velocities up to 18,000 ft. per minute are dealt with. Speed reductions from turbine shafts to electric generator shafts, mill shafts, and ships' propeller shafts need specially designed and accurately made gearing. Similar kind of gearing is needed to increase speed from internal combustion engines, electric motors, etc., required to work blowers, compressors, rotary pumps, and the like.

The arrangement of a gear reduction may be considered from three points of view :—(*a*) a single or double reduction. The limit for single reduction is 1 to 20, and in some exceptional cases 1 : 26, whilst 1 : 50 ratio has been obtained by double reduction; (*b*) turbines may be in 1 to 3 casings; (*c*) by the use of double helical (herring-bone) gears the axial thrust on the shaft is balanced automatically. Plain helical gears, if used, may be arranged to operate so that the axial thrust due to propeller or turbine rotor is balanced. The double helical gear is the more usual.

Turbine gears of the double helical type have been produced having efficiencies of 98 per cent. and 99 per cent., whilst it is common to find efficiencies of 94 per cent. to 96 per cent.

Wear on Teeth.—When running at high efficiency the wear is small—in fact, helical gears running with high-speed machines, such as the De Laval turbine, have been known to show little or no wear after several years of service. Such results can only be expected when materials and manufacture are of the best, as well as with careful use.

Uneven stresses setting up pitting of gear teeth and large amount of wear may occur on marine installations, due to rough weather causing straining in the larger gear wheels.

Turbine Gear Arrangements.

Figs. 16 to 23 show diagrammatically different turbine gear arrangements as follows :—

Fig. 16.—Single casing turbine with single reduction gear.
Fig. 17.—Twin turbine casings, single reduction.
Fig. 18.—Three turbine casings, single reduction.
Fig. 19.—Four turbine casings in groups.

To allow the condenser to be placed below turbine the L.P. casings are arranged so that the pinions mesh near the top of the wheel, as in figs. 22 and 23.

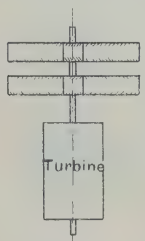


FIG. 16.

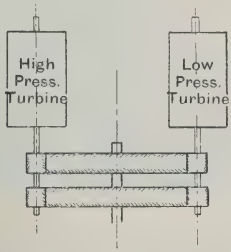


FIG. 17.

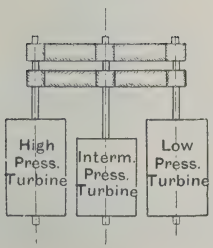


FIG. 18.

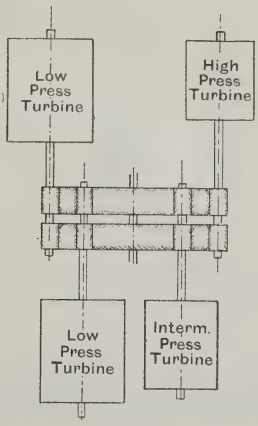


FIG. 19.

Fig. 20 shows a twin turbine double reduction.

Fig. 21 shows an arrangement for securing a more symmetrical stressing of pinions and reduction in twist than with the previous arrangement. The intermediate gears are arranged in the centre of the divided main gears.

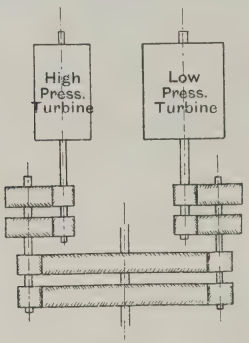


FIG. 20.

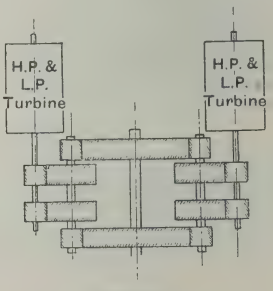


FIG. 21.

Fig. 22 shows single helical gear reduction by Messrs. Brown, Boveri et Cie.*

Fig. 23 shows Messrs. Parsons' gear reduction arrangement.

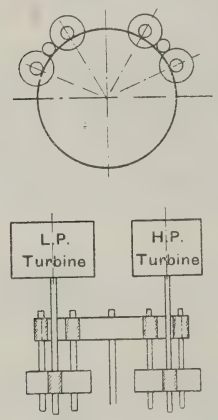


FIG. 22.

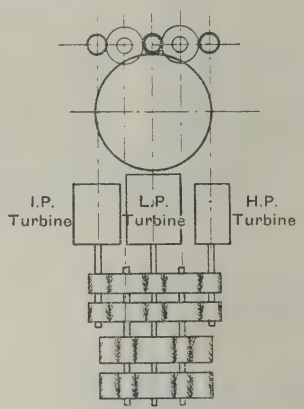


FIG. 23.

One of the factors to be taken into account when deciding upon the tooth pressure is the danger of the oil film failing if this pressure is too high. According to H. M. Martin (*Engineering*, August 11, 1916), the stress on the oil film obeys some such law as $p = bdn$, where

* Nozzles 0·10 in. to 0·15 in. diameter are employed, spaced 1½ ins. to 4 ins. apart. Oil pressure is governed so that the rate of flow is about ·33 to ·67 gals. per minute per inch width of tooth when peripheral speed is 33 ft. per second. When peripheral speed reaches 130 ft. per second, 1·0 to 1·32 gals. per minute per inch width are used. Nozzles having too small a bore should not be used, since clogging is liable to occur. It is a further advantage to have the oil reservoir above the level of the gear wheels. Too high a jet velocity should not be used as atomisation of the oil is likely.

p = load on tooth per inch run in lbs.; d = pitch diameter of pinions in inches; and b = constant; n is given as $\frac{1}{4}$, and b is taken by Messrs. Parsons as 175, but values of 250 have been used.

In most of the examples given the pinions are located above the gear wheels with which they engage. The pinions are divided, each pinion or wheel forming one-half of the double helical gear.

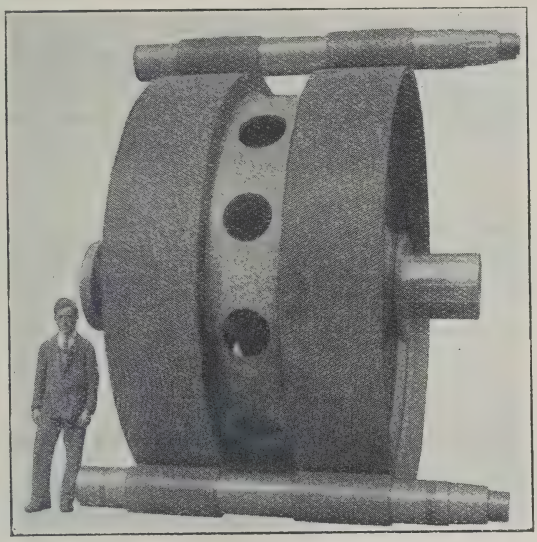


FIG. 24.

An example of a modern single reduction turbine gear made by the Power Plant Co. of West Drayton is shown in fig. 24. Each wheel transmits 35,000 h.p., the speed reduction being 2,880/288 r.p.m.

The figure shows gears of the type which have been fitted to the French cruisers *Duquesne* and *Tourville*.

Turbine Gear-Tooth Design.

An important factor in tooth design is the limiting pressure which can be allowed on the teeth. Points to be considered in fixing the pressure are wear, type of lubrication, and peripheral speed.

Gears for cargo steamers limiting pressure per inch width, 350 to 450 lbs., calculated on the length projected on to the shaft. In naval turbines pressures up to 850 lbs. per in. are used.

Double reduction gears tooth pressure is limited on the first pinion on account of high peripheral speed. About 450 lbs. is allowed on the first pinion, and 650 lbs. to 850 lbs. on the second.*

Considering the bending stress on the teeth due to normal tooth pressure, and assuming it to act at the tooth tips, 5,700 lbs./sq. in. to 7,100 lbs./sq. in. for cargo steamers, and 14,200 lbs./sq. in. for naval vessels, are common values.

The height of the teeth is usually about 0·6 times the circular pitch. The teeth are inclined usually at 30°.

Pitches vary. For small gears, values of $\cdot 12\pi$ to $\cdot 16\pi$ are used, while in the slower running gears, pitches of $\cdot 32\pi$ are used.

---

* Landahn has used tooth pressures up to 1,450 lbs. per in. width on gears running at 164 ft. per second and transmitting 10,000 h.p.—*Techn. Rundschau d. Berl. Tageblatt*, May 1919.

Thickness of tooth at pitch line is usually about $\frac{1}{2}$ circular pitch. Tooth clearance to allow for heating of gears and to ensure proper lubrication is usually ·01 in. to ·02 in. The centre distance is usually increased by this amount to provide clearance.

In order to secure best results the highest degree of accuracy is needed in the manufacture of turbine gears, together with accurate lining up and careful attention while at work.

The teeth are usually cut by special relief ground hobs. The end-mill process is an older method, but is still used where hobbing process is inapplicable.

For turbine gears a 20° pressure angle is used, the normal tooth profile being involute.

The fact that several teeth are in mesh, or partly so, is not taken into account, the whole load being assumed to be carried by one tooth.

The actual length of tooth carrying the load depends upon the deformation, due to bending and twisting. The amount allowed is limited to about ·0008 in. or ·0012 in.

Pinions.

Pinions are usually forged solid with shaft, and 3 to 5 per cent. nickel steel alloy is used; also nickel-chrome alloy in some cases. The number of teeth is not often less than 30 or 35. For cargo steamers and the first reduction gear, pitch-line velocities of 80 ft. per second are employed. For second reduction gear, 28 to 40 ft. per second are used, while naval gears run at 150 ft. per second. Greater pitch-line velocities are sometimes used.

$$\text{Tooth length} = \frac{\text{tangential load on teeth}}{\text{bearing pressure per inch width}}.$$

Wheel Construction.

The wheel body is usually cast iron, having a forged steel rim shrunk on. Some makers have used cast steel entirely.

The Power Plant Co., Ltd., of West Drayton use, for the pinions, alloyed steels (nickel or chrome-nickel steel), and for the wheels above velocities of 3,000 ft. per min., forged steel rims.

The calculation must start from the requisite pinion diameter to transmit a given load without exceeding the very narrow limits within which bending and twisting are allowed.

With gears where the pinion active face exceeds three times the pitch diameter, three bearings are invariably required for the pinion, to resist bending. The slightest inaccuracies in pitching and setting of the wheels, and comparatively small bending and twisting strains are liable to affect turbine gears very adversely, but these effects can be counteracted by special designs, such as the wheels with the pressure compensating device as manufactured by the above company.

Lubrication.

In the case of gears running at high pitch-line velocities, special precautions are necessary to ensure ample oil supply between the teeth. The oil is projected on the approaching side of the gears on to a point just before meshing occurs.

Pitch.

Fine pitches give silent running at high speeds, and a normal pitch, that is, the pitch measured at right angles to the teeth, of 0·583 in. (or $\frac{7}{12}$ in.) has been largely used in this country. For small pinions, or where extreme silence is of importance, 0·4 in. in pitch has been used. For turbine gears the pitch for a pinion having at least thirty teeth may be found from the expression:

$$p = \frac{P}{WK},$$

where,

p = pitch in inches; P = total tooth load in lbs.; W = width in inches; K = constant corresponding to the velocity. (See fig. 25.)

The expression applies to teeth with normal addendum and dedendum.

The materials in turbine gearing are never highly stressed; gears are designed primarily for wear, and this gives a much greater factor of safety than would be employed were they to be designed for considerations of strength alone.

An ample supply of oil is imperative, not only to lubricate the teeth, but also to carry off the heat generated. For this purpose it is usual to have ¼ in. to $\frac{3}{16}$ in. nozzles of about 5 ins. pitch squirting oil under a pressure of 10 to 20 lbs. per sq. in. on to the pinion, some distance in

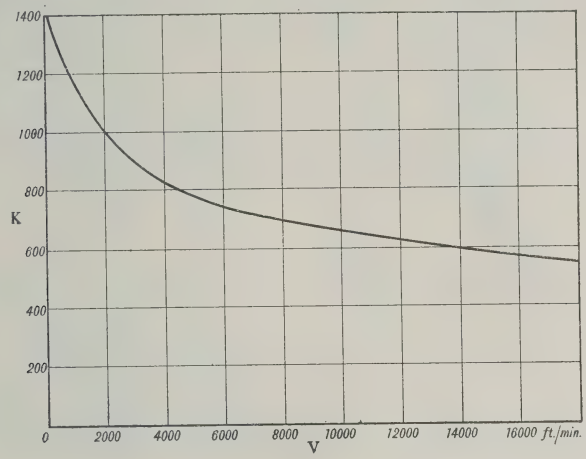


FIG. 25.

advance of the engagement, as this allows some of the oil to disperse and it is not carried forward and pumped out by the gears themselves. The quantity of oil is about 1 gal. per minute per 100 to 150 h.p. transmitted. The case must be well drained, otherwise if the wheel dips into the oil at the bottom much heat will be generated by the churning of the oil.

PINION DISTORTION.*

The distortion of the pinion is made up of two items—the twist due to the torque transmitted and the bending due to the load on the teeth. The load on the tooth is assumed to be uniform, and the effective diameter of the pinion is the pitch diameter.

At any point x in fig. 26, the torque T is :—

$$\frac{pxd}{2} = \frac{2fI}{d} = \frac{\pi f d^3}{16}$$

where f is the shear stress at the circumference of the pinion, and I is the polar moment of inertia or $\frac{\pi d^4}{32}$. Hence it follows that $f = \frac{8px}{\pi d^3}$. If N is the modulus of rigidity, which for steel is 12,000,000 lbs. per sq. in. :—

$$\frac{dy}{dx} = \frac{f}{N} = \frac{8px}{\pi N d^3}$$

$$y = \frac{4p}{\pi N} \cdot \frac{x^2}{d^3}$$

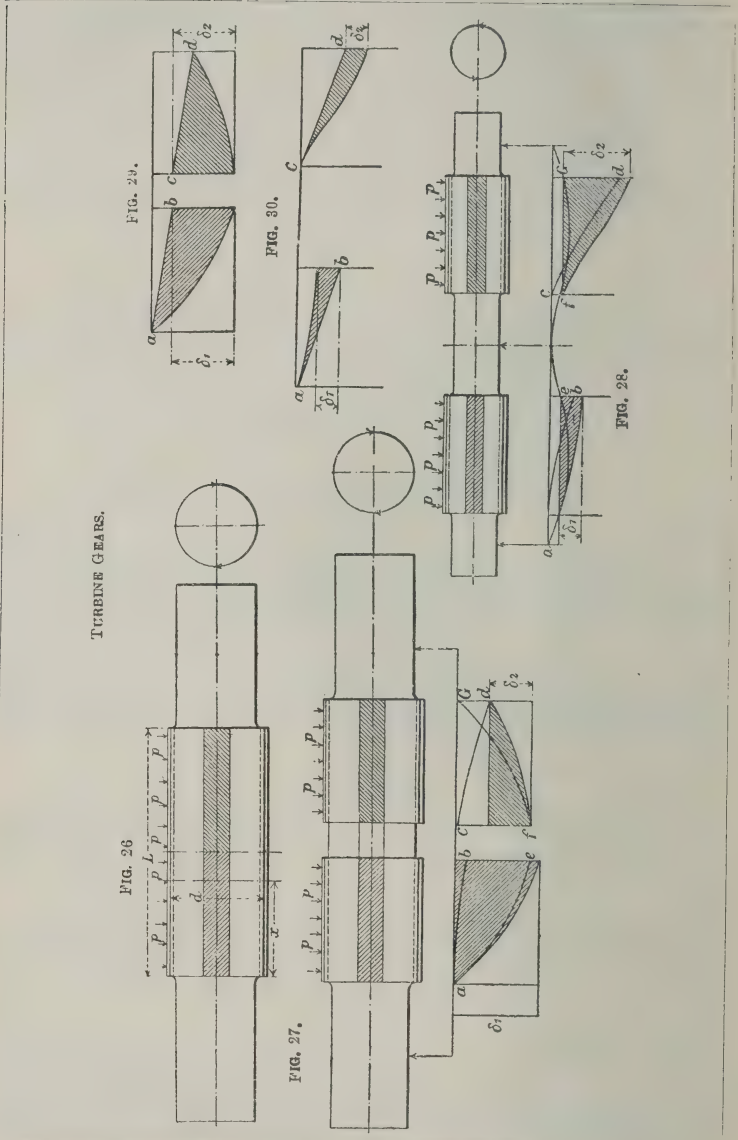
which is a parabola. The twist in the first half of the pinion is :—

$$y_1 = \frac{p}{\pi N} \cdot \frac{l^2}{d^2}$$

and in the second half :—

$$y_2 = \frac{3p}{\pi N} \cdot \frac{l^2}{d^2}$$

* Prof. Gerald Stoney, F.R.S., D.Sc., ' *Trans. Manchester Assoc. of Engineers*,' 1919.


TURBINE GEARS.
FIG. 26
FIG. 27.
FIG. 28.
FIG. 29.
FIG. 30.

There are two cases to consider: the case where there is no centre bearing and the pinion is supported from the ends, as in fig. 27, and the case where there is a centre bearing, as in fig. 28.

The deflection of the pinion can be calculated in the usual way, and is shown by the curves *ae* and *fg*. To this must be added the twist, as shown by the curves *ab* and *cd*. Adding these together, the total distortion is shown by the shaded area, the maximum distortions in each half of the pinion being δ_1 and δ_2. The gap shown in the middle of the pinion is to enable the hob to clear the teeth in each half.

If, in the case of a pinion with a centre bearing, the pinion took up the position *abcd*, shown in fig. 30, the effect of the distortion of the pinion is reduced.

In the case of a pinion without a centre bearing, the effect is only small, as shown in fig. 29.

In the case of a pinion without a centre bearing the bending is the chief consideration, whereas in the case of a pinion with a centre bearing the twist is the chief thing.

WORM GEARING.

There are two types of worm gear—namely, one in which the worm is parallel throughout its whole length and the other in which the worm is curved along its face to suit the curvature of the worm wheel. The former is known as the ' parallel ' worm gear, and the latter as the ' Hindley ' or ' globoidal ' worm gear.

The introduction of machine-cut worm gear of high precision and the increased knowledge of the nature of tooth contacts has resulted in the use of worm gear for the transmission of large horse-powers, with very high efficiencies.

Formerly it was considered correct to make the threads of all worm gear straight-sided on a linear section through the axis of the worm, and to take the pitch line midway down the working depth of the threads; thus, for the parallel gear this section showed the threads to be a pure rack, and for the globoidal gear a similar rack formation but with the threads pointing radially to the

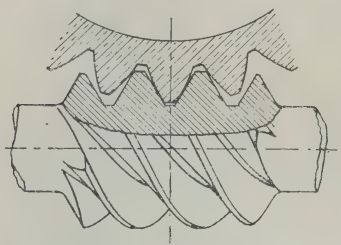


FIG. 31.—Parallel Worm Gear.

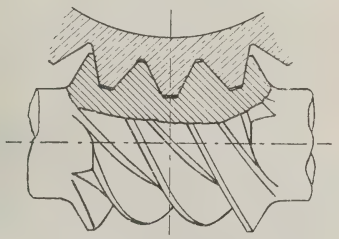


FIG. 32.—Hindley Worm Gear.

centre of the worm wheels. Actually these two types are very closely related because a parallel worm can be regarded as being a globoidal worm which has been made to suit a worm wheel of infinite diameter.

The construction of the threads of the parallel worm to correspond to that of a rack gives involute profiles to the worm-wheel teeth on the central section, as shown in fig. 31, and with the Hindley gear, shown in fig. 32, a shape corresponding to the threads of the worm, consequently giving teeth in simultaneous contact. It would therefore appear as if these two types were quite correct in every way: involute tooth engagement for the parallel worm, and a complete tooth engagement for the Hindley worm. Actually these conditions only apply on the central plane. Neither is the involute tooth profile nor the attractiveness of the Hindley type maintained; their value is entirely lost on all other planes owing to the curvature of the face of the worm wheel. Whenever straight-sided threads are used for worms, whether for the parallel or the Hindley type, the teeth on other planes than the central one are not, and cannot be, wholly conjugate to the threads of the worms, because the hob during cutting unavoidably removes too much metal from the wheel teeth. The result is that in actual practice neither of these worms transmits uniform velocities to the worm wheels, and instead of the contact being mainly on the leaving side it is almost wholly on the entering side, thus giving a higher coefficient of friction and a lowering of the overhaul efficiency of the gears. Worm gear, unless correctly designed, does not transmit a uniform velocity from one member to another. A close approximation is obtained with gears of comparatively fine pitch, and in those of a high lead angle any error is

sometimes concealed by the adjacent thread being in action in such a manner that by sustaining an extra proportion of the load distribution the variable velocity period is tided over. Even the inertia of the parts may prevent a change in the velocity of the engaging pair, but this is detrimental to the gears. To consider a worm gear as a kind of a development from the screw is fundamentally wrong. The pitch surfaces of the worm gear are cylinders lying at right angles to each other, hence they only touch at one point, whereas in a screw and nut the pitch surfaces coincide. Expressed in simple language, one can say that, consistent with the transmission of uniform velocity, the normal to the surfaces from the point of contact must always cut the central plane which contains the axis of the wheel, at a point on the straight line lying on the wheel pitch cylinder, and consequently tangential to the pitch cylinder of the worm. Thus, for every point of intersection between a worm thread and the zone of contact, there is a definite relation between its angular and longitudinal positions. If contact takes place at any other point than this intersection, it does so to the detriment of the transmission of uniform velocity.

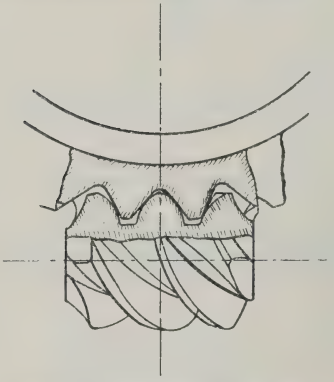


FIG. 33.—' D.B.S. Patent Worm Gear.'

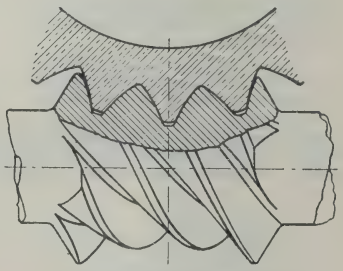


FIG. 34.—Enveloping Worm Gear.

The mathematical investigations carried out by Mr. F. J. Bostock have shown that the usual rack formation of the threads of the parallel and the globoidal worms is entirely wrong. Fig. 33 shows the correct form of worm threads for the parallel worm, and is known as the ' D.B.S. Patent Worm Gear,' manufactured by Messrs. David Brown & Sons (Huddersfield), Ltd. In this gear, instead of the threads being straight-sided on a section containing the axis of the worm, they are straight-sided on a section taken at a predetermined distance away from the axis, and the pitch line, instead of being half-way down the working depth, is taken at the bottom of the working depth or at the throat diameter of the worm wheel. This gear has been largely used for the propulsion of buses and lorries in this country and America, and upon test by the N.P.L. gave an efficiency of 97·3 per cent.

The corresponding correct form of threads for the globoidal type is shown in fig. 34, which was also the result of Mr. Bostock's investigations, and is manufactured by Bostock & Bramley, Leeds.

Although the ordinary involute type of parallel worm gear has certain inherent defects, it is still largely used for drives in which the maximum load-carrying capacity and efficiency are of little importance.

Definitions.

' *Gear Ratio*' :—

$$\frac{\text{No. of teeth in wheel}}{\text{No. of starts on worm}}$$

If worm is single threaded, no. of starts is 1; if double threaded, no. of starts 2, and so on.

' *Worm-pitch cylinder and worm-wheel pitch circle*' (fig. 35).—The worm-pitch cylinder is a virtual cylinder which makes contact with the worm-wheel pitch circle at a point lying on a line through the axis of the worm wheel and perpendicular to the worm axis.

' *Centre distance* ' is the distance between the axis of the worm and the centre of the worm wheel.

'*Axial pitch of worm threads*.'—The distance between the centre lines of two adjacent threads taken on an axial section.

'*Normal circular pitch of worm threads*' is the distance between the centres of adjacent threads measured round the pitch cylinder and normal to the helix.

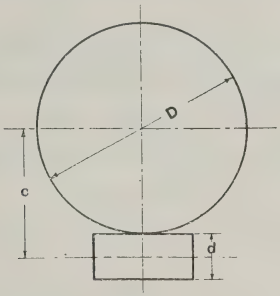


FIG. 35.

'*Worm lead angle*' is the slope of the helix, and is the angle between the helix and a plane perpendicular to the worm axis (fig. 36); lead angle = α.

'*Worm spiral angle*' is the complement of the lead angle = θ (fig. 36).

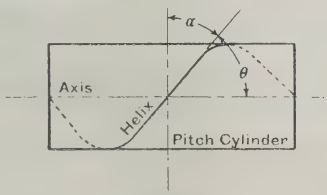


FIG. 36.

'*Axial pressure angle*' is the angle of inclination of worm profile at any point on an axial section to a perpendicular to the worm axis = ϕ (fig. 37).

'*Normal pressure angle*' is the angle between a tangent plane at any point on tooth profile, and a perpendicular to the axis at that point.

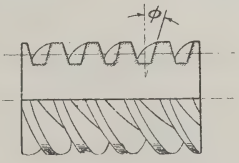


FIG. 37.

'*Lead of worm*' is the linear distance of advancement parallel to the axis of each thread of the worm per revolution of the worm.

'*Circular pitch of wheel teeth*' is the distance between the centres of adjacent wheel teeth measured round the pitch circle.

'*Clearance*' at worm wheel tooth base as for spur gears.

'*Worm thread addendum.*'—Height of thread above pitch cylinder.

'*Worm thread dedendum.*'—Depth of thread below pitch cylinder.

'*Worm crest diameter.*'—The overall dia. of the worm threads.

'*Worm root diameter.*'—The dia. of the worm at the root of the threads.

'*Wheel throat diameter.*'—The diameter of wheel rim at the bottom of the curved tooth face.

'*Wheel tooth throat addendum.*' The distance between the pitch circle and top of teeth on a centre section normal to the wheel axis.

'*Wheel tooth overall addendum.*'—The addendum measured to overall dia. of rim.

'*Wheel teeth dedendum.*'—The distance between the pitch circle and the bottom of the tooth spaces measured radially and on a central plane.

'*Wheel overall diameter.*'—The greatest dia. of wheel rim.

'*Wheel root diameter.*'—The dia. at the base of wheel teeth.

Design.

In the common type of worm gear, in which the straight-sided thread on the axial section is used, the 14½° axial pressure angle is generally adopted. The cross-sectional dimensions of the worm thread are commonly made the same as the 14½° rack teeth—i.e. addendum $= \frac{1}{dp}$; dedendum $= \frac{1}{dp}$ + clearance; clearance = ·155 module. If the lead angle used is greater than 20°, then the proportions are based on the normal pitch. If a 20° lead angle is adopted, then 14½° pressure angle is too small to give satisfactory results. For automobile drives a 30° pressure angle is usually adopted.

If the worm is to be cut on a shaft direct, then the pitch dia. must be such as to give sufficient strength to make the shaft rigid, and strong enough to carry the torque with the requisite margin of safety. The root dia. must be such as to prevent serious deflection when carrying the load.

The width of face of the worm wheel should be as wide as is reasonably possible, compatible with general design of gear.

The throat should not envelop root of worm by more than about 1·5 times the module, otherwise the wheel teeth will be cut to a sharp edge.

Calculations.

To compute gear ratio, revs. of worm and wheel being given:—

R = revs. per min. of worm; r = revs. per min. of wheel;

$$\text{Gear ratio} = \frac{R}{r}.$$

To compute no. of wheel teeth N, when gear ratio is given and no. of worm starts, n,

$$N = \text{Gear ratio} \times n.$$

To compute worm pitch dia., given centre distance and worm lead angle, α.

$$d = \frac{2 \text{ centre distance}}{1 + [\text{gear ratio} \times \tan \alpha]}, \text{ approx.}$$

If centre distance and wheel pitch circle dia. is given,

$$d = 2 \text{ centre distance} - p.c.d. \text{ of wheel.}$$

To compute the circular pitch of wheel teeth (p), given $p.c.d.$ of wheel D and no. of teeth N,

$$p = \frac{\pi D}{N}.$$

To compute the pitch dia. of wheel when circular pitch and no. of teeth in wheel are given,

$$D = \frac{pN}{\pi}.$$

To compute the centre distance when the pitch dia. of worm is given and the pitch circle dia. of wheel,

$$C = \frac{D + d}{2}.$$

To compute the lead of a worm, given the pitch and number of starts of worm (n),

$$\text{Lead} = \text{pitch} \times n.$$

To compute lead of worm, given the pitch dia. of wheel and the gear ratio,

$$\text{Lead} = \frac{\pi D}{\text{Gear ratio}} = \frac{\pi D n}{N}.$$

To compute the worm lead angle, given the pitch dia. of worm and worm lead,

$$\tan \alpha = \frac{\text{Lead}}{\pi d}.$$

To compute normal circular pitch, given the pitch and lead angle α,

$$n.c.p. = p \cos \alpha.$$

To compute worm thread addendum and dedendum on wheel throat addendum, given the pitch of the wheel teeth,

$$\text{Add.} = \cdot 3183p\,; \quad \text{Ded.} = \cdot 3683p.$$

If α exceeds 20°, Add. $= \cdot 3183\ n.c.p.$; Ded. $= \cdot 3683\ n.c.p.$

To compute worm crest dia., given worm pitch dia. and addendum,

Worm crest dia. $= d + 2$ addendum; Worm root dia. $= d - 2$ dedendum.

To compute wheel throat dia., given pitch dia. of wheel and wheel tooth throat addendum,

Wheel throat dia. $= D + 2$ wheel tooth throat addendum.

To compute overall dia. of wheel, given pitch circle dia. of wheel and wheel tooth throat addendum,

Overall dia. of wheel $= D + 3$ wheel tooth throat addendum.

To compute wheel root dia., given pitch dia. of wheel and wheel tooth dedendum,

Wheel root dia. $= D - 2$ wheel tooth dedendum.

Worm-wheel peripheral velocity, $V_w = \cdot 262\ DN'$ ft. per min.

Worm peripheral velocity, $v_w = \cdot 262\ dn'$ ft. per min.

where $N' =$ no. of revs. per min. of worm wheel, and $n' =$ no. of revs. per min. of worm.

Rubbing velocity at pitch point,

$$V_R = \sqrt{V_w^2 + v_w^2} = \frac{v_w}{\cos \alpha}.$$

Let,

$T =$ torque on shaft transmitting H.P. horse-power at N revs. per min.,

Then,

$$T = \frac{63{,}025 \text{ H.P.}}{N} \text{ lb. ins.}$$

Also,

$$T = \frac{\pi d^3}{16} fs,$$

where $d =$ shaft dia. in ins.; $fs =$ safe shear stress for the material.

Neglecting bending and other stresses,

$$d = \sqrt[3]{\frac{16\,T}{\pi fs}}; \quad \text{H.P.} = \frac{2.\pi.N.T.}{33{,}000 \times 12}.$$

FORCES ACTING BETWEEN WORM AND WHEEL.

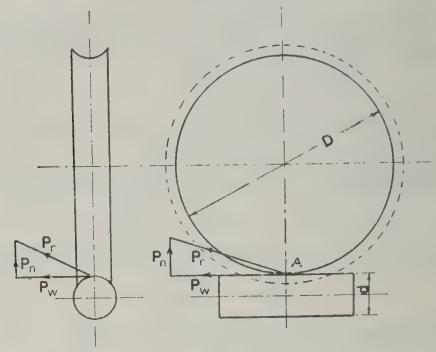


FIG. 38.

It is assumed that the resultant reaction between the worm and wheel passes through the pitch point A.

Resultant reaction = P_r; resultant force acting parallel to wheel axis = P_W; resultant force acting parallel to worm axis = P_w; force acting perpendicular to both axes = P_p; R = Gear ratio. T = torque due to H.P. in lb. ins.

Approximately,

$$P_w = \frac{2RT}{D} = \frac{2TN}{D_n}; \quad P_W = \frac{2T}{d}; \quad P_p = P_w \tan \phi.$$

The efficiency is approximately $= \dfrac{\tan \alpha}{\tan (\alpha + \lambda)}$,

where λ = angle of friction.

The maximum efficiency occurs when the lead angle is about 45°.

Worm and Worm-Wheel Manufacture

WORM.

A comparison of the parallel and globoidal worms will at once show that two distinct methods of production are necessary.

In the former case the worm 'blank' can be run past a rotating cutter whilst the 'blank' is traversed in a helical track. Finishing can be effected by substituting a grinding wheel for the cutter. In the latter case the 'blank' must be rotated at uniform speed whilst the cutter moves about a centre which must be accurately located relative to the 'blank,' and at the same time adjustable.

Worms are usually supplied solid with the shaft, a bearing being provided on each side of the worm and close to it.

To ensure perfect concentricity of shaft journal and worm it is common practice to grind both the journal and the outside of the worm blank before the actual cutting is made.

Special machines are in use for worm milling. The worm is made to move in a helical path before a rotating cutter. A thread space is milled for each longitudinal traverse of the table. The arbor is carried in a swivelling head, which enables it to be set at any angle to meet the spiral of the worm to be cut.

CUTTER FORM GENERATION.

The rotary cutter at first sight appears to be quite a straightforward matter, with its contour made to correspond with the space of the worm thread. Such, however, is far from being the case, as its actual contour is quite different from the thread space which it cuts in the worm blank, as

will be seen from fig. 39. This apparent paradox is caused by the interference of the cutter with the side of the worm thread, and it will be understood when it is remembered that the worm thread is an endless spiral. As a rotary cutter must be circular in shape, and as a circle and spiral cannot coincide, it is obvious that the contour of the circular cutter cannot coincide with the thread space of the spiral worm. The base of the worm thread system is a straight line. In the system of cutter generation a straight edge tool is set to an angle corresponding to the pressure

FIG. 39.—Worm and its Cutter.

angle of the worm to be cut, and is then made to traverse in a spiral path corresponding exactly with the spiral of the worm. Thus a generating tool is mounted in a worm milling machine in the same position and manner as the worm which is to be cut, its straight edge accurately representing one side of the worm thread. When it has generated one side of a white metal blank it is indexed to the other side, thus forming the contour of the required cutter.

WORM WHEEL.

The production of a worm wheel of accuracy is a far more difficult process than that of producing a worm.

Worm wheels are produced by the process of hobbing, the hob being similar in form to the worm. A parallel hob having teeth of constant shape throughout its length provides the best means of securing accuracy, which is essential to obtain the best results from the worm gear.

The hob is made by the same process as the worm, and is hardened and afterwards ground. The production of a hob for the globoidal worm wheel is very difficult, and generally the accuracy of the finished job is not of the highest order. Often for these hollow worm wheels the single cutter is used, but of course the result is only an approximation to the true tooth profile. The cost of the hollow type of worm gear is rendered more expensive by the complications arising from the required shapes, and accuracy is much less likely than with the parallel worm.

HOBS—MANUFACTURE AND SHARPENING.

The teeth of worm wheels are of such construction that they do not lend themselves to operations of planing, shaping, or milling like other forms of gearing: they can only be produced by a process of generation by means of a cutting tool called a 'hob.' Whilst it is true that the spiral angle pitch, the number of threads, and pitch diameter in the hob must correspond with that of the worm, yet both the outside and bottom diameters of the hob are made larger in order to provide requisite clearance in the roots of the teeth. The hob throughout is gashed along its length normal to the spiral angle for the purpose of providing a cutting face, whilst the teeth are relieved in order to provide a cutting clearance. As the shape and contour of the wheel teeth are entirely dependent upon the design and accuracy of the hob, it is necessary that this tool should receive the most careful supervision in all stages of its manufacture and subsequent regrinding. Hobs are generally made with their cutting faces radial; it is therefore essential that they should be sharpened radially, or an alteration in shape will result. The same, of course, applies to ordinary rotary cutters. It will be appreciated that as a hob is sharpened its diameter is reduced, but with a reamer hob the reduction in diameter is only a few thousandths of an inch in the whole length of the teeth, thus uniformity of shape is retained.

WORM WHEEL GENERATING.

Generally the worm wheels are rough turned on automatics and turret machines, and pass from thence to the worm-wheel section for rough cutting of the teeth, after which they are finished, turned, and finally the teeth undergo a reamering or finishing operation. In no small measure is the interchangeability and high efficiency of the wheels due to this reamering operation.

The generating of the teeth of the worm wheel is carried out in a machine as illustrated in fig. 40, the blank being mounted upon a revolving table, which is positively driven at a predetermined speed, according to the ratio of the gears being cut, whilst the hob is caused to travel longitudinally across the face of the wheel. Owing to the spiral of the hob it is necessary

to compensate for its longitudinal travel by means of a differential, whilst the rate of travel or feed can be varied at the will of the operator.

A special feature of the mounting of the worm wheel on the table is made. A hardened and ground micrometer adapter carries the arbor, which supports various other adapters to suit the locating register of the worm wheel blanks, the micrometer being adjusted until the specified distance from the locating face of the blank to the axis of the hob is obtained.

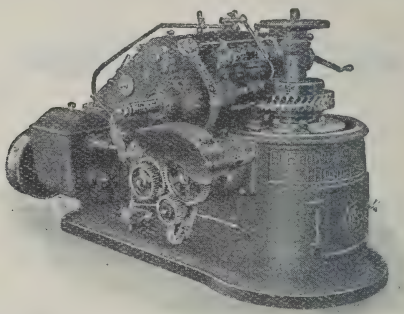


FIG. 40.—Worm Wheel Generator (*David Brown & Sons (Huddersfield), Ltd.*).

To facilitate setting up, and to test the true running of the registers and worm-wheel blank, the drive to the table is arranged so that it can be disconnected in order that the operator may be able to revolve the table by hand and so, by means of a clock indicator, test the accuracy of the mounting.

The head which slides horizontally along the bed towards the worm wheel contains the whole of the mechanism for driving the hob and traversing the hob slide. The latter is mounted on the head at an angle normal to the direction of the stress, so as to present the maximum bearing area to the very best advantage.

The main drive is taken from a single pulley fitted with an internal friction clutch in conjunction with a four-speed gear box. The drive is then transmitted in one case directly to the index change wheels for driving the table, and in the other to the worm wheel mounted on the hob spindle.

It should be mentioned that the dividing wheel is of greater diameter than the largest blank which the machine will cut, and a neat design is effected by allowing the cuttings to fall through the arms of the table into the bed of the machine, from whence the cutting lubricant is drained.

Worm and Wheel Inspection.

For testing the accuracy of the pressure angle a knife edge protractor is placed along the side of the thread and the angle read to plus or minus a few minutes. This is carried out in a worm gear testing machine, illustrated in fig. 41, which has been designed to enable the various testing operations to be performed on both worms and wheels. These can be tested separately as individual pieces, or in combination when mounted together at the specified centre distance. The bed consists of two slides at right angles, one carrying a pair of adjustable centres and the various testing appliances, whilst the other slide carries an adjustable arbor adapter, which is controlled by a micrometer screw, enabling it to be set at any required distance from the other slide, so giving correct centres for the worm and wheel under test. This machine is most conveniently operated, as it can be worked from either side, thus allowing the operator to be in the most advantageous position to carry out all the necessary tests. Other instruments are used on it for testing concentricity and lead.

The process of case-hardening worms is exactly the same as for other work, but although every precaution be taken some little distortion is bound to occur. They therefore require straightening so that their pitch diameter runs true; for this purpose a clock indicator on a testing machine is used in conjunction with the straightening press, and by means of a ball placed in the space this operation can be accurately and quickly performed.

Interchangeability of the worm wheel is ensured by the use of a double ball micrometer, which measures the comparative pitch diameters.

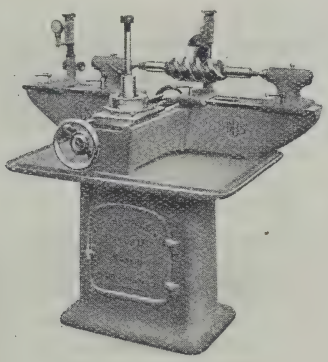


Fig. 41.—Testing Worm for Accuracy of Pressure Angle, Tooth Thickness, and Contour (D. B. S.).

The final inspection is made on a testing machine, where the gears undergo a more rigorous test. The wheels are tested for concentricity of cutting and the distance from the centre of the worm axis to the locating face, after which they are tested together to verify the centre distance.

Worm Gear Application.

Worm reducing gear units are with great advantage coupled direct to electric motors, enabling reductions in speed from 4 : 1 to 100 : 1 in a silent and efficient manner, and when both of them are mounted on the common baseplate they represent a simple, sound, and economical proposition.

On account of their ability to withstand considerable overload for short periods they lend themselves admirably for intermittent work, because for the work they are called upon to do they can be made relatively small; therefore they are of great use in crane work and for starting purposes of electric motors, steam, gas, and oil engines.

Horse-Power Formula.

$$H.P. = K \times A \times T \times r^{\frac{8}{3}} \times R^{\frac{8}{3}} \times S,$$

where,

K = constant; A = projected area of worm wheel in sq. ins.; T = teeth in contact; r = pitch radius of worm in ins.; R = revs. per minute of worm; S = function of lead angle.

For K, use:

Hardened	mild steel worm and	phosphor-bronze wheel	K = 0·14
Soft	" "	" "	K = 0·1
Hardened	" "	cast-iron wheel	K = 0·08
Soft	" "	" "	K = 0·07

For intermittent work multiply the above by 1·3.

For T, use:

Worm Threads	1	2	3	4	5	6	7	8
T	1·4	1·5	1·66	1·8	1·9	2	2·1	2·2

For S, use:

Lead Angle	5°	10°	15°	20°	25°	30°	35°	40°	45°
S	0·09	0·18	0·27	0·36	0·45	0·55	0·65	0·77	0·9

Location.

The parallel worm is not seriously affected by slight inaccuracy in the centre distance, and does not need special precaution with respect to endwise movement.

The globoidal form, however, needs accurate location in all directions in order to work correctly. Further, when dismantling the worm and shaft cannot be removed bodily as is the case with the parallel worm.

Finishing.—It is essential that worm threads have a very hard case, and must be ground to produce a highly polished surface. Hence the selection of suitable material is of prime importance. Generally worms machined from solid steel forgings give best results. Particular care is also required in the hardening processes in order to avoid, as far as possible, serious distortion.

EXAMPLE.—Obtain the dimensions of a worm for a gear ratio of 10 : 1 when the centre distance is 8 ins.

Assume number of worm threads = 5 = n, then no. of teeth in wheel = 50 = N. Assuming lead angle = 45°.

$$\text{Let } \frac{\text{pitch dia. of worm}}{\text{module}} = m. \quad \text{Then } \frac{n}{m} = \tan 45° = 1.$$

$$m = n = 5. \qquad \text{Also N} + m = \frac{2c}{\text{module}}.$$

$$\therefore \quad \text{module} = \frac{16}{55} = \cdot 29, \text{ for which a module of } \cdot 3 \text{ would be used.}$$

Pitch dia. of worm = $1\frac{1}{2}$ ins.; dia. of worm wheel = $6\frac{1}{2}$ ins.

REDUCTION GEAR.

H-R Gear.

(J. Stone & Co., Ltd., Deptford, London, S.E. 14.)

With this gear, using the worm gear as its base, it is possible to give any speed ratio in a very compact size, *i.e.* in the small space of 12 ins. × 12 ins. × 12 ins., any ratio up to 20,000 to 1 can be obtained, while for larger ratios compound gears take up little or, in some cases, no more space than for small ratios.

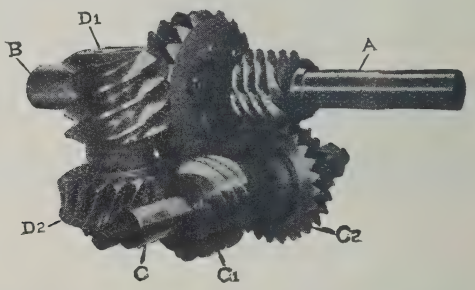


FIG. 42.

The arrangement adopted in a standard type of reduction gear is shown by fig. 42. The driving shaft A transmits power from the prime mover through a worm mounted on the shaft to a worm wheel C, arranged on shaft C. The shaft C is extended to carry a worm C_1, and the power is transmitted through this worm to a worm wheel D_2, mounted on shaft D transverse to the shafts A and C. This shaft in turn carries a worm D_1, which transmits the power through a worm wheel arranged on the driven or slow speed shaft B, so that the essence of the invention is the placing of gear shafts around the periphery of the driving worm and the driven worm wheel, thus permitting the first and final shafts to be arranged co-axially.

The efficiency of the gear is 70 to 90 per cent. in gears having ratios up to 250 to 1. This efficiency is maintained throughout the life of the gear. As the gear is always running in an oil-bath and its design is such that low tooth pressure and efficient helix angles are obtained, it is durable and silent in operation.

There are over 300 standard H-R Gear ratios, from 10 to 1 to 50,000 to 1, to select from.

In addition to secondary motion shaft, intermediate drives, both vertical or horizontal plane, may be incorporated in any standard box.

Ratios from 10 to 1 up to 50,000 to 1 can be obtained in standard gear-boxes. The size of a particular gear-box for a given ratio or load-carrying capacity is very compact and is claimed to be smaller than any other type of gear.

Since each gear ratio represents only the cube of the actual ratio required, only a small ratio on each train of gears is necessary, consequently exceedingly efficient worm lead angles are obtainable; thus with a gear ratio 64 to 1, each train of gears requires a ratio of 4 to 1 and a lead angle of 45°, each gear train having an efficiency of 95 per cent., so that the over-all efficiency would be not less than 85 per cent. With the ordinary type of worm gear having a single worm and wheel, for such a gear ratio only a single start worm could be employed having a lead angle of, say, 7°, which would give an efficiency of not more than 65 per cent. Even if very fine pitch and a 2-start worm were employed, it would be impossible to obtain an efficiency exceeding 75 per cent.

An exceedingly important feature of the H-R Gear, particularly where employed with large ratios and where space restrictions come in, is its load-carrying capacity. The maximum permissible loading on any worm-gear tooth depends entirely upon the rubbing velocity, and it will be seen that in the H-R Gear the rubbing velocity of the final worm and worm wheel is broken down very considerably through the first two trains of gears. In addition, because of the comparatively small ratios on each train, the minimum number of teeth and maximum pitch can be employed. On the very large ratio gears it is possible to load the gear teeth almost up to the yield point of the material of the worm wheel.

The H-R Gear is claimed to be the only absolutely silent gear which provides for a co-axial drive; this, in many cases, is a most important feature.

The gear has been applied to every kind of machinery.

H.-R. Balance Drive Gear.

(J. Stone & Co., Ltd., Deptford, London, S.E. 14.)

Although not providing the highest speed ratios obtainable with the H.-R. Standard Gear, this gear possesses several important additional advantages. The final gear-wheel is driven by two distinct elements which by a special patented arrangement are always in driving contact. By such an arrangement the gear is capable of transmitting double the torque of the ordinary gear. Further, the arrangement secures elimination of the radial load on the final shaft.

The worm-wheel A (fig. 43) is driven by two worms, B1, B2. These latter have connected at

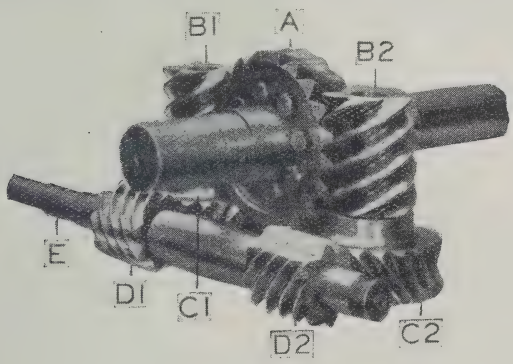


Fig. 43.

their lower extremities worm-wheels C1, C2. C1 and C2 are in engagement with worms D1 and D2 mounted on high-speed shaft E. Since worms B1 and B2 are in engagement on opposite sides of A they rotate in opposite directions, and must therefore have opposing lead angles. This also applies to worm-wheels C1 and C2. Similarly, worms D1 and D2 have opposite leads and oppose one another. The thrusts are therefore balanced, and eliminate the necessity for special precautions in this respect.

All the advantages are secured as with the H.-R. standard gear as regards silence, efficiency, etc.

Max. speed ratio with the balanced gear 250 to 1
Min. „ „ „ „ „ 5 to 1

The gear is specially applicable to heavy duties where shock is present, such as for rolling-mills, main drives for stern wheel paddle-boats, conveyors, etc.

VARIABLE-SPEED GEAR.

The 'VSG' Mark III. Hydraulic Variable-Speed Gear.

(*The Variable Speed Gear Ltd.*, 6 *Broadway, Westminster, London, S.W.* 1.)

This gear consists of two units, viz. an 'A' end or pump unit (fig. 44), and a 'B' end or hydraulic motor (fig. 45). These 'A' and 'B' ends can be opposed to one another directly with a common connecting plate termed a 'valveplate,' forming a type 'C' gear, or they can be separated to any reasonable distance and connected by piping, each unit having a separate valveplate, forming a type 'K' gear.

The flexibility of arrangement possible with the 'VSG' gears is a valuable feature, particularly so where considerations of space enter into account. Oil, which is practically incompressible, is the medium of transmission of power, and as this is supplied under pressure to the most important working surfaces and the whole of the machine casing is filled with oil, lubrication is perfect and wear negligible.

In certain instances machines are arranged to run with empty cases, and when this is so, special arrangements are made for the lubrication of the working parts.

Construction.

The 'B' end or hydraulic motor is generally similar in construction to the 'A' end or pump unit, which is fully described in the 'Pump' section, p. 1274, a number of its components being identical with those of the 'A' end. For the majority of purposes no controls or relief valves are necessary in the 'B' end, and the tilting box is usually fixed, giving the 'B' end a fixed capacity.

Where cooling is desirable, a water cooling coil is fitted in the 'B' end around the cylinder barrel.

Whilst in some cases 'Michell' type bearings are fitted to take the axial and radial thrust behind the socket ring in the 'B' end, in certain equipments 'VSG' 'B' ends are fitted with roller bearings.

Operation of the Gear.

When a 'VSG' gear is ready for running, the entire space within the cases and valveplate of the type 'C,' and valveplates and piping in the type 'K,' is filled with oil.

A definite portion of the oil is enclosed within the 'cylinders' ahead of the pistons, valveplate port passages, and piping of type 'K.' This is the working oil used in transmitting energy, the case oil not being subject to pressure.

Assuming the tilting box with its socket ring is set at the neutral position (*i.e.* perpendicular to the mainshaft), the 'A' end mainshaft, if rotated, will carry with it the socket ring and the cylinder barrel, together with the pistons and connecting rods. The pistons will not reciprocate or move to and fro in the cylinders, and as there will be no delivery from the pump, the 'B' end will remain stationary.

If the control be moved a little, so as to move the top of the tilting box away from the 'valveplate,' and the 'A' end mainshaft is rotated in a clockwise direction (looking on its shaft end) all the pistons as they move down on the right-hand side will be driven in towards the valveplate, expelling the oil contained in the cylinders through the cylinder ports and the large port in the same side of the valveplate. The 'B' end, as is explained later, will then commence to rotate at a speed corresponding to the amount of delivery given by the 'A' end.

The pistons on the left-hand side of the 'A' end will, as they ascend, be drawn away from the valveplate, and their cylinders will fill with oil from the valveplate port facing them.

It should be noted that when a piston reaches the top or highest position in its revolution, and also the lowest position, it for an instant makes no end movement. This corresponds with the cylinder port crossing the top or bottom lands or spaces between the two valveplate ports.

The quantity of oil forced through the valveplate port depends upon the length of stroke imparted to the pistons and consequently upon the angle at which the tilting box stands.

The 'B' end is an inversion of the 'A' end, and the 'B' end socket ring is carried normally at a fixed angle of about 20°, the top of the tilting box lying back from the valveplate, and when the 'B' end mainshaft is rotated the 'B' end pistons will make their full stroke as they pass between the bottom and top pistons.

The oil expelled by the 'A' end pistons through the pressure port of the valveplate enters the 'B' end cylinders facing that port and exerts pressure on the pistons in those cylinders.

Due to the socket ring being housed in the tilting box, the pistons cannot move except by rotating the socket ring in its bearings on the inclined plane, thereby converting the reciprocating motion of the pistons into rotary motion of the socket ring and the shaft to which it is connected.

The socket ring therefore revolves and carries with it the mainshaft to which it is connected by the universal joint. The 'B' mainshaft in turn rotates its cylinder barrel, and the whole of the 'B' end rotating group revolves in the opposite direction to the rotation of the 'A' end mainshaft.

The speed of rotation of the 'B' end mainshaft depends upon the quantity of oil delivered by the 'A' end. For example, if each cylinder has a capacity of, say, 3 cu. ins., one complete revolution of the 'B' end group of 11 cylinders would transfer 33 cu. ins. of oil from the right-hand side to the left-hand side. If the top of the 'A' end tilting box be tilted away from the valveplate

THE 'VSG' MARK III. HYDRAULIC VARIABLE-SPEED GEAR.

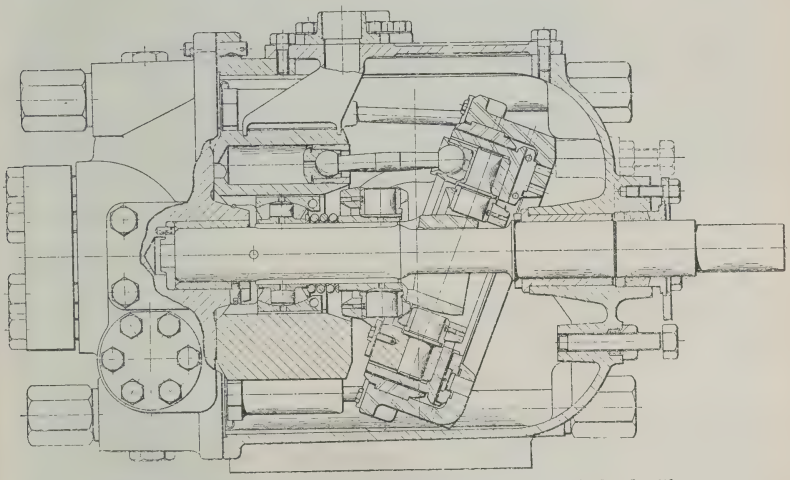


FIG. 44.—Vertical section through 'VSG' Mark III. 'A' end, fitted with lever control.

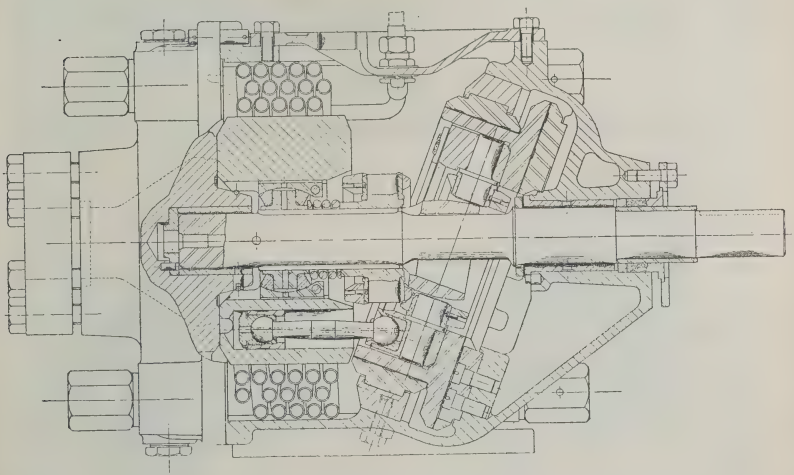


FIG. 45.—Vertical section through 'VSG' Mark III. 'B' end, fitted with Michell type bearings.

sufficiently to cause the socket ring to impart a small stroke to the ' A ' end pistons corresponding to an output from each cylinder of, say, 0·01 cu. in., all the 11 pistons of the ' A ' end will together transfer 0·11 cu. in. of oil from the left-hand to the right-hand side at each revolution of the mainshaft.

On the basis of the capacity of the ' B ' end cylinders per revolution being 33 cu. ins., 300 revs. of the ' A ' end mainshaft at this degree of tilt will be necessary to rotate the ' B ' end mainshaft once. If, however, the ' A ' end socket ring be tilted still farther, the output of the ' A ' end will be greater and the ' B ' end mainshaft will rotate proportionately faster, until at full stroke both ends are at the same speed.

So far reference has been made to tilting the ' A ' end socket ring in one direction only, namely, by moving the top of the tilting box away from the valve plate. If, however, it be moved in the opposite direction, the oil will be discharged through the left-hand port instead of through the right-hand port as previously ; this will cause the ' B ' end group to rotate in the opposite direction to that in which it has previously revolved, and it will now correspond in direction of rotation with the ' A ' end mainshaft—viz. clockwise. The gear is therefore fully reversible.

When the ' A ' and ' B ' ends are of equal capacity, the maximum speed of the ' B ' end mainshaft will be equal to that of the ' A ' end, forward or reverse as required.

For some purposes ' A ' and ' B ' ends of different capacity are combined, usually a large ' B ' end being employed, thereby giving a fixed reduction ratio as well as variable and reverse speeds.

Where a reduction or step-up gear is necessary on either the input or output shaft of the gear, this can, in many cases, be arranged within the gear casing.

Control.

Various types of control can be fitted, the purpose of all of which is to vary the angle of the ' A ' end tilting box, and thereby the delivery of the pump. The most usual forms of control are :—

(1) *Lever.*—A simple lever mounted directly on an extension of one of the tilting box trunnions.

(2) *Wheel or Rotary.*—Comprising a shaft carrying a pinion engaging a toothed sector fitted to the top of the tilting box. On this shaft is mounted a handwheel for direct manual operation, or sprocket, bevel, etc., for distant or mechanical control.

(3) *Automatic or Pressure Control.*—This consists of a hand or mechanically operable control combined with an automatic pressure operated device which regulates the speed of the gear automatically to suit the load and obviates the necessity of using skilled operators to safeguard equipments dealing with widely fluctuating loads.

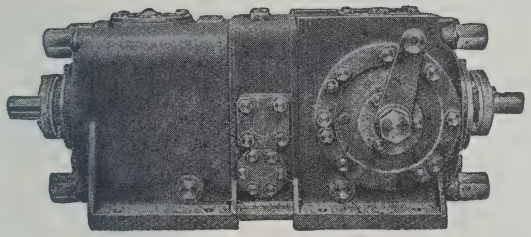
Advantages.

The special properties of the ' VSG ' Hydraulic Variable Speed Gear can be summarised as follows :—

(1) Speed change without declutching from prime mover.
(2) Ability to ' inch ' and rotate at very low speeds.
(3) High torque at low speed.
(4) Perfectly smooth acceleration and retardation at any constant or variable rate.
(5) Quick reversibility without shock.
(6) Complete positive and simple control in either direction.
(7) Steady speed under fluctuating loads.
(8) Enables constant speed non-reversing prime mover to be used.
(9) Protects complete installation from damage due to overloads.
(10) Reduces labour and operating costs.
(11) Repairs and maintenance negligible.

An important feature inherent in the ' VSG ' gear is the braking effect which can be obtained when it is used with a regenerative prime mover, enabling the retardation of masses having large momentum and unbalanced loads to be carried out under absolute control, *e.g.* in the case of a hoist, loads may be lowered at any desired speed, decelerated and held in all positions without the use of mechanical brakes. In fact, the same degree of control is available when lowering as when hoisting.

With electrically driven machinery a distinct advantage is obtained by the use of the ' VSG ' gear, inasmuch as the simplest and cheapest type of motor can be installed, *i.e.* squirrel cage induction or shunt wound DC., also controllers and resistances with their attendant heavy current losses are dispensed with, a light load starter only being necessary. Current peaks when starting heavy loads are eliminated.



Type "C" "VSG Mark III" Variable Speed Gear, fitted with Lever Control.

"VSG"

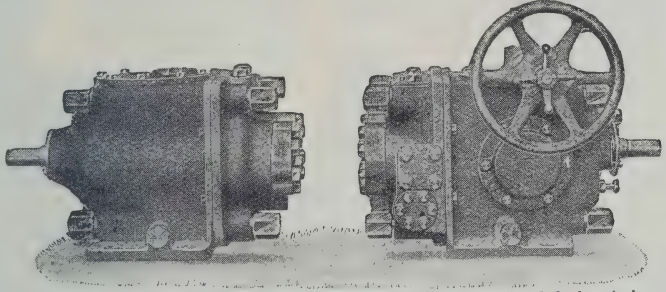
HYDRAULIC VARIABLE SPEED TRANSMISSION GEARS

Infinite variation of speed without steps, forward and reverse. Perfect and easy control. High starting torque. Low maintenance costs. Eminently suitable for controlling machinery for industrial purposes, viz.:—

Annealing Furnaces
Cable-Making Plant
Colliery Plant
Conveyors
Coilers for Belts, etc.
Foundry Plant
Floorcloth Machinery
Iron & Steel Works Machinery
Linoleum Machinery
Paper-Making Machinery
Photographic Paper and Film Treating Plant
Printing Machines
Rubber Machinery
Textile Machines
Wire-Making Machines
Woodworking Machines

Also for Ship Machinery, including Steering Gears, Windlasses, Capstans, Boat Hoists, Winches, etc.
Extensively used for control of Ordnance, Director Firing Gear, Ammunition Hoists, and other special purposes where positive control is essential.

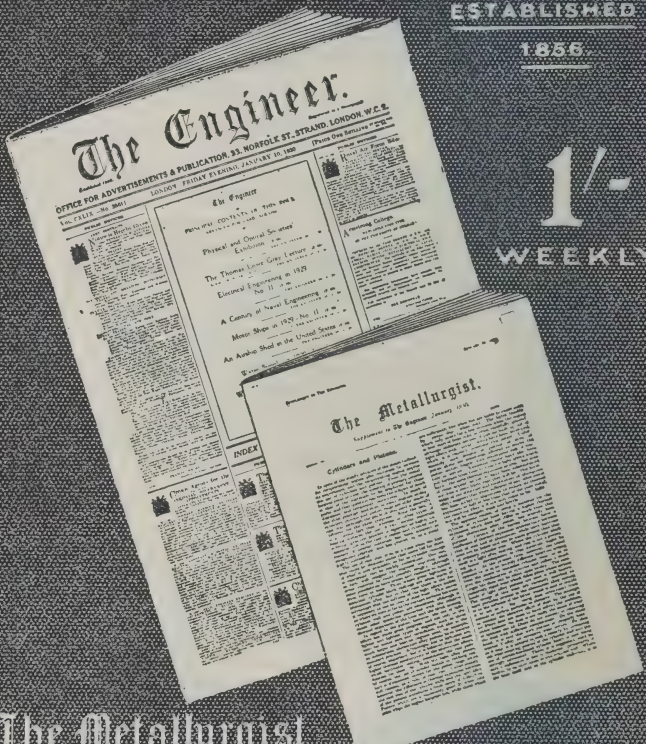
THE VARIABLE SPEED GEAR LTD.
6, BROADWAY, WESTMINSTER, S.W. 1.
Telephone: Victoria 6900. Telegrams: "Varispeed, Sowest, London."



Type "K" "VSG Mark III" Variable Speed Gear, fitted with handwheel control.

The Engineer.

ESTABLISHED 1856.



1/-
WEEKLY

The Metallurgist
IS A FREE MONTHLY SUPPLEMENT
TO The Engineer.

EDITORIAL and PUBLICATION OFFICES
28 ESSEX STREET STRAND, LONDON, W.C.2

ANNUAL SUBSCRIPTION RATE £3 5 0

A Continuously-Variable Chain-Driven Change-Speed Gear.

The P.I.V. (Positively Infinitely Variable) Chain Gear gives any desired speed variation up to six to one with a fully positive drive by means of an all-metal chain. The chain rides in the Vee formed between the opposed conical discs. These are formed with radial teeth, which are engaged by hardened steel laminations or 'slats,' held in the chain and capable of a small transverse movement enabling them to form teeth corresponding to those on the discs. An illustration of the chain showing the tooth formation is shown in fig. 46. The efficiency as shown by an N.P.L. test is 95 per cent. at full load. Gear-boxes, totally enclosed, with automatic lubrication, are made in five sizes up to 10 h.p. For full description, see 'The Engineer,' Aug. 27, 1926.

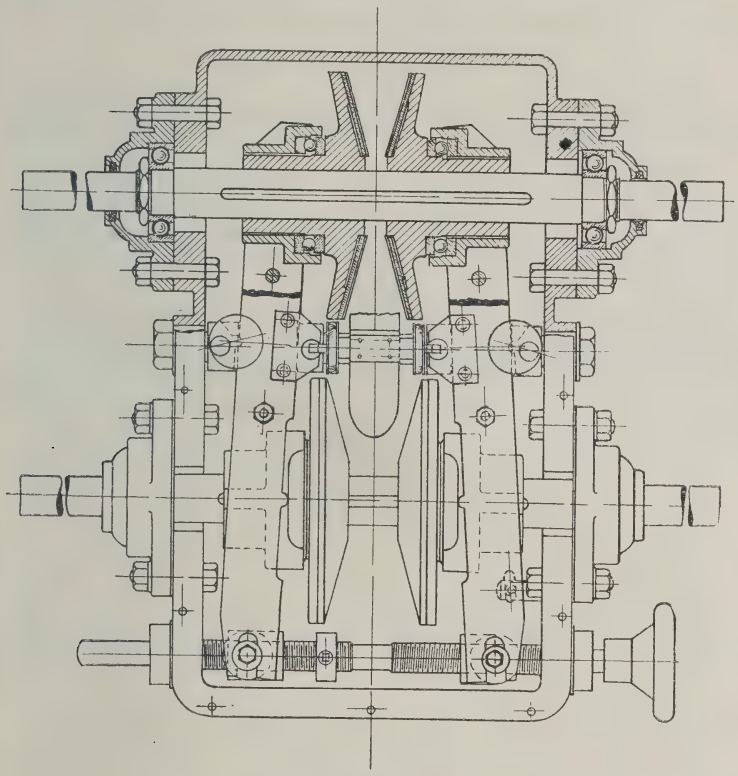


Fig. 46.

See also

The Keenok Pinion and Gear.
'The Engineer,' Sept. 23, 1927.

ADDENDUM.

Spur Gearing.

A report of recent researches conducted in America was presented to the American Gear Manufacturers' Association toward the end of 1930. The report deals with experimental work conducted on the Lewis testing machine to verify or modify the method by which the stress on gear teeth is determined.

Taking the Lewis formula, $W = ysfp$, and the stress formula, $S_v = S_o \left(\frac{600}{600 + V}\right)$ and, putting $yfp = K$, we have $W = K \left(\frac{600}{600 + V}\right) S_o$, which can be rewritten thus:

$$S_o = W \left(\frac{600 + V}{600K}\right) = \frac{1}{K}\left(W + W \frac{V}{600}\right).$$

The first item, W within the brackets, is the static load which can be applied to the gear, while the second item is that usually known as the increment load, which arises due to errors in tooth form, inaccuracy of tooth spacing, etc.

The equation shows the latter to be directly proportional to the static load W, a fact which in the light of these researches is erroneous in the case of high-speed gearing. The increment load depends upon inaccuracies in gear cutting, speed, and nature of material. The most common materials used seem to make the increment load practically independent of the static load. The former is apparently dependent upon the modulus of rigidity of the material used, since in the case of cast-iron gearing the increment load is more dependent on the applied load than is the case with steel gears. This is probably accounted for by the fact that the material with the lower modulus of rigidity is better able to absorb shock.

The increment loads given by the old formula are in many cases much in excess of the new values obtained in these researches, particularly in the case of the highly rigid materials such as the steels now employed. Complete data for calculating the increment load as determined by these new researches is not yet available.

SECTION XIX

PART IV

LINK MOTIONS.

(H. T. Davey, A.M.I.Mech.E.)

The Pantagraph.

This arrangement, fig. 1, is such that a point is constrained to copy to any desired scale the locus of another point. By this means reproductions are obtained. A common application of the pantagraph is in the engraving machine. Referring to fig. 1, AC, CP_1, BE, DE, are four levers,

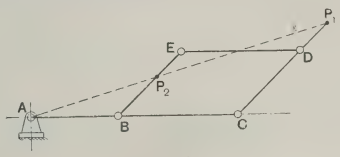


FIG. 1.

having pin joints at BCDE and forming a parallelogram, in which BE is parallel to CP_1 and DE is parallel to BC. The complete mechanism is pivoted to a bracket A. Points P_1 and P_2 are tracing points, P_2 being in link BE, so that A, P_2, and P_1 are in a common straight line. P_1 is guided over a figure to be reproduced, and P_2 traces a reproduction to any desired scale. P_2 can also be the guided point, and P_1 will then trace out the reproduction.

Link BC is kept constant and the length of AC is varied—*i.e.* ratio $\frac{AB}{AC}$ may be varied, and a reproduction to a desired scale be made.

BE is parallel to CD in all positions of the mechanism.

$\frac{AP_2}{AP_1} = \frac{AB}{AC}$ = constant. Hence the locus of P_1 will be similar to that of P_2.

See also p. 90.

Crosby Indicator Link Gear.

This is a multiplying arrangement in which the motion of the indicator piston is kept as small as possible to minimise friction and inertia errors. The actual motion of the piston is

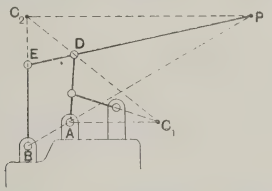


FIG. 2.

multiplied by a link motion (fig. 2). Links must be light to reduce friction and inertia, and P must move in a vertical path.

Points ABP are in the same straight line. C_1 and C_2 are the instantaneous centres of rotation

of links AD and EP respectively. The velocity ratio between P and A must be constant, P travelling in a straight line. Hence C_2P must be horizontal for all positions of the linkage, and the design must be arranged so that this is so, as nearly as possible.

Scott Russell Parallel Motion.

(1.)

In this motion, fig. 3, AB = AO = AC, and the angle BOC always a right angle. The link BC is constrained at point A by OA, which is fixed at O. OA is half the length of BC.

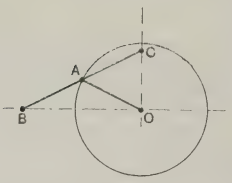


FIG. 3.

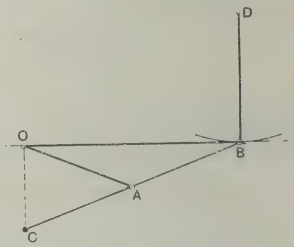


FIG. 4.

Point C moves along a straight line perpendicular to OB, and the locus of C is a reproduction of that of B. A small displacement of B causes a considerable movement of C as angle OAB approaches 180°.

(2.)

Point B, fig. 4, is not guided as in the Scott Russell linkage, but is controlled by a link BD, which in the normal position is perpendicular to OB. BD is relatively large, and hence the arc described by B is for a short length approximately a straight line. Hence C moves in a straight line.

Peaucellier Straight Line Motion.

This arrangement, fig. 5, is mathematically correct. It consists of seven pivoted links. Links

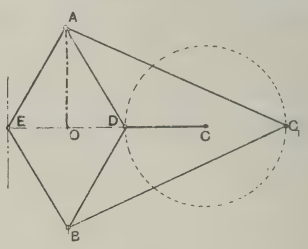


FIG. 5.

AC_1 and BC_1 are equal in length, and C, C_1 are fixed centres of rotation. Also AD = BD = BE = AE, and CD = CC_1.

Now

$$AC_1^2 = OC_1^2 + AO^2, \text{ and } AE^2 = EO^2 + AO^2;$$

therefore,

$$AC_1^2 - AE^2 = OC_1^2 - EO^2 = (OC_1 - EO)(OC_1 + EO) = DC_1.EC_1.$$

Thus since AC_1 and AE are both constant, $DC_1.EC_1$ is always constant, and therefore, since D lies in a circle, the locus of E must be a straight line, which is perpendicular to CC_1.

Kempe Straight Line Motion.

(A. B. and H. R. Kempe.)

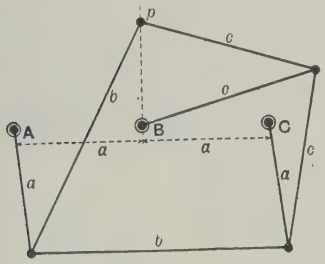


FIG. 6.

This, like the Peaucellier Straight Line Motion, is mathematically correct. The arrangement, fig. 6, is a combination of seven links, with three of the links pivoted at A, B, and C. The relative lengths of the links a, b, and c are such that

$$a : c :: c : b,$$

or,

$$c = \sqrt{ab}.$$

In the fig., a and b are in the proportion of 1 to 2, so that

$$c = \sqrt{1 \times 2} = \sqrt{2}.$$

With this arrangement the point p is constrained to move in an *accurately straight line traverse* away from B, towards B, or beyond B, the movement being perpendicular to AC.

So long as the value of c is such that

$$c = \sqrt{ab},$$

the proportionate value of a to b may be varied as desired, though it is not desirable to have this value very disproportionate, as although the straight traverse is still theoretically preserved, the arrangement becomes unstable from a practical point of view, through lack of *stiffness*.

Watt Link-Gear.

This is an approximate straight line motion. The arrangement, shown by fig. 7 in two forms, consists of two links AC, BD, which are pivoted at A and D, and connected together by a link BC. In the normal position, AC and BD are parallel, the connecting link being perpendicular to the links AC, BD.

A point P may be selected in the link BC, so that it moves in a straight line when small oscillations of AC and BD are made. Actually a point P in this link traces out a lemniscoid and only

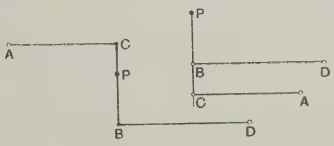


FIG. 7.

a small portion of its locus is a straight line. The required point P to give a straight line reproduction is found by making the lengths CP, PB, inversely proportional to the lengths of the levers.

The Watt link-gear is often combined with the pantagraph (see fig. 1), so as to provide two points having approximately linear motion, as in the beam engine, for example.

Quick Return Motion.

The link motion shown in fig. 8 is that commonly used in shaping machines in which the cutting stroke is made during the slower movement of the tool, while the return or non-cutting stroke is made more quickly. The cutting stroke is accomplished during the slower swing of the slotted link *l*—*i.e.* when the crank-pin *c* is moving in the upper arc of its path; the quick return stroke is made while the crank-pin moves in the lower arc of its path.

The ratio of the gear is equivalent to $\frac{\text{average time of cutting stroke}}{\text{average time of return stroke}}$.

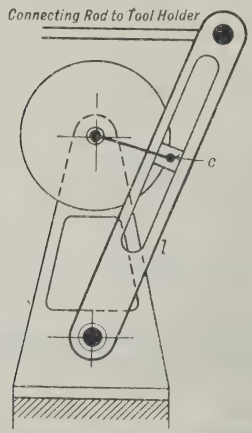


FIG. 8.

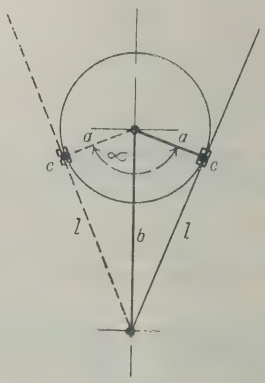


FIG. 9.

When the crank-arm a is at right angles to the slotted link l this link is then in its extreme position either to the left or right of its centre position. The cutting angle is therefore $360° - \alpha$, while the return angle is α (see fig. 9).

Hence the ratio of the gear $R_G = \frac{360° - \alpha}{\alpha}$.

Also, referring to fig. 9 : $a = b \cos \frac{\alpha}{2}$.

If R_G is known, then the lengths of the links can be determined from the given equations.

INDUSTRIAL STROBOSCOPES.

(A. J. Ashdown, Ltd., 119 Victoria Street, London, S.W. 1.)

'Stroboscope' is a general term appertaining to the science of motion study by interrupted vision. Its purpose is to provide a more intimate insight into the behaviour of mechanical and physical motions under conditions where theoretical considerations can sometimes be very misleading, owing to non-possession of all the factors which will govern the particular motion. In the last four years the use of stroboscopes has very considerably extended, and in practically every branch of industry it is used to confirm or modify features in mechanical design.

The principle on which the stroboscope works is that of retentivity of vision. The apparatus permits a succession of glimpses of the moving object to be received which, by this retentivity, blend into one continuous picture to the vision. It is obvious that as the object will frequently be moving at a high velocity it moves during the glimpse, and the latter must therefore be of

such shortness of duration that the object will have moved through but an infinitesimal distance; otherwise a 'blurring' or drawing out of the image will result.

The apparatus is so arranged that these glimpses can be given in exact synchronism with the moving object—in which case the object appears to be motionless—or it can be set at a speed slightly in excess of or less than that of the object, in which case the well-known 'slow-motion' study is permitted.

Stroboscopes are divided broadly into two divisions: in the first, the light illuminating the object is interrupted; in the second, the light remains continuous, and the line of vision itself is interrupted. Both methods are now in a very complete state of development. In the first form a neon-gas tube is used, and some forms of this apparatus are capable of giving amazing results. Such stroboscopes, however, necessitate the use of high-frequency currents of extremely high voltage discharged from a bank of condensers, this taking place between two electrodes

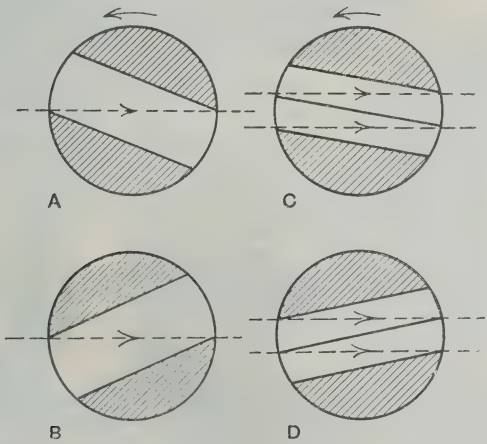


FIG. 10.

within the bulbs. As the heat capacity of the rarefied, inert gas is practically negligible, the tube lights up and darkens to all intents and purposes instantaneously as the current switches on and off. With this form another objection is that, in most cases, fixed contact has to be made with some part of the object to secure the necessary synchronism; also that for satisfactory observation all existing lighting must be obscured so that the full benefit of the timed flashes can be secured. It must be obvious, therefore, that the interrupted-vision type has here a very great advantage, for the reason that all the available normal lighting can be utilised, amplified when necessary by an ordinary continuous source of lighting. However, a mechanical shutter to operate with sufficient rapidity is an extremely difficult problem, and cannot be accomplished by any disc form of the Plateau type.

In 1925 the Ashdown Rotoscope was put on the market; its special feature lies in the shutter, or 'sight-gears' as they are called. In the simplest of these, known as the 'bladed' type, the vision slits are bladed, or 'compounded' as it is called, which results in the large field and ample lighting of a very wide slit being secured with the sharpness or 'definition' of an extremely narrow slit. The principal is shown in fig. 10. A second form is known as the 'heteroptic' shutter, which consists of two cylinders, one rotating within the other, and both revolving at extremely high velocities but at different speeds. By this means a 'phasing' takes place, and the requisite succession of glimpses takes place, each of which has a duration of less than one one-hundred-thousandth part of a second. So sharp and clear is the definition obtained, in fact, that it is possible to read from newspapers as they are coming off the printing-presses at a rate of 600 to 700 per minute.

This apparatus has a speed range in the standard types of from 100 to 40,000 per minute. Thus any object, with no matter what form of motion within this speed range (either rotary, reciprocating, or vibratory), can be intimately studied at an apparent speed of less than one-thousandth the actual.

A most important secondary function of the Ashdown Rotoscope is that it will ascertain, with considerable accuracy, the speed of any motion within the above range, and the extent of any speed variations taking place, without contact in any way with the object—*i.e.* at a safe distance.

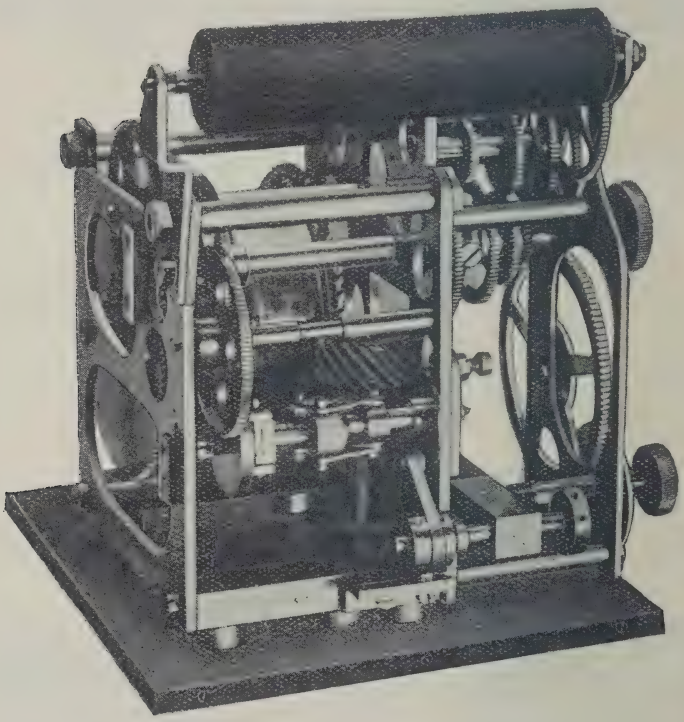


FIG. 11.—The Ashdown Rotoscope.

SECTION XX

PART I

BEARING PRESSURES—HEATING OF BEARINGS—BALL AND ROLLER BEARINGS (pp. 961-976)

(Revised by Professor Dempster Smith, M.B.E., M.Sc., M.I.Mech.E., and Prof. John Goodman, M.Inst.C.E.)

PART II

LUBRICATING OILS, LUBRICATION, AND LUBRICATORS (pp. 977-992)

(Contributed by Harold Moore, M.Sc., M.Inst. Pet. Tech., F.C.S., A.I.M.E.)

PART III

VISCOSITY AND VISCOUS FLOW OF OILS (pp. 993-999)

PART IV

SHAFTING AND SHAFTS—FLEXIBLE COUPLINGS (pp. 1001-1008)

PART V

SPRINGS (pp. 1009-1016)

The Bearings for heavy loads

The components of the Hoffmann Bearing are proportioned to give greatest load capacity. The rolling elements are as large as possible consistent with adequate race thickness, and a sturdy separating cage of yellow anti-friction metal is employed.

Whatever the conditions of speed and load, there is a Hoffmann Bearing to do the job efficiently.

We have a technical staff which is always ready to collaborate with regard to your bearing problems.

HOFFMANN
Ball and Roller Bearings

The Hoffmann Mfg. Co. Ltd. Chelmsford, Essex.

F. 961

SECTION XX

PART I

BEARING PRESSURES—HEATING OF BEARINGS—BALL AND ROLLER BEARINGS

(Revised by Professor Dempster Smith, M.B.E., M.Sc., M.I.Mech.E.)

BEARING PRESSURES.*

Pressures on Bearings.

The following table shows the wide variation of pressure in bearings :—

	Lbs. per Sq. In.
Light and fast-running cotton spindle	2 to 5
Light shafting	80 to 100
Heavy shafting	300 to 600
Marine engine crank pins	600 to 900
Crosshead pins (horizontal engines)	1,200 to 1,300
Crank pins ,, ,,	800 to 950
Crank shaft (steam load) ,, ,,	100 to 150
,, ,, (combined load) ,, ,,	150 to 200
Second motion bearings ,, ,,	100 to 200
Crosshead slides ,, ,,	40 to 50
Intermediate piston rod slides ,, ,,	20 to 40
Main bearings (vertical high-speed engines)	200 to 300
Crank pins ,, ,,	400 to 550
Crosshead pins ,, ,,	1,000 to 1,200
Crosshead slippers ,, ,,	40 to 50
Eccentric straps ,, ,,	60 to 70
Crosshead pins (locomotive)	1,400 to 1,800
Crank pins ,,	1,000 to 1,500
Axle bearings (loco. engine and tender)	200 to 250
Railway carriage bearings	300 to 350
Babbitt metal on cast-iron slide	200 to 250

MAXIMUM SAFE PRESSURES ON BEARINGS.

Nature of Bearing.	Safe Load. Lbs. per Sq. In.
Ordinary bearings for shafting :—	
Wrought-iron on cast-iron	250
Wrought-iron on gunmetal	300
Mild steel on cast-iron	300
Mild steel on gunmetal	370
Mild steel on white metal	500
Cast-steel on gunmetal	600
Flywheel shaft	250

At unit pressures of 250 lbs. per sq. in. for cast-iron bearings, and 145 lbs. for bronze bearings, there is difficulty in keeping the boxes from seizing the shaft.

* For Bearing Metals, see METALLURGY, Section XL.

Professor W. C. Unwin carried out a series of experiments with Delta White Antifriction Metal, of which particulars are given. These tests show that this metal heats but little, even with high-bearing pressures :—

Mean Revolutions of Shaft per Minute	Mean Pressure per Square Inch of Journal Surface in lbs.	Mean Temperature in Deg. Fahr.	Mean Moment of Friction in lbs.-in.	Net Pressure on One Step in lbs.	Friction on One Step at Journal Surface in lbs.	Co-efficient of Friction
123	128·35	62·65	9·25	477·5	6·17	0·01292
	256·70	64·00	16·26	955	10·84	0·01135
	385·05	65·75	24·23	1432·5	16·15	0·01128
	513·40	67·70	40·87	1910	27·25	0·01428
315	128·35	73·80	12·00	477·5	8·00	0·01675
	256·70	76·20	17·25	955	11·50	0·01204
	385·05	80·05	26·00	1432·5	17·33	0·01210
	513·40	86·40	44·63	1910	29·75	0·01558

COEFFICIENTS OF FRICTION AND SAFE MAXIMUM PRESSURES FOR VARIOUS LUBRICANTS.

Lubricant.	Coefficient of Friction.	Max. Safe Pressure in Lbs. per sq. in. on Nominal Area.
Olive oil	·00172	520
Lard oil	·00172	570
Sperm oil	·00208	570
Mineral oil	·00176	625
Mineral grease	·00233	625

Heating of Bearings.

SIGNALS FOR INDICATING HEATING OF BEARINGS.

The double iodide of silver and mercury, $AgIHgI_2$, is a pale lemon-yellow powder at ordinary temperatures, and changes to a vivid carmine colour at about 90° to 100° C., the change in both directions being fairly sharp. The copper mercuric iodide, $Cu_2I_2 2HgI_2$, is a vivid scarlet-vermilion at ordinary temperatures, and at 60° to 70° C. it changes to a chocolate-brown. Both these compounds are adapted for use under suitable conditions, but a mixture of the two, consisting of 85 per cent. of the copper salt and 15 per cent. of the silver salt, is more sensitive and gives a greater colour range from vermilion to almost black, the transition being exceedingly sharp.

The compounds may be prepared in a variety of ways. A simple method of preparing the copper salt is by well triturating in a mortar 380 grms. of cuprous iodide with 910 grms. of mercuric iodide, sufficient water being added to form a paste. When thoroughly mixed, the mass is dried in the water-oven, and the dried mass ground to an impalpable powder. Another method is to dissolve the mercuric iodide in excess of potassium-iodide solution, and precipitate the double salt by the addition of copper-sulphate solution. The silver salt can be prepared in a similar manner by double precipitation. Whichever method is adopted the product should be well washed to eliminate soluble impurities, as these appear to retard the colour-change, as well as to affect its character.

To apply the mixture, convert it into a varnish paint or enamel, by incorporating the powder with a colourless varnish medium that will stand moderately high temperatures without softening. A good white-shellac spirit-varnish, such as is used on electrical machinery, has been found suitable for most occasions, but for high temperatures a medium such as is employed in the high-temperature aluminium paints used for steampipes, and which will resist temperatures up to 200° C., is preferable. About 100 grms. of compound to 70 c.c. of medium will be found a mixture of suitable consistence.

A convenient method of applying this heat-detector is first to paint a white circle or ring on the bearing or other place which it is desired to keep under observation, using a zinc-white paint. This white circle is applied to the journal or housing of the bearing, and the detector-paint is applied as a vivid red bull's-eye in the middle of this, thus forming a striking object which is visible from considerable distances. It must be applied to the block of metal as near as possible to the point at which the heat is likely to be generated. It is sometimes desirable to apply the paint to the shaft itself, just where the latter extends beyond the bearing-metal. A convenient method is to apply a broad band of white paint with a narrower band of the red detector paint in the middle. This red band in the middle of the white one is easily seen as the shaft rotates. In any case, after the paint is dry it should be protected with a coat of colourless oil-proof varnish, so that the surface may be occasionally wiped free from dust and dirt. The visibility of the paint against its white background is high, the bright scarlet colour being visible from long distances without effort, and the change at 60° to 70° C. to black is easily observed. It is generally easy to arrange matters so that the detector is placed in a position where it is visible almost without conscious effort; and experience has proved that a workman is easily trained to cast his eye occasionally on the detector, while he could not be depended upon to climb to the spot and feel the bearing on the remote chance of finding it heating. When the paint changes colour there is something wrong with the bearing, but generally speaking it will be found that the matter is taken in time, and that no damage is done, the bearing simply requiring cleaning and adjustment without serious stoppage and expense. (*H. T. Pinnock*, '*Journal of the Chemical Society.*')

BALL AND ROLLER BEARINGS.

(Contributed by John Goodman, M.Inst.C.E.)

BALL BEARINGS.

In the earliest cylindrical or radial bearings the balls bore on four points as shown at A, fig. 1,* and corresponding thrust bearings were made, as shown at F. It is impossible, under such conditions, to secure a pure rolling motion for the balls, so that grinding inevitably takes place.

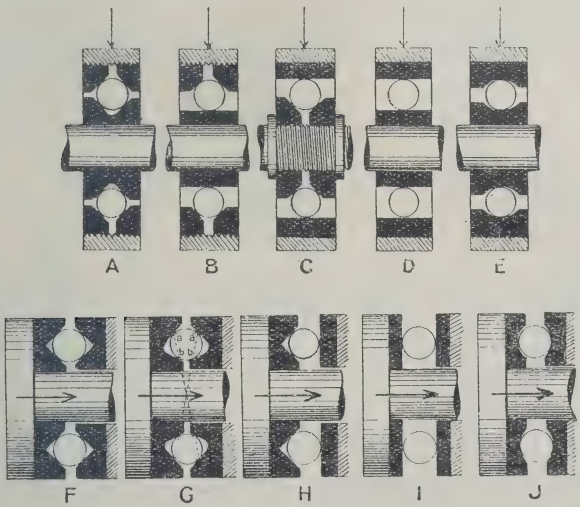


FIG. 1.

* Fig. 1 is reproduced from Professor Goodman's 'Mechanics applied to Engineering,' by the permission of the publishers, Messrs. Longmans, Green & Co.

The modification shown at G was then tried, but only with partial success. Three point bearings such as those shown at B, C and H, and also many others, were tried, but in every instance the balls grated and developed deep scratches after running for a short time; they were then abandoned for two-point bearings (the outcome of experiments by the writer), shown in D and I. The friction was found to be less, the bearing was cheaper to make, the balls had a true rolling motion, and, consequently, showed no signs of scratching, grinding, or wear; the bearings ran smoothly, and were in every respect a success. In general, a cage was used for keeping the balls in their relative positions, but in some instances it was found that it might be dispensed with. Other experimenters, notably Professor Stribeck, of Berlin, modified the two-point bearing by providing grooved ball races, as shown in E and J. Balls running in such races will support considerably greater loads than when running on plain surfaces which, of course, is a great advantage. The load that may be placed on a bearing of given dimensions, other things being equal, depends upon the relative radius of the groove and that of the balls, the smaller the difference between them the greater the load-carrying capacity; in modern bearings the radius of the groove is about 5 per cent. greater than the radius of the balls. The safe load also depends upon the number and the diameter of the balls, but it is, of course, useless to increase the diameter of the balls at the expense of the races, which must be sufficiently robust to retain their shape under load.

Radial Ball Bearings.

A considerable amount of thought and ingenuity has been expended on the problem of introducing the greatest possible number of balls into a radial bearing.

In the bearing D there is no difficulty in inserting the full number of balls, but in bearings which have grooved tracks as shown at E it is necessary, when assembling, to displace the one race relatively to the other, as shown in fig. 2, when the bearing can be filled to the extent of half the full number of balls *plus* one. The section through such a bearing is shown in fig. 3.

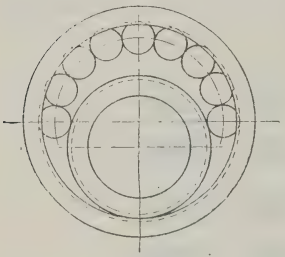


Fig. 2.

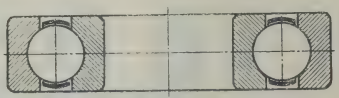


Fig. 3.

Extra balls can be inserted through transverse filling slots in the sides of the grooved tracks, the depth of which must be less than that of the groove, so that the slot does not extend to the middle of the track or in any way interfere with the pathway of the balls; but this condition is not so easily fulfilled as it may appear at first sight. The balls may be quite clear of the filling slot when the bearing is unloaded, but when it is loaded both the ball and the track are distorted to such an extent as to render it possible that the side of the track may be intersected by the filling slot; it is obvious that this is more liable to occur when the radius of the track closest approaches that of the balls. Thus on the one hand a small difference between the radius of the track and the ball is aimed at for the sake of increasing the safe load on the balls, and on the other a larger radius of track has to be used in order to prevent the balls from fouling the filling slot. Thus it is a matter of opinion as to whether the bearing provided with filling slots and a larger number of balls running in a wider track will actually carry a greater load than the bearing without such slots and with a closer fitting track with a smaller number of balls.

In order to overcome the difficulty of inserting the full number of balls, the 'Skefko' bearing is made with the outer ball track a portion of a sphere, the centre of which coincides with the centre of the shaft (fig. 4).* By this arrangement the outer race will swivel relatively to the inner (see fig. 5), when a full number double row of balls can be inserted.

* A filling slot materially reduces the capacity of the bearing to take end-thrust.

The swivelling property of the 'Skefko' bearing also has other advantages. If the housing shifts relatively to the shaft due to a lack of rigidity in the machine frame, or if the shaft itself is out of truth, the bearing will still run perfectly.

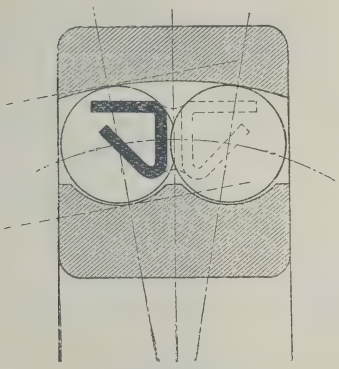


FIG. 4.

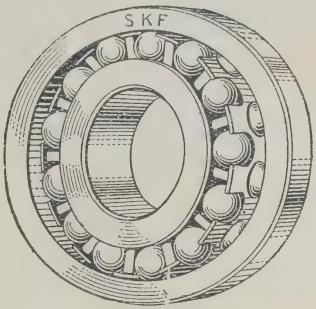


FIG. 5.

In fig. 6 is shown a 'Hoffmann' self-aligning outer race and shell and self-aligning end covers to accommodate errors in mounting. When a bearing is not completely filled with balls it is obvious that a cage must be used or the balls may crowd together and cause trouble. For moderate speeds a light sheet metal cage may be used, but for high speeds a more robust

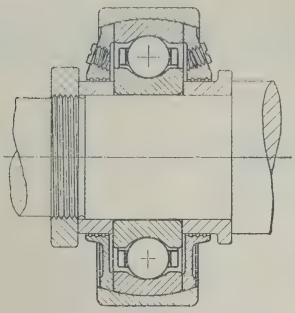


FIG. 6.

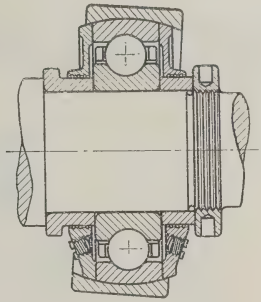


FIG. 7.

cage is necessary, accurately machined all over and fitting on the inner ball race ring to prevent the cage from running eccentrically. At very high speeds the centrifugal force on an eccentrically running cage becomes serious and is certain to give trouble. White metal and aluminium do not possess sufficient strength or hardness; steel, gun-metal, or bronze give the best results.

For long line shafts the 'Hoffmann' bearing, shown in fig. 8, is mounted in a housing slung on a rocking pin to accommodate expansion of the shafting. The inner race is clamped to the shaft by an adapter or split taper sleeve.

In fig. 9 is shown a bearing hanger for the same purpose by the Skefko Ball Bearing Co. Ltd.

When bearings are required for loads beyond the capacity of a standard ball journal bearing, it is usually cheaper and better practice to use a standard roller journal bearing; or several

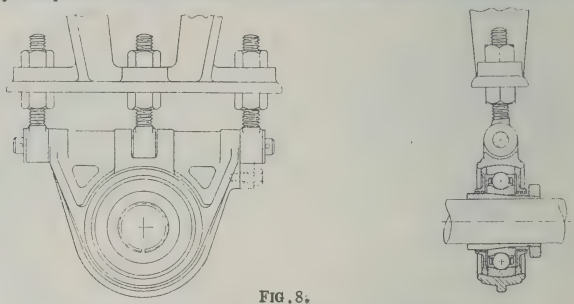


FIG. 8.

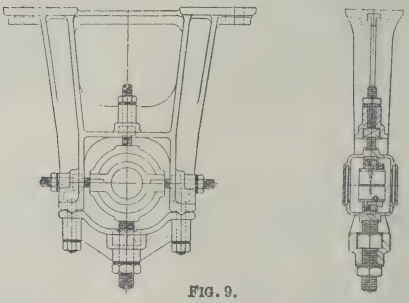


FIG. 9.

standard bearings can be mounted side by side, but there is a difficulty in ensuring that each bearing takes its fair share of the load. Goodman overcomes this difficulty by backing them with linoleum, vulcanised fibre, or hard wood. The housing is spherically mounted to allow it to adjust itself to any want of alignment (fig. 10).*

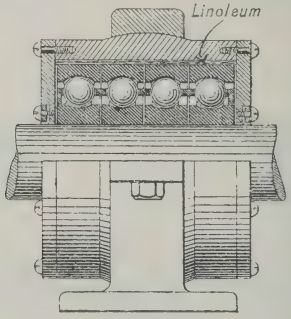


FIG. 10

* Fig. 10 is reproduced from a paper on 'Roller and Ball Bearings,' *Proceedings of the Institution of Civil Engineers*, vol. clxxxix, 1912, by the permission of the Council.

In all cases, even with single bearings, it is good practice to use a semi-hard backing when mounting ball and roller bearings. The backing acts as a cushion, thereby reducing noise, and it enables the outer ring to adjust itself to any slight lack of alignment.

Thrust Ball Bearings.

In thrust bearings there is no difficulty in getting in the full number of balls. A cage is always used. Radial bearings with grooved races will take a certain amount of end thrust, depending upon the radial load on the bearing. For numerical data on this question such a book as Macaulay's 'Ball and Roller Bearings' should be consulted.

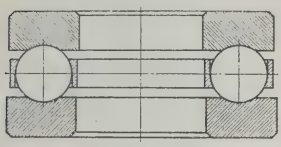


Fig. 11.

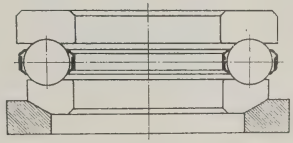


Fig. 12.

If the end thrust and radial load are large, two bearings should be used, the one to take the radial load and the other the end thrust.

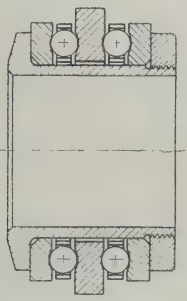


Fig. 13.

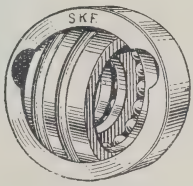


Fig. 14.

Fig. 11 shows a flat ring thrust bearing, and fig. 12 a thrust bearing with a spherical seat to accommodate errors in mounting.

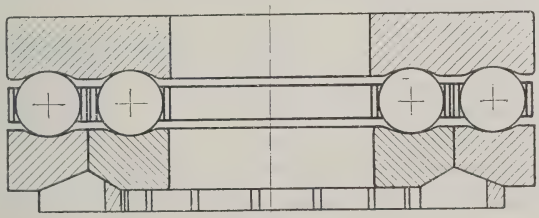


Fig. 15.

Fig. 13 shows a 'Hoffmann' double thrust ball bearing mounted on a sleeve, arranged to take thrust in either direction. The 'Skefko' bearing for the same purpose is shown in fig. 14.

For larger thrust loads multiple ring bearings are used mounted on a seating ring. The 'Hoffmann' bearing of this type, shown in fig. 15, is provided with an elastic seating ring to ensure the correct proportionate distribution of the load between both rows of balls; the 'Skefko' bearing in fig. 16.

Goodman mounts standard thrust rings on a linoleum backing, as shown in figs. 16 & 17.*

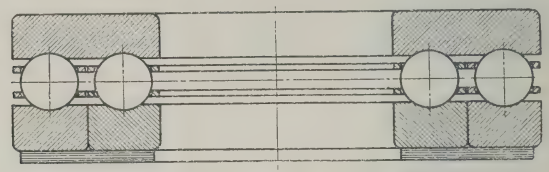


FIG. 16.

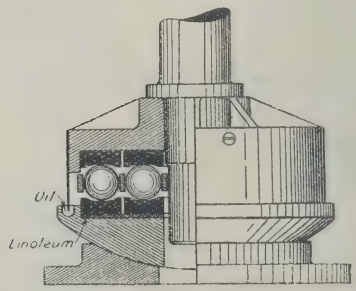


FIG. 17.

Permissible Loads on Ball Bearings.

The safe working load for a given bearing is much greater when the bearing is running at a low than at a high speed. The ruling factor as regards the working load for a ball bearing is the speed of rotation of the balls themselves; thus in the case of two shafts, A and B, each fitted with thrust bearings having the same number and diameter of balls, and both running at the same number of revolutions per minute, if the diameter of the ball path in A were greater than in B, the safe working load would be greater for shaft B than for A.

Let P = The maximum permissible working load in pounds;

d = The diameter of the ball in inches;

D = The diameter of the ball path in inches;

n = The number of revolutions per minute of the shaft;

k = A constant depending upon the form of bearing;

c = A constant;

m = The number of balls in the bearing;

x = A coefficient depending upon the revolutions per minute of the shaft.

Stribeck's formula for radial ball bearings is:—

$$P = 12{\cdot}8\ x\ m\ d^2$$

* Fig. 17 is reproduced from a paper on 'Roller and Ball Bearings,' *Proceedings of the Institution of Civil Engineers*, vol. clxxxix, 1912, by the permission of the Council.

where x has the following values :—

n 0	10	150	300	500	1,000	1,200
x 48·4	36·3	29·04	24·2	20·9	15·62	14·96
n 1,500	2,000	2,500	3,000	4,000	5,000	10,000
x 13·64	12·1	10·56	9·68	8·58	7·7	5·06

Goodman's formula for ball bearings is :—

$$P = \frac{k\ m\ d^2}{nD + cd},$$

where k and c have the following values :—

Type of Bearing.		k	c
Radial.	Flat (i.e. not grooved) races	1,000,000	2,000
	Hollow races of ordinary quality	2,000,000	2,000
	Hollow races of best quality	2,500,000	2,000
Thrust.	Flat (i.e. not grooved) races	500,000	200
	Hollow races of ordinary quality	1,000,000	200
	Hollow races of best quality	1,250,000	200

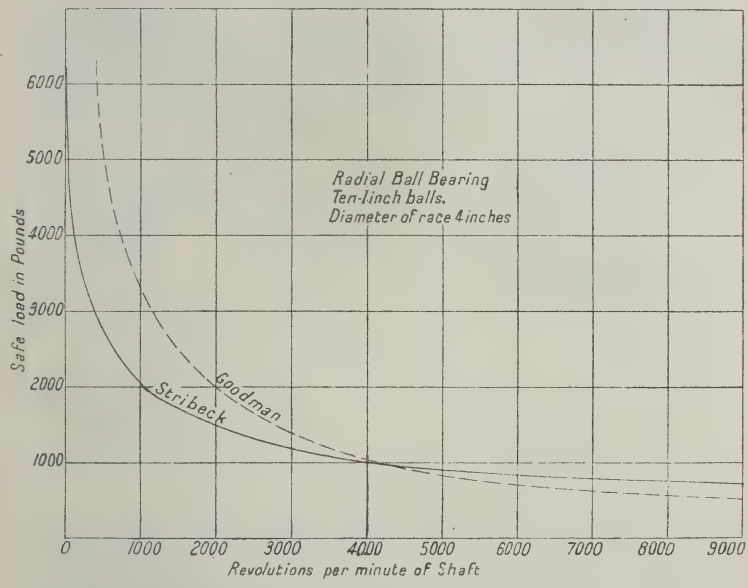


FIG. 18.

A comparison of the results obtained by the two formulæ is shown in fig. 18.

Load tables based on these or similar formulæ are published by most manufacturers. But in all cases other factors should be taken into account, such as the number of hours per day the bearing is likely to run, whether a steady or a fluctuating load, and whether it will be possible to keep it free from dirt, dust, and damp. When bearings are required for special conditions of running it is advisable to consult the manufacturers.

In the 'Skefko' bearing the races must be regarded as flat, but since there are at least twice as many balls in this type of bearing as in other bearings of the same external dimensions, the theoretical load carrying capacity is the same. It is claimed that this bearing will work satisfactorily with a small amount of lateral thrust.

Advantages and Disadvantages of Ball Bearings.

Advantages.

The friction is usually less than one-tenth of that encountered in ordinary bearings as commonly lubricated.

The friction at starting is no greater than when running.

The friction does not increase with the speed.

They require an extremely small amount of oil, the cost of which is negligible.

They occupy much less lateral space on the shaft than an ordinary bearing.

If the bearing is properly treated the wear is extremely small.

Disadvantages.

The first cost is usually greater than that of an ordinary bearing, but this is not invariably the case.

If overloaded, or if lubricated with unsuitable oil the balls pit and peel, and then they make a great deal of noise, especially at high speeds.

They require extreme care in fitting on to the shaft and in the housing.

Power Saving and Reduction of Starting Effort.

The power saving amounts to at least 90 per cent. of the loss due to friction of the best plain bearings, whilst the reduction of starting effort of no less than 99 per cent. means that the acceleration is more rapid, which is of particular importance in machinery where frequent stops and starts are essential. Further, the reserve of power usually required for starting machinery is unnecessary, and the size of the power unit can therefore be reduced.

The enormous saving in power, and still more remarkable reduction in starting effort, are illustrated by a test carried out by the National Physical Laboratory, the results of which are given in the following table:—

	Standard Plain Hanger.		'Hoffmann' Ball-Bearing Hanger (Patent).
	With Needle Lubricator.	With Oil Ring Lubricator.	
Starting effort.	830 in.-lbs.	770 in.-lbs.	0·9 in.-lbs.
Revs. per Minute.	Coefficient of Friction.		
80	0·014	0·015	0·0013
130	0·015	0·014	0·0014
250	0·016	0·013	0·0015
500	0·017	0·012	0·0016

Roller Bearings.

For heavy loads, particularly where the bearing is subjected to severe shocks, a roller bearing should be used in preference to a ball journal bearing.

The development of the roller bearing has been more difficult and consequently slower than the ball bearing owing to the tendency of the rollers to creep axially, the resultant end-thrust causing friction and consequent wear in the cage and rollers. This creep is invariably caused by

inaccuracy of the tracks and rollers; the smallest error along the length of these alters the relative peripheral speed and destroys pure rolling motion.

Four distinct types of roller bearings have been developed :—

(1) With cylindrical rollers in which the length is considerably greater than the diameter.
(2) With taper roller.
(3) With cylindrical rollers in which the length is equal to the diameter.
(4) With slightly barrelled rollers in which the length is equal to the diameter.

Bearing in mind the basic fact that 'Friction causes Wear,' the following table of coefficients of friction obtained experimentally by Professor Goodman (*Proc. Inst. Civil Engineers*, vol. clxxxix.) affords a sound method of comparison of the merits of the various types, and it will be seen that bearings with short cylindrical rollers are now made combining the efficiency of ball bearings with greatly increased load capacity :—

Type of bearing.	Ball.	Short Cylindrical Roller. III.	Taper Roller. II.	Long Cylindrical Solid Roller I.	Long Cylindrical Flexible Roller I.
Coefficient of friction	0·0010 0·0015	0·0010 0·0015	0·0020 0·0040	0·0030 0·0100	0·0040 0·0070

(1) *Long cylindrical roller bearing.*—This type of bearing has not proved satisfactory for arduous duty on account of the tendency of the rollers to 'skew.' The difficulty of manufacturing the rollers accurately as regards diameter and parallelism is obviously increased in proportion to the length, and the superimposed bending stresses caused by 'skew' also increase proportionately.

In the 'Hyatt' roller bearing the solid roller is replaced by right- and left-hand helical springs arranged alternately; it is claimed that 'skewing' difficulties are obviated by the flexibility of the rollers.

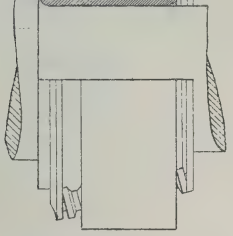


FIG. 19.

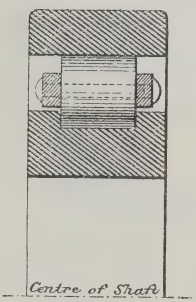


FIG. 20.

(2) *Taper roller bearing.*—Several makes of this type of bearing are now on the market, and it is claimed as advantages that the bearing will support both journal and thrust loads, and can be adjusted to take up wear.

On account of its design the bearing cannot be used singly except where there is a constant end-thrust in one direction.

While this type of bearing is adjustable when new and unworn, it should not be forgotten that under journal load only one half of the stationary race is loaded, and any wear in the track of this race will make it oval, whilst the revolving race and rollers may wear evenly and remain round.

Adjustment subsequent to wear is therefore analogous to forcing a round plug into an oval hole. One of the most successful bearings of this type is the Timken (see fig. 19); it is largely used for motor-car work.

(3) *Short cylindrical roller bearing.*—This type of bearing, fig. 20, was originated and developed in England by the Hoffmann Manufacturing Co., but is now made by many firms. The rollers are solid and have a length equal to their diameter. They are accurately finished on diameter to the same limits as balls, viz. plus and minus 0·0001 ins., and the ends are ground perfectly square. The extreme accuracy of these rollers is illustrated by the fact that they are supplied for use as gauges under a National Physical Laboratory Certificate of Accuracy.

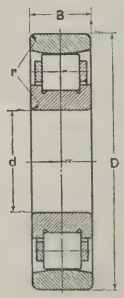


FIG. 21.—Cylindrical Roller Bearing, with convex outer race.

To guide the rollers, the inner race is provided with two lips to form a channel-shape track which, in conjunction with the accuracy with which the tracks and rollers are produced, effectually prevents any tendency to 'skew.' The outer race is a plain annular ring, and the rollers are separated by a cage of solid yellow anti-friction alloy metal.

With these bearings contact is obtained with both tracks along the whole length of the roller, from 70 to 130 per cent., and this 'line contact' gives a load capacity greater than that of a ball bearing of similar size, according to the conditions and nature of the load. The bearings are made with the same external dimensions (bore, outside diameter, and width) as ball journal bearings, and are interchangeable with them with only slight alterations in the mounting.

As the outer race has a plain cylindrical track, the bearing cannot take end-thrust of any sort, and to prevent its slipping sideways off the rollers, the outer race must be definitely clamped endways. This is a definite advantage where a roller bearing is mounted in conjunction with a

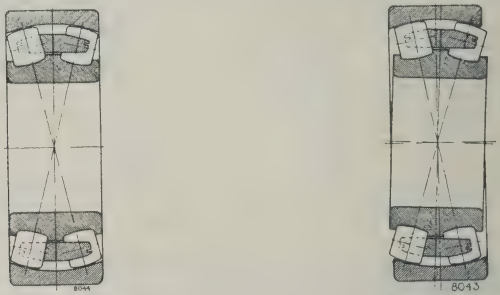


FIG. 22.—Double-row Self-aligning Roller Bearing.

ball bearing, and the outer races revolve, as both can be made an interference fit in the housing to prevent creep. Apart from this, the bearings should be mounted exactly similar to ball bearings.

The Skefko Ball Bearing Co. Ltd. make the outer race slightly convex in order to allow of a small

angular movement of the inner race relative to the outer, which has proved to be very successful in practice in cases in which there is a difficulty in maintaining an exact alignment between the shaft and housing (see fig. 21), but of course it will not take any end-thrust. Where end-thrust has to be dealt with in addition to radial load the bearing shown in fig. 22 is used. It has been

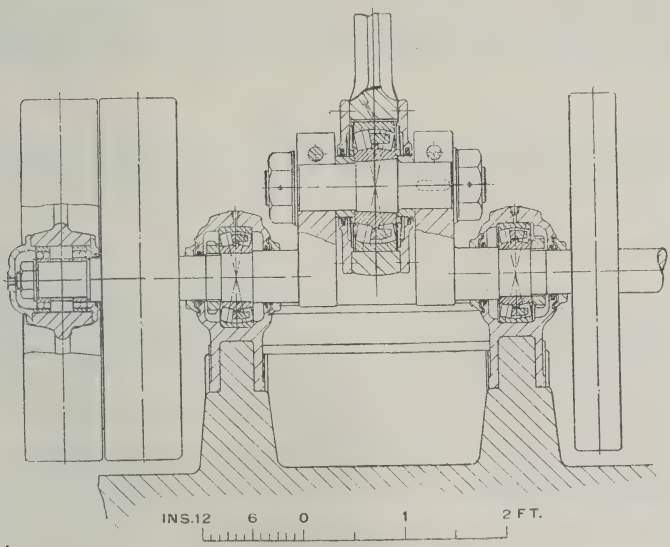


FIG. 23.

successfully used for heavy duty such as railway axle-boxes, trams, crushers, calenders, etc. On account of its construction it is able to negotiate thrust load in either direction simultaneously with radial load. This bearing can deal with a radial load of about 100 per cent. greater than a ball bearing of equivalent dimensions, and has been applied to a large number of passenger coaches where the speeds attained have been between 60 and 70 m.p.h.

ANTI-FRICTION BEARINGS FOR LOG FRAME-SAWS.*

Log frame-saws of the modern type have several saw-blades in series, mounted in a suitable frame made to slide up and down within the main frame of the machine by means of a crank and connecting rod. The type which is of particular interest here is the single-crank type, and fig. 23 illustrates a bearing arrangement which has been found effective.

The size of the crank-pin bearing, which is of the spherical double-row roller type, is as follows:

Bore, tapering, 140 mm. diameter at the smallest end;
Outside diameter, 380 mm.;
Width, 120 mm.

The connecting-rod is of wrought nickel steel, and the thickness of the material of the big-end housing varies between 30 to 50 mm. The crank radius is 250 mm. (approximately 10 ins.). If the width of the saw-frame is between 500 and 600 mm., its weight, complete with blades and accessories will be about 615 lbs., and the weight of the connecting-rod complete, including the outer ring of the roller-bearing, will be about 485 lbs. When the frame is running at a speed of

* J. B. Dahlerus, *Proc. Inst. Mech. Eng.*, 1925.

350 r.p.m., this being the highest speed attained so far, the maximum acceleration force on the crank-pin bearing will be about 37,600 lbs., or approximately 16¾ tons.

In calculating the maximum load coming on the bearing, the force required for sawing should also be added. This amounts normally to about 900 lbs. The calculated load is not, of course, constant, but varies intermittently throughout every revolution.

With a bearing of the dimensions given in this particular instance, the factor of safety is approximately two over the published figures, and has been found to meet the conditions admirably. The service on these machines is practically continuous day and night, so that the requirements in respect of durability and safety in operation are very severe.

General Applications.

Other machines where these bearings have been successfully applied include: ball mills; tube mills; pulp beaters for paper making; pulverizers; disintegrators; chocolate refining rolls; an alternator of 2,500 kw. at 1,000 r.p.m.; 3-bearing rotary converters; road wheels of motor buses; cross rolls in steel plant.

Mounting Ball and Roller Bearings.

The revolving race of a ball or roller journal bearing must be made an interference fit on its seating to prevent creep or slow rotation of the race relative to the shaft or housing, and it is advisable to clamp this race firmly endways. The stationary race should be allowed just sufficient clearance on its seat to permit it to take up its correct position endways relative to the revolving race, and sufficient end clearance should be arranged for to prevent the stationary race coming in contact with a shoulder and thus putting a permanent side-thrust on the bearing.

The writer of this article considers that the stationary race should be mounted with a soft backing such as linoleum, a hard wood ring, or hard fibre in order to reduce noise and to allow of a small movement if the housing gets out of alignment.

In obtaining the necessary fits it is essential that the seatings and abutments of the housing and shaft be perfectly round and true. The hardened steel races have a comparatively thin section, and should be regarded purely as tracks for the balls, as they will conform readily to inaccuracies of their seatings.

The use of set screws, keys, or similar devices for fixing the races and preventing creep cannot be too strongly deprecated, as they readily distort the races, causing overloading of the bearings. In any case, it will be found these are of very little use as they shear or fret away rapidly.

Where there is no definite end-thrust the shaft may be located endways by clamping the stationary race of one bearing only in a solidly connected length of shaft. The most lightly loaded bearing should be selected for this purpose.

The revolving race of a ball thrust bearing should be made a push fit on the shaft, and it is essential that the shoulder against which it abuts is perfectly smooth and square. It should be of such diameter as to extend at least half-way across the track of the race to adequately support this and prevent deflection under load. The stationary race must be held concentric, and where it is not possible to ensure a perfectly square seating, a self-aligning bearing should be used.

It is essential that a ball thrust bearing be not called on to take any journal load whatsoever, and a journal bearing of some kind must be mounted immediately alongside to ensure that the two ball races are held concentric. A ball or roller bearing is, of course, to be recommended for this, as it will not wear slack and allow the shaft to drop out of concentricity with the housing.

For high speeds it is important to keep the pitch diameter of the races and the size of the balls as low as possible, and where thrust bearings are required for speeds above those listed by manufacturers, it is advisable to consult the makers.

Split housings should as a general rule be avoided, and it is generally found an advantage to make the housings in the form of plain, eye-bored castings, and fit loose end covers for protection and retention of lubricant.

Perhaps the most frequent cause of failure of a ball bearing is lack of protection against dirt and water. The highly polished surfaces of the balls and races render them peculiarly liable to corrosion and rust. Grit or dirt in a bearing is crushed between the balls and tracks, forming an abrasive compound which causes rapid wear.

For ordinary conditions it is sufficient to make the protecting end cover as wide as possible where it meets the shaft, and to bore it out with a diametric clearance of about 0·010 in. A series of annular grooves should be turned in the bore, which will become filled with grease and form grease seals.

For special conditions, where there is excessive dust or moisture present, extra protection is necessary, and throwers should be fitted on the shaft overhanging the end covers so that the rotation of the thrower will tend to throw off the dirt or water and prevent its having direct access to the aperture between the end cover and the shaft.

Where it is not possible to obtain accurate alignment between two housings, self-aligning bearings should be used, but it should be noted that these can only swivel about one centre, and will not accommodate actual lateral displacement errors, so that where there are more than two bearings per shaft the self-aligning feature is of use only to accommodate want of squareness of the housing. For self-alignment of two or more bearings mounted in a rigid housing it is essential that all the bearings swivel about one point.

Manufacture of Ball and Roller Bearings.

The success of ball and roller bearings depends upon extreme accuracy of manufacture and fitting; also on the quality of the material. All must be of the very best. The manufacture of such bearings requires great skill and special plant, hence it is the height of folly for firms not equipped for such work to attempt to make their own bearings.

Some makers favour special low-carbon steel, deeply carburised on the tracks only; others prefer high-carbon chrome alloy steel. Since highly satisfactory bearings are turned out from both kinds of steel, it is impossible to say which is the better.

For special purposes, such as to resist corrosion or chemical action, stainless steel is sometimes used, but as regards load-carrying capacity it is not equal to the standard materials.

After rough finishing, both balls and races are carefully heat treated and then finished by grinding and polishing to a very high degree of accuracy. Balls are produced to guaranteed limits, both as to diameter and sphericity, of + or — 0·0001 in. The ball tracks also have to be finished to the same degree of accuracy. The bores and outside diameters of the races of best quality bearings have a tolerance of only 0·0005 in., from which it will be evident that the bearings are to a very large extent interchangeable.

The reason why extreme degrees of accuracy in workmanship are necessary for ball and roller bearings will be apparent when it is remembered that the approach of two surfaces when separated by a ball is extremely small for safe loads, hence if there be only very small hills on either the balls or tracks the increase of load as the ball passes is very severe. Articles dealing with this question will be found in *Engineering*, Aug. 3 and 24, 1923, also June 19, 1925.

When under load the balls and tracks are deformed until there is a measurable but very small area of contact, the stresses on this area are consequently very high, which are repeated many millions of times during the life of the bearing.

Lubrication of Ball and Roller Bearings.

In a ball or roller bearing the intensity of pressure between the rolling elements and the track is such as to entirely squeeze out every trace of lubricant, and actual metal-to-metal contact occurs. This does not mean that a ball or roller bearing will run satisfactorily without any lubricant, which is necessary to lubricate the cage containing the balls or rollers, and to protect the tracks and rolling elements from corrosion. In a high-class bearing the greatest care is taken in obtaining a surface as free from imperfections and scratches as it is possible to produce commercially. This extreme care is necessary to obtain durability, a low coefficient of friction, and silence in running. Unfortunately, surfaces of this character are particularly susceptible to corrosion, and it is essential, therefore, that the lubricant be free from all solid impurities or free acid, and the free alkali should not exceed 0·1 per cent. Lubricants with an animal or vegetable base are all liable to develop free acid over a period and are therefore unsuitable.

For normal conditions a pure mineral grease, perfectly neutral, and of a consistency rather greater than that of vaseline, is the most suitable lubricant for ball and roller bearings. It must not readily absorb moisture, nor contain solidifying matter such as French chalk, graphite, etc., which would act as a lapping compound and cause rapid wear and deterioration of the bearing. At high speeds the lubricant used in a bearing is subjected to severe churning action, and unless a good quality grease be used, it will be found there is a tendency for the grease to disintegrate, the oil running out of the bearing, leaving a hard, gummy residue which clogs and finally breaks up the cage and thus destroys the bearing.

In exceptional cases where the bearing is exposed to an abnormally high temperature, grease with a high melting-point should be used. This is obtainable suitable for use up to a temperature of 300° F. Its consistency, however, is very stiff at normal temperatures, and it should therefore not be used where there is no external heat.

If arrangements can be made to prevent leakage, oil may be used for lubrication, and, indeed, at high speeds is preferable to grease. It has, however, the disadvantage that where the bearing is likely to remain stationary for any length of time the oil will drain away from the upper surfaces, leaving them exposed to the corrosive action of the atmosphere.

Provision must be made for replenishing the lubricant. This, however, is only necessary at long intervals, which vary according to the conditions under which the bearing is running. The grease may be inserted by a Stauffer or other lubricator, or by means of a grease pump, but it is of the utmost importance that the grease used, and the hole through which the grease is inserted, are perfectly clean; the latter should be kept plugged. It cannot be over-emphasised that the best of bearings can be ruined in a few days by the action of dirt or water.

SECTION XX

PART II

LUBRICATING OILS, LUBRICATION, AND LUBRICATORS.

LUBRICATING OILS.

(Contributed by Harold Moore, M.Sc.Tech., M.Inst. Pet. Tech., F.C.S., A.I.M.E., and R. B. Hobson, A.M.C.T., A.M.I. Pet. Tech.)

At the present time most machinery is lubricated either by means of mineral oils or by mixtures containing a large proportion of mineral oil with a small amount of either animal or vegetable oil added. The mineral oils used are almost entirely derived from petroleum, but a certain amount of low-viscosity oils, such as spindle oils, are prepared from shale oil, and certain tar and lignite products are used for grease manufacturing. Where coal tar and lignite are available in quantity, coal tar and lignite products are used as substitutes for petroleum products. As a general rule the petroleum product is the superior article, and it is only for exceptional purposes that coal tar or lignite products are found to possess equal values with the petroleum oils. For grease-making, however, anthracene oil, a heavy distillate of coal tar, is used to a considerable extent in this country.

Lubricating oils may be divided into three classes: (1) the straight run mineral oils, which are certain fractions obtained by the distillation of petroleum; (2) certain blended oils, which are mineral oils prepared by the mixture of two or more different varieties of mineral oil; and (3) compounded oils, which contain mixtures of mineral oil, together with either animal or vegetable oil.

Lubricating oils of very high viscosity are prepared by distilling off a large portion of the oil, and then utilising the residue as lubricant. The residues are usually dark, and are known as 'dark steam refined cylinder stock.' For very high-class oils of high viscosity it is necessary to remove a certain amount of the bituminous content of these residues, and this is effected by filtering the cylinder stock through fuller's earth, bauxite, or adsorbent clays, when the bituminous bodies are absorbed and the black, opaque cylinder stock is converted into 'filtered cylinder stock,' usually of a greenish colour and semi-transparent. Such cylinder stocks are generally mixed with the distillate to give blended oils of any desired viscosity.

The distillates used for lubrication are generally refined, either by treatment with sulphuric acid or by filtration through fuller's earth, bauxite, or clay. The characters of the lubricants obtained from the crude oil, though to some extent affected by the treatment to which the oils have been subjected, is mainly dependent upon the nature of the oil which is used in the first place for the preparation of the lubricant.

The most important properties by the estimation of which the valuation of lubricating oils is effected, are the 'specific gravity,' 'flash-point,' and 'viscosity,' and in special cases it is also necessary to determine the coke content, demulsification properties, acidity, content of saponifiable oil, and a large number of minor tests. The main importance of the specific gravity is its value as denoting the base of an oil. The 'open flash-point' is determined by heating the oil in an open cup, observing the temperature, whilst a small flame is periodically brought to bear upon the surface of the oil, the open flash-point being that temperature at which a small sheet of flame flickers over the surface of the oil and immediately goes out. If the oil is heated to a higher temperature after the observation of the open flash-point, it will eventually reach a temperature at which sufficient inflammable vapour is generated to maintain a flame on the surface of the oil. This temperature is known as the 'burning-point' or 'fire-test.' Special instruments have been devised for the determination of flash-points, in which the oil is enclosed in a small cylindrical vessel, so that the vapour is not liable to be affected by currents of air. The results given by such instruments are called 'closed flash-point' tests. The Abel instrument consists of a vessel surrounded by a water-jacket to ensure gradual increase in temperature, and is designed for the testing of kerosene (lamp oil), the flash-point of which is required by governmental regulation to be above 73° F. by the Abel test. As the water-jacketed instrument is

unsuitable for testing flash-points at high temperatures, it is usual to employ either the Gray or the Pensky-Marten instrument for the testing of lubricants. Both of these instruments are very similar to the Abel apparatus, except in that a cast-iron jacket is employed in place of the water jacket, and that stirring arrangements are provided to allow of the agitation of the oil whilst heat is being applied. In the Gray instrument a bevel-driven paddle is used, whilst the Pensky-Marten has a paddle driven by a flexible shaft.

The property of viscosity is the resistance to motion, or, in common terms, the 'thickness' of the oil. As the viscosity varies according to temperature, it is necessary that a viscosity should be determined at some fixed temperature, in this country usually at 70°, 100°, 140°, 200°, or 212° F. As the rate of change in viscosity with increase of temperature varies with different types of oil, it is often necessary to plot a curve obtained by the viscosity of oils at various temperatures. The instruments usually used for measuring this property are the Redwood, Engler, Saybolt, and Barbet viscometers. The results of determinations made on these instruments are quite different, but the principle on which they work is very much the same in each case. It consists in allowing a quantity of oil to fall through a jet or tube, and in measuring either the time taken for a fixed quantity to pass through the instrument, or in measuring the quantity passed in a fixed time. The Redwood instrument is used in this country; the Engler is very extensively used on the Continent; the Saybolt is the standard American instrument; whilst the Barbet is generally used in France.

During recent years it has been frequently suggested that measurements of viscosity should be made in absolute units instead of in arbitrary units obtained in the various commercial testing appliances. The absolute unit of viscosity is the amount of force in dynes required to move a unit area of plane surface with unit speed relative to another plane surface, from which it is separated by a layer of lubricant of unit thickness. It is usual to employ C.G.S. units, and the result is obtained in dynes per square centimetre per second. This unit has been named the 'poise.' As the absolute unit is a comparatively large unit of measurement, and would be inconvenient for commercial use, it is probable that, if absolute viscosity measurements came into general use, the unit taken would be the 'centipoise,' or one-hundredth part of the absolute unit.

Mineral oils possessing similar viscosities at any particular temperature may not possess similar viscosities at a higher or at a lower temperature, the rate of fall in viscosity with increase in temperature being dependent upon the chemical nature of the base oil. For example, a paraffin base oil does not show nearly such a rapid fall in viscosity with increase in temperature as do the asphaltic and naphthenic base oils, and to compare the viscosity of oils it is really necessary to prepare a curve by plotting the viscosity against the temperature.

Though these three tests are the ones most commonly used in commerce, it is necessary to employ a large number of other tests for various particular purposes. For example, oils required for service on internal combustion engines should be tested by the coke test, to measure their liability to produce carbon in the engine. Oils required for use in turbines should be tested to ensure that they will readily separate from water without forming an emulsion. It is also necessary to ensure that the oils are free from mineral acids, and that they do not contain excessive quantities of organic acids. When compounded oils are under examination, it is necessary to determine how much of the saponifiable oil is present. It may also be necessary to test for inorganic matter by the determination of the ash content, or for water, or for mechanical impurities.

In addition to the properties mentioned above, there is the property of 'oiliness,' on which subject a great deal of research has been conducted during the last few years. This is undoubtedly a surface tension effect, and is the property of maintaining a film when subjected to pressure between two plane surfaces. This is a property which cannot be readily measured, but its influence is well known. Certain of the saponifiable oils, that is to say, animal and vegetable oils, are much more satisfactory as regards this property than are mineral oils, and it is frequently observed that when a saponifiable oil or a compounded oil is employed for any particular purpose it is possible to use a product of lower viscosity than if a straight mineral oil were being used for the same purpose. The asphaltic and naphthenic oils are slightly superior to the paraffin base oils as regards this property.

In drafting specifications for the supply of lubricants for any particular purpose, it is essential that note should be taken of the nature of the mineral oil employed as a base. It is quite insufficient to simply specify the usual properties of gravity, flash-point, and viscosity.

The colour of a lubricant, which denotes the degree to which it has been refined, is a matter of the greatest importance. Clear oils are preferable to turbid oils, and, as a general rule, oils of light colour are more valuable than darker oils.

Owing to the use of different standard instruments for the determination of the viscosities of oils, it is sometimes necessary to be able to convert values given in one standard into the corresponding values in another standard. The three tables, p. 1103, serve for the conversion of values between the Redwood, Engler, and Saybolt systems.

Viscosity in C.G.S. units $= 0{\cdot}00353 \left(3{\cdot}6 \text{ Red.} - \frac{\delta^2}{\text{Red.}}\right)$, approximately, where 'Red.' is the Redwood value, and δ is the density. *(H. R. K.)*

MULTIPLYING FACTORS FOR THE CONVERSION OF ENGLER NUMBERS TO SAYBOLT OR REDWOOD SECONDS.

Engler Number.	Factor for Conversion to Saybolt Seconds.	Factor for Conversion to Redwood Seconds.	Engler Number.	Factor for Conversion to Saybolt Seconds.	Factor for Conversion to Redwood Seconds.	Engler Number.	Factor for Conversion to Saybolt Seconds.	Factor for Conversion to Redwood Seconds.
1·00	28·1	26·7	1·70	32·5	29·0	2·90	36·1	30·4
1·05	28·4	27·0	1·80	33·0	29·2	3·00	36·2	30·5
1·10	28·8	27·2	1·90	33·5	29·4	3·50	36·7	30·7
1·15	29·1	27·5	2·00	33·9	29·6	4·00	37·1	30·9
1·20	29·5	27·6	2·10	34·2	29·7	4·50	37·3	31·1
1·25	29·8	27·8	2·20	34·5	29·9	5·00	37·4	31·2
1·30	30·2	28·0	2·30	34·8	30·0	6·00	37·5	31·3
1·35	30·5	28·2	2·40	35·1	30·1	8·00	—	31·3
1·40	30·8	28·3	2·50	35·3	30·2			
1·45	31·1	28·5	2·60	35·5	30·3	50·00	37·5	31·3
1·50	31·4	28·6	2·70	35·7	30·3			
1·60	32·0	28·8	2·80	35·9	30·4			

MULTIPLYING FACTORS FOR THE CONVERSION OF SAYBOLT SECONDS TO ENGLER NUMBERS OR TO REDWOOD SECONDS.

Saybolt Seconds.	Factor for Conversion to Engler Number.	Factor for Conversion to Redwood Seconds.	Saybolt Seconds.	Factor for Conversion to Engler Number.	Factor for Conversion to Redwood Seconds.	Saybolt Seconds.	Factor for Conversion to Engler Number.	Factor for Conversion to Redwood Seconds.
28	0·0357	0·95	50	0·0314	0·91	110	0·0276	0·84
30	0·0352	0·95	55	0·0308	0·90	120	0·0274	0·84
32	0·0346	0·95	60	0·0302	0·89	130	0·0272	0·84
34	0·0342	0·94	65	0·0297	0·88	140	0·0271	0·84
36	0·0337	0·94	70	0·0293	0·87	160	0·0269	0·84
38	0·0333	0·93	75	0·0289	0·86	180	0·0268	0·84
40	0·0330	0·93	80	0·0286	0·86	200	0·0267	0·84
42	0·0327	0·92	85	0·0284	0·86	250	0·0267	0·84
44	0·0323	0·92	90	0·0282	0·85	300	0·0267	0·84
46	0·0320	0·92	95	0·0280	0·85			
48	0·0317	0·91	100	0·0279	0·85	1,800	0·0267	0·84

MULTIPLYING FACTORS FOR THE CONVERSION OF REDWOOD SECONDS TO SAYBOLT SECONDS OR TO ENGLER NUMBERS.

Redwood Seconds.	Factor for Conversion to Saybolt Seconds.	Factor for Conversion to Engler Numbers.	Redwood Seconds.	Factor for Conversion to Saybolt Seconds.	Factor for Conversion to Engler Numbers.	Redwood Seconds.	Factor for Conversion to Saybolt Seconds.	Factor for Conversion to Engler Numbers.
28	1·05	0·0377	50	1·13	0·0344	120	1·20	0·0324
29	1·05	0·0372	55	1·14	0·0340	130	1·20	0·0323
30	1·06	0·0368	60	1·15	0·0337	140	1·20	0·0322
32	1·06	0·0364	65	1·16	0·0335	160	1·20	0·0321
34	1·07	0·0361	70	1·16	0·0333	180	1·20	0·0320
36	1·07	0·0358	75	1·17	0·0331	200	1·20	0·0320
38	1·08	0·0355	80	1·18	0·0330	250	1·20	0·0320
40	1·09	0·0353	85	1·18	0·0329	300	1·20	0·0320
42	1·09	0·0351	90	1·19	0·0328			
44	1·10	0·0349	95	1·19	0·0327	1,500	1·20	0·0320
46	1·11	0·0347	100	1·19	0·0326			
48	1·12	0·0345	110	1·19	0·0325			

PROPERTIES OF LUBRICATING OILS.

Oil.	Sp. Gr.	Flash Pt. F°.		Viscosity Redwood Seconds at F°.					Cold Test. F°.
		Clsd.	Open.	70	100	140	200	250	
Pennsylvanian 200 neutral	0·872	415	425	374	158	74	43	–	30
Russian No. 2	0·900	355	–	405	162	69	41	–	below 0°
Polish spindle oil	0·884	305	325	110	63	44	–	–	35
Polish red Oil	0·910	385	402	525	200	82	45	–	35
Californian red oil	0·932	336	342	713	228	82	43	–	below 0°
Paraffin base red oil	0·908	408	420	655	240	99	47	–	35
Californian red oil	0·935	387	399	1390	392	121	50	–	below 0°
Russian No. 1 engine oil	0·911	380	395	1200	405	130	54	–	below 0°
Roumanian red oil	0·923	380	–	980	272	95	46	–	10
Roumanian red oil	0·935	395	–	2220	550	160	56	–	–
Polish red	0·921	430	445	1260	427	142	56	–	32
Pennsylvanian 550E filtered cylinder oil	0·894	480	–	–	–	382	115	62	55
Pennsylvanian cylinder oil	0·895	515	–	–	–	530	141	72	40
Pennsylvanian bright stock	0·891	485	–	–	2085	612	168	78	14
Pennsylvanian bright stock	0·891	510	–	–	1860	550	153	73	32
Western bright stock	0·915	540	–	–	4550	1093	234	102	30
Pennsylvanian long residuum	0·882	420	440	1740	640	225	75	–	25
Pennsylvanian 600 S.R. cylinder	0·900	505	535	–	2730	622	155	–	–
Western 600 S.R. cylinder	0·933	505	–	–	–	904	193	86	32
Russian O.O.	0·912	417	–	2610	760	220	72	–	–
Roumanian cylinder	0·952	485	–	–	–	845	150	–	10

ACTION OF OIL ON METALS.

From experiments with common metals immersed in various oils for twelve months at normal atmospheric temperatures, the following results were obtained :—

Class of Oil.	Not affected.	Slightly affected.	Moderately affected.	Much affected.
Mineral lubricating	Copper, Zinc	Brass	—	Lead
Olive	—	Tin	Lead	Copper, Brass
Rape	Brass, Tin	Iron	—	Copper
Tallow	—	Tin	Iron	Copper
Lard	—	Zinc	Lead	Copper
Cottonseed	—	Lead	—	Tin
Sperm	—	Brass, Copper	Lead	Zinc
Whale	Tin	Brass	—	Lead
Seal	—	Brass	Iron	Copper

As fatty oils have a destructive effect upon metals at high temperatures in the presence of moisture, lubricants compounded of such oils and of hydro-carbon mineral oils, although serviceable for internal combustion engines, are not desirable for the internal lubrication of steam engines. *(Mechanical World Year Book.)*

TABLE SHOWING THE PHYSICAL AND FRICTIONAL PROPERTIES OF VARIOUS STANDARD OILS COMMONLY USED FOR LUBRICATION.

(T. J. Pullin and J. Veitch Wilson.)

Lubricant.	Description.	Specific Grav. at 60° Fah.	Flash Point Fah.	Average Viscosity at				Normal Working.		Viscosity at Working Temp.
				60° F.	70° F.	120° F.	212° F.	Drag.	Temp. Fah.	
		1.	2.	3.	4.	5.	6.	7.	8.	9.
Water (distilled)	—	1,000	—	21	21	20½	boils	—	—	—
Sperm (refined) .	Southern . .	881	482	51	44½	30	24	18	189	25
,, ,, .	Arctic . . .	878	490	52	45	31	24½	18·5	194	25
Olive . . .	Gallipoli . .	916	457	97	79	38	262	26·5	202	26½
,, . . .	Smyrna . .	917	418	107	84	39	26	25	198	25½
,, . . .	Saffi . . .	915	430	89½	72	37	25½	26	202	26
Neatsfoot . .	British . .	916	458	108	83	40	26	27	208	26½
Lard Oil . .	Prime American	916	525	100	80	39	26	27·5	210	26½
Tallow Oil . .	British . .	914	456	109	87	40	26	27	204	26½
Refined Rape Oil	'Special' . .	916	535	116	93½	43	27	32	214	27
,, ,, .	German . .	914	498	114	88½	42	27	32	213	27
,, ,, .	English . .	918	506	100	82	40	27	28	207	27
Castor Oil . .	French First .	964	526	—	974	143	33	41	232	31½
Tallow (solid) .	British . .	—	502	solid	solid	43	27	—	—	—

NOTE.—The viscosities given in this table were determined by Coleman's Viscosimeter.

SPECIFICATIONS FOR LUBRICATING OILS FOR USE ON HEAVY-OIL ENGINES.*

(Diesel and other high-compression types.)

Details.	2- and 4-stroke cycle Heavy-oil Engines.	
	Specification A. 100 to 240 r.p.m.	Specification B. Above 240 r.p.m.
Specific gravity at 60° F. (water = 1)	Not exceeding . ·885	Not exceeding . ·882
Closed flash point	Not below . 400° F.	Not below . 400° F.
Viscosity in Redwood seconds :		
At 70° F. . . .	Not exceeding . 1250	Not exceeding . 700
At 200° F. . . .	Not less than . 56	Not less than . 50
Cold test, or pour point . .	Not higher than 35° F.	Not higher than 35° F.
Colour (by transmitted light) . .	Clear red	Clear red

The oil shall be a *Straight Mineral Oil.*

Method of testing : Inst. Pet. Tech. No tolerances allowed.

See also B.E.S.A. Spec. No. 210–1924.

* Issued by Diesel Engine Users' Association.

Simple Tests for Lubricating Oil.

Colour.—A good oil, before use, will appear of a clear colour, and will not separate into layers of different shades, or deposit sediment after being allowed to stand for a few days. It will not rust or corrode in any way any pieces of ordinary polished tool steel suspended in it.

Emulsion.—When mixed with warm water and shaken in a bottle the oil will soon separate entirely from the water and show no soapy emulsion. The test does not apply to marine engine oil nor to black cylinder stock oils which are blended, for obvious reasons, to produce this very emulsion.

Wax Content.—At ordinary temperatures it is a difficult matter to decide whether wax, either in the form of solid paraffin or some other substance, is or is not present. The readiest test is to cool the oil with ice, when the wax will have a tendency to separate out, and so cause the oil to appear cloudy.

Filtration of Oil.

When oil is properly filtered it can be used indefinitely without losing any of its lubricating qualities.

Recovery of Lubricating Oil.*

Settlement and Filtration Method.

Fig. 1 shows a waste oil filter made by Messrs. A. C. Wells & Co.; these filters consist of three compartments, any of which can be readily removed from the other to facilitate cleaning.

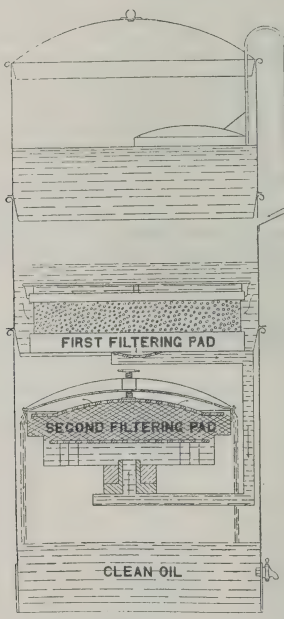


Fig. 1.

The top chamber receives the dirty oil, which after admission is allowed to settle for about a day, the time varying with the degree of foulness attained by the oil. The top chamber is

* Abstract of paper by Alfred J. Wilson, A.M.I.Mech.E., A.M.I.Pet.Tech., read before the Diesel Engine Users' Association, 21st October 1920.

provided with a floating syphon which when set in operation delivers the settled oil drop by drop into the second compartment, whence it passes by gravity through a special filter pad into the lowest compartment. The descending oil is collected off the underside of the filter pad in the second stage and led into a tube which terminates in a further filter pad. This cleans the oil by upward filtration, the recovered oil falling into the bottom of the third, or lowest, compartment, whence it is drawn off for re-use as required. By heating the oil with steam coils or electric heaters the capacity of the filter can be practically doubled.

Boiling Water Method.

Compound oils can be cleaned by passing them over boiling water.

One of the most satisfactory examples of this system is that installed at the Generating Station of the Chelsea Electricity Supply Company. The waste oil from the engines, excluding that drained from the compressors, is collected into a common tank, whence it is pumped into a larger tank placed at a height of about 8 ft. above floor level. A pipe from this elevated tank is led to a cylindrical open-top tank of about 50 gallons capacity which stands on the floor beneath. This latter tank (fig. 2) is the one used for the boiling treatment, and is filled with water to a height of 12 ins. or thereabouts. The water is raised to boiling-point by means of a steam coil immersed therein. The dirty oil is then slowly fed into the tank by gravity and the

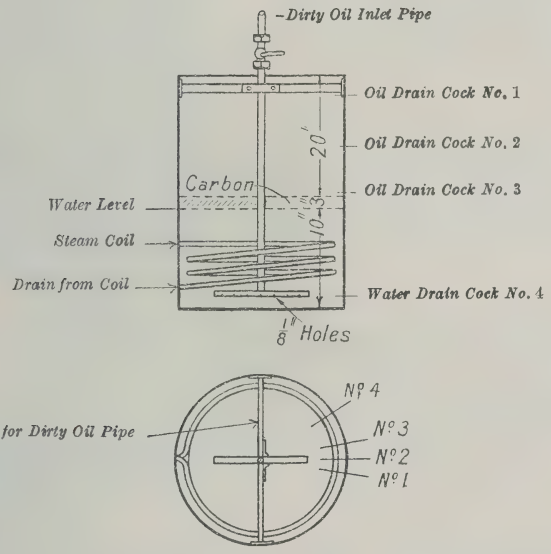


FIG. 2.

flow is regulated by a valve, so that the mixture of oil and water is maintained at a temperature which causes a yellow froth to appear on the surface. When about 30 gallons of dirty oil have been so introduced the supply is shut off and also the steam heater. Settling then takes place and is allowed to continue for some hours, the treated oil being drawn off through cocks at different levels and put into a filter, leaving a deposit of carbon particles on the top of the water. The filter to which the oil is transferred is divided into an upper and a lower compartment, the treated oil when cooled being drawn off by a floating suction pipe and passed into the lower compartment. This arrangement ensures that oil from the upper surface only of the cooled oil shall be passed into the lower compartment, from whence it is drawn off for re-use. No filtering medium is used, the carbon, etc., which has escaped removal in the boiling tank settling out in he first compartment of the filter.

Other methods are those in which :—

(1) The oil is allowed to settle either with or without warming. Objections to this method are the length of time and the large number of settling tanks required.

(2) The oil is warmed and then cleaned by chemical means.

(3) The oil is passed through a centrifugal cleaner.

LUBRICATION.

Crank Chambers.

In engines of the Willans type, in which lubrication is effected by a bath of water with a thin layer of oil on the top, it is generally safer to use a pure hydrocarbon of good body, preferably of the charcoal refined class, which represents the highest lubricative efficiency obtainable in mineral oil. It is important that the oil selected, especially if it is to be used in conjunction with hard water, should be free from tendency to emulsify, as, should this occur in the crank case, lubrication will be much impaired.

For engines lubricated by force, as in the Belliss and some others, an oil of moderate body is required, as, assuming that the bearings are in good condition, the pressure of the pump ought to provide a continual film of oil on each journal. It is also important that this oil should separate freely from water, as the formation of an emulsified magma would interfere with the circulation of the oil.

Crankshaft Bearings.

It is impossible to lay down definite rules in regard to the oil to be used on these bearings, which vary greatly in size, in speed, and in the method of applying the oil. Speaking generally, for large engines lubricated by syphons or by pipes, an oil of good body containing a percentage of fatty oil, ought to be used. It is important to note, in comparing a fatty oil with a mineral oil, that their respective efficiency cannot be determined by body or viscosity alone, as, on account of some hidden virtue in fatty oils, it will be found that a fatty oil of moderate body will reduce friction to a greater extent and will last longer than a mineral oil of much superior body, showing that body by itself as not the final criterion.

Ring Lubrication

introduces another condition apart from the direct question of the immediate efficiency of the oil. In ordinary lubrication the oil is slowly fed to the bearing at a rate calculated to make good the loss by waste or use. The oil, therefore, practically serves only once, but in the case of ring lubrication the oil may remain in use for months with only occasional slight additions to replace waste ; but during all the time that the oil is in use it is being taken up by the rings and spread over the journal and bearing in a thin film continually exposed to the action of the atmosphere. Under these conditions an animal or vegetable oil would, by the absorption of oxygen from the atmosphere, be slowly thickened and become gummy and unsuitable. Therefore, in this case, it is advisable to forgo the advantages offered by fatty oils and to select a pure hydrocarbon oil of suitable body and high boiling point which will not too readily lose bulk by evaporation or be chemically affected by the action of the atmosphere.

Notes on Ring Lubrication.

Sluggish action of the oil-rings of motors in a large New Jersey factory, leading to improper lubrication of the bearings, was cured by bevelling the sides of the rings. On testing the motors it was noticed that the rings would creep along at a very slow speed, two or three revolutions a minute on a machine of several horse-power, instead of the regular speed of about thirty to fifty revolutions. It was finally discovered that under certain conditions a ring would start with the machine and perhaps come up to normal speed, but would soon work itself towards one side of the slot and slow down, hugging closer and closer until there was only a very thin film of oil between the ring and the flat part of the journal. Three rings were removed and their sides were slightly bevelled, when the trouble entirely disappeared.

(*P. G. Bernholz*, '*Electrical World.*')

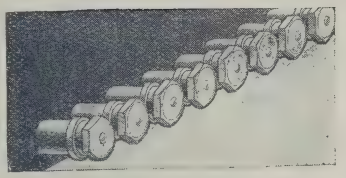
Regular inspection of the oil-rings on generators, converters, or motors should be made by the operator to determine if they are turning properly. This is especially necessary when a machine is started up, as the rings are apt to stick at this time. The fact that the oil need be

TECALEMIT

LUBRICATION SYSTEMS FOR ALL TYPES OF INDUSTRIAL MACHINERY

CENTRAL LUBRICATION

Positive Lubrication under high pressures (150-200 lbs. per sq. in.) may be effected from one central pump. By the single action of a small hand lever, oil is discharged from each main outlet of the pump to as many as 8 to 10 points. A separate discharge chamber and plunger serves each main outlet and the lubricant is controlled at the bearings by special regulator plugs. Pumps are supplied in varying sizes with any number up to five main outlets.



GROUPED NIPPLE SYSTEM

A convenient and simple method of lubricating any number of bearings from one position. The nipples are fitted on a battery plate which is easily accessible with any type of Tecalemit gun.

Lubricant is forced through special smooth bore brass piping to every bearing; there is no risk of any point being overlooked or neglected, and the lubrication of the whole machine is rendered a quick, clean and safe operation.

TECALEMIT LTD., GREAT WEST ROAD, BRENTFORD, MIDDLESEX

F. 984

THE IRONMONGER

HARDWARE AND METAL TRADES ADVERTISER.

ESTABLISHED 1859.

PUBLISHED WEEKLY

28 ESSEX STREET, STRAND, LONDON, W.C.2

For over seventy years The Ironmonger has been the leading newspaper published in the interests of buyers of hardware, metals and mill furnishings.

Its news service is invariably up-to-date and its information reliable, while its weekly record of current prices of staple commodities gives to its pages a world-wide reputation as a valuable buyers' guide for the purchasing departments of engineers, shipbuilders, collieries, etc. etc.

All this information is obtainable for the cost of a year's subscription, £1, which sum includes a copy of The Ironmonger Diary and Hardware Buyers Guide, published each December.

To anyone interested a specimen copy of The Ironmonger will be sent on application accompanied by trade card.

changed only once or twice in a year does not necessarily imply that the bearings are fool-proof and need no attention on the part of the operator. If the rings stick and fail to revolve, the bearing will get no oil. This frequently happens, resulting in an overheated bearing, and sometimes in enforced shutdown of the machine. Consequently, it is an important duty of the operator to make regular inspection of these rings to see that they are turning. If they are not, they should be started with a small hardwood stick, which should be kept on hand for this purpose. At the same time, the operator should see that the gauge glasses on bearings are at least half-full of oil, and that there is no leakage around the glasses. In case of any apparent leakage, lack of oil, or chronic sticking of any particular oil-ring, the matter should be reported to the foreman. When inspecting oil-rings on a running rotary, operators should wear goggles to protect their eyes from possible flashovers.

The rings should be turned and bored truly from extra thick commercial brass tube, and made approximately twice the diameter of the journal it is required to lubricate, and the width about one-half of the shaft diameter. As regards thickness, it has been proved by experiment that the thicker the ring, with its consequently increased weight, the more dependable is the rotation with the shaft, the oil allowing the greater surface of the heavier ring to pass through it with less increase of friction compared with the frictional drive of the heavy ring by the shaft.

Steam Turbines.

The only parts calling for lubrication in turbines are the glands and the external bearings. In the larger sizes both the glands and the bearings are provided with jackets through which water may be circulated, when this is necessary, to keep the bearings cool. In the earlier types of turbines, the glands were packed in the ordinary manner and called for the use of oil of a cylinder character to reduce friction. In modern practice the glands do not call for lubrication, as they are fitted with special metallic ring packing which does not make metallic contact with the shaft. A little water is injected and serves to condense any escaping steam at the high pressure end and the admission of air at the low pressure, or vacuum, end and also in a measure acts as a lubricant.

The lubrication of turbine bearings is usually effected by the circulation of oil by a pump which forces the oil from a tank in the base of the engine through a system of pipes to the bearings, from which it returns by other pipes to the tank through which cold water circulates in pipes to reduce the temperature of the oil.

The conditions calling for consideration in the lubrication of turbines seem to be the abnormally high temperature of the bearings, due to the transmission of heat by the shaft from the turbine, the probability that, despite most careful precautions, water may find its way into the oil, and the fact that the oil may remain in use for many months without change and with comparatively little addition of fresh oil. These conditions call for oil of considerable body, which should not be liable to emulsify in contact with water, and which should not be injuriously affected by long use and contact with air. (See also Section XXX, Part IV.)

Dynamos.

which are generally provided with ring bearings, should be dealt with on the lines indicated in 'Ring Lubrication,' p. 984.

Air Compressors.

See p. 1190.

Shafting

requires oil of moderate body, which may be either a compound oil, if the best results are required or a pure mineral oil if the strain on the bearings is not too great and the system of lubrication ensures a constant and regular supply of oil.

Abrasive Wheels.

See p. 1105.

Automatic Tools.

The high speeds adopted in tools of the turret lathe type, and the great pressure and high temperature developed at the cutting point in large automatic lathes, call for oils of superior quality to save the edge of the tool and the surface of the metal on which it operates; but

although the old-fashioned soap-suds are unsuitable, from their rusting action on the tools and on the metal of the job, it has been found that, in many cases, satisfactory results are, and may be, obtained by the use of a soluble oil, or soluble paste dissolved in water, which keeps the cutters cool and in good condition, and does not rust the job.

Automobiles.
See p. 1970.

Cutting Lubricants.
See p 1078.

Drills.
See p. 1033.

Lubrication at High Pressures.

Two examples of equipment operating under heavy bearing pressures are the boring mill and the metal press. Each is a distinct problem, since in the first the pressures are steady, while in the second they are intermittent. The first condition is more complex because of the bearing temperature. However, the pressures existing on the down-stroke of a metal press may be from 4,000 to 5,000 lbs. per sq. in. for a brief period.

For average conditions—*i.e.*, with pressure below 400 lbs. per sq. in.—a highly refined non-emulsifying straight mineral oil having a viscosity of from 180 to 200 secs. Saybolt at 100° F. will normally meet operating requirements. In certain operations, however, thrust pressures in excess of 1,000 lbs. per sq. in. may occur. Such a condition requires a straight mineral steam-cylinder oil of a viscosity of from 130 to 160 secs. Saybolt at 210° F.

See also p. 994.

Lubrication Oil Grooves.

The function of grooves is primarily to distribute oil from the source of supply over the complete length of the bearing. Wherever possible, they must be confined to the unloaded portion of the bearing; the loaded part should be left free from interruptions. It appears that an angle of about 45° to 60° must be had between the line of oil admission and the load line to allow of building up an efficient load-carrying oil-film.

An oil-groove should be well chamfered at its edges, so as to help the oil-film to develop rather than to scrape the film from the journal.

Graphite Lubrication.

Graphite, plumbago, and blacklead are mineralogically similar, but whereas blacklead often consists of a large proportion of clay and very little pure blacklead, on the other hand graphite (especially those grades prepared for very special lubricative purposes) may consist of as much as, and even more than, 99 per cent. of pure graphitic carbon.

The selection and preparation of graphite for lubrication requires the observance of two principal considerations, viz., the structure of the graphite itself and the character and amount of other matter incorporated with it. For instance, a flake graphite, comprising 95 per cent. of pure graphitic carbon having the necessary unctuousness, but mixed with 5 per cent. of hard siliceous rock or quartz, would not be satisfactory for any form of lubrication. Unfortunately, many purchases of such delcterious admixtures are made because the price is lower than, for example, a flake graphite, consisting also of 95 per cent. unctuous graphite with 5 per cent. smooth foliated micaceous matter. The latter, for many purposes, is every bit as good as a 99·8 per cent. or 99·9 per cent. pure flake graphite, and is comparatively much cheaper than grades of such high purity, but its cash value is many times that of the one containing rock or quartz (though the difference in actual cost between the 'quartz' grade and the 'soft mica' grade may only be a few pence per lb.). The damage which such an abrasive mixture will undoubtedly do to any kind of machinery is serious. Engineers should therefore exercise very particular supervision over these purchases.

As already stated, the important points are :—

(1) The structure of the graphite, and (2) the character and amount of the impurities in it.

Flake graphite is necessarily the best for lubrication, when consideration is given to the manner in which its lubricative duties are performed. It is required to associate itself and become attached to, and form part of, one or both of the rubbing surfaces, and impart an even and smooth finish to these surfaces, by filling up any tiny depressions. The way confetti clings to the floor is a good illustration in this connection of the advantage of a flat or flake structure.

In the well-known 'Foliac' brand of flake graphite (of *Graphite Products, Ltd.*), the flake structure is combined with foliations which act similarly to a pack of cards under pressure, by slithering apart and presenting a much larger smooth and slippery surface. This foliated or laminated structure is a distinct feature of these products, and it is claimed that a much lower coefficient of friction is obtained with them than with, e.g., the amorphous varieties of graphite or plumbago, whether carried to the surface in colloidal form or not. In connection with colloidal or deflocculated graphites it must be remembered that acidity, alkalinity, or other disturbing cause is necessary in order that the colloid may flocculate into particles large enough to attach themselves to bearing surfaces.

Recent examinations of representative samples of 'Foliac' flake graphites showed the following results :—

Foliac No. 1,	assayed,	96·00	per cent.	carbon.
Foliac No. 2,	,,	95·50	,,	,,
Foliac No. B1371,	,,	96·13	,,	,,
Foliac No. 100,	,,	99·85	,,	,,
Foliac No. 101,	,,	99·75	,,	,,

Because graphite has a higher specific gravity than oils (about 2·2 to, say, 0·9 or so for oil) it is impossible to permanently suspend it in ordinary lubricating oils, at any rate without special and complicated treatment; therefore, in order to convey flake lubricating graphites to bearing surfaces with certainty and regularity, suitable greases with the graphite in intimate admixture are best. Many instances are recorded where suitable flake graphite greases have outlasted other kinds (without flake graphite) five or six times and with less friction.

The Use of Graphite.

The right kind should be selected for a specific purpose. As examples, the large flake is best suited to large areas of surface, whether bearings, cylinders or valve faces; and the small flake is best for smaller areas of surface. At the same time, for intricate, fine or fast moving motions, it is well to select a specially prepared fine flake graphite for these purposes.

All surfaces in frictional contact should be well rubbed with the selected flake graphite for any particular class of machinery, before assembly. This rubbing is best done by dipping a clean oily pad in the graphite, and rubbing it well into the surfaces. The smoothness and consequent reduction of friction is most noticeable, and it is a very long time before this 'graphite skin' or polish is lost.

The removal of the flake graphite after this rubbing in can only be done by filing or grinding—in other words, to remove it, the surface or the metal itself has to be removed.

For regular use after assembly of parts, as already stated, one of the flake graphite greases is best, but occasional doses of flake graphite can be fed with the ordinary lubricating oil with most satisfactory results. It will be obvious that the large flake graphite mixed with a thin spindle oil will settle much faster than a very fine flake graphite in a bodied engine or heavy machine oil.

Too much should never be used, especially when mixed with oils—from 2 per cent. to 5 per cent, according to the service required, is usually ample. With greases, the proportion may be larger.

Graphite, particularly the flake variety (if free from abrasive matter), is an exceedingly useful and beneficial product—but it should not be used to excess, as too much is likely to give trouble.

Feeding Graphite.

In the first place, when erecting or repairing engines, or whenever the cylinders and valves are opened, the working surfaces should be freely covered with a thick mixture of graphite and cylinder oil. Flake graphite cannot satisfactorily be fed through the usual sight-feed displacement lubricators for the reason that its high specific gravity causes it to settle in the oil and to clog up the connecting pipe. A reasonable doubt might be entertained as to whether graphite can readily become attached to the frictional surface if it is so fine that it passes through the pores of filter paper, as is claimed by some, which would indicate that the particles of graphite are surrounded by an unbreakable oil film. But if these particles could become attached to the frictional surface, efficient but not lasting aid to lubrication would certainly result. Experience seems to show that there is a limit to a desirable subdivision of graphite, and if this is so the settling of graphite in cylinder lubricators must be overcome by mechanical means, since even the smallest particles of graphite which can be seen will settle in oil. There are, however, cylinder lubricators on the market for feeding graphite, the best known of which relies on continuous stirring to secure suspension of the graphite, even in the delivery pipe, by means of a spiral wire which is constantly turning.

Graphite as a Lubricant for Cylinders.

As a cylinder lubricant, flake graphite has come to be widely used, as it is quite unaffected, either chemically or physically, by any temperature encountered in a cylinder and retains its high lubricating value even when at a red heat.

Wells and Scott-Taggart, in their book 'Cylinder Oil and Cylinder Lubrication,' write concerning the oil polish as follows: 'In order to satisfy themselves as to the working of an oil, most engineers open the cylinders at intervals and note the condition of the surfaces. Satisfaction is generally felt if they find the inside of the cylinder polished like silver, or if it has a bright, smooth surface, "like a mirror," as some express it. Now, this is not by any means the condition the cylinder ought to be in, if it is lubricated with oil alone. Suppose an engineer found, on opening the cylinders of his engine, that bright streaks existed, he would conclude at once that the bright spots were the result of friction . . . Strange to say, if he found the whole surface of his cylinder polished, he would not think it was friction. A well-lubricated cylinder ought to present a rather dark-looking surface, smooth, but with no signs of bright polish, even if coated with oil. The slightest rubbing with a polishing rag would produce a bright surface on such a cylinder, thereby proving that the lack of a polish is due to the absence of friction.'

In addition to the 'oil polish' referred to above, there is the smoothing and polishing effect produced in ammonia cylinders of refrigerating machinery, the 'water polish' frequently observed in marine engines and in pump cylinders, and the 'graphite polish.' This last is different from the others, which are really friction polishes, obtained by rubbing off the roughness of the grain of the metal. Graphite polishes by filling in the grain of the metal, just as wood filler and varnish polish a surface of wood, and cannot therefore be construed as an evidence of friction.

The question of the best manner of separating graphite from exhaust steam is frequently raised by those not familiar with its use. In answer it may be said that any separator, grease extractor, or settling tank that will remove even a fair percentage of cylinder oil will remove all the graphite that goes out with the exhaust. Graphite passing into a separator is apt to adhere to the baffle plates, and it is good practice to remove and clean them from time to time.

Where measures are not taken to separate or otherwise eliminate such graphite as may leave the cylinder with the exhaust steam, certain small amounts may pass into the boiler with the feed water. Graphite in a boiler is an advantage rather than a detriment. If the feed water contains scale-forming salts, the graphite will combine with the scale to make it softer and more easily crumbled. The heat transfer will not be affected, for graphite is a very good conductor of heat. (See also 'Foliac Boiler Graphite,' p. 1627.)

Graphite as a Lubricant for Gas Engines.

An authority on gas-engine cylinder lubrication has stated that the perfect lubricant for this work should either be consumed entirely or else not at all. Flake graphite meets this last requirement fully, since the intense heat encountered in a gas-engine cylinder has no effect whatever on graphite. There are many ways of feeding graphite for gas-engine cylinders, as there are many types of gas engines, but space does not admit of a full description of the different methods. Inhalation of dry graphite from a closed container or cup attached to the inlet pipe into the gas and air mixing chamber has been successful, the pulsations due to alternate compression and reduction of pressure allowing a little graphite to feed through the hole (say $\frac{3}{32}$ in. diameter) in the bottom of the container.

Difficulty has sometimes been encountered with graphite by its short-circuiting sparking points in jump-spark ignition. This usually means that too much graphite has been used, and reducing the quantity removes the trouble.

Lubrication Troubles from Uncleaned Castings.

When a new engine or turbine is first set to work, there is certain to be trouble with bearings in perhaps a majority of cases. This is because so many constructors neglect the proper cleansing of castings. Castings are never properly cleaned of sand even externally by the still common methods of steel brushing and coke rubbing, and in steam ports and passages it is hopeless to expect cleanliness from such antiquated methods. 'Sandblasting' with chilled-steel pellets will clean a casting externally, but cannot be relied on for the internal cleansing of ports or oil receptacles. Thus it is that lubrication troubles are so usual when starting up new sets. Some engineers strongly favour the pickling of castings as the most effective way of removing sand. The castings are first roughly cleansed and fettled, and then hosed with weak hydrochloric acid. Where sand is present this spreads the acid by capillary action, and the result of a few hours of acid action is to remove all sand and leave the iron surfaces clean and free also from the hard siliceous scale which is so destructive of turning tools and other tools, especially millers, and is also most injurious to lathe-bed surfaces and other working surfaces of all machine tools. When the acid treatment has done its work, the castings are washed by warm water, and this may be dosed with soda to neutralise any unspent acid.

Note on Lubrication.

Special attention should be given to all internal parts such as box beds, which act as oil receptacles, as well as to steam and exhaust passages. The oil of any circulating system becomes dirty with use. Dirt comes over from the boilers with the steam and finds its way into turbine bearings in spite of all the constructive care to guard against this. The atmospheric dust gets in somewhere, and there may be leakage at oil coolers. There is also the microscopic metallic waste from bearings. All these sources of dirt are quite enough to call for filtration, apart from the gross carelessness which is shown by the presence of foundry sand. Oil may sometimes be at fault if it has been chosen wrongly, but more usually sand, which ought never to have found itself so far from the foundry, is to blame. Organic oils deteriorate with age owing to their property of absorbing oxygen: this causes them to become gummy or resinous, and thus to lose their lubricating quality. Mineral oils, which are not, chemically speaking, oils, do not absorb oxygen, and are free from this deterioration.

LUBRICATORS.

The V.S.M. Automatic Forced Lubrication System.

In the V.S.M. System of Lubrication (*Messrs. Vickers, Ltd.*) the oil is forced in between the shaft and its bearings at the point of maximum pressure. This is done by means of a pump, which is only about $\frac{3}{8}$-in. diameter, with a stroke of about $\frac{5}{16}$ of an inch. Preferably one such pump is provided for each bearing. The pump works in a closed circuit of oil, and the same oil used over and over again. *Whenever the pressure per square inch between the shaft and its bearing increases, due to the weight of the shaft or to external stresses, the pump automatically raises the pressure of oil to meet and overcome this resistance, and forces it through to form a positive and unfailing film between the two surfaces exposed to maximum pressure.*

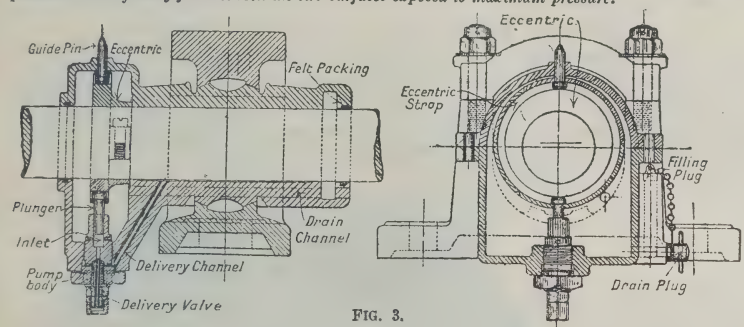


FIG. 3.

Fig. 3 shows the forced lubrication system applied to a $2\frac{1}{2}$-in. swivel bearing. The pump is surrounded by a reservoir of oil, which has the effect of flooding the inlet openings in the pump body. In this way a suction valve is dispensed with, and the pump plunger is, in effect, its own valve.

The action of the pump is as follows: When the plunger rises to the top of its stroke the inlet openings in the pump body are fully exposed, and oil from the reservoir flows in through these openings, filling the available space in the pump chamber. In making the return stroke the pump plunger gradually passes the inlet openings and finally cuts off the connection between the pump chamber and the external reservoir. The remainder of the downward stroke forces the oil imprisoned in the pump chamber past the non-return delivery valve into the delivery channel, and thence into the bearing on the surface exposed to maximum pressure. This action is repeated with every revolution of the shaft, and in large slow-running shafts the eccentric can be replaced by a cam giving two or more pump strokes per revolution as required.

Each stroke of the pump delivers a definite volume of oil to the bearing, and at the same time displaces an equal volume from the bearing. The displaced oil falls into the chambers provided at each end of the bearing, and flows back, by simple gravity only, into the reservoir surrounding the pump.

In cases where the V.S.M. System of Lubrication has to be applied to bearings and structures already in existence, and where it is impossible to incorporate the V.S.M. apparatus in the design and structure of each bearing, a multiple pump arrangement is adopted consisting of a number of units up to twelve in number. The principle of the arrangement is exactly the same as that shown in fig. 3. To ensure that each bearing gets its proper supply of oil, a separate pump is provided for it, with delivery pipe.

The multiple pump can be driven by a belt or chain, or it can be coupled up to another shaft. The eccentrics operate the pumps exactly as described. The oil flows by gravity from the reservoir past the stop valve through channels to the regulating valves. The oil drops by gravity into the suction side of the pump, and is then forced past the delivery valve on the way to the bearing to be lubricated. Glass tubes are provided for convenience to exclude dust and yet at the same time to enable one to watch the amount of oil passing to each bearing.

In the application of the V.S.M. system to an axle-box for railway rolling-stock the eccentric pump and accessories are substantially the same as those used for ordinary fixed bearings. With fixed bearings, however, the pressure on the bearing itself is usually on the under side, but in railway axle-bearings the pressure is on the top side, and in addition the bearing on the under side is entirely cut away. In consequence it is necessary to provide a short pipe, preferably of copper, to convey the oil from the pressure side of the pump to the top of the bearing at the point of maximum pressure. This arrangement permits the oil pump to be fixed at or near the bottom of the oil reservoir, ensuring an ample supply on the inlet side of the pump. The copper pipe leading from the pressure side of the pump to the top of the bearing is made long with bends, so that any relative movement between the top bearing and the pump may take place in the bends without stressing the unions at both ends of the pipe. Preferably the pump itself is designed as a separate piece from the axle-box for the purpose of manufacture, inspection, and more convenient application to the axle-box.

With a railway axle-bearing having an oil reservoir the great difficulty is to prevent the surplus oil escaping along the axle through the inner end of the axle-box, and so rapidly emptying the reservoir. Another difficulty is to prevent the dust entering through the same space and acting as an abrasive on the axle and its bearings, and causing the rapid wear of both. To overcome these difficulties a special ring has been provided to act as a dust excluder and oil retainer.

The Hulburd Valveless Force-Feed Mechanical Lubricator.

(Hulburd Patents Ltd., 26, Park Road North, Acton, London, W. 3.)

The Hulburd Valveless Force-Feed Mechanical Lubricator (for locomotive cylinders, steam chests, axle-boxes, etc.), fig. 4, is a thoroughly reliable instrument of robust construction and extreme simplicity ; it thus affords adequate guarantees as to durability and low-working costs. It is entirely without valves, either suction or delivery, and, being positive in action, it makes air locks impossible, and will start to work automatically without the preliminary hand manipulation necessary in the case of previously existing types.

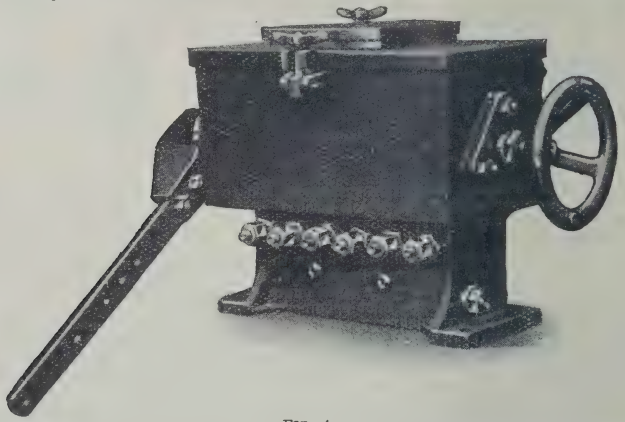


Fig. 4.

The working parts of the valveless lubricator have been reduced to the lowest possible number. As they are all self-lubricating and, being enclosed, efficiently protected from grit or dirt, long life is assured and wear reduced to a minimum.

The design of the valveless lubricator permits the use of the thickest oil or a mixture of oil and graphite or even grease, the absence of valves and restricted passages enabling it to deal with lubricants of practically any consistency ; thus the usual provisions for warming the container, being unnecessary, are entirely dispensed with.

The principle of action of the valveless lubricator is that of two opposed plungers working in a common cylinder. Of these plungers one has a continuous reciprocating motion imparted to it, and the other, normally carried with the first and partaking of its movement, is arrested and held stationary at suitable intervals so that they mutually separate from and approach each other, thereby in turn closing the inlet and outlet ports and producing the suction and discharge strokes which are thus rendered effective without valves of any description.

There being no delicate springs or valves to be put out of action by every small obstruction, and no narrow and intricate passages to become blocked, the lubricator is rendered unchokeable, and all working parts, being within the oil container, are not only effectively protected from dust and dirt, but are also continuously lubricated without extraneous attention, so that neglect under that head has not to be reckoned with, the whole being entirely automatic.

The valveless lubricator is positive in its action, and ensures a uniform rate of feed against all pressures and under all conditions; it delivers, as regulated, a definite charge of lubricant at every stroke which is forced in its entirety to the desired point.

One of these lubricators, after running the equivalent of some 40,000 miles at the rate of 60 miles per hour, and forcing oil against an average pressure of 175 lbs. per sq. in., was dismantled, and it was impossible to detect any wear on any of the working parts; the pump plungers after being measured on a micrometer were found to be exactly as they were before the test started; also during the test the amount of oil delivered was carefully measured at intervals and found to be absolutely constant throughout.

The 'Tecalemit' High-Pressure Lubrication System.

(*Tecalemit Ltd.*, 10 *Little Portland Street, London, W.* 1.)

This system consists of lubricating plugs (fig. 5) fitted with a non-return valve, which takes the place of the Stauffer type or drop-feed lubricator. These plugs can be supplied with any required male or female thread, or can be supplied for fitting to plain bored holes.

Fig. 5.—Hook-on plugs.

A 'Gun,' which is operated by hand, is used to supply the lubricant to these plugs. This consists of a lubricant receptacle (fig. 6) of $2\frac{1}{2}$, 5, 10, or 17 ozs. capacity, fitted with a handle which forces the lubricant into a small high-pressure cylinder. When the gun is applied to the plug or lubricating point, either by sliding it on in the case of the 'Automatic' type, or by simply pressing it on in the case of the 'Tecalemit Z' type, a leak-proof connection between the gun and the plug is automatically made, and, by applying further pressure, the high-pressure cylinder telescopes into a small piston fitted inside the nosepiece, and forces the lubricant under a pressure of 5,000 lbs. to the square inch through the plug and on to the bearing.

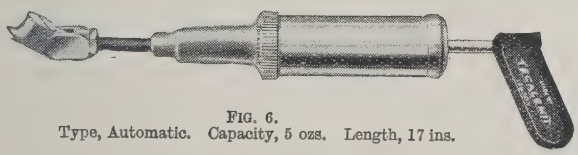


Fig. 6.
Type, Automatic. Capacity, 5 ozs. Length, 17 ins.

If there are sufficient lubricating points to justify the outlay, there is the Tecalemit transportable workshop pump, capacity 17 lbs. of lubricant, on four small wheels, or the 11 lbs. capacity knapsack pump, arranged to be strapped on the operator's back. The former type is only suitable for the Tecalemit slide-on nozzle, whilst the latter can be used with either the slide-on or the push-on nozzles. The pressure obtainable is 5,000 lbs. per square inch.

A special pattern of portable lubricating tank, capacity 100 lbs., is made, which is suitable for any type connection. With this tank a high output of lubricant under low pressure can be

obtained, or, by screwing on one of the nosepieces delivered with the apparatus, a large number of points can be lubricated under 5,000 lbs. pressure with two turns of the handle. This machine is in considerable demand for large workshops, tramway depots, etc.

Notes on Lubrication and Lubricators.

Much money is wasted and poor operating results are obtained through paying good money for good lubricants and applying them in a haphazard manner by means of unsuitable devices. A good lubricating device is almost as important as a good lubricant, and different conditions require different types of oil or grease cups. These conditions should be investigated and a proper device adopted.

In general, leaky connections on a lubricator will cause irregular feeding of the oil. Should the sight-feed glass become clogged, it may be due to setting up too tightly on the bottom nut when the glass was installed. This often causes the rubber gasket to be squeezed out of place and perhaps into the glass. If the oil-feed cone should become displaced or off centre, it may cause the globules of oil to move up the sides of the glass, and perhaps to stick there, causing the glass to fill with oil. The glass should then be removed and the cone filed down, or it will be found effective to solder a fine copper wire to the side of the cone, leading it upward through the centre of the glass, nearly to the top. The oil drop will then follow the wire, and the glass will remain clean. It is not advisable to allow a hydrostatic lubricator to run empty of oil, owing to the danger of the sight-feed and gauge glasses becoming hot, thus causing oil to stick to the sides of the glass.

SECTION XX

PART III

VISCOSITY AND VISCOUS FLOW OF OILS.

VISCOSITY OF OILS.

Oils of a particular description may vary largely in kind; 'shale' oil, for example, may vary from petrol to heavy oil, and other descriptions may also vary; the names of the oils referred to in the following table cannot therefore be taken as being characteristic of their viscosity.

TABLE I.—VISCOSITY TESTS OF VARIOUS OILS.

(Referred to in Tables II, III, and IV.)

No.	Description.	Origin.	Colour	Density (δ) at 20° C.	Redwood Viscosity Number at 20° C.
1	Shale	—	Dark brown	0·8530	8·25
2	Fuel	Burma	Dark brown	0·8890	14·8
3	Refined machine	—	Pale yellow	0·8630	21·8
4	Olive	—	Yellow	0·9130	69·3
5	Refined spindle	Russia	Pale brown	0·8955	76·3
6	Fuel	Texas	Black	0·9310	126·0
7	Engine	Russia	Brown	0·9055	255·0
8	Fuel	Roumania	Black	0·9495	579·0

TABLE II.—CHANGE OF VISCOSITY BY CHANGE OF TEMPERATURE.

No.	Percentage rate of change of viscosity per 1° C.*									
	20°	25°	30°	35°	40°	45°	50°	55°	60°	65°
1	3·2	3·0	2·8	2·6	2·5	2·4	2·4	2·4	2·5	2·6
2	4·7	4·1	3·8	3·5	3·1	2·9	2·7	2·6	2·5	2·6
3	—	4·5	3·8	3·4	3·2	3·3	3·4	3·6	3·9	4·4
4	4·7	4·4	4·2	4·0	3·7	3·4	3·2	3·2	3·5	4·1
5	7·0	5·7	5·3	5·5	5·7	—	—	—	—	—
6	—	—	6·1	5·1	4·8	4·6	4·4	4·4	4·4	4·4
7	—	—	5·6	5·4	5·9	—	—	—	—	—
8	—	—	6·5	6·1	6·2	6·7	—	—	—	—

* The values given in the table must be divided by 1·8 to obtain the percentage rate of increase of viscosity per 1° F.

TABLE III.—DENSITY (δ).

No.	Temperature ° C.						
	10°	20°	30°	40°	50°	60°	70°
1	0·8595	0·8530	0·8465	0·8395	0·8330	0·8265	0·8200
2	0·8955	0·8890	0·8825	0·8760	0·8690	0·8625	0·8560
3	0·8695	0·8630	0·8565	0·8500	0·8440	0·8375	0·8315
4	0·9195	0·9130	0·9065	0·9000	0·8935	0·8865	0·8800
5	0·9015	0·8955	0·8890	0·8825	0·8760	0·8700	0·8635
6	0·9375	0·9310	0·9245	0·9180	0·9115	0·9050	0·8985
7	0·9115	0·9055	0·8990	0·8930	0·8870	0·8810	0·8845
8	0·9560	0·9495	0·9430	0·9365	0·9300	0·9235	0·9170

TABLE IV.—VISCOSITY IN C.G.S. UNITS.*

No.	Temperature ° C.												
	10°	15°	20°	25°	30°	35°	40°	45°	50°	55°	60°	65°	70°
1	0·1075	0·0905	0·077	0·066	0·057	0·0498	0·0438	0·0387	0·0343	0·0304	0·0270	0·0238	0·0208
2	—	0·216	0·168	0·135	0·111	0·0925	0·0785	0·0675	0·0590	0·0515	0·0453	0·0400	0·0350
3	—	—	0·257	0·200	0·163	0·1365	0·116	0·0985	0·0835	0·070	0·0585	0·0475	0·0375
4	1·38	1·075	0·840	0·670	0·540	0·440	0·363	0·304	0·258	0·221	0·187	0·155	0·124
5	—	1·375	0·920	0·680	0·520	0·400	0·300	0·224	—	—	—	—	—
6	—	—	—	1·100	0·795	0·610	0·478	0·378	0·302	0·244	0·195	0·157	0·126
7	—	—	—	2·050	1·515	1·160	0·880	0·640	—	—	—	—	—
8	—	—	—	5·150	3·650	2·680	1·970	7·440	1·000	—	—	—	—

CHANGE OF VISCOSITY BY CHANGE OF PRESSURE.

Experiments made by the National Physical Laboratory have shown that at a pressure of 9,000 atmospheres, the viscosity of animal oils was about 4 times as much as at atmospheric pressure, but in the case of mineral oils the increase was as much as 16-fold.

VISCOUS AND TURBULENT FLOW.

(1)

Although there are two kinds of flow, viz. 'stream line' or 'viscous' flow, and 'turbulent' flow, for all practical conditions, heavy oils, Mexican oils, and others, follow the law governing viscous flow.

The frictional loss due to purely viscous flow is given by the equation

$$H = 0{\cdot}0066875\ \frac{TVP}{d^2}$$

where,

H = loss of head per 100 ft. of piping;
T = viscosity measured in secs. by Redwood No. 2 Viscometer;
V = velocity in ft. per sec.;
P = weight in lbs. per cub. ft.
d = diameter of pipe in ins.
(*G. & J. Weir, Ltd., Cathcart, Glasgow.*)

If

Red. = Redwood viscosity number;
W = weight in tons per hour of oil passing through pipe;
δ = oil density;

then,

$$T = 4{\cdot}895\ \frac{\text{Red.}}{\delta};\quad V = 1{\cdot}83\ \frac{W}{d^2\,\delta}\ \text{ft. per sec.};$$

* See p. 978. The 'viscosity' of a substance is measured by the tangential force on unit area of either of two horizontal planes at unit distance apart, one of the planes moving with unit velocity relatively to the other.

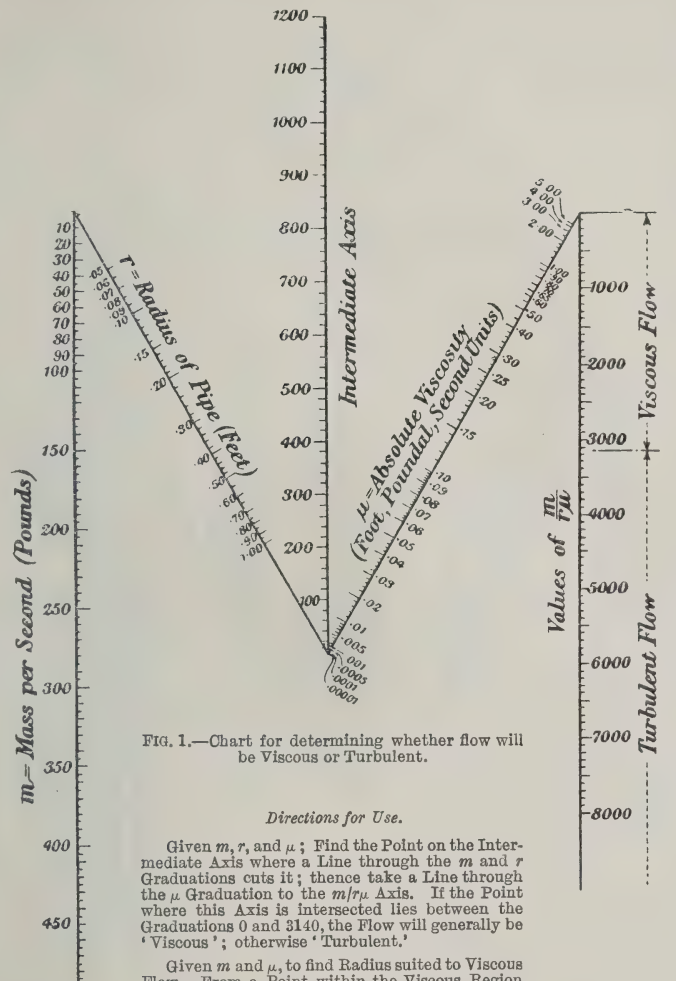


FIG. 1.—Chart for determining whether flow will be Viscous or Turbulent.

Directions for Use.

Given m, r, and μ; Find the Point on the Intermediate Axis where a Line through the m and r Graduations cuts it; thence take a Line through the μ Graduation to the $m/r\mu$ Axis. If the Point where this Axis is intersected lies between the Graduations 0 and 3140, the Flow will generally be 'Viscous'; otherwise 'Turbulent.'

Given m and μ, to find Radius suited to Viscous Flow. From a Point within the Viscous Region on the $m/r\mu$ Axis take a Line through the μ Graduation to the Intermediate Axis. A Line thence to the m Graduation will give the Radius for a Pipe in which the Flow will be 'Viscous.'

('*Engineering.*')

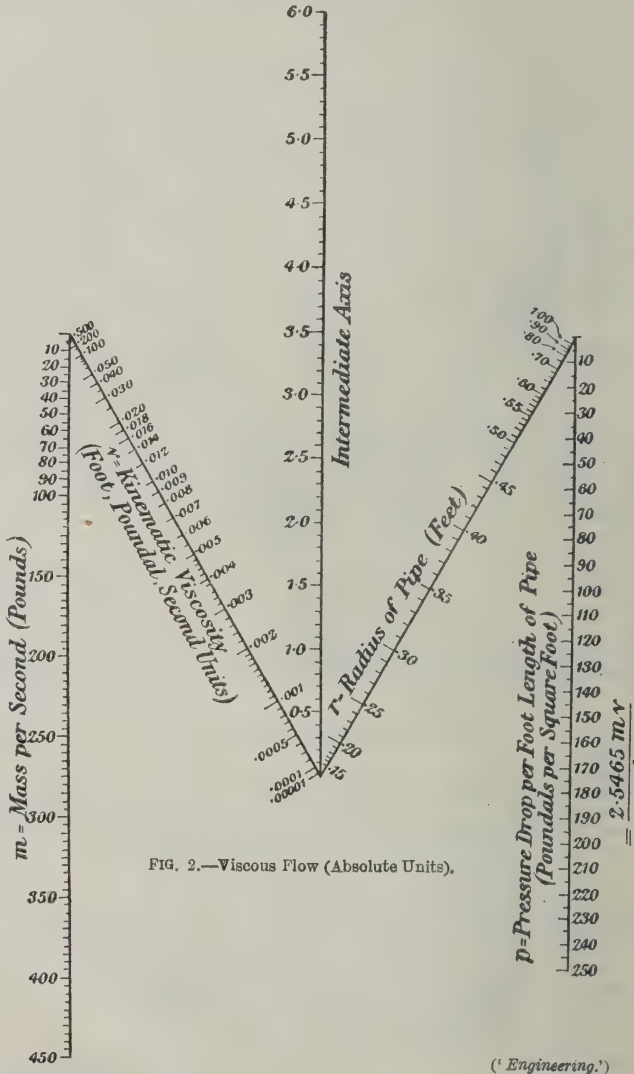


FIG. 2.—Viscous Flow (Absolute Units).

('*Engineering*.')

TURBULENT FLOW.
(ABSOLUTE UNITS.)

First Term in Lees' Equation,

$$p = \frac{\mu^{0.35}\, m^{1.65}}{55{\cdot}08\, r^{4.65}\, \rho} + \frac{{\cdot}0001824\, m^2}{r^5\, \rho}$$

= Pressure Drop in Poundals per Sq. Ft., per Lineal Ft. of Pipe = $p_1 + p_2$, say.

Add the Reading, p_1, on the Right-Hand Axis of this Chart, to the Reading, p_2, on the Right-Hand Axis of the Chart for the Second Term in the above Equation. The Sum is the Total Pressure Drop.

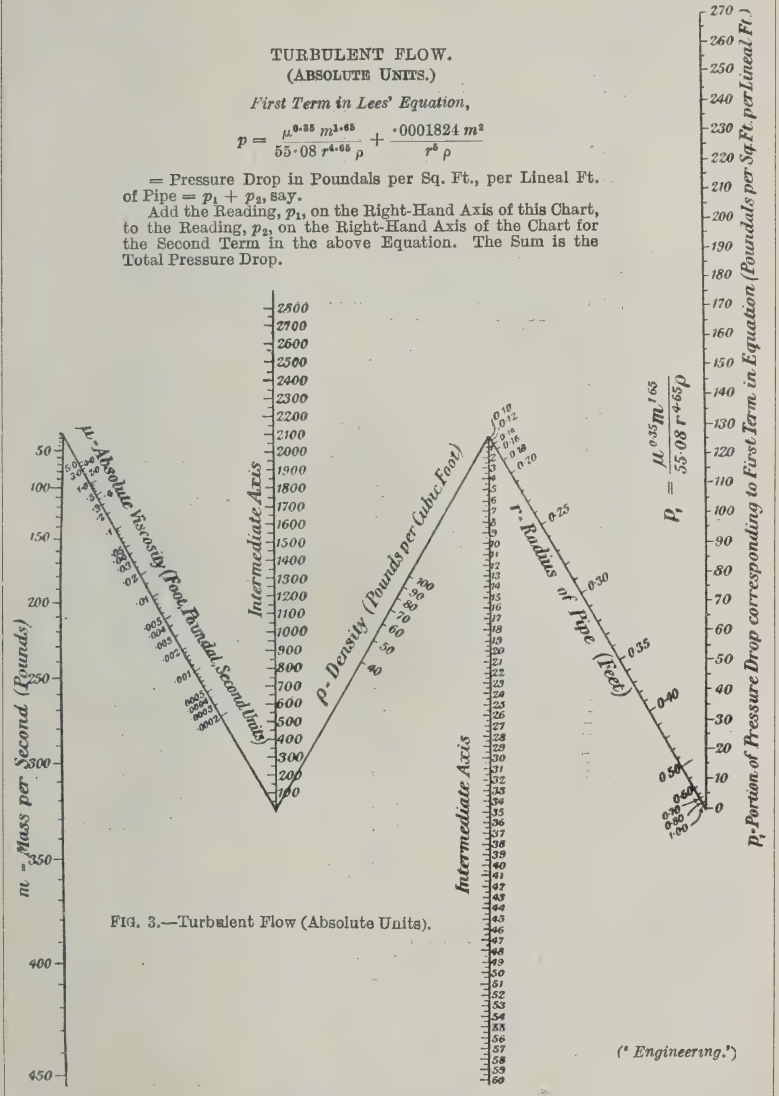


FIG. 3.—Turbulent Flow (Absolute Units).

('Engineering.')

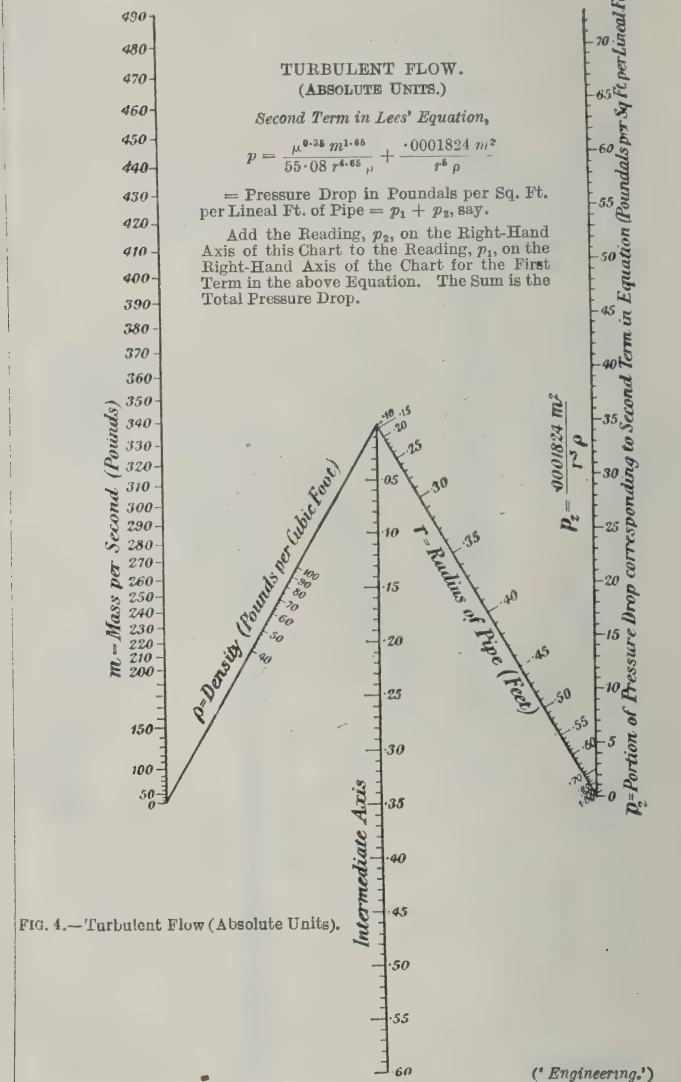


FIG. 4.—Turbulent Flow (Absolute Units).

('Engineering.')

$$H = 2{\cdot}09\ \frac{\text{Red. V.}}{d^2} = 3{\cdot}73\ \frac{\text{Red. W.}}{d^2\,\delta} \qquad (H.\ R.\ K.)$$

Example.—124 tons (W) per hour of No. 8 oil, Table I. p. 993, is to be pumped through a straight length of 9-in. diameter (d) pipe, 50 ft. long. What is the frictional loss (H) ?

From Table I., Red. = 579, $\delta = 0{\cdot}9495$, therefore,

$$H = 3{\cdot}73\ \frac{579 \times 124}{9^2 \times 0{\cdot}9495} = 3{\cdot}73\ \frac{579 \times 124}{6561 \times 0{\cdot}9495} = 43 \text{ ft.};$$

or,

$$43 \times \frac{50}{100} = 21{\cdot}5 \text{ ft. for 50 ft.}$$

See also 'The Design of Oil Fuel Pipe Lines,' by W. G. Watkins, *Engineering*, *Dec.* 19 *and* 26, 1924.

(2)

Oils of various degrees of viscosity are frequently pumped through pipes from the oilfields to ships or to distant storage tanks. To determine the power required in a given case, and to choose the most economical installation of pumping plants and pipes, bearing in mind the life of the field, it is necessary to know the frictional resistance encountered by the liquids when so transported. Such resistance will follow one or other of two laws, according to the viscosity and velocity of the liquid, and it is desirable to have at hand ready means of determining which law to apply in a given case. Having determined this, one proceeds to ascertain the magnitude of the resistance. If the flow is viscous, or, as it is sometimes called, stream-line, all the particles then moving parallel to the axis of the pipe, the resistance can be calculated in terms of the viscosity, density, and velocity. If, however, the flow is turbulent, frictional resistance cannot be found from first principles, and resort must be had to experiment, as in the case of the flow of water. Extensive experiments have been carried out with the object of discovering a law connecting viscosity, density, and velocity when the flow is of a turbulent character, and, in this connection, the work done at the National Physical Laboratory, Teddington, takes a foremost place. Various formulæ have been deduced from these experiments, and, of these, probably none is so reliable as that of Professor C. H. Lees; the charts, figs. 1, 2, 3, and 4, are based on his formula.

As the handling of the formulæ for both stream-line and turbulent flow is somewhat difficult in practice, especially when trials must be made of several different schemes, attention has been directed to the use of graphic methods for displaying the results. The alignment charts shown were prepared for the purpose by Mr. George Higgins, formerly lecturer in Civil Engineering at the University of Melbourne.

The chart shown in fig. 1 may be used for determining which of the two laws will be followed when the quantity flowing, its viscosity, and the size of the pipe are known. Directions for the use of this chart are given at its foot, and other uses to which it can be applied are also indicated. For liquids of high initial viscosity, the practice on the oilfields is to apply heat at intervals along the pipe line, and thus reduce the resistance to flow by reducing the viscosity. When this is done, the average viscosity between heating stations is estimated, and the resistance calculated for this average.

The chart fig. 2 is to be used when the chart fig. 1 indicates that the flow will be 'viscous,' *i.e.* stream-line. Given the quantity and kinematic viscosity, *i.e.* the ratio of the absolute viscosity to the density, then a straight line drawn through the axes for mass and kinematic viscosity at the appropriate graduations, will cut the intermediate axis in a certain point. From this point, a straight line drawn through the appropriate point on the radius axis will cut the right-hand axis at a graduation corresponding to the loss of pressure per unit length. It will be seen that swinging the latter line up or down about the said point on the intermediate axis as centre, will show how the adoption of one radius or another will affect the loss of pressure, and the most suitable radius can thus be more quickly found than by repeated calculations using the formula.

Charts figs. 3 and 4 are to be used when the chart fig. 1 indicates that the flow will be 'turbulent.' Here the complexity of the formula rendered it necessary to separate the factors of viscosity and density, and also to construct two charts, the results of which are to be added together. The method of using these two charts is quite similar to that explained when describing the chart fig. 2, except that the chart fig. 3 involves the drawing of three lines across the graduated axes instead of two. Their employment dispenses with much troublesome tentative calculation when the object is to determine the most suitable combination of pumps and pipes in a given case.

In practice, the lines are not, as a rule, actually drawn across the axes, a straight edge or thread being employed; a very suitable appliance for the purpose is a transparent set-square.

(Abstract—'*Engineering*,' *April* 27, 1928.)

PUMPS FOR VISCOUS LIQUIDS.

See Section XXV, Part II.

SECTION XX

PART IV

SHAFTING AND SHAFTS.

Horse-power of Shafting.

WROUGHT-IRON SHAFTING.

Wrought-iron shafting is now but seldom used. The diameter required to transmit any orse-power may be calculated from the formula :—

$$\text{Diameter of a wrought-iron shaft, in inches, at bearings} = 5\sqrt[3]{\frac{\text{Horse-power}}{\text{Number of revolutions per minute}}}$$

STEEL SHAFTING.

If D = shafting diameter ; R = revs. per min. ; H.P. = horse-power transmitted, hen,

$$\text{H.P.} = D^3 \times R \times 0{\cdot}02.$$

POWER THAT STEEL SHAFTING WILL TRANSMIT AT VARIOUS SPEEDS.

Revolutions per Minute.	Diameter in Inches.									
	1½	1¾	2	2¼	2½	2¾	3	3½	4	4½
	Horse-power Transmitted.									
50	3·3	5·3	8·0	10·9	15·6	20·8	27	43	64	91
60	4·0	6·4	9·6	13·1	18·8	25·0	32	51	77	109
70	4·7	7·5	11·2	15·2	21·9	29·1	38	60	89	128
80	5·4	8·5	12·8	17·4	25·0	33·3	43	69	102	146
90	6·0	9·6	14·4	19·6	28·1	37·4	49	77	115	164
100	6·7	10·7	16·0	21·8	31·2	41·6	54	86	128	182
110	7·4	11·8	17·6	23·9	34·4	45·8	59	94	141	200
120	8·1	12·9	19·2	26·1	37·5	49·9	65	103	154	219
130	8·7	13·9	20·8	28·3	40·6	54·1	70	111	166	237
140	9·4	15·0	22·4	30·5	43·8	58·2	76	120	179	255
150	10·1	16·1	24·0	32·6	46·9	62·4	81	129	192	273
160	10·8	17·1	25·6	34·8	50·0	66·5	86	137	205	292
170	11·5	18·2	27·2	37·0	53·1	70·7	92	146	218	310
180	12·2	19·3	28·8	39·2	56·3	74·9	97	154	230	328
190	12·8	20·4	30·4	41·3	59·4	79·0	103	163	243	346
200	13·5	21·4	32·0	43·5	62·5	83·2	108	172	256	365
225	15·2	24·1	36·0	49·0	70·3	93·6	122	193	288	410
250	16·9	26·8	40·0	54·4	78·1	104·0	135	214	320	456
275	18·6	29·5	44·0	59·8	85·9	114·4	149	236	352	501
300	20·3	32·2	48·0	65·3	93·7	124·8	162	257	384	547
325	21·9	34·8	52·6	70·7	101·6	135·2	176	279	416	592
350	23·6	37·5	56·0	76·2	109·4	145·6	189	300	448	638
375	25·3	40·2	60·0	81·6	117·2	156·0	203	322	480	683
400	27·0	42·9	64·0	87·0	125·0	166·4	216	343	512	729
425	28·7	45·6	68·0	92·5	132·8	176·8	230	364	544	775
450	30·4	48·2	72·0	97·9	140·6	187·2	243	386	576	820
475	32·1	50·9	76·0	103·4	148·4	197·6	257	407	603	866
500	33·7	53·6	80·0	108·8	156·2	208·0	270	429	640	911

POWER THAT STEEL SHAFTING WILL TRANSMIT AT VARIOUS SPEEDS—*continued.*

Revolutions per Minute.	Diameter in Inches.									
	5	5½	6	6½	7	7½	8	9	10	12
	Horse-power Transmitted.									
50	125	166	216	275	343	422	512	729	1,000	1,728
60	150	200	259	330	412	506	614	875	1,200	2,074
70	175	233	302	385	480	591	717	1,021	1,400	2,419
80	200	266	346	439	549	675	819	1,166	1,600	2,765
90	225	299	389	494	617	759	922	1,312	1,800	3,110
100	250	333	432	549	686	844	1,024	1,458	2,000	3,456
110	275	366	475	604	755	928	1,126	1,604	2,200	3,802
120	300	399	518	659	823	1,012	1,229	1,750	2,400	4,147
130	325	433	562	714	892	1,097	1,331	1,895	2,600	4,493
140	350	466	605	769	960	1,181	1,434	2,041	2,800	4,838
150	375	499	648	824	1,029	1,265	1,536	2,187	3,000	5,184
160	400	532	691	879	1,097	1,350	1,638	2,333	3,200	5,530
170	425	566	734	934	1,166	1,434	1,741	2,479	3,400	5,875
180	450	599	778	989	1,235	1,519	1,843	2,624	3,600	6,221
190	475	632	821	1,044	1,303	1,603	1,945	2,770	3,800	6,566
200	500	665	864	1,099	1,372	1,687	2,048	2,916	4,000	6,912
225	563	749	972	1,236	1,543	1,898	2,304	3,280	4,500	7,760
250	625	832	1,080	1,373	1,715	2,109	2,560	3,645	5,000	8,640
275	688	915	1,188	1,510	1,886	2,320	2,816	4,009	5,500	9,504
300	750	998	1,296	1,648	2,058	2,531	3,072	4,374	6,000	10,368
325	813	1,081	1,404	1,785	2,229	2,742	3,328	4,739	6,500	11,232
350	875	1,165	1,512	1,922	2,401	2,953	3,584	5,103	7,000	12,096
375	938	1,248	1,620	2,060	2,572	3,164	3,840	5,468	7,500	12,960
400	1,000	1,331	1,728	2,197	2,741	3,374	4,096	5,832	8,000	13,824
425	1,063	1,414	1,836	2,334	2,915	3,585	4,352	6,197	8,500	14,688
450	1,125	1,497	1,944	2,472	3,087	3,796	4,608	6,562	9,000	15,552
475	1,188	1,580	2,052	2,609	3,258	4,007	4,864	6,926	9,500	16,416
500	1,250	1,664	2,160	2,746	3,430	4,218	5,120	7,290	10,000	17,280

NOTE.

The faster a shaft revolves under the same horse-power, the less is the torsional strain upon it.

Hollow Shafts.

Hollow shafts resist torsion better than solid ones of the same area of metal.

If D = outer diameter;

d = inner diameter,

then,

$$\text{Breaking torsion force, in lbs.} = \frac{(D^4 - d^4) \times \text{Constant}}{\text{Leverage in feet} \times D^3}$$

Distance between Supports in Ordinary Shafts.

If d^2 = diameter of shaft, in inches;

l = distance between supports, in feet,

then,

$$l = 5\sqrt[3]{d^2},$$

where no power is taken off between the supports.

FRANCIS'S TABLE, WHERE POWER IS TAKEN OFF BETWEEN SUPPORTS.

Diameter of Shaft, in Inches.	Distance between Bearings, in Feet.	
	Wrought-iron Shafts.	Steel Shafts.
2	15·46	15·89
3	17·70	18·19
4	19·48	20·02
5	20·99	21·57
6	22·30	22·92
7	23·48	24·13
8	24·55	25·23
9	25·53	26·24

Where power is taken off by rigger between supports, each case should be separately considered ; 7 to 12 feet is usual.

Speed of Shafting.

Shafting for metal-cutting machines . . . 200 revolutions per minute.
" for wood-working . . . . . 250 " "
" for cotton and woollen mills . . . 300 " "

Hollow shafting has been run at 600 revolutions per minute.

DIAMETER OF AXLE, CRANK-PIN, AND SHAFT.

Diameter of Axle.

d = diameter in inches, in the bearing ; and the length of the bearing, $1{\cdot}5d$.
W = weight in pounds, acting in the bearing.

Waterwheels.

$d = \frac{\sqrt{W}}{18}$, for cast iron.

$d = \frac{\sqrt{W}}{21}$, for wrought iron.

$d = \frac{\sqrt{W}}{26}$, for steel.

Common Machinery in good order.

$d = \frac{\sqrt{W}}{24}$, for cast iron.

$d = \frac{\sqrt{W}}{28}$, for wrought iron.

$d = \frac{\sqrt{W}}{34}$, for steel.

EXAMPLE.—A waterwheel weighs 58,680 lbs., and is supported in two bearings. Required the diameter of the wheel axles. The weight acting in each bearing will be $\frac{58680}{2} = 29{,}340$ lbs., and diameter $d = \frac{\sqrt{29340}}{21} = 8{\cdot}15$ inches of wrought iron.

Diameter of Crank-pin.

$d = \sqrt{\frac{F}{1200}}$, for wrought iron. $\quad d = \sqrt{\frac{F}{1570}}$, for steel.

Length of pin $= \frac{3d}{2}$.

EXAMPLE.—The pressure on pin = 27,000 lbs. Required the diameter and length of pin of wrought iron.

$$d = \sqrt{\frac{27000}{1200}} = 4\tfrac{3}{4} \text{ inches.}$$

Length of pin $= 4\frac{3}{4} \times \frac{3}{2} = 7$ inches.

Diameter of Shaft.

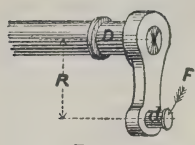


FIG. 1.

D = diameter of shaft, in inches. R = Radius of crank, in feet.
F = force from the steam piston, in pounds.

$$D = \sqrt[3]{\frac{FR}{80}}, \quad F = \frac{D^3 \times 80}{R}, \quad \text{for wrought iron.}$$

$$D = \sqrt[3]{\frac{FR}{120}}, \quad F = \frac{D^3 \times 120}{R}, \quad \text{for steel.}$$

EXAMPLE.—The force on the piston in a steam-engine is F = 45,600 lbs., applied direct on a crank of R = 3 feet radius. Required the diameter of the shaft, the latter being of wrought iron.

$$D = \sqrt[3]{\frac{45600 \times 3}{80}} = 11{\cdot}96 \text{ inches.}$$

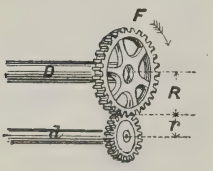


FIG. 2.

$$D = \sqrt[3]{\frac{H}{n} \times 125}, \quad \text{for wrought iron.}$$

$$D = \sqrt[3]{\frac{H}{n} \times 83{\cdot}3}, \quad \text{for steel.}$$

H = the horse-power transmitted. n = the number of revolutions per minute.

$$D : d :: \sqrt[3]{R} : \sqrt[3]{r}, \quad \text{or,} \quad d = D\sqrt[3]{\frac{r}{R}}.$$

EXAMPLE.—A cog-wheel of R = 6·5 feet radius is to gear with a pinion of r = 1·25 feet radius, and to transmit an effect of 231 horses (H) with 42 revolutions per minute (n). Required the diameter of the wheel and pinion shafts. The force F is acting uniformly at the periphery.

Diameter of wheel-shaft $D = \sqrt[3]{\frac{231}{42} \times 125} = 8{\cdot}8$ inches.

Diameter of pinion-shaft $d = 8{\cdot}8\sqrt[3]{\frac{1{\cdot}25}{6{\cdot}5}} = 5{\cdot}1$ inches.

TO DETERMINE WHAT DIAMETER OF STEEL SHAFT WILL MAINTAIN THE TWISTING MOMENT BELOW A REQUIRED LIMIT.

δ = twist or deflection in degrees; F = twisting force in lbs., applied at pin d (Fig. 1) at right angles to crank; R = radius of twisting force in inches; L = length between arms of twist in inches; D = diameter of shaft in inches, then

$$D = 0{\cdot}08347\sqrt[4]{\frac{FRL}{\delta}} = K\sqrt[4]{FRL}.$$

TABLE OF VALUES OF δ AND K.

$$\left(K = \frac{0{\cdot}08347}{\sqrt[4]{\delta}}\right)$$

δ	K.	δ	K.	δ	K.
0·10	0·1484	1·50	0·0755	5·00	0·0558
0·15	0·1341	2·00	0·0703	5·50	0·0545
0·20	0·1248	2·50	0·0665	6·00	0·0534
0·25	0·1180	3·00	0·0635	7·00	0·0514
0·50	0·0991	3·50	0·0612	8·00	0·0497
0·75	0·0896	4·00	0·0591	9·00	0·0482
1·00	0·0835	4·50	0·0574	10·00	0·0470

Example—

What diameter, D, of steel shaft would be required to bring the twisting moment below 0·25 degree (δ), the values of D, F, R, and L being

$\delta = 0{\cdot}25$; F = 4,000 lbs.; R = 6 ins.; L = 48 ins.

From table, K (corresponding to $\delta = 0{\cdot}25$) = 0·1180, therefore

$$D = 0{\cdot}1180 \sqrt[4]{4{,}000 \times 6 \times 48} = 3{\cdot}87 \text{ ins.}$$

Twisting Angle of Steel Shafts.

To find the angle of twist or deflection, δ, of a given shaft, the value of K is found from

$$K = \frac{D}{\sqrt[4]{FRL}}$$

and the corresponding value of δ in the table gives the required result.

Standard Conditions and Rules for the Shafts of Marine Steam Engines.

(Issued by the Board of Trade* and published by His Majesty's Stationery Office, Adastral House, Kingsway. Price 1*d.* net.)

(Abstract.)

MATERIALS FOR SHAFTING.

Ingot steel for shafts shall be generally of 28 to 32 tons per sq. in. standard quality and be such that after forging or rolling the sum of its tensile strength, in tons per sq. in., and percentage of elongation, measured on the British Engineering Standards Association test-pieces C, D, or E, shall be not less than 57.

The webs of built crank shafts may be forged or rolled from ingot steel, wrought iron, or scrap mild steel, or they may be steel castings.

The material of cast steel webs generally must be such that the tensile strength does not exceed 32 tons per sq. in., and the sum of the tensile strength in tons per sq. in. and the percentage of elongation measured on the British Engineering Standards Association test-pieces C, D, or E, shall be not less than 50.

RULES FOR THE DIAMETERS OF SHAFTS.

Turbine-driven Shafting.

1. Turbine-driven installations shall have shafts the diameters of which are not less than is given by the following rule:—

$$d = \sqrt[3]{\frac{\text{S.H.P.}}{R}} \times F.$$

Where d is the diameter of the intermediate shaft in inches;

S.H.P. is the maximum designed shaft horse-power;

R is the number of revolutions per minute at that power;

F is (1) 64 for ocean-going and home-trade ships, such as would require a No. 1 or No. 2 Certificate of the Board of Trade if carrying passengers. (2) 58 for ships making short coasting voyages or working on estuaries, rivers, lochs or lakes, such as would require a No. 3, No. 4, or No. 5 Certificate of the Board of Trade if carrying passengers.

Shafting of Reciprocating Engines.

2. Reciprocating steam-engine installations shall have shafts the diameters of which are not less than are given by the following rules:—

$$d = \sqrt[3]{\frac{D^2 \times S \times p}{f(r+2)}}.$$

Where d is the diameter of the intermediate shaft in inches;

D is the diameter in inches of the low-pressure cylinder, or the equivalent diameter where two or more low-pressure cylinders are used;

S is the stroke of the piston in inches;

* Accepted by Board of Trade, Lloyd's Register, British Corporation, and Bureau Véritas, at Conference for the Unification of Rules for the Construction and Dimensions of Shafting for Marine Steam Engines, April 16, 1923.

p is the working pressure in the boiler in lbs. per sq. in.;

r is the ratio of the swept volume of the low-pressure cylinder or cylinders to that of the high-pressure cylinder or cylinders;

is a coefficient from the following table, and has two values for each type of engine depending upon the nature of the service as defined in the foregoing paragraph.

Type of compound, triple or quadruple reciprocating engine.	Service of Ship.	
	No. 1 (1).	No. 1 (2).
(*a*) 2 cranks at 90°	1900	2100
(*b*) 2 cranks at 180°	1350	1500
(*c*) 3 cranks at 120°	2150	2400
(*d*) 4 cranks balanced	2150	2400
(*e*) 4 cranks at 90°	2100	2300

(2) Crank shafts shall be in diameter not less than $1{\cdot}05 \times d$.

(3) Thrust shafts transmitting torque shall be at the collars not less in diameter than $1{\cdot}05 \times d$. Thrust shafts may be tapered down outside the collars to the diameter d required for the intermediate shaft.

(4) Tube shafts shall be in diameter not less than $1{\cdot}05 \times d$, and any part of a tube shaft within the tube which may be exposed to sea-water shall be in diameter not less than $1{\cdot}075 \times d$.

(5) Wheel shafts of geared turbine-driven installations shall be in diameter not less than $1{\cdot}05 \times d$, but where there is only one pinion gearing into the wheel, or where there are two pinions which are set to subtend an angle at the centre of the shaft of less than 120° the diameter of the wheel shaft at the wheel and the adjacent journals shall be not less than $1{\cdot}1 \times d$. Abaft the journals the shaft may be tapered to $1{\cdot}05 \times d$.

Tail Shafts.

3. Tail shafts shall be not less in diameter than is given in the following rule:—

$$d_t = d + \frac{P}{K},$$

where,

d_t = diameter of tail shaft in inches;

d = diameter of intermediate shaft required;

P = diameter of screw propeller in inches;

K = 144 where a continuous liner is fitted;

K = 100 where a continuous liner is not fitted.

Tail shafts which run in stern tubes may have the end forward of the stern gland tapered down to a diameter at the coupling flange equal to $1{\cdot}05 \times d$.

Crank Webs of Built Shafts.

1. Crank webs of built shafts should have dimensions not less than the following:—

$$h = 0{\cdot}625 \times d$$

$$t = \sqrt{\frac{0{\cdot}12 \times d_c^2}{h}}$$

where,

d_c = diameter of crank shaft in inches;

h = thickness of web measured parallel to the axis;

t = thickness of metal around the eye-holes measured radially.

The Rules also relate to 'Bronze Liners on Shafts' and 'Coupling Flanges and Bolts.

British Standard Specification for Shafting for Marine Purposes.

See No. 3021—1922.

Fitting Oil Snap-Rings on Shafts.

Some shaft bearings are made with snap-on sling rings, instead of wipers, to throw the oil off the shaft at a point inside the housing, and thereby prevent its creeping out of the bearing ends along the shaft. These snap-on rings are generally slipped on the shafts before they are laid into the bearings, but it occasionally happens that the shaft must be laid in first and the

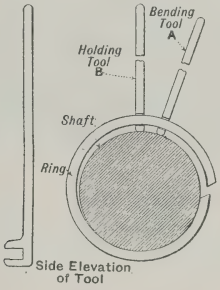


FIG. 3.

rings snapped on afterwards. To make the rings 'hug' the shafts, the tools shown in fig. 3 serve the desired purpose satisfactorily. The ring is held back with the holding tool B and then by pushing forward on the bending tool A, the ring can be made to assume a smaller circular form. The rings usually have enough temper to make them spring tightly against the shaft when the tools are removed. The tools may be made from flat stock by just filing out for the 'notch,' or they may be made from round stock by welding the grip lugs on.

(*'Power.'*)

Flexible Couplings.

Flexible couplings are now being used in all cases where it is not possible to ensure shafting being exactly in line.

Experiments over a large number of factories and works have shown that the power absorbed by shafting forms quite a large percentage of that generated. This loss is chiefly due to malalignment, and can be saved by introducing suitable flexible couplings; the same remark also applies

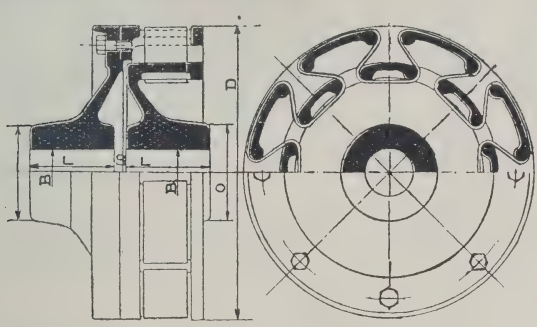


FIG. 4.

in the case of two machines whose shafts are coupled together, and apart, of course, from the actual loss of power sustained, trouble and expense is entailed due to the unnecessary stresses in shafting and bearings.

The flexible shaft coupling shown by fig. 4 is designed generally to absorb shocks and stresses, and for taking up malalignment and wear in bearings.

The coupling, fig. 4, consists of four parts :

(*a*) The outer disc, which has a suitable boss to take the shaft of the driving member.

(*b*) The belt ring, which is secured to the outer disc.

(*c*) The inner half, which has a suitable boss for taking the shaft of the driven member.

(*d*) A special cotton belt, which passes alternately around the lugs on the inner half and on the belt ring, securing together the two halves of the coupling.

The outer disc can be taken apart without removing the belt. A special advantage of the arrangement is that one shaft is electrically insulated from the other. The coupling, however, is not suitable for reversing drives.

Among the many applications of this type of coupling may be mentioned the following :—

For generators driven by steam, gas, oil, and petrol engines, or motors ; for motor-driven centrifugal fans ; for direct connected motor-driven shafting in textile and other mills ; for motor-driven rolling-mills ; for direct connection between motors and machine tools, and for connecting one length of shafting with another.

TABLE OF PARTICULARS OF FLEXIBLE SHAFT COUPLINGS.

Outside Dia.	Capacity B.H.P. R.P.M.	Max. Safe Speed in R.P.M.	Bore Diameter.		Boss Diameter.		Boss Length.	Clearance.	Overall Length.
			Max.	Min.	Max.	Min.			
D.			B.	B.	O.	O.	I.	S.	
Inches.			Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
8	0·01	1700	2	1	$3\frac{1}{2}$	3	2	$\frac{3}{4}$	$4\frac{3}{4}$
10	0·02	1350	$2\frac{1}{4}$	$1\frac{1}{2}$	4	3	$2\frac{1}{2}$	$\frac{3}{4}$	$5\frac{3}{4}$
$11\frac{3}{4}$	0·03	1160	$2\frac{1}{2}$	$1\frac{1}{4}$	$4\frac{1}{2}$	$3\frac{1}{2}$	3	$\frac{3}{4}$	$6\frac{3}{4}$
$13\frac{3}{4}$	0·05	1000	$2\frac{3}{4}$	$1\frac{3}{4}$	$5\frac{1}{4}$	4	$3\frac{1}{2}$	$\frac{3}{4}$	$7\frac{3}{4}$
$15\frac{3}{4}$	0·075	870	3	$1\frac{1}{2}$	$5\frac{1}{2}$	$4\frac{1}{2}$	4	$\frac{3}{4}$	$8\frac{3}{4}$
$17\frac{3}{4}$	0·10	770	$3\frac{1}{4}$	$1\frac{3}{4}$	6	5	$4\frac{1}{2}$	$\frac{3}{4}$	$9\frac{3}{4}$
$19\frac{3}{4}$	0·15	700	$3\frac{3}{4}$	2	$6\frac{3}{4}$	$5\frac{1}{2}$	5	$\frac{3}{4}$	$10\frac{3}{4}$
$21\frac{3}{4}$	0·27	630	$4\frac{1}{4}$	$2\frac{1}{4}$	$7\frac{3}{4}$	6	$5\frac{1}{2}$	$\frac{3}{4}$	$11\frac{3}{4}$
$23\frac{1}{2}$	0·46	580	$4\frac{3}{4}$	$2\frac{3}{4}$	9	$7\frac{1}{2}$	6	$1\frac{1}{4}$	$13\frac{1}{2}$
$25\frac{1}{2}$	0·60	540	5	3	9	$7\frac{1}{2}$	$6\frac{1}{2}$	$1\frac{1}{4}$	$14\frac{1}{2}$
$27\frac{1}{2}$	0·75	500	$5\frac{1}{4}$	$3\frac{1}{4}$	$9\frac{1}{2}$	8	7	$1\frac{1}{4}$	$15\frac{1}{2}$
$31\frac{1}{2}$	1·12	435	6	$3\frac{3}{4}$	10	$8\frac{1}{2}$	$8\frac{1}{2}$	$1\frac{1}{4}$	$17\frac{3}{4}$
35	1·60	390	$6\frac{3}{4}$	$4\frac{1}{2}$	11	$9\frac{1}{2}$	$9\frac{1}{2}$	$1\frac{1}{4}$	$20\frac{1}{4}$
39	2·13	350	$7\frac{1}{2}$	$5\frac{1}{2}$	12	10	$10\frac{3}{4}$	$1\frac{1}{2}$	23
43	2·70	320	8	$5\frac{1}{2}$	14	11	$11\frac{1}{4}$	$1\frac{1}{2}$	24
47	3·80	290	$8\frac{1}{4}$	6	$14\frac{1}{2}$	12	12	$1\frac{1}{2}$	$25\frac{1}{2}$
55	5·25	250	$9\frac{1}{2}$	7	$17\frac{1}{2}$	$16\frac{1}{2}$	13	2	28
63	8·0	220	$10\frac{3}{4}$	8	$21\frac{1}{2}$	19	14	2	30
71	10·0	190	12	9	23	$20\frac{1}{2}$	15	2	32

SECTION XX

PART V

SPRINGS.

HELICAL.

If r = mean radius of spring, in inches, measured from axis to centre of wire; d = diameter, in inches, of wire, if *round* in section; h and b = depth and breadth respectively, in inches, of wire, if *rectangular*; b = breadth and depth, if *square*; n = number of coils of which spring consists; c = coefficient or modulus of transverse elasticity of wire, in lbs. per square inch; W = any load, in lbs., not exceeding the greatest safe load; Δ = corresponding extension or compression, in inches; W_s = greatest safe *steady* load, in lbs.; Δ_s = greatest safe corresponding extension or compression, in inches; f = greatest safe shearing stress per square inch, in lbs.; then

$$\Delta = \frac{W n r^3}{c d^4} 64, \qquad \text{or,} \quad W = \frac{\Delta c d^4}{64 n r^3}, \quad \text{for } \textit{round} \text{ wire}$$

$$\Delta = \frac{W n r^3}{b^3 h^3} 6\pi, \qquad \text{or,} \quad W = \frac{\Delta c \dfrac{b^3 h^3}{b^2 + h^2}}{6\pi n r^3} \quad \text{for } \textit{rectangular} \text{ wire}$$

$$\Delta = \frac{W n r^3}{c b^4} 12\pi, \qquad \text{or,} \quad W = \frac{\Delta c b^4}{12\pi n r^3}, \quad \text{for } \textit{square} \text{ wire}$$

$$\Delta_s = \frac{f n r^2}{c d} 4\pi, \qquad \text{and,} \quad W_s = \frac{f d^3 \pi}{r\, 16}, \quad \text{for } \textit{round} \text{ wire;}$$

$$\Delta_s = \frac{f n r^2}{c \dfrac{b h}{\sqrt{b^2 + h^2}}} \left(\frac{\pi}{2}\right)^3 \frac{3}{\sqrt{2}}, \text{ and,} \quad W_s = f \frac{\dfrac{b^2 h^2}{\sqrt{b^2 + h^2}}}{r} \left(\frac{\pi}{4}\right)^2 \frac{1}{\sqrt{2}}, \quad \text{for } \textit{rectangular} \text{ wire}$$

$$\Delta_s = \frac{f n r^2}{c b} \left(\frac{\pi}{2}\right)^3 3, \qquad \text{and,} \quad W_s = \frac{f b^3}{r} \left(\frac{\pi}{4}\right)^2 \frac{1}{2}, \quad \text{for } \textit{square} \text{ wire.}$$

$$\text{Greatest safe sudden load} = \frac{W_s}{2}; \qquad \text{Resilience} = \frac{W_s \Delta_s}{2}$$

$\frac{1}{64} = \cdot0156$; $6\pi = 18\cdot85$; $\frac{1}{6\pi} = \cdot0531$; $12\pi = 37\cdot70$; $\frac{1}{12\pi} = \cdot0265$; $4\pi = 12\cdot57$;

$\frac{\pi}{16} = \cdot196$; $\left(\frac{\pi}{2}\right)^3 \frac{3}{\sqrt{2}} = 8\cdot22$; $\left(\frac{\pi}{4}\right)^2 \frac{1}{\sqrt{2}} = \cdot436$; $\left(\frac{\pi}{2}\right) 3 = 11\cdot63$; $\left(\frac{\pi}{4}\right)^2 \frac{1}{2} = \cdot308$.

The value of c for steel in lbs. per square inch may be taken as 16,000,000. The greatest safe value (f) which is certain not to impair the elasticity of the spring by frequent repetition may be taken at 50,000.

From the formula $W_s = \frac{\cdot196 f d^3}{r}$, if $f = 50{,}000$, we get,

$$d = \sqrt{\frac{W_s r}{\cdot196 f}} = \sqrt[3]{\frac{W_s 2r}{\cdot392 f}} = \sqrt[3]{\frac{W_s 2r}{19{,}600}}; \quad \text{Board of Trade rule, } d = \sqrt[3]{\frac{W_s 2r}{8{,}000}}$$

A spring with its mean diameter small as compared with the size of the bar will allow a higher stress with less set than one of larger mean diameter; size of bar being the same, a spring of small diameter may safely stand higher stress than one of large diameter; springs of too small diameter compared with size of bar are so injured in coiling as to weaken the steel; the ratio of mean diameter of spring to diameter of bar should never be less than 3. For a good grade of steel the following values have been found safe:

Ratio of Mean Diameter to Diameter of Bar.	Safe Torsional Stress.	
3	112,000	Bars less than $\frac{1}{4}$ in. in diameter.
8	85,000	
3	110,000	Bars $\frac{7}{16}$ in. to $\frac{3}{4}$ in in diameter.
8	80,000	

Ratio of Mean Diameter to Diameter of Bar.	Safe Torsional Stress.	
3	105,000	Bars $\frac{13}{16}$ in. to $1\frac{1}{4}$ in. in diameter.
8	75,000	

For extension springs the value of stress is only two-thirds these values.

(*C. H. Benjamin and R. A. French.*)

Steel Helical Springs—Round Wire.

(Contributed by R. A. Sturgeon, Assoc.M.Inst.C.E.)

Table of Safe Loads and Scales.

The stresses from which the safe loads are calculated are given for each size of wire. The scale—i.e. the ratio of load to deflection produced—is based on a single coil spring. By dividing the figure given in the table by the number of coils in the spring, the load required to extend or compress the spring 1 in. is obtained. The value of G is assumed to be 12,000,000 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Scale. Lbs. per In.
16 Gauge—0·064 in. (=$\frac{1}{16}$ in.+) diam. Safe Stress—70,000 lbs. per sq. in.		
$\frac{3}{8}$	19·2	477
$\frac{7}{16}$	16·5	301
$\frac{1}{2}$	14·4	201
$\frac{9}{16}$	12·8	141
$\frac{5}{8}$	11·5	103
$\frac{3}{4}$	9·6	59·6
$\frac{7}{8}$	8·2	37·6
1	7·2	25·2
$1\frac{1}{8}$	6·4	17·7
$1\frac{1}{4}$	5·8	12·9
$1\frac{3}{8}$	5·2	9·7
$1\frac{1}{2}$	4·8	7·5
$1\frac{3}{4}$	4·1	4·7
2	3·6	3·1
15 Gauge—0·072 in. (=$\frac{5}{64}$ in.−) diam. Safe Stress—70,000 lbs. per sq. in.		
$\frac{3}{8}$	27·4	764
$\frac{7}{16}$	23·5	481
$\frac{1}{2}$	20·5	322
$\frac{9}{16}$	18·2	226
$\frac{5}{8}$	16·4	165
$\frac{3}{4}$	13·7	95·6
$\frac{7}{8}$	11·7	60·1
1	10·3	40·3
$1\frac{1}{8}$	9·1	28·3
$1\frac{1}{4}$	8·2	20·6
$1\frac{3}{8}$	7·5	15·5
$1\frac{1}{2}$	6·8	11·9
$1\frac{3}{4}$	5·9	7·5
2	5·1	5·0
14 Gauge—0·08 in. (=$\frac{5}{64}$ in.+) diam. Safe Stress—70,000 lbs. per sq. in.		
$\frac{3}{8}$	37·5	1,165
$\frac{7}{16}$	32·2	734
$\frac{1}{2}$	28·2	491
$\frac{9}{16}$	25·0	345
$\frac{5}{8}$	22·5	252
$\frac{3}{4}$	18·8	146
$\frac{7}{8}$	16·1	91·7
1	14·1	61·4
$1\frac{1}{8}$	12·5	43·1
$1\frac{1}{4}$	11·2	31·5
$1\frac{3}{8}$	10·2	23·6
$1\frac{1}{2}$	9·4	18·2
$1\frac{3}{4}$	8·0	11·5
2	7·0	7·7
$2\frac{1}{4}$	6·2	5·4
$2\frac{1}{2}$	5·6	3·9
13 Gauge—0·092 in. (=$\frac{3}{32}$ in.) diam. Safe Stress—70,000 lbs. per sq. in.		
$\frac{3}{8}$	57	2,038
$\frac{7}{16}$	49·0	1,283
$\frac{1}{2}$	42·8	860
$\frac{9}{16}$	38·1	603
$\frac{5}{8}$	34·3	440
$\frac{3}{4}$	28·5	255
$\frac{7}{8}$	24·5	160
1	21·4	107
$1\frac{1}{8}$	19·0	75·5
$1\frac{1}{4}$	17·1	55·0
$1\frac{3}{8}$	15·6	41·3
$1\frac{1}{2}$	14·3	31·8
$1\frac{3}{4}$	12·2	20·1
2	10·7	13·4
$2\frac{1}{4}$	9·5	9·4
$2\frac{1}{2}$	8·6	6·9
12 Gauge—0·104 in. (=$\frac{7}{64}$ in.−) diam. Safe Stress—70,000 lbs. per sq. in.		
$\frac{3}{8}$	82	3,327
$\frac{7}{16}$	71	2,096
$\frac{1}{2}$	62	1,404
$\frac{9}{16}$	55	986
$\frac{5}{8}$	49·5	719
$\frac{3}{4}$	41·2	416
$\frac{7}{8}$	35·3	262
1	30·9	175
$1\frac{1}{8}$	27·5	123
$1\frac{1}{4}$	24·7	89·8
$1\frac{3}{8}$	22·5	67·5
$1\frac{1}{2}$	20·6	52·0
$1\frac{3}{4}$	17·7	32·7
2	15·5	21·9
$2\frac{1}{4}$	13·7	15·4
$2\frac{1}{2}$	12·4	11·2
11 Gauge—0·116 in. (=$\frac{7}{64}$ in.+) diam. Safe Stress—70,000 lbs. per sq. in.		
$\frac{1}{2}$	86	2,173
$\frac{9}{16}$	76	1,526
$\frac{5}{8}$	69	1,113
$\frac{3}{4}$	57	644
$\frac{7}{8}$	49·1	405
1	42·9	272
$1\frac{1}{8}$	38·2	191
$1\frac{1}{4}$	34·3	139
$1\frac{3}{8}$	31·2	104
$1\frac{1}{2}$	28·6	80·5
$1\frac{3}{4}$	24·5	50·7
2	21·5	33·9
$2\frac{1}{4}$	19·1	23·8
$2\frac{1}{2}$	17·2	17·4
3	14·3	10·1
10 Gauge—0·128 in. (=$\frac{1}{8}$ in.) diam. Safe Stress—70,000 lbs. per sq. in.		
$\frac{1}{2}$	115	3,221
$\frac{9}{16}$	102	2,262
$\frac{5}{8}$	92	1,649
$\frac{3}{4}$	77	954
$\frac{7}{8}$	66	601
1	58	403
$1\frac{1}{8}$	51	283
$1\frac{1}{4}$	46·1	206
$1\frac{3}{8}$	41·9	155
$1\frac{1}{2}$	38·4	119
$1\frac{3}{4}$	32·9	75·1
2	28·8	50·3
$2\frac{1}{4}$	25·6	35·3
$2\frac{1}{2}$	23·1	25·8
3	19·1	14·9
9 Gauge—0·144 in. (=$\frac{9}{64}$ in.+) diam. Safe Stress—70,000 lbs. per sq. in.		
$\frac{1}{2}$	164	5,160
$\frac{9}{16}$	146	3,623
$\frac{5}{8}$	131	2,642
$\frac{3}{4}$	109	1,529
$\frac{7}{8}$	94	963
1	82	645
$1\frac{1}{8}$	73	453
$1\frac{1}{4}$	66	330
$1\frac{3}{8}$	60	248
$1\frac{1}{2}$	55	191
$1\frac{3}{4}$	47·0	120
2	41·0	80·6
$2\frac{1}{4}$	36·5	56·6
$2\frac{1}{2}$	32·8	41·3
3	27·3	23·9

STEEL HELICAL SPRINGS—ROUND WIRE—*continued.*

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Scale. Lbs. per In.
8 Gauge—0·16 in. (=$\frac{5}{32}$ in.+) diam. Safe Stress—70,000 lbs. per sq. in.		
$\frac{1}{2}$	225	7,863
$\frac{9}{16}$	200	5,521
$\frac{5}{8}$	180	4,027
$\frac{3}{4}$	150	2,330
$\frac{7}{8}$	129	1,467
1	113	983
$1\frac{1}{8}$	100	690
$1\frac{1}{4}$	90	503
$1\frac{3}{8}$	82	378
$1\frac{1}{2}$	75	291
$1\frac{3}{4}$	64	183
2	56	123
$2\frac{1}{4}$	50	86·2
$2\frac{1}{2}$	45	62·9
3	37·5	36·4
7 Gauge—0·176 in. (=$\frac{11}{64}$ in.+) diam. Safe Stress—70,000 lbs. per sq. in.		
$\frac{5}{8}$	240	5,896
$\frac{3}{4}$	200	3,412
$\frac{7}{8}$	171	2,148
1	150	1,439
$1\frac{1}{8}$	133	1,011
$1\frac{1}{4}$	120	737
$1\frac{3}{8}$	109	554
$1\frac{1}{2}$	100	426
$1\frac{3}{4}$	86	268
2	75	180
$2\frac{1}{4}$	66	126
$2\frac{1}{2}$	60	92·1
3	50	53·3
$3\frac{1}{2}$	42·8	33·6
4	37·5	22·5
6 Gauge—0·192 in. (=$\frac{3}{16}$ in.+) diam. Safe Stress—70,000 lbs. per sq. in.		
$\frac{5}{8}$	311	8,351
$\frac{3}{4}$	259	4,833
$\frac{7}{8}$	222	3,043
1	195	2,038
$1\frac{1}{8}$	173	1,432
$1\frac{1}{4}$	156	1,044
$1\frac{3}{8}$	142	784
$1\frac{1}{2}$	130	604
$1\frac{3}{4}$	111	380
2	97	255
$2\frac{1}{4}$	86	179
$2\frac{1}{2}$	78	130
3	65	75·5
$3\frac{1}{2}$	56	47·5
4	48·6	31·8
5 Gauge—0·212 in. ($\frac{7}{32}$ in.+) diam. Safe Stress—70,000 lbs. per sq. in.		
$\frac{5}{8}$	419	12,413
$\frac{3}{4}$	349	7,183
$\frac{7}{8}$	300	4,523
1	262	3,030
$1\frac{1}{8}$	233	2,128
$1\frac{1}{4}$	210	1,551
$1\frac{3}{8}$	191	1,165
$1\frac{1}{2}$	175	898
$1\frac{3}{4}$	150	565
2	131	379
$2\frac{1}{4}$	116	266
$2\frac{1}{2}$	105	194
3	87	112
$3\frac{1}{2}$	75	70·7
4	65	47·3
4 Gauge—0·232 in. (=$\frac{15}{64}$ in.) diam. Safe Stress—70,000 lbs. per sq. in.		
$\frac{5}{8}$	549	17,801
$\frac{3}{4}$	458	10,302
$\frac{7}{8}$	392	6,487
1	343	4,345
$1\frac{1}{8}$	305	3,052
$1\frac{1}{4}$	275	2,225
$1\frac{3}{8}$	250	1,675
$1\frac{1}{2}$	229	1,287
$1\frac{3}{4}$	196	813
2	172	543
$2\frac{1}{4}$	152	381
$2\frac{1}{2}$	137	278
3	114	161
$3\frac{1}{2}$	98	101
3 Gauge—0·252 in. (=$\frac{1}{4}$ in.) diam. Safe Stress—70,000 lbs. per sq. in.		
$\frac{3}{4}$	587	14,341
$\frac{7}{8}$	503	9,030
1	440	6,049
$1\frac{1}{8}$	391	4,249
$1\frac{1}{4}$	352	3,097
$1\frac{3}{8}$	320	2,327
$1\frac{1}{2}$	293	1,792
$1\frac{3}{4}$	251	1,129
2	220	756
$2\frac{1}{4}$	195	531
$2\frac{1}{2}$	176	387
3	147	224
$3\frac{1}{2}$	126	141
4	110	94·5
2 Gauge—0·276 in. (=$\frac{9}{32}$ in.—) diam. Safe Stress—67,900 lbs. per sq. in.		
$\frac{3}{4}$	746	20,636
$\frac{7}{8}$	639	12,993
1	559	8,704
$1\frac{1}{8}$	497	6,113
$1\frac{1}{4}$	447	4,457
$1\frac{3}{8}$	407	3,343
$1\frac{1}{2}$	373	2,579
$1\frac{3}{4}$	319	1,624
2	279	1,088
$2\frac{1}{4}$	248	764
$2\frac{1}{2}$	223	557
3	186	322
$3\frac{1}{2}$	160	203
4	140	136
1 Gauge—0·3 in. (=$\frac{19}{64}$ in.+) diam. Safe Stress—66,000 lbs. per sq. in.		
$\frac{7}{8}$	799	18,137
1	699	12,150
$1\frac{1}{8}$	622	8,534
$1\frac{1}{4}$	559	6,221
$1\frac{3}{8}$	509	4,674
$1\frac{1}{2}$	466	3,600
$1\frac{3}{4}$	400	2,267
2	349	1,519
$2\frac{1}{4}$	310	1,067
$2\frac{1}{2}$	279	778
3	233	450
$3\frac{1}{2}$	200	283
4	175	190
$4\frac{1}{2}$	155	133
0 Gauge—0·324 in. (=$\frac{21}{64}$ in.—) diam. Safe Stress—64,100 lbs. per sq. in.		
1	860	16,530
$1\frac{1}{8}$	765	11,610
$1\frac{1}{4}$	688	8,463
$1\frac{3}{8}$	626	6,359
$1\frac{1}{2}$	574	4,898
$1\frac{3}{4}$	492	3,084
2	430	2,063
$2\frac{1}{4}$	382	1,451
$2\frac{1}{2}$	344	1,058
3	287	612
$3\frac{1}{2}$	246	386
4	215	258
$4\frac{1}{2}$	191	181
5	172	132
$5\frac{1}{2}$	156	99
00 Gauge—0·348 in. (=$\frac{11}{32}$ in.+) diam. Safe Stress—62,200 lbs. per sq. in.		
1	1,033	22,000
$1\frac{1}{8}$	918	15,451
$1\frac{1}{4}$	826	11,264
$1\frac{3}{8}$	751	8,462
$1\frac{1}{2}$	688	6,518
$1\frac{3}{4}$	590	4,105
2	516	2,750
$2\frac{1}{4}$	459	1,931
$2\frac{1}{2}$	413	1,408
3	344	815
$3\frac{1}{2}$	295	513
4	258	344
$4\frac{1}{2}$	229	241
5	206	176
$5\frac{1}{2}$	188	132
000 Gauge—0·372 in. (=$\frac{3}{8}$ in.) diam. Safe Stress—60,000 lbs. per sq. in.		
1	1,215	28,725
$1\frac{1}{8}$	1,080	20,175
$1\frac{1}{4}$	972	14,721
$1\frac{3}{8}$	883	11,050
$1\frac{1}{2}$	810	8,511
$1\frac{3}{4}$	694	5,360
2	607	3,591
$2\frac{1}{4}$	540	2,522
$2\frac{1}{2}$	486	1,839
3	405	1,064
$3\frac{1}{2}$	347	670
4	304	449
$4\frac{1}{2}$	270	315
5	243	230
$5\frac{1}{2}$	221	173
0000 Gauge—0·4 in. (=$\frac{13}{32}$ in.—) diam. Safe Stress—58,000 lbs. per sq. in.		
$1\frac{1}{4}$	1,297	26,970
$1\frac{1}{4}$	1,167	19,661
$1\frac{3}{8}$	1,061	14,772
$1\frac{1}{2}$	973	11,378
$1\frac{3}{4}$	834	7,165
2	730	4,800
$2\frac{1}{4}$	648	3,371
$2\frac{1}{2}$	584	2,458
3	486	1,422
$3\frac{1}{2}$	417	896
4	365	600
$4\frac{1}{2}$	324	421
5	292	307
$5\frac{1}{2}$	265	231
6	243	178

STEEL HELICAL SPRINGS—ROUND WIRE—*continued.*

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Scale. Lbs. per In.	Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Scale. Lbs. per In.	Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Scale. Lbs. per In.
00000 Gauge—0·432 in. (= $\frac{7}{16}$ in.—) diam. Safe Stress—55,400 lbs. per sq. in.			000000 Gauge—0·464 in. (= $\frac{15}{32}$ in.—) diam. Safe Stress—53,700 lbs. per sq. in.			0000000 Gauge—0·5 in. (= $\frac{1}{2}$ in.) diam. Safe Stress—50,000 lbs. per sq. in.		
$1\frac{1}{8}$	1,559	36,692	$1\frac{1}{8}$	1,686	35,599	$1\frac{1}{8}$	1,782	36,063
$1\frac{1}{4}$	1,403	26,749	$1\frac{1}{4}$	1,533	26,746	$1\frac{1}{4}$	1,634	27,784
$1\frac{3}{8}$	1,275	20,096	$1\frac{3}{8}$	1,405	20,601	$1\frac{3}{8}$	1,400	17,493
$1\frac{1}{2}$	1,169	15,479	$1\frac{1}{2}$	1,204	12,973	2	1,225	11,179
$1\frac{3}{4}$	1,002	9,748	2	1,054	8,691	$2\frac{1}{4}$	1,089	8,230
2	877	6,530	$2\frac{1}{4}$	937	6,104	$2\frac{1}{2}$	980	6,000
$2\frac{1}{4}$	779	4,586	$2\frac{1}{2}$	843	4,450	3	817	3,472
$2\frac{1}{2}$	701	3,344	3	703	2,575	$3\frac{1}{2}$	700	2,187
3	584	1,935	$3\frac{1}{2}$	602	1,622	4	612	1,465
$3\frac{1}{2}$	501	1,218	4	527	1,086	$4\frac{1}{2}$	545	1,029
4	438	816	$4\frac{1}{2}$	468	763	5	490	750
$4\frac{1}{2}$	390	573	5	422	556	$5\frac{1}{2}$	445	564
5	351	418	$5\frac{1}{2}$	383	418	6	408	434
$5\frac{1}{2}$	319	314	6	351	322	7	350	273
6	290	242	7	301	203	8	306	183
			8	263	136	9	272	129

Multiplying Constants.

1. For compression springs with little variation of load multiply *Safe Loads* by 1·25.

2. For Springs, especially tension springs subjected to considerable variation of load suddenly applied, multiply *Safe Loads* by 0·67.

3. For Square Wire Springs where the Side equals the diameter given in the table—multiply

Safe Load by 1·06
Scale by 1·5

4. For Brass Wire Springs (round) multiply the Scale by 0·42. (Only for *Hard* Brass.)

Note.—In getting out Compression Springs allow for the fact that one coil at each end is idle. These two coils do not affect the Scale, but do affect the closed length.

PHOSPHOR-BRONZE SPRINGS.

The formulæ adapted for phosphor-bronze wire such as is manufactured by the Phosphor-Bronze Smelting Company, of Philadelphia, are as follows :—

$$W = \frac{7{,}000\, d^3}{D - d}; \quad F = \frac{8\, P\, (D-d)^3}{6{,}200{,}000\, d^4}.$$

When P = W, then

$$F = \frac{7\, (D-d)^2}{775\, d}$$

W = carrying capacity, in lbs. ; d = diameter of wire, in ins. ; D = outside diameter of coil in ins. ; F = deflection in one coil, in ins. ; P = load, in lbs.

These formulæ for springs of all diameters, ranging from d = No. 10 B. & S. gauge to $d = \frac{5}{8}$ in., are accurate within 10 per cent. They apply equally well to tension springs wound closely without initial tension.

(*H. R. Gilson.*)

CLOCK TYPE SPRINGS.

(1)

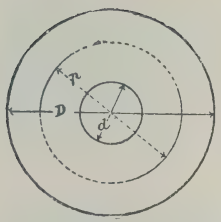


FIG. 1.

Let
d = diameter of inmost coil = spindle + t ; D = diameter of outer coil = barrel − t ; b = breadth of spring steel ; t = thickness of spring steel ; L = total length ; I = moment of inertia of section ; n = number of coils when unwound ; n' = number of coils when wound ; s = turns from unstrained state ; E = coefficient of elasticity ; a = total angle of wind ; p = diameter of inner coil unwound, or of outer coil wound ; M = moment of spring in inch pounds.

For D = 3d the following obtains :—

$$tn = \cdot1273\,D; \qquad d = 0\cdot33\,D;$$

$$t = \frac{\cdot0787\,D}{n'-n}; \qquad n = \frac{\cdot1273\,D}{t}; \qquad n' = \frac{\cdot206\,D}{t};$$

$$I = \frac{bt^3}{12}; \qquad L = \frac{\cdot3491\,D^2}{t}; \qquad \text{or} \quad L = n \times 2\cdot7416\,D; \qquad a = 2\pi s;$$

$$M = \frac{Ebt^3a}{12L}; \qquad D = \sqrt{\frac{\cdot000859\,M\,(n'-n)^4}{bs}}; \qquad p = \sqrt{\tfrac{1}{2}(D^2+d^2)}.$$

d = usually 30t. For any spiral, $L = \frac{N}{2}(C+c)$, C and c being the circumferences of extreme coils. Maximum practicable, D = 160t.

(2)

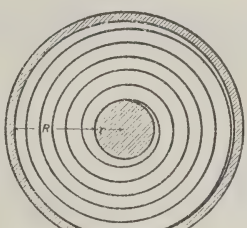


FIG. 2.

l = length of spring ; b = breadth of spring : = thickness of spring : each in inches. E = Young's modulus of elasticity in lbs. per sq. in. ; F = safe tensile strength of material in lbs. per sq. in. ; N = number of turns spring is capable of taking. R = radius of box containing spring ; r = radius of arbor : each in inches. T = maximum torque of spring.

$$= \frac{F}{E} \cdot \frac{(R+r)(R+3r)}{R-r}; \qquad b = \frac{6E^2}{F^3} \cdot \frac{T(R-r)^2}{(R+r)^2(R+3r)^2};$$

$$l = \frac{\pi E}{4F} \cdot \frac{(R-r)^2}{R+r}; \qquad N = \frac{E}{4F} \cdot \frac{(R-r)^3}{(R+r)^2(R+3r)}.$$

For steel, E = 32,000,000 ; F = 224,000 ;

therefore

$$t = \cdot007\,\frac{(R+r)(R+3r)}{R-r}, \qquad b = \cdot547\,\frac{T(R-r)^2}{(R+r)^2(R+3r)^2},$$

$$= 112\cdot2\,\frac{(R-r)^2}{R+r}. \qquad N = 35\cdot7\,\frac{(R-r)^3}{(R+r)^2(R+3r)}.$$

(*J. St. Vincent Pletts :* '*The Engineer*,' *July* 18, 1913.)

VOLUTE FLAT SPRINGS.

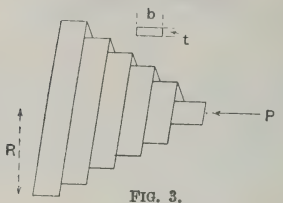


FIG. 3.

If

P = carrying capacity of spring in lbs. = greatest safe *steady* load; S = safe stress in bending, in lbs. per sq. in.; b = breadth of spring metal; t = thickness of spring metal; R = radius of spring at base; L = length of spring; G = torsional modulus of elasticity; S_s = safe shearing stress in lbs. per sq. in.; Δ = deflection of spring in ins.; C for various ratios of $\frac{b}{t}$ is as follows:

$\frac{b}{t}$	1	2	4	8
C	3·56	3·50	3·35	3·21

then,

$$P = \frac{2Sbt^2}{9R}\ ;\quad \Delta = \tfrac{1}{2}CR^2L\frac{(b^2+t^2)P}{Gb^3t^3} = \tfrac{1}{2}CRL\frac{(b^2+t^2)S_s}{Gb^3h},$$

the torsional modulus of elasticity for steel being 12,600,000.*

See also 'Kent's Mechanical Engineers' Handbook.' (New York: John Wiley & Sons, Inc.; London: Chapman and Hall, Ltd.)

CONICAL ROUND WIRE SPRINGS.

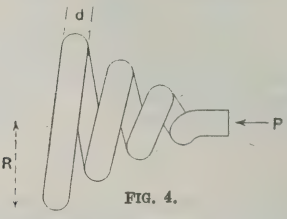


FIG. 4.

If

d = diameter of wire; H = solid height of spring, assumed to equal d × number of coils, then,

$$P = \frac{\pi Sd^3}{16R} = \cdot 196\frac{S_sd^3}{R};$$

$$\Delta = \frac{16PLR^2}{\pi Gd^4} = \frac{RLS_s}{Gd};$$

$$L = 1\cdot 571\frac{H}{d}(D_1 + D_2).$$

If

D_1 = mean diameter of largest coil; D_2 = mean diameter of smallest coil; p = average horizontal pitch of coils; d = diameter of wire,

then,

$$\Delta = 0\cdot 00332\frac{D_1^3 - D_2^3}{pd};\quad P = 31{,}400\frac{d^3}{D_2},$$

the torsional modulus of elasticity for steel being 12,600,000.*

See also 'Machinery Handbook.' (New York: The Industrial Press; London: The Machinery Publishing Co., Ltd.)

* See Semi-Steel, p. 184.

Heat Treatment of Springs.

Experiments made by the Automatic and Electric Furnaces, Limited, show that springs heated to 800 degrees and afterwards tempered to 350 degrees give the best results. Overheating is shown to be detrimental; springs which are heated to 900 degrees and quenched, then reheated to 800 degrees and tempered at 350 degrees, show that the bad effects of overheating are not neutralised by reheating to 800 degrees and requenching. These effects can be neutralised by thoroughly annealing after the first overheating and then by reheating to the correct temperature and quenching.

A spring that has not been overheated, and that has been correctly tempered, can be loaded up to within 5 per cent. of its breaking point without showing any set after the first loading.

Handy Kink for Making Spiral Springs.

Take a piece of fibre about 2 by 6 ins. and bore four or five holes in one end, ranging in size from $\frac{1}{8}$ to $\frac{3}{8}$ in., as in fig. 5. Near the large holes bore several small ones, say $\frac{1}{16}$ in. in diameter. By using nails or pieces of small rod, any size spring can be made, according to the size of the holes.

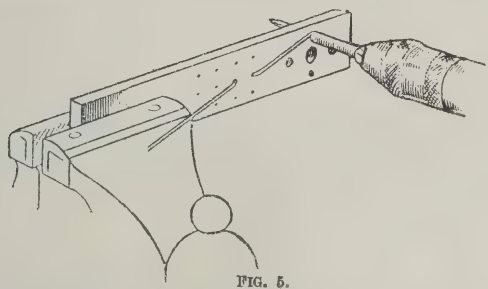


FIG. 5.

Put the block in a vice and thread the spring wire through the holes to give it tension. Take a nail the size of the spring you want to make, cut the head off, and put it into the hole where it fits best. Wrap a few turns of the wire around the nail, and grip the wire and nail in an ordinary brace, as in the figure. Then simply wind the wire around the nail until the desired length of spring is obtained. (*J. W. Fager, 'Power.'*)

Winding a Compression Spring.

Small music wire compression springs can be easily made by the following method. Having selected a mandrel A, fig. 6, the exact diameter of the interior of the desired spring, and wire B, obtain a piece of hard wood C and bore a hole in it in which the mandrel will fit, as shown in the illustration. Within $\frac{1}{32}$ inch from this hole bore another hole, the size of the wire to be used for the spring. Place the mandrel in the chuck of the lathe, slip the piece of wood over the mandrel, and insert the wire in the smaller hole, pushing it through until its end catches in the jaws of the chuck. Now press the piece of wood against the jaws of the chuck and commence winding the spring, first by pulling on the lathe belt, and then by throwing on the power when the coiling operation is well started. During the winding operation, the piece of wood should be pressed firmly against the coils so that the spring will have the appearance of being close wound. When the completed spring is removed from the mandrel, it will elongate or expand, thus forming a compression spring that is the size of the mandrel on which it is wound.

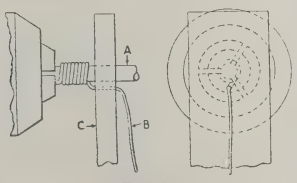


FIG. 6.

If, instead of leading the wire through a small hole as described, a notch is cut in the hole which fits over the mandrel, and the wire fed through this, the inside diameter of the spring will be smaller than the outside diameter of the mandrel. To make the spring a loose fit on the mandrel, the wire should be led through a small hole located at a greater distance from the large hole. If the spring is straightened out by pulling it off from the mandrel, the wire will be badly kinked, but if re-wound by the

methods described, a good spring will be produced. To remove the strains in the wire spring that result from the winding operation, it should be heated carefully, if made from steel, until it is a uniform blue colour. If the spring is made of phosphor-bronze or brass wire, it should be covered with tallow or beeswax and heated until it smokes; this will remove the strains, so that it will maintain its form. (*J. H. B.*, '*Machinery*.')

See also 'Researches on Springs,' 3.—Torsional Fatigue Tests on Spring Steels. Department of Scientific and Industrial Research—Engineering Research; Special Report, No. 9. Price 1s. net. Postage extra. His Majesty's Stationery Office; London, Adastral House, Kingsway, W.C. 2.

SECTION XXI

(Contributed and revised by Professor Dempster Smith, M.B.E., M.Sc.Tech., M.I.Mech.E.)

PART I

MACHINE TOOLS — BROACHING — DRILLING — MAGNETIC CHUCKS — POWER REQUIRED BY MACHINE TOOLS — TOOL STEEL — TOOLS — SAVING WORN PARTS — HARDNESS OF MATERIALS (pp. 1019–1102)

PART II

GRINDING, ABRASIVES, AND POLISHING APPLIANCES (pp. 1103–1124)

(Revised and amplified by Alfred B. Searle)

PART III

STEAM AND POWER HAMMERS (pp. 1125–1136)

SECTION XXI

PART I

MACHINE TOOLS — BROACHING — DRILLING — MAGNETIC CHUCKS—POWER REQUIRED BY MACHINE TOOLS—TOOL STEEL — TOOLS — SAVING WORN PARTS — HARDNESS OF MATERIALS.

(Revised and amplified by Professor Dempster Smith, M.B.E., M.Sc.Tech., M.I.Mech.E.)

METAL CUTTING MACHINES.

LATHES.

With a cutting angle of 80° for cast iron (tool angle 75°) and of 70° for soft steel (the best cutting angles in practice), the horse-power needed at the point of the tool is about 2 h.p. per pound of chips per minute. To this must, of course, be added the power needed to drive the tool light, which, in the case of light cuts, is a large proportion of the whole, and is the main reason why a heavy cut and a slow speed are more economical of power than a light cut at a higher one. Where the chips come off in the same shape and fashion, and the tools are equally sharp, the net horse-power expended in the actual cutting operation is practically the same per pound of metal removed whether a light or a heavy cut is taken. The power to be provided will, of course, depend upon the heaviest cut which the tool is to take. This will generally be heavier in a turret lathe than in a common lathe operating on work of the same diameter.

For the common lathe the heaviest cut is $\frac{1}{40}$th of the height of the centres in depth, and the traverse is $\frac{1}{160}$th of the same height. The area cut is then $\frac{h^2}{6,400}$, where h is the height of the centres. The speed of cutting may be in feet per minute,

$$\text{For soft steel, } v = \frac{10}{\sqrt[3]{a}}. \qquad \text{For cast iron, } v = \frac{3{\cdot}5}{\sqrt[3]{a}}.$$

where v is the cutting speed in feet per minute, and a is the area of cut in square inches.

Keener tools should be used on a material like copper than on steel, and keener on steel than on cast-iron.

In regard to the proper proportioning of the step-pulley and change-gears, the highest speed should be that enabling the tool to give its maximum possible output on the smallest diameter which it is profitable to turn in the lathe, and this may be a diameter of one-eighth the height of centres. The lowest speed should be such that the maximum output is also attained on the maximum diameter which can be swung on the lathe. The geometric ratio between successive speeds should be 1·08 in the case of a 6-inch lathe and 1·24 in the case of a 72-inch lathe. That is to say, if R be the maximum of revolutions per minute provided for and r the smallest, then, in the case of a 6-inch lathe, $R = r(1{\cdot}08)^n$, where $n + 1$ is the total number of speed changes.

The highest and lowest Spindle Speeds and their ratio.—These data are fundamental to the design of the fast headstock, and the following figures, rationally deduced, have been confirmed by practice: —

Height of centres . .		6 ins.	12 ins.	18 ins.	36 ins.	60 ins.	
Spindle speeds for carbon steel lathes	Greatest .	300	150	102	50	30	Ratio
	Least .	6	3	1·7	0·5	0·16	
		50	50	60	100	180	
Spindle speeds for high-heat steel lathes	Greatest .	600	300	200	100	60	Ratio
	Least .	80	10	4	1	0·5	
		10	30	50	100	120	

The Belt drive.—To enable them to transmit the great powers now required, much higher belt speeds have become necessary. Speeds through the air of 2,000 ft. or 3,000 ft. per min. are now used, and unduly wide belts avoided. The efficiency of the belt drive is also improved by using large pulleys running at moderate speeds, rather than small pulleys running at very high rates of revolution.

A simple rational formula for the power required for actual cutting when operating with ordinary high-speed steel on mild steel is :—

$$\text{Cutting HP} = \frac{\text{Force} \times \text{speed of cutting}}{33{,}000}.$$

The force for mild steel is 90 tons per square inch and the standard cutting speed is $\frac{10}{\sqrt{a}}$ where a is the area of cut (cut × traverse) in square inches.

$$\text{Thus C.HP} = \frac{2240 \times 90 \times a}{33{,}000} \times \frac{10}{\sqrt{a}} = 60\sqrt{a}.$$

For standard lathes the test cut in terms of the height of centres is

$$\frac{h}{40}, \text{ and the traverse is } \frac{h}{160}, \text{ or } a = \frac{h^2}{6{,}400}$$

$$\text{Then C.HP} = 60\sqrt{a} = 60\sqrt{\frac{h^2}{6{,}400}} = 0{\cdot}75\,h.$$

Adding 10 per cent. say for friction in the journals, gears, etc., then gross HP = 0·83 h.

(Nicolson & Smith.)

High-Speed Lathe Centres.

Owing to trouble experienced with carbon steel centres on repetition work, an experiment was made by the Ready Tool Co., of U.S.A., with the use of high-speed steel centres. These centres were used on the same machines and for the same work as carbon-steel centres, they showed no signs of either burning or wearing after five weeks of hard usage, and as a result it was definitely decided to put them on the market.

Experiments have shown that on lathe work at both high and low speeds the life of high-speed centres is easily ten times that of carbon-steel centres. In grinding machines, high-speed centres also seem to stand the wear of the abrasive which gets into the cutting compound and deteriorates the carbon-steel centres very rapidly.

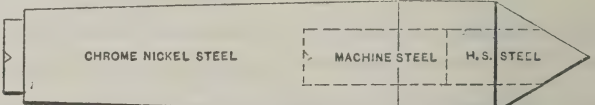


Fig. 1.

The centres, being made from solid high-speed steel, were, of course, expensive, but by welding the front or wearing end of the high-speed steel centre to a chrome-nickel steel shank, a satisfactory centre can be made at a comparatively low cost. These centres are shown by fig. 1 ; the object in designing the centres was to reduce the first cost, especially in large sizes, and still retain all the advantages of a high-speed steel centre.

The centre consists of a chrome-nickel steel shank having a high-speed steel point. This point is welded to a machine-steel extension piece to provide an unusually deep seat in the shank and ensure holding the point securely. The hole in the shank is reamed slightly tapering, and the point or plug is ground, after hardening, to the same taper as this hole. In assembling the plug and shank, the latter is first heated to about 1100° or 1200° F., and then the plug is forced into the hole under a heavy pressure, after which the shank is cooled by plunging it into cold water. As a result of this combination shrinkage and force fit, the plug or centre is gripped very tightly by the shank.

Lathe Tool Holders.

It is rarely possible to hold a too or cutter securely by a plain parallel shank or stem fitting alone, without any sort of screw or binding clamp. Fig. 2 shows a tool holder made by Sir W. G. Armstrong, Whitworth & Co., Ltd., Manchester, for their railway wheel lathes. The tool has a head *A*, about 1½ inches in diameter, which is tapered slightly to provide for front clearance.

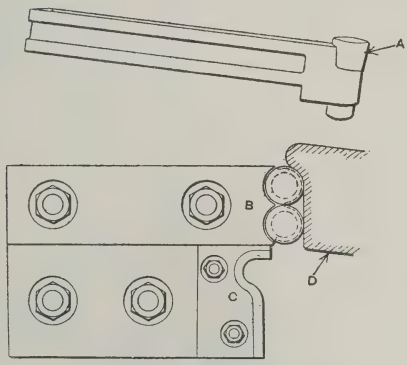


Fig. 2.

The stem is cylindrical and a tight fit in the holder. The top of the head is dished slightly to give the required rake. In machining the holder, the end is countersunk to fit the head, after which the front portion is cut away; this leaves a close-fitting supporting wall at the back of the cutter. When the cutting edge is dulled, the tool *A* is simply loosened and turned around far enough to bring a new cutting edge into position. When the entire edge has become dull, the tool is re-ground.

PLANING MACHINES.

Vickers' Patent Automatic Reversing Drive.

In this system, fig. 3, as applied to a planer, the motor itself reverses and is directly geared to the driving screw or rack-wheel of the machine. There are no driving pulleys to reverse, and the inertia of the driving system is much reduced, permitting quick reversal without the excessive waste of energy shown by a power diagram of the usual type of belt machines. There are no belts to wear. The cost of belts averages 30*l.* to 40*l.* per annum on belt-driven planers of large size. The speed of cutting can be instantly varied between the limits of the slowest cut and of the quick return. Thus, in a planer having slowest cutting speed 15 ft. per min., and (constant) quick return 60 ft. per min., the cut can be varied from 15 ft. to 60 ft. per min. by small intervals, and without altering the quick return speed, by simply turning the handle of the speed controlling switch to the proper position. The speed of the return stroke can be varied in a similar manner. The cutting speed can be instantly suited to the material from the hardest steel to softest steel and wrought iron, thus increasing the general output of the machine. The tool will plane accurately up to a line.

VICKERS' PATENT AUTOMATIC REVERSING DRIVE.

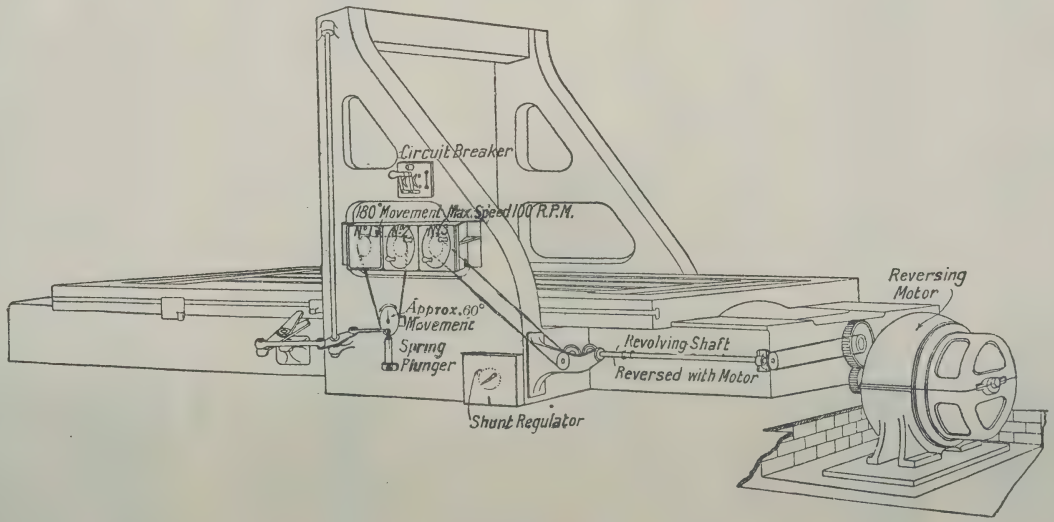


FIG. 3.

The following is a list of speeds of cutting and return stroke in use with planers which have been converted to this system of drive :—

Size of Planer.	Cutting Speeds.	Return Speeds.	No. of Tools.	Horse-power of Motor.
Feet.	Ft. per Min.	Ft. per Min.		
3 × 3 × 20	30/120	120/30	3	10
4 × 4 × 12	30/120	120/30	3 or 4	15
5 × 5 × 12	30/120	120/30	4	20
6 × 6 × 14	25/100	100/25	4	25
7 × 7 × 18	25/100	100/25	4	30
8 × 8 × 16	20/80	80/20	4	40
9 × 9 × 20	15/60	60/15	4	45
10 × 10 × 20	15/60	60/15	4	45
*12 × 12 × 24	10/40	40/10	Up to 8	50

* Armour-plate planing machines.

PLANING MACHINE TESTS.

Material.	Cutting Speed. Feet per Minute.	Depth of Cut.	Feed.	Authority.
		In.	In.	
Steel slabs . . .	25	$\frac{1}{2}$	$\frac{1}{8}$	Armstrong-Whitworth, using A.W. Steel.
Steel horn blocks .	35	$\frac{1}{2}$	$\frac{1}{8}$	Armstrong-Whitworth, using A.W. Steel.
Hard cast iron . .	40	$\frac{1}{4}$	$\frac{1}{8}$	Richard Hornsby & Sons, Ltd.
Cast iron . . .	36	$1\frac{1}{2}$	$\frac{1}{7}$	Armstrong-Whitworth.
Cast iron . . .	55	$\frac{1}{2}$	$\frac{3}{16}$	Armstrong-Whitworth.

PLANING SPEEDS AND FEEDS.

Material.	Cutting Speed. Ft. per Min.	Feed Ins. per Stroke.	Depth of Cut. Ins.
Roughing Cuts.			
Cast iron	37–48	$\frac{3}{16}$	$\frac{7}{16}$
Cast steel	28–48	$\frac{5}{32}$	$\frac{1}{4}$
Machine steel	28–48	$\frac{5}{32}$	$\frac{1}{4}$
NOTE.—Leave $\frac{1}{32}$ in. of metal for removal by subsequent cuts. When so set, the tools tend to dig in and actually leave about 0·026 in. of surplus stock.			
Straightening Cuts.			
Cast iron	45	$\frac{3}{16}$ to $\frac{1}{4}$	0·018
Cast steel	45	$\frac{1}{8}$	0·018 to 0·021
Machine steel	45	$\frac{1}{8}$	0·018 to 0·021
Finishing Cuts.			
Cast iron	28–35	$\frac{1}{4}$ to $1\frac{1}{2}$	0·003 and 0·001
Cast steel	45	$\frac{1}{32}$	0·004
Machine steel	45	$\frac{1}{32}$	0·004
NOTE.—For cast iron, it is good practice to divide the work up into two operations, taking off 0·003 in. with the first cut and 0·001 in. with the second cut. This prevents a tendency for the tool to dig in at the high spots.			

SPEED TABLE FOR PLANING MACHINES.

(Bateman's Machine Tool Co. Ltd.)

Size of Planer.	Length of Stroke taken.	Time of 10 Cycles.	No. of Feet in 10 Cycles.	Average Speed.	Maximum Speeds. Ft. per min.	
	Ft. Ins.	Min. Secs.		Ft. per Min.	Out.	Re-turn.
24 ins. by 24 ins. by 6 ft.	6 3½	1 8	126	111	80	225
30 ins. by 30 ins. by 8 ft.	8 0	4 20	160	37	22	218
(With three-speed gear-box)	8 0	2 21	160	68	45	218
	8 0	1 41	160	96	65	218
36 ins. by 36 ins. by 12 ft.	12 0	6 0	240	40	20	175
(With three-speed gear-box)	12 0	3 34	240	67	40	175
	12 0	2 46	240	87	62	175
36 ins. by 36 ins. by 20 ft.	20 3	10 12	405	40	24	163
(With three-speed gear-box)	20 3	6 14	405	65	42	163
	20 3	4 46	405	85	61	163
42 ins. by 42 ins. by 20 ft.	22 6	6 39	450	68	45	145
42 ins. by 42 ins. by 14 ft.	14 0	3 56	280	71	50	155
42 ins. by 42 ins. by 12 ft.	11 0	2 36	220	84	58	175
48 ins. by 48 ins. by 8 ft.	8 0	2 4	160	77	52	160
60 ins. by 60 ins. by 12 ft.	12 0	6 0	240	40	25	150
(With three-speed gear-box)	12 0	3 50	240	62½	42	150
	12 0	2 50	240	85	60	150
72 ins. by 72 ins. by 16 ft.	16 0	7 40	320	41	25	110
(With three-speed gear-box)	16 0	5 40	320	56	39	110
	16 0	4 50	320	66	50	110

For taking the roughing and straightening cuts it is usually found that high-speed steel is the most satisfactory material, owing to its greater productive capacity ; but where considerable accuracy and perfection of finish is required in performing the final planing operations, far more satisfactory results will usually be obtained by making the cutting tools of carbon steel, because this material can be ground to a keener edge, and experience has shown that the tools retain this edge longer and are thus able to produce a smoother and more accurate finish on the work.

MILLING MACHINES.*

One of the chief points is to make the cutters of such shape that all the fitting surfaces and cutting edges can be easily and accurately ground. In fig. 4 an ordinary grooving cutter is shown so designed. It will be noticed that the hole is chambered out so that the cutter fits the spindle at the two narrow surfaces, A and A′, only. These surfaces can be more easily ground to standard size than could a bearing the full width of the cutter. After the hole has been accurately finished it is an easy matter to mount the cutter on a mandrel and grind the surfaces BB and B′B′, and also the cutting edges, CC, C′C′, and DD all at the same setting, as indicated in fig. 4. Nothing now remains to be done but to back off the teeth in the way indicated in figs. 5 and 6. Such a cutter, when mounted on a true spindle, provided with a solid collar, against which the cutter is held, is self-setting and accurate.

In determining the correct angle for the face of the teeth, the diameter and shape of the cutter should be taken into account. For small-sized cutters it will, in order to obtain a strong shape, generally be found best to make the faces truly radial, as shown in figs. 4, 5, and 6. This also is more convenient when the cutter is to have teeth both on the sides and periphery. Cutters with the teeth so formed have worked upon both wrought and cast iron perfectly well, and at the same time they are not so liable to seize hold and bite into the work as cutters which have the faces of their teeth undercut. The pitch of the teeth is, again, partly determined by the diameter and class of cutter. In those cutters which are cut at the sides, the pitch at the circumference has often to be made wider than is advisable, in order to give clearance room for the emery wheel

* Abstract of a paper read before the Manchester Association of Engineers.

when grinding the converging side teeth (see fig. 6). A pitch of from $\frac{5}{16}$ inch to $\frac{3}{8}$ inch will usually be found to give good results. In all except very narrow cutters it is advisable to cut the teeth on the periphery spirally. This ensures a smoother action of the machine and produces better work. In deciding whether to make the cutters right- or left-hand spirals, attention should be paid to the direction in which they are to revolve. For example, in a vertical machine whose spindle revolves to the right hand, as in an ordinary drilling machine, the cutters should be cut

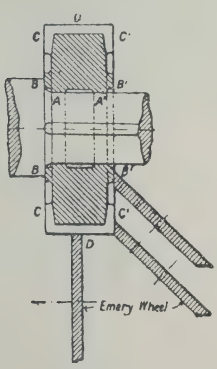


FIG. 4.

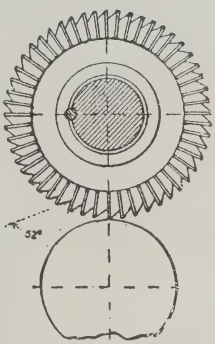


FIG. 5.

with a left-hand spiral. This has the effect of forcing the work down on to the table and the spindle up against its end bearings, as the cutter and the work act as a screw and a nut. In cutters of small size high-class tool steel is most suitable, although great care is required in heating it gradually and evenly. The bottom of the spaces between the teeth should be slightly rounded, and the keyways made semicircular, in order to avoid starting-places for fracture.

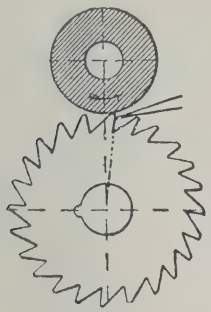


FIG. 6.

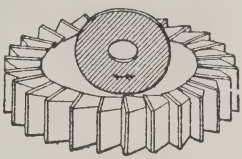


FIG. 7.

For cutters of larger sizes and strong sections Howell's case-hardening steel may be used with great advantage.

Too great stress cannot be laid upon the necessity of keeping milling cutters sharp, and both machines and cutters in such a condition that the cutters shall run true. In grinding milling cutters it is essential that the spindle and emery wheel of the grinding machine should run true and be accurately balanced, as, owing to the high speeds at which they are run, it is clear that any inaccuracy in these respects is highly detrimental. In order to prevent the emery wheel from

glazing, it is necessary to keep it perfectly free from oil or perspiration, and care should be taken not unduly to force the emery wheel against the work, as excessive pressure rapidly generates heat, which not only softens and glazes the emery wheel, but also draws the temper of the thin edges of the cutter, often without showing any discoloration on the edge just ground. For general cutter grinding the emery wheels should be fairly coarse and free-cutting, otherwise they are apt to soften the cutter, which may be known by a fine blue burr, to be seen on the faces of the teeth, the metal along this line being softened. For ordinary-sized cutters an emery wheel $2\frac{1}{2}$ inches in diameter by $\frac{5}{16}$ inch or $\frac{1}{4}$ inch wide will be found most suitable, the speed of which may vary from 3,000 to 4,500 revolutions per minute, but either too fast or too slow a speed will cause the wheel to burn and soften the cutter. The finishing cut should be very small in amount, especially in cutters of large diameter, otherwise the wear in the diameter of the emery wheel will materially affect the height of the cutter teeth, those last ground being the highest.

For general work, in which ordinary cylindrical cutters can be used, cutters of small or medium size, say up to $3\frac{1}{2}$ inches in diameter, will be found to possess important advantages.

The plain cylindrical cutter is the cheapest and most effective wherever it can be used.

Speed of Milling Cutters.

Table I. shows examples taken from everyday practice, working on wrought-iron forgings.

TABLE I.

Diameter of Cutter.	Depth of Cut.	Width of Cut.	Revolutions of Cutter per Minute.	Cutting Speed in Feet per Minute.	Feed in Inches per Minute.
Inches.	Inch.	Inches.			
2	$\frac{1}{4}$	$\frac{3}{4}$	150	78	$2\frac{1}{2}$
$3\frac{1}{2}$	$\frac{1}{4}$	$3\frac{1}{2}$	70	64	2
$7\frac{1}{2}$	$\frac{1}{4}$	5	35	68	$1\frac{1}{2}$

The following particulars of milling work which was done at the North London Railway shops were furnished by Mr. J. C. Park, the locomotive superintendent.

TABLE II.

CUTTERS 3 INCHES IN DIAMETER.

Speed and Feed.	Brass.	Wrought Iron.	Tool Steel.	Cast Iron.	Mild Steel.
Speed of cutters in feet per minute	112	60	33	18	33 to 60
Feed in inches per minute	1 in. and upwards	$\frac{3}{4}$ in. to 1 in.	$\frac{1}{2}$ inch	$\frac{1}{2}$ in. to $\frac{5}{8}$ in.	$\frac{3}{4}$ in. to 1 in.

TABLE III.

Class of Work.	Diameter of Cutter.	Depth of Cut.	Width of Cut.	Revolutions per Minute.	Cutting Speed in Feet per Minute.	Feed in Inches per Minute.
	Inches.	Inches.	Inches.			
Mild steel coupling rods, roughing cut	..	$\frac{1}{2}$	5	..	50	$\frac{3}{4}$
Finishing cut	..	$\frac{1}{32}$	5	..	60	1
Steel rails	7	1	2	15	32	$\frac{1}{7}$
Wrought-iron spring buckles	12	$1\frac{1}{2}$	2	15	47	$\frac{1}{7}$
Cast-steel horn-blocks	6	$\frac{1}{4}$	$2\frac{5}{8}$	14	22	$\frac{1}{4}$
Wrought-iron nuts (with two side cutters)	6	$\frac{1}{16}$ to $\frac{1}{8}$	..	30	47	$\frac{1}{2}$
Brass flanges	3	..	..	80	62	$1\frac{1}{2}$
Soft brass window frames	$1\frac{1}{2}$	$\frac{1}{8}$	..	90	35	$2\frac{1}{2}$
Cast-iron steam-chest covers	3	..	..	25	20	$\frac{1}{2}$

Tables IV., V., and VI., furnished by Messrs. John Musgrave & Sons, Bolton, are the results of an extended experience in milling.

TABLE IV.

AVERAGE SPEED AND FEEDS OF CYLINDRICAL MILLING CUTTERS FOR MACHINE STEEL.

Diameter of Mill in Inches.	Depth of Cut in Fractions of an Inch.	Cut $\frac{1}{4}$ Inch Wide.		Cut $\frac{1}{2}$ Inch Wide.		Cut 1 Inch Wide.		Cut 2 Inches Wide.		Cut 4 Inches Wide.	
		Revolutions per Minute.	Revolutions per Inch of Feed.	Revolutions per Minute.	Revolutions per Inch of Feed.	Revolutions per Minute.	Revolutions per Inch of Feed.	Revolutions per Minute.	Revolutions per Inch of Feed.	Revolutions per Minute.	Revolutions per Inch of Feed.
$\frac{1}{2}$	Full diameter	360	110	330	150						
	$\frac{1}{8}$	400	110	365	110	335	110				
	$\frac{1}{16}$	440	110	400	110	365	110				
	Finishing	480	85	440	85	400	85				
$\frac{3}{4}$	Full diameter	240	100	220	120						
	$\frac{1}{8}$	280	100	255	100	230	100				
	$\frac{1}{16}$	300	100	275	100	250	100				
	Finishing	320	75	290	75	265	75				
1	Full diameter	180	90	165	100						
	$\frac{1}{8}$	200	90	185	90	170	90	150	90		
	$\frac{1}{16}$	220	90	200	90	185	90	165	90		
	Finishing	240	68	220	68	200	68	180	68		
$1\frac{1}{4}$	Full diameter	144	80	130	90	120	90				
	$\frac{1}{8}$	160	80	150	80	130	80	120	86		
	$\frac{1}{16}$	175	80	160	80	145	80	132	80		
	Finishing	190	60	175	60	160	60	145	60		
$1\frac{1}{2}$	Full diameter	120	70	112	80	100	80	92	80		
	$\frac{1}{4}$	130	70	120	70	108	70	96	70		
	$\frac{1}{8}$	140	70	130	70	116	70	106	70		
	Finishing	160	53	150	53	130	53	120	53		
2	$\frac{1}{2}$	95	60	88	60	80	60	72	60	58	60
	$\frac{1}{4}$	100	60	92	60	84	60	75	60	60	60
	$\frac{1}{8}$	105	60	96	60	87	60	80	60	64	60
	Finishing	120	45	108	45	100	45	90	45	72	45
$2\frac{1}{2}$	$\frac{1}{2}$	75	50	70	50	64	50	58	50	47	50
	$\frac{1}{4}$	80	50	74	50	68	50	60	50	49	50
	$\frac{1}{8}$	85	50	77	50	71	50	64	50	51	50
	Finishing	95	37	88	37	80	37	72	37	58	37
3	$\frac{1}{2}$	65	40	60	40	54	40	48	40	38	40
	$\frac{1}{4}$	68	40	63	40	57	40	51	40	41	40
	$\frac{1}{8}$	72	40	66	40	60	40	54	40	43	40
	Finishing	80	30	74	30	67	30	60	30	48	30
4	$\frac{1}{2}$	48	35	44	35	40	35	36	35	29	35
	$\frac{1}{4}$	50	35	47	35	43	35	38	35	30	35
	$\frac{1}{8}$	53	35	49	35	47	35	40	35	32	35
	Finishing	60	26	55	26	50	26	45	26	36	26
5	$\frac{1}{2}$	38	30	35	30	32	30	29	30	23	30
	$\frac{1}{4}$	41	30	37	30	34	30	30	30	24	30
	$\frac{1}{8}$	43	30	39	30	36	30	32	30	26	30
	Finishing	48	23	44	23	40	23	36	23	29	23
6	$\frac{1}{2}$	32	25	30	25	26	25	24	25	19	25
	$\frac{1}{4}$	34	25	31	25	28	25	26	25	21	25
	$\frac{1}{8}$	36	25	33	25	30	25	27	25	22	25
	Finishing	40	19	37	19	33	19	30	19	24	19
7	$\frac{1}{2}$	28	22	25	22	23	22	21	22	17	22
	$\frac{1}{4}$	30	22	27	22	25	22	22	22	18	22
	$\frac{1}{8}$	31	22	28	22	26	22	23	22	19	22
	Finishing	34	17	31	17	29	17	26	17	21	17
8	$\frac{1}{2}$	24	20	22	20	20	20	18	20	14	20
	$\frac{1}{4}$	26	20	23	20	22	20	19	20	15	20
	$\frac{1}{8}$	27	20	25	20	23	20	20	20	16	20
	Finishing	30	15	27	15	25	15	22	15	18	15

TABLE V.

AVERAGE SPEEDS AND FEEDS FOR CYLINDRICAL MILLING CUTTERS FOR CAST IRON.

Diameter of Mill in Inches.	Depth of Cut in Fractions of an Inch.	Cut ¼ Inch Wide.		Cut ½ Inch Wide.		Cut 1 Inch Wide.		Cut 2 Inches Wide.		Cut 4 Inches Wide.	
		Revolutions per Minute.	Revolutions per Inch of Feed.	Revolutions per Minute.	Revolutions per Inch of Feed.	Revolutions per Minute.	Revolutions per Inch of Feed.	Revolutions per Minute.	Revolutions per Inch of Feed.	Revolutions per Minute.	Revolutions per Inch of Feed.
	Full diameter	450	110	410	110	375	110				
½	⅛	500	110	455	110	415	110	375	110		
	1/16	550	110	500	110	455	110	410	110		
	Finishing	600	85	550	85	500	85	450	85		
	Full diameter	300	100	275	100	250	100				
¾	⅛	350	100	320	100	290	100	265	100		
	1/16	375	100	340	100	310	100	280	100		
	Finishing	400	75	365	75	330	75	300	75		
	Full diameter	225	90	205	90	190	90	170	90		
1	⅛	250	90	330	90	210	90	190	90		
	1/16	275	90	250	90	230	90	205	90		
	Finishing	300	68	275	68	250	68	225	68		
	Full diameter	180	80	165	80	150	80	135	80		
1¼	3/16	200	80	185	80	165	70	150	70		
	1/16	220	80	200	80	180	70	165	70		
	Finishing	240	60	220	60	200	60	180	60		
	Full diameter	150	70	140	67	125	67	115	67		
1½	¼	160	70	150	67	135	50	120	50		
	⅛	175	70	160	67	145	50	133	50		
	Finishing	200	53	185	50	165	40	150	40		
	½	120	60	110	45	100	40	90	40	72	40
2	¼	125	60	115	45	105	40	94	40	75	40
	⅛	130	60	120	45	108	40	100	40	80	40
	Finishing	150	45	135	34	125	30	112	30	90	30
	½	95	50	88	36	80	32	72	32	58	32
2½	¼	100	50	92	36	84	32	75	32	61	32
	⅛	105	50	96	36	88	32	80	32	64	32
	Finishing	120	37	110	27	100	24	90	24	72	24
	½	80	40	74	30	67	26	60	26	48	26
3	¼	85	40	78	30	71	26	64	26	51	26
	⅛	90	40	82	30	75	26	67	26	54	26
	Finishing	100	30	92	22	83	20	75	20	60	20
	½	60	35	55	22	50	19	45	19	36	19
4	¼	63	35	58	22	53	19	47	19	38	19
	⅛	66	35	61	22	55	19	50	19	40	19
	Finishing	75	26	69	17	62	15	56	15	45	15
	½	48	30	44	17	40	16	36	16	29	16
5	¼	51	30	47	17	43	16	38	16	30	16
	⅛	54	30	49	17	45	16	40	16	32	16
	Finishing	60	23	55	13	50	12	45	12	36	12
	½	40	25	37	14	33	13	30	13	24	13
6	¼	43	25	39	14	35	13	32	13	26	13
	⅛	45	25	41	14	37	13	34	13	27	13
	Finishing	50	19	46	11	41	10	37	10	30	10
	½	35	22	31	12	29	11	26	11	21	11
7	¼	37	22	33	12	61	11	28	11	22	11
	⅛	39	22	35	12	33	11	29	11	23	11
	Finishing	43	17	39	9	36	8	32	8	26	8
	½	30	20	27	10	25	9	22	9	18	9
8	¼	32	20	29	10	27	9	24	9	19	9
	⅛	34	20	31	10	28	9	26	9	20	9
	Finishing	38	15	34	7	31	6	28	6	22	6

TABLE VI.

AVERAGE SPEEDS AND FEEDS OF FACE MILLING CUTTERS.

Depth of Mill in Ins.	Depth of Cut in Fractions of an Inch.	Cast Iron.		Machine Steel.		Cast Steel.	
		Revolutions per Minute.	Revolutions per Inch of Feed.	Revolutions per Minute.	Revolutions per Inch of Feed.	Revolutions per Minute.	Revolutions per Inch of Feed.
4	Scale	36	16	29	32	24	32
	$\frac{1}{16}$	40	12	32	24	27	24
	Finishing	45	8	36	12	30	12
5	Scale	29	13	23	26	19	26
	$\frac{1}{16}$	32	10	26	20	21	20
	Finishing	36	7	29	10	24	10
6	Scale	24	11	19	22	16	22
	$\frac{1}{16}$	27	8	21	16	18	16
	Finishing	30	6	24	8	20	8
7	Scale	21	9	17	18	14	18
	$\frac{1}{16}$	23	7	19	14	15	14
	Finishing	26	5	21	7	17	7
8	Scale	18	8	14	16	12	16
	$\frac{1}{16}$	20	6	16	12	13	12
	Finishing	22	4	18	6	15	6
10	Scale	14	7	11	14	9	14
	$\frac{1}{16}$	16	5	12	10	10	10
	Finishing	18	3	14	5	12	5
12	Scale	12	6	9	12	8	12
	$\frac{1}{16}$	13	4	10	8	9	8
	Finishing	15	$2\frac{1}{2}$	12	4	10	4
16	Scale	9	5	7	10	6	10
	$\frac{1}{16}$	10	3	8	6	7	6
	Finishing	11	2	9	3	8	3
20	Scale	7	4	5	8	5	8
	$\frac{1}{16}$	8	$2\frac{1}{2}$	6	5	5	5
	Finishing	9	$1\frac{1}{2}$	7	$2\frac{1}{2}$	6	$2\frac{1}{2}$
24	Scale	6	3	4	6	4	6
	$\frac{1}{16}$	7	2	5	4	4	4
	Finishing	8	$1\frac{1}{2}$	6	2	5	2

Generally speaking, the speed of milling cutters of 6 inches diameter and upwards, provided there is no great depth of material to be cut away, may be as follows :—

Material.	Speed in Feet per Minute.	Depth of Cut in Inches.
Steel	36	$\frac{1}{2}$
Iron	48	1
Cast iron	60	$1\frac{2}{3}$
Brass	120	$2\frac{2}{3}$

CUTTER DIMENSIONS.

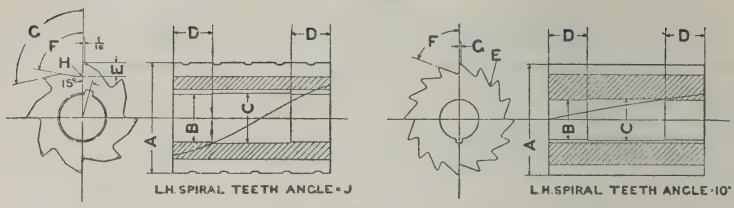


FIG. 8.

Coarse Pitch.
Dimensions in Inches.

Diam. of Cutter. A	B	C	D	E	F	G	H	J	Key-way.	No. of Teeth.
2½	1	1 3/16	1½	3/8	80°	103°	1/8 R	26½°	5/32 × 3/32	8
3	1¼	1 5/16	1½	3/8	80°	103°	1/8 R	26½°	3/16 × 1/8	8
3½	1½	1 9/16	1¼	7/16	80°	103°	1/8 R	26°	1/4 × 5/32	8
4	1¾	1 13/16	1¼	7/16	71°	94°	3/16 R	25½°	1/4 × 5/32	10
4½	2	2 1/16	1½	1/2	71°	94°	3/16 R	26½°	5/16 × 3/16	10

Fine Pitch.
Dimensions in Inches.

Diam. of Cutter. A	B	C	D	E	F	G	Key-way.	No. of Teeth.
2½	1	1 1/16	1½	3/64 R	70°	3/64	5/32 × 3/32	14
3	1¼	1 5/16	1½	3/64 R	70°	3/64	3/16 × 1/8	14
3½	1½	1 5/16	1½	3/64 R	70°	3/64	3/16 × 1/8	16
4	1½	1 9/16	1½	1/16 R	65°	1/16	1/4 × 5/32	18
4½	1½	1 9/16	1½	1/16 R	65°	1/16	1/4 × 5/32	20

The above table gives particulars of a range of cutters suitable for heavy roughing work, and also corresponding fine pitch cutters used for light milling.

(*P. V. Vernon* (*Alfred Herbert & Co.*))
(*Paper read before the Manchester Association of Engineers*, 1912.)

Experiments to determine the effect of cutting speed on the life of a milling cutter were mentioned by E. Buckingham in a discussion on a paper on milling that was read recently before the American Society of Engineers. Tests were made at speeds of 70 and 80 feet per minute with high-speed steel cutters on small nickel-steel forgings. Very consistent results showed definitely that at 70 feet per minute the cutters would mill twice as many pieces as when running at 80 feet per minute. Therefore two cutters exactly similar, except that one has 21 and the other 24 teeth, would show a great difference in their life between grindings if each were set to remove the same amount of material in the same time. Under the conditions mentioned above the 24-tooth cutter would mill twice as many pieces before regrinding were necessary. The author of the original paper referred to experiments conducted with spiral milling cutters. One of the cutters experimented with was of double spiral type, in which successive teeth are of opposite spiral hand. The results when compared with those obtained from a spiral cutter with teeth all of one hand, showed that with the double spiral there was a saving of from 20 to 30 per cent. in the power required. (*'Power.' February* 16, 1922.)

The cutting speed will, when tapping and screwing with carbon steel tools and copious streams of lubricant, be about 10 ft. per minute.

A Milling Cutter for Copper.

The diameter of the cutter (fig. 9) should be of such proportions as will allow the placing of eighteen teeth of 1-in. pitch upon a 6-in. diameter circle. This will give excellent results, and is the result of a series of exhaustive tests.

Each tooth should be hacked off slightly on either side, as shown shaded at A, and have a rake angle of about 10°.

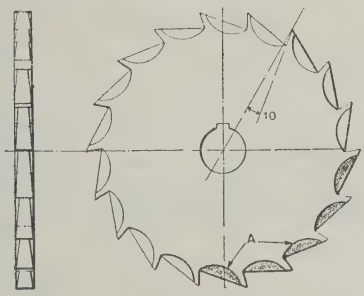


FIG. 9.

Should a cutting lubricant be found to be necessary, a quantity of lard oil, thinned with turpentine, is recommended. (*Mechanical World*, *April* 3, 1925.)

BROACHING.

Recently the finishing of holes, oil gutters, keyways, etc., by broaching has received considerable attention. Any class of material appears to lend itself to this process provided the broach is of suitable design. Broaches under 1½ ins. diameter are frequently made of high-speed steel containing from 14 per cent. to 18 per cent. tungsten. Such steel, however, is hard throughout and likely to fail under shock. For this reason, some firms make their broaches entirely of vanadium steel, which contains neither tungsten nor chromium. This steel is hardened at 750° C. and quenched in oil and tempered at 420° C. For general work a mild steel containing about 0·15 per cent. carbon, carburised at 900° C., reheated to 800° C. and dipped in water has given very satisfactory results. The reheating before quenching gives a tough core. For broaching hard steel, a mild steel containing from 2 per cent. to 3 per cent. nickel and about 0·14 per cent. carbon, carefully case-hardened, reheated to 770° C. and quenched in whale oil, has proved suitable. In making a broach, choose a bar with at least ¼ in. excess in diameter of the finished broach, so as to avoid small surface cracks and skin stresses. Broaches above 2 ins. diameter are frequently built up, in which case the teeth are formed on hardened steel rings, and these are securely mounted on a mild steel core. The teeth of the broach should be periodically examined when in use. Should any of these be chipped or dulled, these must have the cutting edge restored. Should a tooth break it must be ground down and the cut distributed amongst the next four teeth or so.

In the case of white metal an accurate hole and fair finish may be obtained with a broach having fine pitched teeth (say, from ⅛ in. to 1 in. in broaches from 1½ ins. to 6 ins. diameter), and each cutting tooth increasing by 0·0005 in. The cutting teeth are preceded by a plain part (to act as a pilot) about 2 in. long and 0·002 in. below the diameter of the first cutting tooth and diameter of the hole. From 6 to 12 teeth at the end of the broach are parallel and about 0·002 in. larger than the finished hole, to allow for creep. This dimension will vary with the springiness of the member. The teeth should have a parallel land of 0·02 in. and a top-take of about 10°.

Design of Broach.

The broach may be secured to the cross-head of the machine by means of a screw or a flat key, or a buttress-thread and pad, a plain cylindrical stem with cotter. An example of the latter is shown in fig. 10. This method is extensively used, on account of the ease with which the pin can be withdrawn and the security it affords, in that the pin is sheared before any damage is done to the broach or machine.

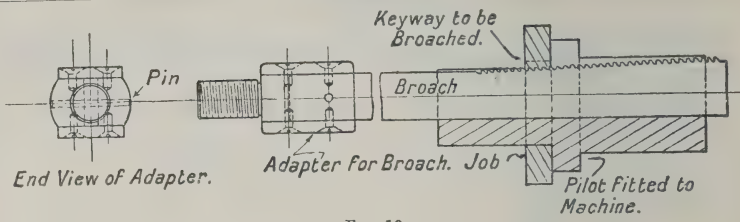


FIG. 10.

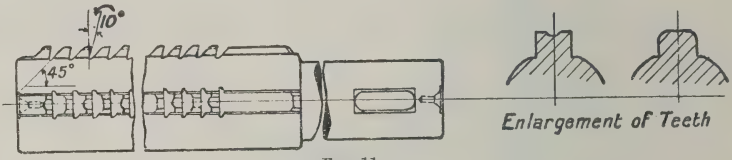


FIG. 11.

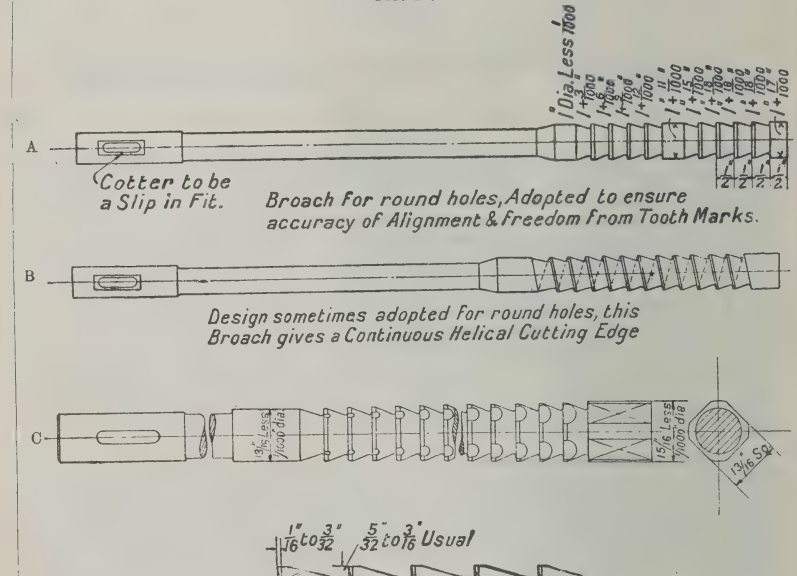


FIG. 12.—Types of Broaches.

All broaches should be provided with a pilot. The pilot is made a sliding fit in the hole or slot, and leads the cutting edges centrally to their work. It is a very important factor, as it not only helps to produce accurate work, but prevents damage to the broach. In the case of very long broaches, pilots or steadies are provided at intervals along the broach itself, as shown at A, fig. 12. It is advisable that the broach should have four or five teeth at the end for sizing only. These may be of a smaller pitch than the normal, as the cut removed per tooth is exceedingly fine. C, fig. 12, shows a square broach with progressive increasing cut per tooth which in this design is permissible.

Pitch of Teeth.

The pitch of the teeth is regulated by the number of teeth engaged, the cut taken per tooth, the material operated upon, and the material of which the broach is made. It is clear that the cutting force should not exceed the tensile strength of the broach or strength of the teeth. In the case of broaching, a circular, square or hexagonal hole in mild steel, it is usual to allow of three teeth being in action, each tooth taking $\frac{3}{1000}$ in. cut, that is, 9 teeth thousandths. This is suitable, say, for a broach 1 in. dia., but for a broach $\frac{1}{2}$ in. dia. only 5 teeth thousandths should be allowed, whilst for a 2 in. broach 18 teeth thousandths can be allowed, and a proportionately larger amount as the diameter of the broach increases. With softer material more teeth can, of course, be engaged. It is desirable that the teeth should have a top rake of about 10°, a land of about $\frac{1}{32}$ in., and a side clearance of about 3°, as shown in fig. 8. Proportions of teeth in common use are shown at D, fig. 12.

It is difficult to obtain uniformity of cutting per tooth on the first trial, and the broach may be pulled through a sample piece, the amount removed per tooth being noted, and the teeth honed down where necessary. On rectangular and circular work the teeth are sometimes spiral as shown at B in fig. 12. This design has the advantage of giving a side rake to the cutting edges, and it is claimed that it tends to neutralise any lateral thrust. Occasionally the teeth are staggered, and where the material to be cut is very hard the teeth may be fluted or nicked. An example of the latter is shown in fig. 11, where the alternate teeth are arranged to remove the corner and body of the groove. This arrangement breaks up the chip and materially reduces the stress on the broach.

In fig. 10 is shown the design of a broach adopted for cutting single keyway in a member, and the pilot which is fitted to the machine to guide the broach. Where more than one key is required, arrangements are made to turn the member round, but the spacing obtained under such conditions is rarely accurate, and a better method is to provide for a double spline broach as shown in fig. 11. It is always desirable, where there is more than one gutter, to have a broach with the necessary number of splines.

Speed of Cutting.

The speed of cutting varies with the material operated upon, and for soft brass may be as high as 20 ft. per minute, and for medium hard steel about 5 ft. per minute. The cutting speed is relatively of little importance, and it is generally advisable to adopt about half the above speeds.

Lubrication.

When a highly-polished finish is required on ductile materials it is desirable to use a cutting compound, and on such materials one pint of oil mixed with 4 ozs. of flowers of sulphur has given good results. The J. N. Lapointe Co. recommends the following mixture when broaching mild steel :—

Mineral lard oil, 3 gallons ; water, 50 gallons ; and soda ash, 2½ lbs. The whole of the lard oil and soda ash should be mixed with 10 gallons of water and the remaining proportion of water added before putting into use.

For heavy duty use castor oil.

DRILLS AND DRILLING.

Cutting Edges of Drills.

The cutting edges of drills are formed by grinding the end facets at an angle as shown by A, C, B, fig. 13, and this angle varies from 5° for drilling hard metal, such as steel, to 20° for

soft metal, such as brass or copper. The angle of one cutting edge to the other varies from 45° for steel to about 35° or 40° for soft metals.

Drills for work of ordinary hardness are tempered to an orange purple; but if the metal to be

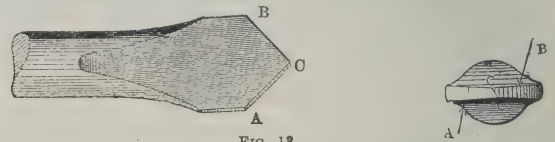


FIG. 13.

cut is very hard a straw colour is preferable, or the drill may be left as hard as it leaves the water; that is to say, hardened, but not tempered. In these cases great care must be exercised and the speed of the drill reduced.

Twist Drills.

A twist drill possesses three cutting edges, A, B, and C, as shown in fig. 14.

The point A, being the least effective, since it cannot be made anything like as keen as the other two, B and C, is a hindrance to clean and rapid cutting and is one of the most important factors to be considered when arriving at the amount of feed progression per inch of travel.

In order to obtain the greatest possible duty from a twist drill, it is imperative that it should be

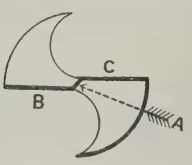


FIG. 14.

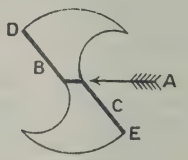


FIG. 15.

ground perfectly true, so that the point of the drill shall be central and in line with the axis on which it revolves. The cutting edges must be of equal length and at the same angle to the axis of the drill. If the point of the drill is not central to the axis, the two cutting edges are not at the same angle, the result being that all the duty falls on one cutting edge, and the hole made will still be larger in diameter than the drill, there being a tendency for the cutting edge B to push, or crowd, the drill over to the opposite side of the hole.

From the foregoing it will be obvious that the more correctly a drill is ground, the longer it will last without re-grinding; the greater its amount of feed may be to take an equal depth of cut, and the nearer the diameter of the hole bored will be to that of the drill.

Another important point when the maximum duty is expected from a twist drill is the relationship between the speed and feed, which varies more or less with different materials and with different grades of hardness of like materials.

When a drill has been in operation for a short time it should be withdrawn from the machine and examined as to its cutting edges, fig. 15. Should the cutting edges B C show more wear towards their extreme diameters D E, this would show the speed is too great as compared with the feed.

The grooves of a twist drill make either a constant angle with the axis or an angle that increases gradually from point to shank. The latter form is supposed to facilitate the escape of the cuttings, though opinions differ, but it has the serious effect of rendering the cutting angle more obtuse as the drill becomes ground back. When the angle is constant it ranges from 24° to 32° with the axis, measured at the periphery, 26° being an average. In the increase-twist drills it ranges roughly between 30° and 25°, the angle lessening by that amount from point to shank. The thickness of the web is increased from point to shank to enable it to resist torsional stresses. The diminished area for the escape of chips consequent on this thickening is counteracted by increasing the twist.

'FIRTH'S SPEEDICUT' HIGH-SPEED TWIST DRILLS.

(Thos. Firth & Sons, Ltd., Sheffield.)

The chart (fig. 16) indicates speeds and feeds suitable for drilling mild steel; for cast iron use the same feed per minute but decrease the speed 20 per cent. Both speeds and feeds may be modified to suit variation of material.

The feeds and speeds given on the chart are approximate, and represent ordinary working speeds for average depths of holes.

The chart shows that a 1-in. drill should run under the following conditions.

Mild Steel.	*Cast Iron.*
Speed, 280 revs. per min.	Speed, 224 revs. per min.
Feed, 4¾ ins. per min.	Feed, 4¾ ins. per min.
Feed per rev., ·0175.	Feed per rev., ·022.

If the material is thin, as in the case of boiler plate, both feed and speed can be increased. If the material is hard, decrease both the feed and speed. Always run with lubricant if possible except in the case of cast iron. If no lubricant is used, decrease both feed and speed about 25 per cent. When possible, always keep the feed per rev. the same as given on the chart for the

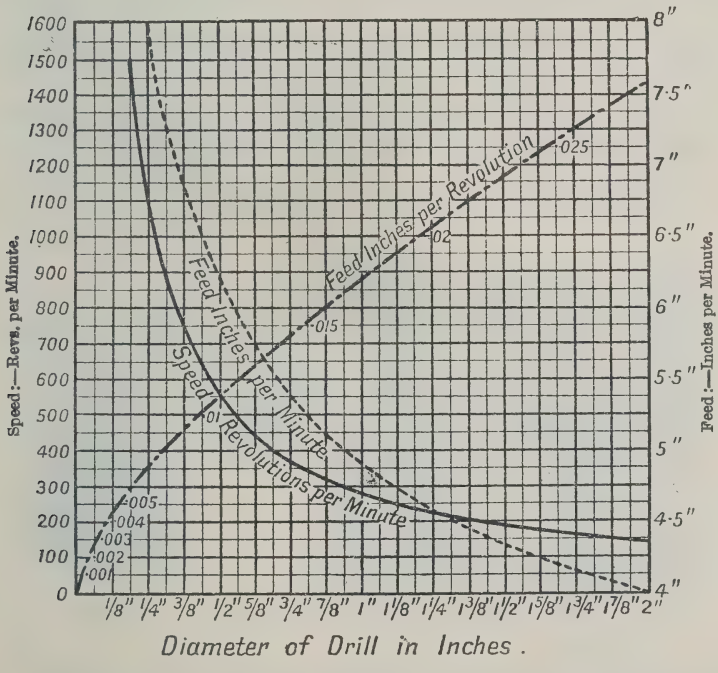


FIG. 16.

two different classes of material, i.e. steel and cast iron. Never decrease the feed below a minimum of, say, 3 ins. per min. for a 1-in. drill, the others in like proportion.

SPEEDS FOR ORDINARY CARBON STEEL DRILLS.

Diameter of Drill in Inches.	Revolutions per Minute.			Diameter of Drill in Inches.	Revolutions per Minute.		
	Soft Steel and Wrought Iron.	Cast Iron.	Brass.		Soft Steel and Wrought Iron.	Cast Iron.	Brass.
1/8	768	610	1,536	1	96	76	192
5/16	512	405	1,024	1 1/8	85	66	170
1/4	384	304	768	1 1/4	77	61	154
5/16	307	243	614	1 3/8	70	55	140
3/8	256	203	512	1 1/2	64	50	128
7/16	219	174	438	1 5/8	59	47	118
1/2	192	152	384	1 3/4	55	43	110
9/16	170	135	340	1 7/8	51	41	100
5/8	154	122	308	2	48	38	96
11/16	139	111	278	2 1/4	42	34	84
3/4	128	102	256	2 1/2	38	30	76
13/16	118	93	236	2 3/4	35	27	70
7/8	109	87	218	3	31	25	62
15/16	102	81	204				

Feed revolutions for the above speeds, on wrought iron and steel: up to and including 1/4 in. diameter, 1/150 in.; from 1/4 in. up to 7/16 in. inclusive, 1/130 in.; from 7/16 in. up to 5/8 in. inclusive, 1/110 in.; from 5/8 in. up to 1 in. inclusive, 3/80 in.; from 1 in. up to 1 1/2 in. inclusive, 1/50 in.; from 1 1/2 in. upwards, 1/75 in. For cast iron these feeds may be increased 20 per cent.

For small holes in brass the feed should be from 0·0015 in. to 0·003 in. per revolution; 1/8 in. to 1/4 in. diameter, 0·003 in. to 0·006 in. per revolution; 1/4 in. to 1/2 in. diameter and upwards 0·006 in. to 0·020 in. per revolution. When drilling mild steel the feed should be approximately one-third the above, and harder metals slower in proportion.

Should the drill be running too fast the outer edge of the lip becomes quickly dulled; should the central stem become dull, or the lips chip, the feed is too great.

GRINDING DRILLS.

When grinding the drill points, the following rules should be observed:—

I. Both cutting lips, fig. 17, should be inclined at the same angle with the axis of the drill, and must be of equal length. The point angle of 59° has been universally adopted as best suited for average conditions.

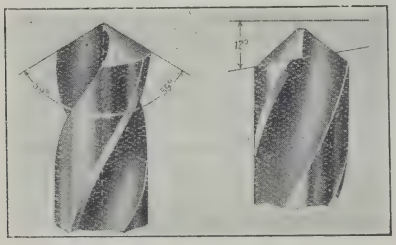


FIG. 17. FIG. 18.

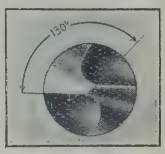


FIG. 19.

II. The drill point must have the proper clearance or contour of surface back of the cutting edges, and this clearance must be identical on both sides. Approximately a 12° clearance angle, fig. 18, combined with the centre angle of 130°, which will give a constantly increasing clearance toward the centre, fig. 19, has proved best for average conditions.

Some of the undesirable conditions resulting from drill points improperly ground are as follows:—

(1) If both lips are not ground at the same angle with the axis, fig. 20, one lip will fail to counteract the tendency of the other to spring away from the cut; consequently, one lip will do more work than the other, which will result in its becoming dull more rapidly than if both lips were cutting equally, and it will be subjected to an abnormal torsional strain.

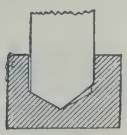


FIG. 20.

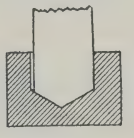


FIG. 21.

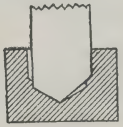


FIG. 22.

(2) When the cutting lips of a drill have the same point angle but are of different lengths, fig. 21, the point of the drill will be off centre, or eccentric. As a result, the hole will be oversize to an extent equal to double the amount of this eccentricity.

(3) If the drill point is ground both with lips at different angles and of different lengths, fig. 22, there will be a combination of the undesirable results described in the two preceding paragraphs.

(4) Fig. 23 shows a side and end view of a drill with the proper angle of point (59°) and the proper angle of clearance at the periphery (12°), but with insufficient clearance at the point or centre.

(5) Fig 24 shows a drill with insufficient clearance both at the periphery and at the centre. The line ABC is at an angle of 12°, but there is no clearance immediately back of the cutting edges (BC), and the excess of clearance at the heel (AB) is of no benefit.

(6) Fig. 25 shows what is liable to happen to a drill ground with insufficient clearance, or as described in paragraphs Nos. 1 to 5, especially if an attempt is made to obtain maximum production.

FIG. 24. FIG. 25. FIG. 26.

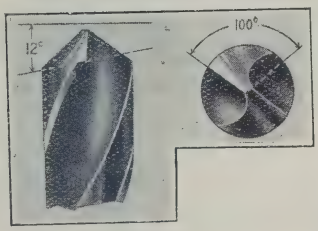


FIG. 23.

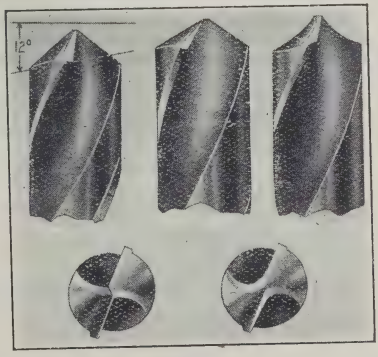


FIG. 27. FIG. 28.

(7) Fig. 26 shows a drill with a clearance angle of about 12°, but it does not have the proper contour back of the cutting edges. This manner of grinding leaves the cutting edges thin and weak, causing them to crumble away under heavy feeds.

Most twist drills are made with a gradual increase in the thickness of the web or centre of the drill toward the shank. As the drill becomes shorter and the web thicker, greater force

is required to drive it. To overcome this, it is good practice to thin the web by grinding away the excess thickness, reducing it to its original dimensions. This grinding must not extend too far up the flute of the drill, and care must be exercised that the cutting lips are not injured and the same amount is ground out of each groove. Fig. 27 shows a drill with the web properly thinned. In fig. 28 the grinding is excessive, leaving the web entirely too thin and liable to crumble. When this happens a split drill is practically inevitable.

Paragraphs 1 to 5 and the preceding paragraph show the cause of drills splitting up the centre, and no manufacturer should be called upon to replace a split drill unless a flaw is evident in the steel.

Twist drills are made with a slight taper from point to shank, so that the largest diameter is always across the corners of the cutting lips. This prevents the drills from binding in the work when they are sharp. If the outer corners are allowed to become badly worn, the drills will bind and cannot perform satisfactorily. Whenever the outer corners of the cutting lips show wear, the drills should be reground and every particle of worn surface removed, or the drill will continue to bind and very quickly be damaged beyond repair.

In grinding high-speed drills care should be taken not to overheat them, and when heated they should never be plunged into cold water. Doing so is likely to cause small surface cracks, which reduce the efficiency of the drill and may result in serious damage to it. Forcing the grinding on a wet grinder may also bring about the same condition.

It is hardly possible to do grinding as accurately by hand as by using a good twist-drill grinding machine.

Broken or damaged tangs of drills are generally the result of an imperfect fit of the drill shank in its socket, which may be caused by a worn-out socket, dirt or chips accumulating in the socket, or bruises on the shank of the drill.

In any case, the driving power of the taper is reduced or destroyed, resulting in an abnormal strain being put upon the tang.

A drill of either carbon or high-speed steel that can be filed is not necessarily too soft for service; in fact, if drills were tempered so that a good file would make no impression, they would be entirely too hard for general use. If there is any doubt regarding the temper, try the drills in actual service.

Assuming the drill is properly ground, when the corners of the cutting lips wear away rapidly it is an indication that the speed is too great. If the cutting edges roughen or break out in minute particles it indicates that the feed is too great.

It is seldom that very small drills can be run at more than a fraction of the speed necessary to obtain the best results, and excessive breakage is inevitable. They should be run true and the cutting edges kept sharp; a fine-grade emery stone is best suited for this latter purpose.

MORSE TAPER MEASURES FOR TWIST DRILLS.

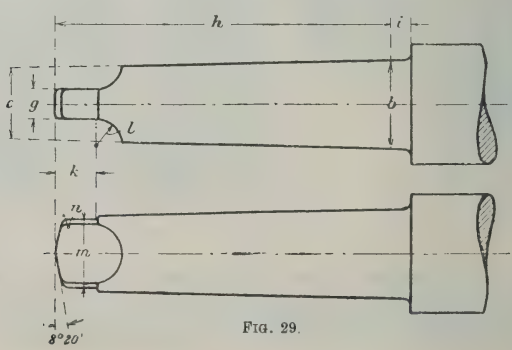


FIG. 29.

Morse Taper No.	*b*		*c*		*g*		*h*		*i*		*k*	*l*	*m*	*n*	Taper.	
	mm.	in.	mm.	in.	mm.	in.	mm.	in.	mm.	in.	mm.	mm.	mm.	mm.	per 100 mm.	in 12-In.
1	12,05	·475	9,03	·356	5,1	$\frac{13}{64}$	60,5	$2\frac{3}{8}$	5	$\frac{3}{16}$	8	5	8,4	1,3	5	·600
2	17,75	·700	14,09	·556	6,3	$\frac{1}{4}$	73	$2\frac{7}{8}$	5	$\frac{3}{16}$	9,5	6,5	13,4	1,6	5,017	·602
3	23,8	·938	19,26	·759	7,9	$\frac{5}{16}$	90,5	$3\frac{9}{16}$	5	$\frac{3}{16}$	11	7	18,5	2	5,017	·602
4	31,25	1·231	25,31	·997	11,9	$\frac{15}{32}$	114,5	$4\frac{1}{2}$	6,5	$\frac{1}{4}$	13	8	24,5	2,5	5,191	·623
5	44,4	1·748	36,74	1·446	15,9	$\frac{5}{8}$	146	$5\frac{3}{4}$	6,5	$\frac{1}{4}$	16	9,5	35,7	3	5,25	·630
6	63,35	2·494	52,76	2·077	19	$\frac{3}{4}$	203	8	8	$\frac{5}{16}$	22	12,5	50,8	3,8	5,216	·626

British Standard Specification for Twist and Straight Flute Drills.

(No. 328—1928.) (*Extract.*)*

Tables are given in this specification of the standard diameters, overall lengths, and flute lengths for the various types of drills, and the dimensions of the Morse taper shanks. Standard limits are laid down for the permissible variation in the diameter of the drills, and performance tests are specified for both carbon and high-speed drills from No. 60 to 2 ins. diameter, inclusive. The performance tests consist in drilling a series of holes, as continuously as possible, in steel billets of a given analysis. The penetration per minute and the number and depth of the holes to be drilled are specified, but the revolutions per minute and the feed per revolution are left to the discretion of the maker. The drills must withstand these tests without seizing, choking, or fusing, and the points of the drills and the lips must be in good condition and fit for further service without regrinding on completion of the tests. Standard nomenclature and definitions for the different types of drills in common use are also included.

Notes on Drilling.

The power for a given diameter of drill and feed is proportional to the cutting speed.

The horse-power is proportional to the torque, and for a given drill and speed does not increase as fast as the feed.

Since the torque is practically proportional to the diameter of the drill squared, the horse-power, for a given feed and cutting speed, is directly proportional to the diameter of the drill.

The horse-power per cubic inch of metal removed is inversely proportional to the feed and independent of the drill and cutting speed.

The work required to drill a given hole, when one drill is used, is greater than that required to drill the same hole in two operations with drills of different diameters. The greater the difference in the drill diameters, the greater the saving in work; speed and feed remaining the same throughout. This is due to the fact that the mean cutting angle of the single drill is greater than the average angle in use for the two drills and that the stress is proportional to the angle.

With twist drills of the usual proportions, the cutting angle is not sufficiently keen to drag the drill into the work when enlarging a hole in cast iron or steel.

The horse-power when operating on soft cast iron or medium steel varies as $t^{0\cdot7}$ (t = feed per rev.) for a given drill and speed.

The horse-power for a given feed and speed does not increase as fast as the diameter but varies as $d^{1\cdot8}$.

The torque and horse-power when drilling medium steel is about 2·1 times that required to drill soft cast iron with the same drill speed and feed.

The end thrust when operating on cast iron or steel does not increase in proportion to the feed for a given diameter of drill or in proportion to the diameter for a given feed.

While the chisel point scarcely affects the torque it is accountable for about 20 per cent. of the end thrust.

Lubricated trials on steel when compared with the dry tests show a diminution in the torque and horse-power, varying from 28 per cent. with the $\frac{1}{40}$ feed to 8 per cent. with the $\frac{1}{5}$ feed.

* By permission of the British Engineering Standards Association.

This may be due to the lubricant washing away the small metal chips which tend to jam between the walls of the hole and the drill, and to the preserved cutting edge. The diminished frictional resistance of the shaving across the lip together with above reduces the end thrust by about 25 per cent. for all feeds.

As the result of a series of experiments on twist drills, recently undertaken at the engineering experiment station of the Illinois University, the following lubricants are recommended:—For tool steel—lard oil, machine oil, turpentine, soda-water, kerosene; for soft steel—lard oil, machine oil, soda-water; for cast iron—dry or compressed air; for brass—dry or paraffin oil; for aluminium—kerosene, soda-water, and aqualine and soda-water.

The drill most commonly adopted in practice has an included angle, at the point, of 120°. If this angle is increased, the torque diminishes but the end thrust increases, while if this angle is decreased, the reverse is the result. So far as economy in power is concerned the torque is the factor to consider, as the feeding horse-power is only about 1 per cent. of the whole, in small drills, and very much less for the larger sizes. From this point of view the drill with the larger point angle is to be preferred. The accompanying increased end thrust, however, strains the machine parts in proportion. When the point of the drill breaks through the metal at the bottom of the hole, a considerable portion of the end load is removed. The strain due to that load is released, thereby causing the drill to advance more than its rated feed, and possibly to break. The drill with the greater included angle will be most likely to give trouble in this direction, both on account of the increased strain and torque, and conversely.

By decreasing the spiral of the drill a keener cutting angle with a decreased end thrust and torque can be obtained without altering the point angle above the accepted standard, 120°. This, however, would in turn affect the durability of the drill.

With a small included point angle there is little metal to support the cutting edge at the chisel point, and trouble due to blunting of this part is to be expected.

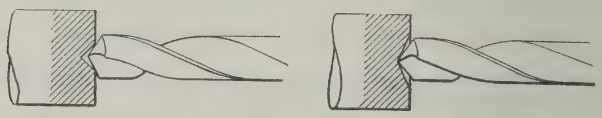
The endurance of a drill may be increased by from 300 per cent. to 1,000 per cent. by rounding the sharp corners at the periphery.

In estimating the time required to drill a hole of given depth, the length of the drill point must be taken into account. The length of the point for different included point angles is:

$$90° = 0{\cdot}5d\,; \quad 120° = 0{\cdot}29d\,; \quad 150° = 0{\cdot}134d.$$

In the course of a paper read before the Institution of Production Engineers, Mr. L. C. Keen gave the following advice:—Do not attempt, on drills below $\frac{3}{16}$-in. diameter, to drill a depth more than three times the diameter of the drill, without drawing out to clear the chips; if the drill as well as the work is revolved, however, it can be easily fed in to a depth of five times the diameter, at one traverse.

True drilling is usually dependent upon good centering, and it may be advisable to draw attention to the well-known but often ignored rule, namely, that the centering tool must have a lesser cutting angle than the drill point, otherwise the drill will strike on the point, wobble, start out of centre, and sometimes break.



A *B*

FIG. 30.

Good centering. Bad centering.

Fig. 30 shows good centering at *A* and bad at *B*. As a general rule the centering drill should ave an included angle of 100° against the standard angle for the drill point of 119°.

(*Machinery.*)

See also '*Smith & Poliakoff: Proc. Inst. of Mech. Engrs.* 1909.')

A high efficiency can be obtained by fitting all running bearings of drilling machines (including the spindle bearings) with annular ball bearings, which never require adjustment. The fact that the spindles themselves have no sleeve or rubbing bearing whatever enables even the smallest sizes of high-speed drills to be run at their proper speeds. At the same time the saving in friction all over the machine permits the feed to be quicker, and enables larger holes to be drilled than has been hitherto possible with ordinary sensitive machines.

Vulcanised fibre is being applied to the thrust bearings of light, sensitive, drilling machines. Ball bearings cause considerable trouble if one of the balls fractures or chips, and it has been found that a fibre disc answers the purpose admirably. The permeability of the fibre to oil renders it self-lubricating, and its life under these conditions is remarkable. (*Machinery.*)

Small Drill Protector.

The function of this protector is to stiffen the shank of the drill, leaving exposed a length o the drill sufficient only for the depth of the hole to be drilled. The protector, fig. 31, consists of a length of hard brass, drilled from one end to the other, and with about three-quarters of its length slotted on one side, the internal diameter being equal to the diameter of the drill to which it is to be applied. To use the protector, the drill is inserted therein with the cutting point projecting from the chamfered end, the exposed length being equal to the required depth of the hole to be drilled. The split end of the protector is inserted into the drill chuck so that it is

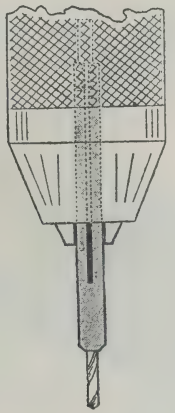


Fig. 31.

secured for about half an inch of its length, the chuck then being tightened up in the usual manner. If it is required to drill a hole of greater depth than usual all that is necessary is to release the chuck and slide the protector farther up the drill for the extra length required and then tighten up the chuck again.

The protectors are supplied suitable for short or long length drills in sizes varying from $\frac{1}{32}$ up to $\frac{1}{4}$ in., from 20 to 66 wire gauge sizes and from 1 to 2 mm.

Lubricant for Drilling and Tapping Copper.

Milk, 50 parts; Water, 45 parts; Turpentine, 2·5 parts; Paraffin, 2·5 parts.

Portable Electric Drills.

The Van Dorn Drill.

These tools, fig. 32, are made by the Van Dorn & Dutton Co., Cleveland, Ohio. They are designed so as to be easy to handle, and are light and compact. The machines designed to take drills up to ⅝-in. diameter may be operated by either direct or alternating current. Another advantage of the design lies in the fact that the drill spindle is offset, so that holes may be drilled close to a flange or similar projection.

The Gear Case is a light aluminium casting reinforced with ribs to resist distortion. This case is not only an efficient housing for the gears, but forms a grease chamber from which the gears and main spindle bearing receive ample lubrication. Each end of armature shaft runs in a ball journal bearing fitted in a steel housing, and the thrust is also taken on a ball thrust washer. The gear spindles run in high-grade phosphor bearings.

The Gears, accurately generated, are made of special heat-treated steel, hardened and ground.

The Motor is series wound. The adoption of this type of winding renders the speed automatic, the pressure used to force the drill into the work governing the speed of the drill. The armature and field coils are form wound and thoroughly insulated.

The Armature is of the slotted drum type built up of soft steel laminations, these being made from special steel adapted to the purpose and uniformly insulated. On the DC.2 and larger machines the armature is built on a hollow shaft to preserve lightness.

Ventilation.—One of the distinctive features of the Van Dorn drills is the system adopted for thoroughly cooling the motor. It consists of a large aluminium fan of special design, screwed on the armature shaft. It is so constructed and placed in such a position as to draw out the hot air at one end of the machine, to be replaced by cool from the other. A constant draught is therefore maintained, ensuring a low temperature.

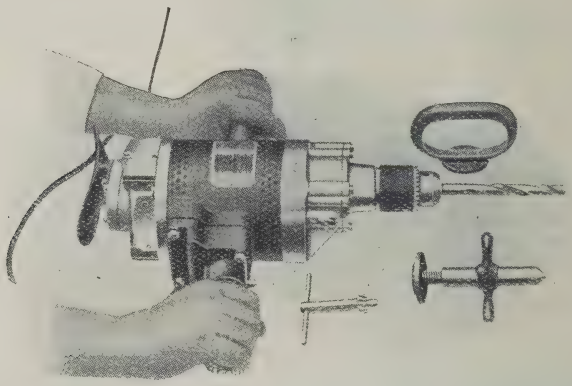


Fig. 32.

The Switch is of the quick-break type with a capacity of 5 per cent. above the duty required. By the use of release attachments the switch has the same rapidity in breaking as in the making contact. The larger sizes are provided with a quick-acting self-releasing attachment.

A Safety Fuse, consisting of a small piece of fuse wire held by screwed terminals, is fitted to machines up to and including ½-in. capacity, and may be quickly replaced if necessary; this prevents the possibility of damage to the machine from overloading or from a short circuit, which is generally caused by the operator omitting to clean the commutator.

STANDARD WORKS

Surveying as Practised by Civil Engineers and Surveyors

Including the Setting-out of Works for Construction and Surveys Abroad, with many Examples taken from Actual Practice. A Handbook for Use in the Field and the Office, intended also as a Text-book for Students. By JOHN WHITELAW, A.M.Inst.C.E. Eighth Edition, thoroughly Revised and Enlarged, by Colonel Sir GORDON R. HEARN, C.I.E., D.S.O., Assoc.Inst.C.E., and late Chief Engineer, Indian State Railways. 594 pages. 297 Illustrations. Demy 8vo, cloth *Net* **16s.**

Reinforced Concrete Bridges

The Practical Design of Modern Reinforced Concrete Bridges, including Notes on Temperature and Shrinkage Effects. By W. L. SCOTT, M.Inst.C.E., Chief Engineer, Considère Constructions, Ltd. Third Edition, Revised and Enlarged. 280 pages. 140 Diagrams. 33 Photographic Plates. Royal 8vo, cloth ... ... [*Nearly ready.*

Water Supply of Towns and the Construction of Waterworks

A Practical Treatise for the use of Engineers and Students of Engineering. By W. K. BURTON, A.M.Inst.C.E., &c. Fourth Edition, Revised and Enlarged, by J. E. DUMBLETON, A.M.Inst.C.E., Author of "Wells and Bore-holes for Water Supply." In Two Volumes. Vol. I.—**Collection and Purification Works**; Vol. II.—**Works for Distribution.** With Folding Plates, Diagrams, and Illustrations. Crown 4to, cloth Per Vol. *Net* **25s.**

Water Channels

By G. HIGGINS, M.C.E., University of Melbourne, M.Inst.C.E., M.Am.Soc.C.E., formerly Lecturer on Civil Engineering at the University of Melbourne. 144 pages. Illustrated by 32 Figures, 7 Diagrams, and 16 Plates. Demy 4to, cloth ... ... *Net* **21s.**

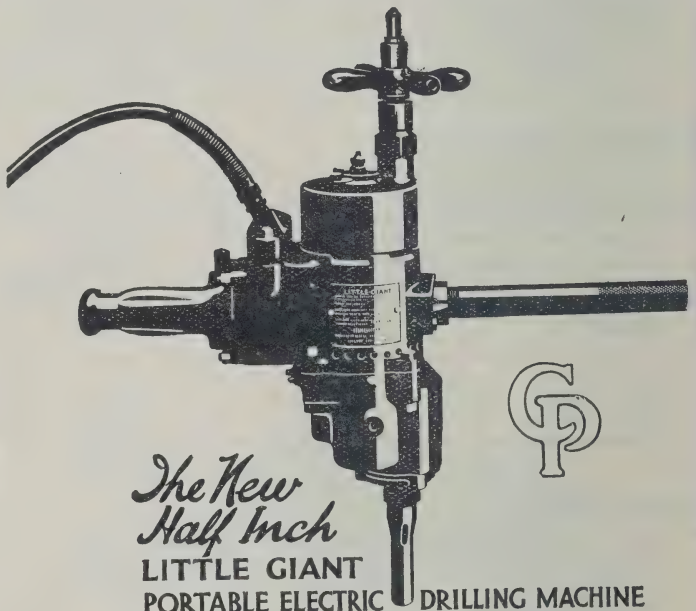
Mechanical Handling and Storing of Material

Being a Treatise on the Automatic and Semi-Automatic Handling and Storing of Commercial Products. Third Edition, thoroughly Revised. By GEORGE FREDERICK ZIMMER, A.M.Inst.C.E., with a Foreword by Sir JOHN PURSER GRIFFITH, M.A.I., M.R.I.A., Past-President of the Institution of Civil Engineers. 820 pages, with over 1,100 Illustrations (including numerous Folding Plates) specially prepared for the Work. Super royal 8vo, strongly bound in Buckram ... ... *Net* £3 3s.

CROSBY LOCKWOOD & SON
Stationers' Hall Court, London, E.C. 4.

LITTLE GIANT

PORTABLE ELECTRIC TOOLS



The New Half Inch

LITTLE GIANT

PORTABLE ELECTRIC DRILLING MACHINE

Little Giant Portable Electric Drills supplied for drilling $\frac{1}{4}$″ to 2″ in metal.

Little Giant Portable Electric Grinders in various types and sizes also supplied.

Write for Electric Catalogue No. 16, giving full particulars of our Portable Electric Tools.

THE CONSOLIDATED PNEUMATIC TOOL CO., LTD.

EGYPTIAN HOUSE, 170 PICCADILLY, LONDON, W.1

Telegrams: "Caulking, Piccy, London." *Telephone: Gerrard 9215*

Branches all over the World

F. 1043

MINIMUM DRILLING PERFORMANCE OF UNIVERSAL PORTABLE ELECTRIC DRILLS.

Carbon-steel drills in 0·20 per cent. to 0·30 per cent. carbon steel.

Drill Size. In.	Speed.	Rate of Feed. In. per Min.	H.P. at 1 H.P. per Cub. In. Metal Removed per Min.*
$\frac{1}{4}$	800	$1\frac{1}{2}$	0·08
$\frac{5}{16}$	650	$1\frac{3}{8}$	0·10
$\frac{3}{8}$	525	$1\frac{1}{4}$	0·13
$\frac{1}{2}$	375	1	0·19
$\frac{5}{8}$	300	$\frac{3}{4}$	0·23

THE LITTLE GIANT ELECTRIC DRILL.

(*The Consolidated Pneumatic Tool Co., Ltd.*)

This drill, fig. 33, is made in many sizes and in A.C., D.C., and universal types. The armature is built up on a one-piece shaft and pinion, hardened and ground, ensuring perfect alignment of pinion at all times. On large sizes, the armature core and commutator are mounted on sleeves, and may be removed from the shaft. Forced ventilation is obtained by means of a fan mounted on the armature shaft. The coils of the armature are former wound, heavily insulated, and carried in insulated slots in the armature core. Other important features are: The use of ball

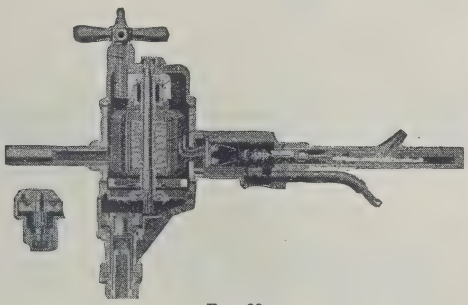


FIG. 33.

bearings, long feed screw (entering into the body of supporting bracket), thus giving the same rigidity as a telescopic feed; also a quick-acting control switch, making slow break impossible and preventing destructive burning of contacts, and a removable, and therefore easily replaced, tool socket in the spindle.

THE LITTLE GIANT ELECTRIC HAMMER DRILL.

(Universal Type.)

(*Consolidated Pneumatic Tool Co. Ltd.*)

This tool (fig. 34) is equipped with a universal motor, and will operate on direct or alternating current. It is designed for drilling in stone or concrete where a hammer blow is necessary to do effective work.

The blow is delivered on the head of the drill bit by a piston governed by compressed air, consequently the electric motor is not subjected to serious mechanical strains or vibration. The tool is well balanced and all parts readily accessible. A switch is conveniently located in the

* Extensive tests have shown this to be a reliable basis for determining power requirements.

handle for the control of the electric current. All bearings throughout are of the latest ball type provision being made for the easy lubrication of all revolving and reciprocating parts.

Wound for 110 or 220 volts (Universal Type, to operate on either D.C. or single-phase A.C.).

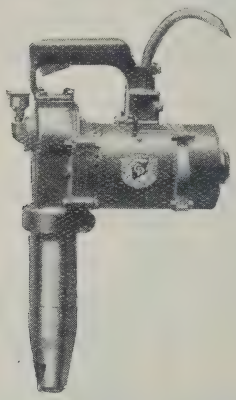


FIG. 34.

Size	No. 1	Weight . .	24½ lbs.
Maximum capacity, in stone or concrete	1 in.	Overall length	17½ in.

Equipment.—Three yards of flexible cable. Two drill steels. Specify type, diameter, and length when ordering. If the tool is required for light chipping, two flat chisels will be supplied instead of drill steels.

Energy consumption . . . 310 watts.

THE LITTLE GIANT AIR DRILL.

(The Consolidated Pneumatic Tool Co., Ltd.)

Fig. 35 shows a Little Giant Air Drill in section. This particular machine, which is representative of the majority of air drills on the market, has four cylinders arranged in pairs diagonal to a two-throw crankshaft, thus two cylinders diagonally opposed operate on one crank,

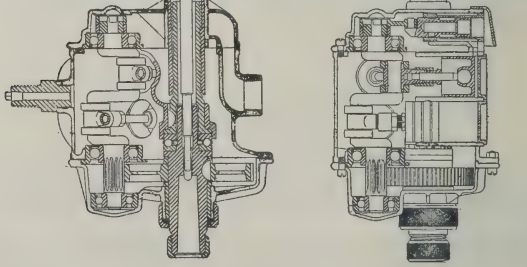


FIG. 35.

given the maximum power for the minimum weight and overall dimensions; such an arrange ment also gives best mechanical balance and freedom from vibration.

One end of the case carries a feed screw, breast plate or grip handle, the other the drill chuck. On each side (usually carried from stout bosses on the drill case) are two handles, usually of steel pipe for lightness, one handle acting as air inlet and having a cock or throttle incorporated therein. See fig. 36.

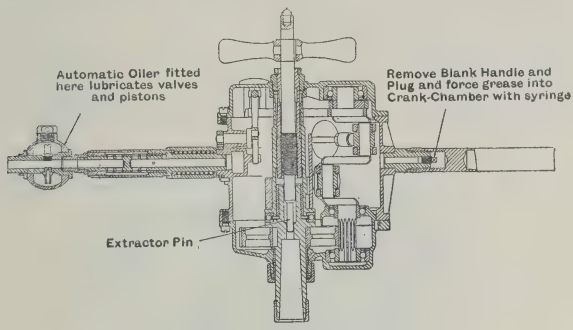


FIG. 36.

Bearings are usually of the ball or roller type, as these suit the high speeds and powers which are developed in such relatively small frames. Sometimes even the big ends of the connecting rods are fitted with balls or rollers, though this is exceptional.

As these machines develop very considerable power (from 1 to 5 h.p.) for their size and weight, it is necessary to use high-grade materials, consequently special steels are used, and all parts are specially heat treated and ground to a finish.

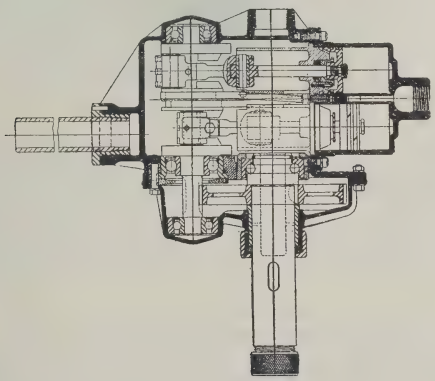


FIG. 37.

Generally, piston or corliss type valves are used, operated by eccentrics of the usual type. Occasionally, a flat plate type of valve is employed, and sometimes a rotary valve has been used with success, the latter being driven by a spur wheel from the main shaft.

A type of tool made by the Consolidated Pneumatic Tool Company is the 'Red Giant,' fig. 37, which contains a twin cylinder double-acting engine. This engine develops equal power to a four-cylinder type, and has crosshead guides, piston valves, and a two-throw balanced crankshaft.

One problem with these small high-speed air engines is the lubrication of the parts, as the drill has to operate in any position that the user may find convenient. Sometimes it operates with the crankshaft horizontal, sometimes vertical, and often at any intermediate position which is necessary on the job, and the difficulties are still further complicated by the necessity of providing the crank case with a 'breather.'

Fig. 38 shows the arrangement introduced by the Consolidated Pneumatic Tool Company. By this arrangement the atmosphere is admitted automatically at the top end of the crank case only, *i.e.* at the end where there is no oil. A tube, which is open to the atmosphere through a **T**-piece in the middle and may be closed at either end by a double cone-shaped plug, which is free to move inside the tube, is fitted approximately parallel to the crankshaft. When the

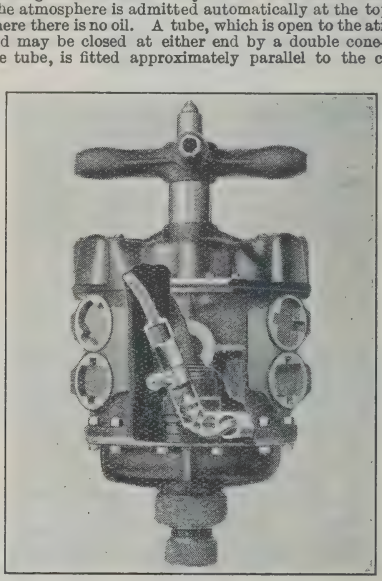


FIG. 38.

machine stands with the drill point downwards, the oil is obviously at the lower end of the crank case, and the lower end of this tube is submerged in oil. The cone plug also falls to this end and closes the tube, so that no oil can be blown into it, and at the same time leaves the top end free to breathe to the atmosphere through the **T**-connection, to the outer air.

If the tool is turned upside down, the oil flows to the other end of the crank case; the cone plug drops to the other end also and now closes that end.

If the machine is operated with the crankshaft horizontal, the cone plug takes no definite position, and the crank case may breathe through either end, and no oil gets blown out.

Drilling Appliances.

THE 'SELSON' GRAVITY DRILLING MACHINE VICE.

In this arrangement, fig. 39, the jaws are opened by sliding them up the angle guides, the maximum opening of the upper jaws being 8 ins., the lower 2½ ins. When the work is set, gravity, supplemented by the pressure of the drill, forces the jaws downwards in the guides, ensuring an absolutely firm hold of the work and making the vice practically an automatic tool.

A slight turn of the handle prevents the work from becoming slack when the pressure on the drill is decreased at the bottom of the hole.

The vice can be used loose on the machine table, except in extreme cases. Slots are, however, provided for bolting it down.

The vice will securely hold all round work from ½ in. to 8 ins. diameter. Work is ensured against turning, even with slight pressure on the drill. Special shapes are held as firmly as regular ones.

There are but five main parts in the vice, which adds to its simplicity.

Size of base, 12 ins. × 6½ ins. Approximate weight, 45 lbs.

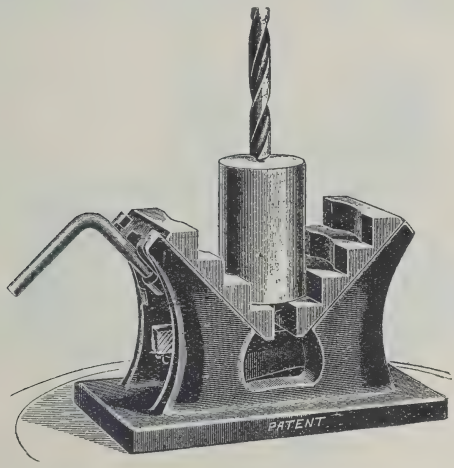


FIG. 39.

V-BLOCK WITH ADJUSTABLE CLAMP.

A compact method of fitting a clamp to grip work in a V-block for marking or for drilling purposes, is to make the block with uprights composed of two plates attached to the ends of the block (fig. 40). Three pinholes are drilled through the uprights, into which an ordinary headed pin is slid to sustain the cranked clamp seen. This is tightened down upon the work by setting up the screw. (*'Machinery.'*)

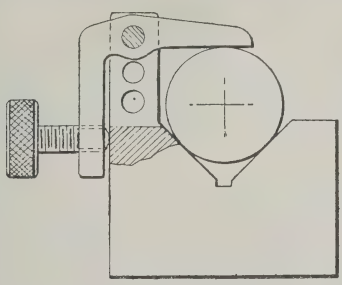


FIG. 40.

THE 'BROOKE' QUICK-CHANGE UNIVERSAL DRILL JIG.

This arrangement (fig. 41) is for the purpose of drilling pin holes in bolts, studs, pins, etc., for split pins, etc. It consists of a baseplate, on which are mounted two pillars carrying, at the top, a steel cross beam, which serves as a clamp for holding the work and also as a guide to the drill by means of suitable bushes. The jig can either be used on a table with a plain surface or bolted down, as required. For plain service tables, a locating pin is supplied to fit the base-

plate and hole in the centre of drilling machine table, keeping the guide bush in the cross beam always in line with the drill spindle.

The work is held in a V-block, provided with a groove on each side, fitting the square portions of the pillars, which thus act as guides, enabling the block to be rapidly adjusted to the required position indicated by pointer and rule, and afterwards locked in place by a small lever instead of the hexagon nut shown in the fig. This block admits bolts from 1½ in. to ¾ in. diameter,

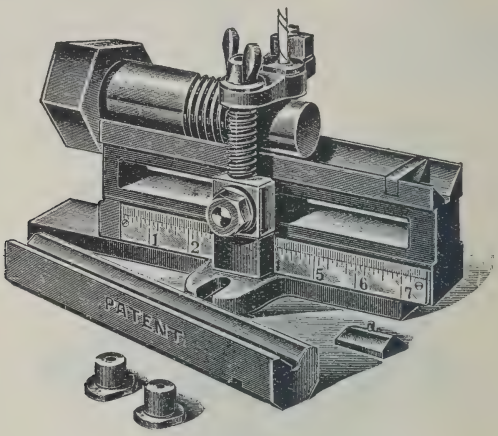


FIG. 41.

and for smaller work an additional V-block is supplied, which fits into the larger one, and can be used for objects down to ½ in. diameter. For studs and other parallel work, steps are provided on the right of both V-blocks, against which the end of the stud rests.

MAGNETIC CHUCKS.

The Magnetic Chuck is now largely used for holding machine parts of iron and steel for grinding, turning, and planing operations. Many pieces, such as milling cutters, thin saws, air-compressor valve discs, chain links, etc., can be more effectively held by this method than by any other. Not only is there a great saving in the time ordinarily required for setting up, but the likelihood of distorting the job due to unequal straining of the holding down bolts is eliminated.

It must be understood that the holding power of a magnetic chuck depends as much on the shape and section of the piece to be held as on the magnetic chuck itself. If the piece is small in section the hold will be very light, no matter what the section or intensity of the chuck magnets may be. The hold is entirely dependent upon the number of magnetic lines that can be passed through the section of the piece, and not on the total magnetism that the chuck can produce. Some makers claim a holding power of 100 lbs. per sq. in., this having been attained on a perfectly surfaced cube of Swedish iron applied between the chuck poles. This affords no useful measure of the holding value of a magnetic chuck, as theoretically a small magnet either of steel, wrought iron or even cast iron, if perfect at the joints, would give (on such a cube) 100 lbs. per in. On the other hand, if two such cubes were placed together the pull would not be proportionately increased, as the magnet being small the section of the first cube would absorb all the magnetism it could supply. Thus a chuck, say 30 ins. by 8 ins., which would pull 100 lbs. on any sq. in. of its surface could not, under any circumstances, pull 30 ins. by 8 ins. by 100 lbs., or nearly 11 tons, on its entire surface. The pitch of the poles should be arranged to suit the class of work to be held; if the pieces are of heavy section, the magnet must be of heavy section to get the best results.

Fine Grinding.—For fine grinding it is well to reduce the magnetic pull of a chuck face, either by inserting resistances in the coil circuit, or by the interposition of non-magnetic material

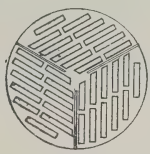


FIG. 42.

between the work and the chuck face. The lighter the pull the more accurate the resultant work. Specially constructed chucks of a very light but even pull are necessary for thin saws, light piston rings, etc. Grinding with water, soda and other liquids makes it absolutely essential that the magnetic chucking apparatus must be impervious to water. In the Humphreys chucks

FIG. 43.

FIG. 44.

FIG. 45.

FIG. 46.

FIG. 47.

one of the leading features is the insulating method employed. The wire itself, before winding, is insulated with a special liquid having very high insulating properties and capable of withstanding weak acids, alkalies, saline solutions, or alcohol. After winding and coating with a special fabric for mechanical protection, the coils are heated to 200° F., and placed in a heated

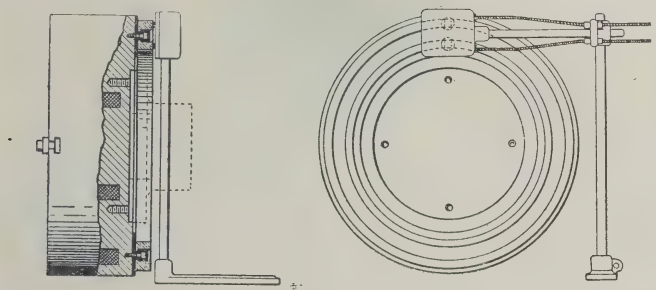


FIG. 48.

tank from which all air and moisture is then extracted. After being subjected to this high vacuum for some time, the tank, without breaking the seal, is filled with a liquid asphaltum compound at high pressure, rendering the coils absolutely proof against moisture, etc. The coils are then placed in position over the cores of the magnets, and the whole chuck body filled and made solid with the asphaltum compound.

Fig. 42 shows the face of a Humphreys (Pat.) circular chuck suitable for piston rings, saws, milling cutters, and general work. These chucks are constructed in sizes ranging from 6½ ins. diameter to 72 ins. diameter. Other designs of chuck faces for special work are shown in figs. 43,

44, 45, 46, and 47; the finer pole designs being used for very small pieces, and the simpler 8-pole design for larger section pieces.

The method of current supply is shown in fig. 48; this is used for dry grinding only. The chuck is fitted to a lathe or grinding machine spindle exactly as an ordinary mechanical chuck.

The brass slip-ring running with the chuck itself collects the current from a contact box, which is held by a tubular support from the lathe or machine bed. The contact box contains a pair of carbon brushes actuated by springs; these brushes are totally enclosed. The method is simple and effective, especially in cases where the chuck is only used occasionally and the apparatus has to be removed and set up each time. Fig. 49 shows an arrangement adopted by the above makers for both horizontal and vertical machine spindles. A hollow spindle machine

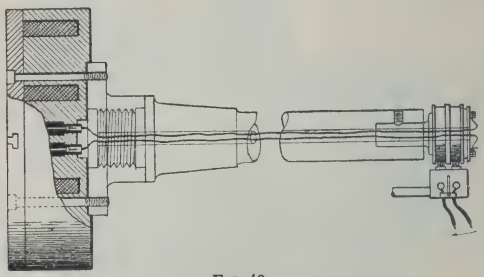


FIG. 49.

is necessary in this case, and the advantage is that all water suds or soda mixture is kept entirely away from the running contacts. The current is carried from the contact rings through the spindle, and is conveyed to the chuck windings by means of a pair of plugs fitting directly into the back of the chuck.

Magnetic chucks are not adaptable to alternating current, and must be used on the voltage for which they are designed. If a higher voltage be applied there is a likelihood of burning out the coils, and with a lower voltage the full power of the chuck is not developed. Fig. 50 shows

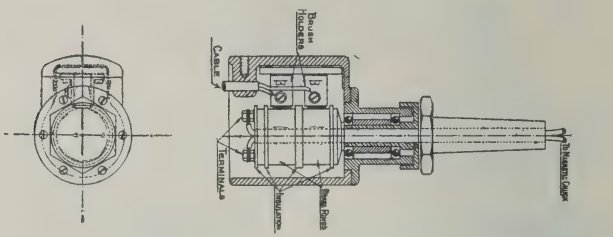
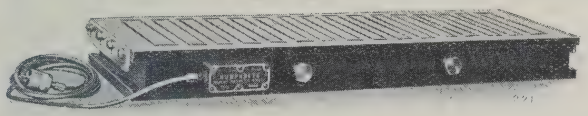


FIG. 50.

the contact arrangement when used with a vertical spindle; it is totally enclosed, the shank is tapered and fitted with a nut to facilitate removal. An oil throw or guard is provided to keep off the bearing oil.

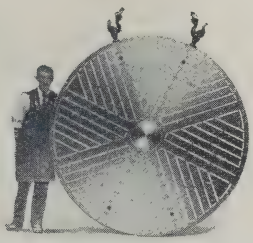
Flat Chucks.—In fig. 51 is shown the details of the 'Humphreys' standard flat chucks, suitable for grinding, planing, and milling machines. These are constructed in sizes up to 72 ins. by 10 ins.; they are fitted complete with a rise and fall parallel strip and an adjustable end-stop. Movable fingers or clamps may be set to support work set on edge; these fingers are secured by set screws in a T-slotted bar, which may be fixed conveniently at any height. The end-stop is necessary to take the thrust of a milling or planing cut; in most grinding operations it is not necessary to use it.

HUMPHREYS
PATENT MAGNETIC CHUCKS



Rectangular Pattern 6″ to 72″ long

CONSTRUCTED ENTIRELY OF HIGH PERMEABILITY MAGNET STEEL.



Circular Pattern 6″ to 84″.

DEMAGNETIZING apparatus for hardened steel.

J. H. HUMPHREYS & SONS,
Engineers,

Telegrams:
'Humphreys, Oldham.'

OLDHAM.

THE METALLURGIST.

Gratis Supplement to The Engineer.
Appears in the last issue of each month.

A paper of 16 pages 11 × 8 devoted to the Science of Metallurgy.

Special articles by scientists and experts.

Critical reviews of metallurgical work in Great Britain and abroad.

THE paper for the works' metallurgist.

The issues of The Engineer containing THE METALLURGIST may be ordered separately from Newsagents, or

The Manager,
THE ENGINEER
28 Essex Street, Strand, London, W.C.2.

Price, including Postage, **15/- per annum.**

Fig. 52 shows a variation of this form of chuck for machining taper pieces. This chuck is carried on a sub-base hinged at one end, and has a fine knurled screw adjustment at the other. The index graduation is in inches per foot. When the setting is at zero the chuck can be used for ordinary flat work.

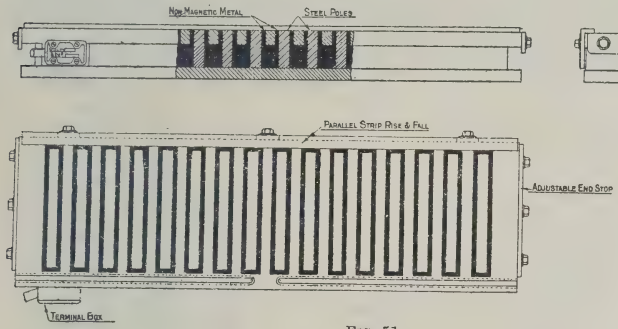


Fig. 51.

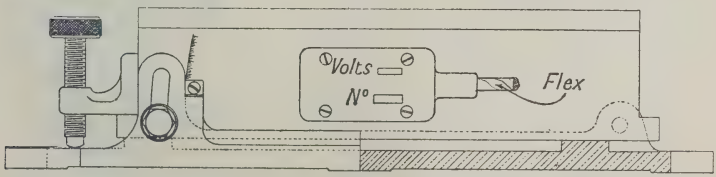


Fig. 52.

To facilitate the grinding of square sections on the chucks shown in figs. 51 and 52, an attachment has been recently devised. This consists of a rectangular block made up of steel plates of very high magnetic permeability interspaced with non-magnetic material. Placed on the chuck face without any connections, either mechanical or electrical, it brings into operation a vertical magnetic face by means of which work can be set up and ground square.

To grind a rectangular piece, grind two opposite faces on the magnetic chuck in the ordinary manner, then by holding the piece by one of the ground surfaces against the side of the attachment and supported underneath to the requisite height, a perfect square can be obtained.

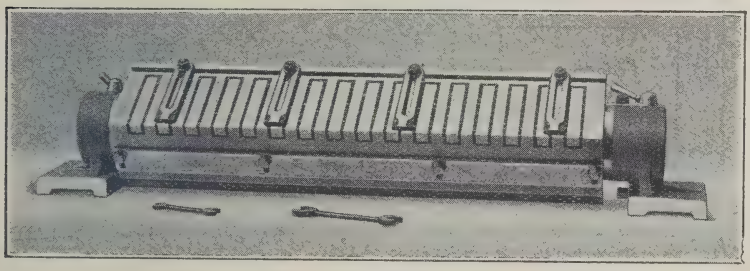


Fig. 53.

A new Humphrey Swivelling Chuck is shown in fig. 53. A graduated ring enables the operator to set the chuck at any desired angle.

The poles on this chuck are carried right up to the edge, thus permitting narrow pieces to be held on the side as well as on the top face of the chuck

Switches.—One of the most important pieces of apparatus in connection with the magnetic chucks are the switches used for operating. When the current from a magnetic chuck is cut off, a back current is induced in the chuck coils. This induced current is of a very high voltage, and if the switch gear is not perfectly designed to deal with it, the insulation of the chuck, however well designed and carried out, will inevitably break down. The switch must be double-pole, that is, it must operate on both the positive and negative supply wires at the same time, also it must be capable of admitting current to the chuck in either direction, and in order to deal with the induced back current when switching the chuck off, it must be provided with a non-inductive resistance which must automatically be put in circuit with the chuck prior to breaking the circuit. Fig. 54 shows the 'Humphreys' No. 7 Demagnetising Switch. To operate it the

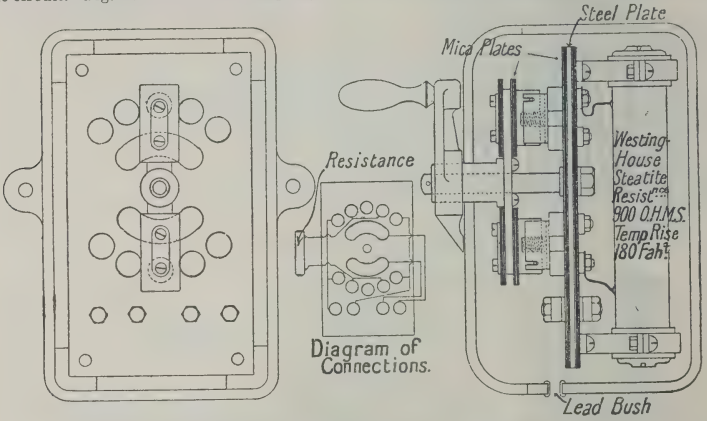


FIG. 54.—No. 7 Demagnetising Switch, Slow Break, Capacity 2 Amps.

switch handle must be moved past the 'Off' position and held momentarily on the 'Reverse,' going back quickly to the 'Off' position; this is for the purpose of demagnetising the chuck sufficiently to allow of the work to be easily removed. The view on the left of the figure shows the arrangement of contacts. For very large chucks a special oil-break switch is furnished; the method of operation is similar to that of the No. 7 switch, but the contacts are immersed in oil and the non-inductive resistances are carried in a separate iron case forming the base of the switch.

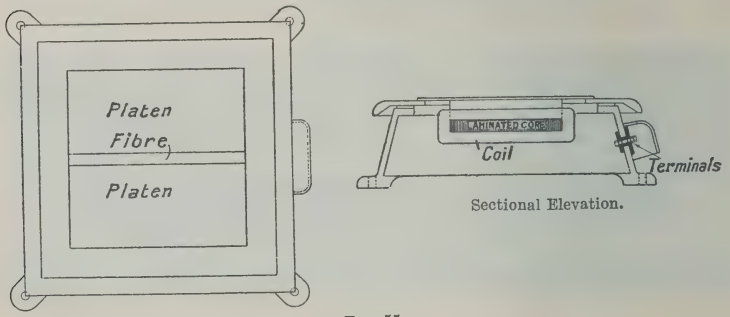


FIG. 55.

The sectional view on the right-hand side of the figure shows the non-inductive resistance. The main plate is mica insulated; in several sizes of switches this main plate consists of a slab of slate.

The arrangement of the connections, etc., is shown in the central view of fig. 54.

Demagnetisers.—Consequent on the use of magnetic chucks is the permanent magnetic condition imparted to the finished work, which with milling cutters and similar articles would prove a decided objection. A magnetised milling cutter would collect to itself the cuttings as produced, and though this might not impair its efficiency as a cutter, it would be a source of annoyance to the operator. Fig. 55 shows a sectional elevation and plan of a demagnetiser by J. H. Humphreys & Sons. There are two working platens, and all that is necessary is to pass the work over these platens, when it will be instantly demagnetised. As shown in the section drawing, immediately under the working platens is a powerful laminated magnet, which takes from the transformer a low period, low voltage, alternating current. The transformer carries

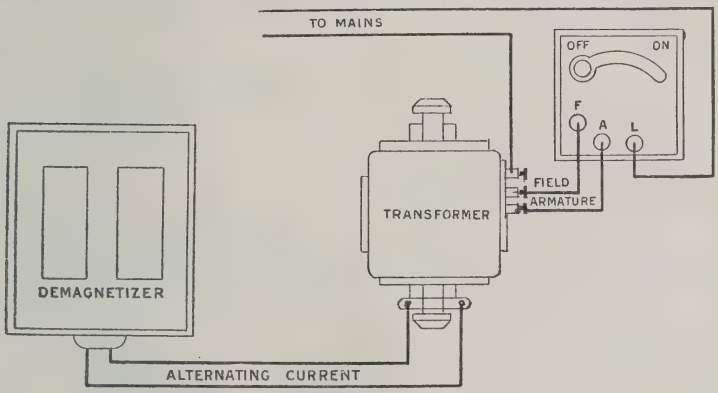


FIG. 56.

a double winding, one connected to a pair of slip-rings at one end of the shaft, the other being an ordinary motor winding which is connected to the mains through a starting rheostat exactly as an ordinary electric motor. This arrangement is very convenient, no fixing is required, and the demagnetiser can be placed in any position independent of the transformer.

The whole apparatus, fig. 56, includes demagnetiser, rotary transformer, and starting rheostat.

Felt Wipers on Slides.

To prevent grit working under the slides of machine tools owing to rounded corners, felt wipers may be fitted as shown in fig. 57. The felt is held in place by 1-in. angle iron secured to

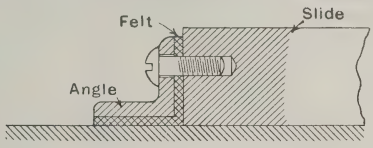


FIG. 57.

the slide by button-head screws. In practice the felt retains the oil and helps to keep the slide lubricated. (*Machinery.*)

POWER REQUIRED FOR MACHINE TOOLS.

LATHES.

Heavy Crank-shaft Lathes with Twin Saddles and Duplex Rests.

Height of centres in inches	42	48	60	66
H.P.—Main drive	40	45	50	60
H.P.—Auxiliary	15	15	20	20

LATHES (*continued*).

Wheel Lathes for Locomotive Wheels (High Speed).

Diam. of Faceplate.	H.P. required for Two Tools.	Diam. of Faceplate.	H.P. required for Two Tools.
Ft. Ins.		Ft. Ins.	
3 0	10	5 0	18
3 6	12	6 0	21
4 0	14	7 0	23
4 6	16	8 0	25

The largest diameter swung may be taken as 4 ins. above the faceplate diameter. For lathes fitted with duplex rests and 4 tools in operation, increase the power given above by 50 per cent., while with 6 tools in operation double the power given is required.

Tyre Boring Lathes suitable for use with High Speed Steel.

Diam. of Faceplate.	H.P.	Diam. of Faceplate.	H.P.
Ft. Ins.		Ft. Ins.	
3 6	20	6 0	35
4 0	25	7 0	35
5 0	30		

Axle Lathes (High Speed).

Height of Centres.	H.P.	Height of Centres.	H.P.
Ins.		Ins.	
8½	15	12	25
10	20	15	30

The machine is provided with a compound rest on each side of the central driving head.

Standard Lathes suitable for use with High Speed Steel.

Height of Centres.	H.P. required.	Twin Duplex Rests.	
		Height of Centres.	H.P. required.
Ins.		Ins.	
6	5	36	30
9	7½	42	35
12	12	48	40
15	15	60	50
18	20	84	75
24	30	96	100

Surfacing and Boring Lathes (High Speed).

Diam. of Faceplate.	H.P.	Diam. of Faceplate.	H.P.
Ft.		Ft.	
2	7	5	20
2½	9	6	25
3	12	7	30
3½	15	8	30
4	17½	10	35

DRILLING AND BORING, SCREWING AND TAPPING MACHINES.

Drilling and Boring Machines.

Class of Machine.	Diam. of Spindle in ins.	H.P.
Sensitive	11/16	1
,,	13/16	1½
,,	1	2
Vertical	1½	3
,,	2	4
Vertical and Radial	2¼	7½
Radial and Horizontal	2¾	9
,, ,,	3	10
,, ,,	3¼	12
,, ,,	3½	15

Bolt Screwing and Nut Tapping Machines.

Size of Machine in ins.	H.P.
¼ to 1¼	1½
¼ to 2	2½
½ to 3	4

Tube Screwing Machines.

Size of Tube.	H.P.
Up to 2 ins. diam.	1
Up to 3 ins. diam.	2
Up to 4½ ins. diam.	3
Up to 10 ins. diam.	4
Up to 12 ins. diam.	5

Shaping and Slotting Machines, Boring and Turning Mills.

Shaping Machines.		Slotting Machines.		Vertical Boring and Turning Mills with two Cutter Bars on Cross-side.	
Stroke.	H.P. per head.	Stroke.	H.P.	Diameter Admitted.	H.P.
Ins.		Ins.		Ft.	
6	1½	6	1½	3	7½
8	2	8	3½	4	10
10	3	10	5	5	12½
12	5	12	7½	6	15
15	5	18	10	8	20
18 Double Head.	7½	24	15	10	25
24	10	30	20	15	35
30	12½	36	25	20	50
		54	30		
		72	35		
		Locomotive Frame Plate Slotting Machine 10 h.p. per head.			

Planing Machines.

Planing Machines (High Speed).		Vertical and Horizontal Planing Machines (High Speed).	
Size in Ft. and Ins. Width × Height.	H.P. required.	Size in Ft. and Ins. Vertically × Horizontally.	H.P. required.
Ft. Ins. × Ft. Ins.		Ft. × Ft. Ins.	
1 9 × 1 9	2½	7 × 10 6	18
2 0 × 2 0	3½ Two tool boxes on cross-slide.	9 × 12	20
2 6 × 2 6	6	12 × 15	25
3 0 × 3 0	8	15 × 20	30
3 6 × 3 6	10	18 × 24	35
4 0 × 4 0	20 Two tool boxes on cross-slide and one on the front upright.	20 × 25	45
5 0 × 5 0	25		
6 0 × 6 0	30		
7 0 × 7 0	35		
9 0 × 9 0	45		

Punching Machines.

Diameter of hole in inches	¾	1	1⅜	1½	1⅞
Plate thickness in inches	⅝	⅝	1	1¼	1⅝
H.P. required	1	2½	5	7½	10

PUNCHING, SHEARING, AND CROPPING MACHINES.

Hole Punched.		Flat Bar Sheared at one Stroke.	Diameter Cropped.	H.P. required.
Diameter.	Plate Thickness.			
Ins.	Ins.	Ins. Ins.	Ins.	
1	$\frac{5}{8}$	$3\frac{1}{4} \times \frac{5}{8}$	$1\frac{1}{2}$	6
$1\frac{3}{16}$	$\frac{13}{16}$	4×1	$1\frac{3}{4}$	8
$1\frac{5}{16}$	$\frac{7}{8}$	$4\frac{3}{8} \times 1$	$2\frac{3}{8}$	10
$1\frac{3}{8}$	1	$4\frac{3}{4} \times 1\frac{1}{8}$	$2\frac{1}{2}$	12
$1\frac{1}{2}$	$1\frac{1}{4}$	$5\frac{3}{8} \times 1\frac{1}{4}$	3	16
The above is suitable for 20 strokes per minute.				

GRINDING MACHINES.

Plain Grinding Machines. (Heavy duty.)		Universal Grinding Machines.		Crank-shaft Grinding Machines.	
Maximum Swing.	H.P. required.	Maximum Swing.	H.P. required.	Maximum Swing.	H.P. required.
Ins.		Ins.		Ins.	
$7\frac{1}{2}$	5	8	3	12	12
10	$7\frac{1}{2}$	10	5	15	15
12	10	12	5	18	20
15	15	15	$5–7\frac{1}{2}$	24	25
20	20	20	$7\frac{1}{2}–10$		
24	25				
30	30				

Internal Cylinder Grinding Machines.		Vertical Spindle Surface Grinding Machines.	
Normal Capacity. (Diameter × length.)	H.P. required.	Diameter of Wheel.	H.P. required.
Ins.		Ins.	
6×15	3	12	15
9×18	5	14	20
18×42	$7\frac{1}{2}$	18	25

MILLING SPEEDS AND FEEDS.

With regard to milling speeds and feeds, the amount of reliable experimental data available is very meagre and the machinist must for the present content himself with compiling data for himself. The subject is very complicated, and he will require to make the fullest possible observations at least at the commencement. For this purpose, he will note the following: dia. and width of cutter, revs. per min., depth and width of cut, feed in inches per min., number of teeth in the cutter, type of cutter, feed per tooth, material and nature of the work. To measure the capacity of the machine by the feed per tooth alone may give misleading results.

TESTS MADE ON A NO. 22 'SINGLE PULLEY' HORIZONTAL MILLING MACHINE MADE BY MESSRS. ALFRED HERBERT LTD., COVENTRY.*

$4\frac{1}{2}$-in. High-speed Cutter, 10 Teeth, 2-in. Hole.

Block (Cast Iron and Mild Steel) 8 ins. wide at 60 turns = 70 ft. per minute.

(Power required to run motor and machine light, ·7 net H.P., has been deducted from the recorded H.P.s to give the figures in the table.)

Depth of Cut.	Feed Measured	Net H.P.	(Cast Iron) Metal Removed.		Feed Measured	Net H.P.	(Mild Steel) Metal Removed.	
			Cubic Ins. per Min.	Cubic Ins. per H.P. Min.			Cubic Ins. per Min.	Cubic Ins. per H.P. Min.
Ins.	Ins.		Ins.	Ins.	Ins.		Ins.	Ins.
·24	$17\frac{5}{8}$	20·3	33·80	1·66	$11\frac{3}{8}$	30·8	21·84	·71
·25	$17\frac{1}{2}$	23·0	35·00	1·52	$10\frac{7}{8}$	29·6	21·75	·73
·26	$17\frac{3}{8}$	26·1	36·15	1·38	$11\frac{1}{4}$	39·8	23·14	·58
·28	$17\frac{3}{16}$	34·6	38·52	1·11	$9\frac{1}{8}$	28·6	20·44	·71
·30	$13\frac{7}{8}$	23·7	33·30	1·40	$9\frac{1}{16}$	29·6	21·67	·73
·32	$13\frac{1}{2}$	29·6	33·92	1·15	$8\frac{7}{8}$	30·8	22·72	·74
·34	$11\frac{5}{8}$	24·9	31·55	1·27	$8\frac{1}{16}$	42·1	24·31	·58
·36	$11\frac{1}{2}$	27·3	33·12	1·21	$7\frac{5}{16}$	30·8	21·06	·68
·375	$13\frac{1}{8}$	22·8	40·50	1·77	$7\frac{1}{8}$	28·6	21·37	·74
·40	$11\frac{3}{16}$	27·3	35·84	1·32	$6\frac{1}{16}$	27·3	19·40	·71
·42	$11\frac{3}{8}$	26·1	38·22	1·46	6	29·6	20·16	·68
·44	$9\frac{1}{2}$	23·7	32·12	1·35	5	24·9	17·60	·71
·46	$9\frac{3}{16}$	24·9	33·34	1·32	5	27·3	18·40	·67
·48	$9\frac{3}{16}$	27·3	35·33	1·29	$4\frac{15}{16}$	28·6	18·96	·66
·50	$9\frac{1}{2}$	30·8	37·00	1·20	$4\frac{15}{16}$	30·1	19·75	·66
·55	$7\frac{1}{8}$	28·6	33·00	1·15	$4\frac{9}{16}$	30·8	21·45	·69
·60	$6\frac{1}{16}$	23·7	29·76	1·25	$4\frac{13}{16}$	37·1	23·10	·62
·625	$9\frac{3}{16}$	24·9	45·93	1·84	$3\frac{15}{16}$	29·0	19·68	·68
·70	$7\frac{7}{16}$	35·1	41·60	1·19	$3\frac{15}{16}$	30·8	22·05	·71
·75	$6\frac{3}{16}$	28·6	38·80	1·35	$3\frac{1}{4}$	28·6	19·50	·68
·80	$6\frac{1}{16}$	29·6	38·79	1·31	$3\frac{1}{4}$	32·0	20·80	·65
·85	$4\frac{7}{8}$	27·3	33·15	1·21	$2\frac{5}{8}$	26·1	17·85	·68
·90	$4\frac{15}{16}$	29 6	35·54	1·20	$2\frac{9}{16}$	30·8	18·45	·60
·95	$4\frac{3}{4}$	35·1	36·10	1·03	$2\frac{1}{8}$	24·9	16·15	·65
1	4	27·3	32·00	1·17	$2\frac{1}{8}$	29·6	17·00	·57
1·05	$3\frac{7}{8}$	37·1	32·54	·86	$2\frac{3}{16}$	24·9	17·32	·69
1·1	$3\frac{5}{16}$	28·6	29·04	1·02	$2\frac{1}{16}$	28·6	18·15	·64

METAL COLD SAWING. A.W. HIGH-SPEED STEEL.

Inserted-Tooth Saw.	Material.	Cutting Speed.	Time to Cut Through.
Diam. in Ins.		Ft. per Min.	Mins.
36	Steel (joist, 20 ins. deep by $7\frac{1}{2}$ ins. wide; $\frac{3}{4}$ in. web)	122	17
36	Steel (rail, 100 lbs. per yard)	90	2
36	Forged steel (bar 3 ins. diam.)	90	$2\frac{1}{2}$
36	Forged steel (bar 10 ins. diam.)	90	11
36	Forged steel (bar $7\frac{1}{2}$ ins. by 9 ins.)	84	9

* P. V. Vernon (Alfred Herbert & Co., Ltd.), before the Manchester Assoc. of Engrs., 1912.

MISCELLANEOUS.

	H.P.
Double punch, running light	2·0
Punching both ends 1 in. hole, $\frac{7}{8}$ in. plate, 28 punches per minute at each end	5·0
Punch and shear, running light	2·0
Punching hole as above, and shearing $\frac{7}{8}$ in. plate, $5\frac{1}{2}$ in. cut, 28 strokes per minute	6·5
Large plate-bending rolls, running light	5·5
Rolling plate $\frac{9}{16}$ in. thick, 4 ft. 4 ins. wide, and 16 ft. 6 ins. long, endways on (4 ft. 4 ins.)	6·9
Sideways on (16 ft. 6 ins.)	12·2
Plate $\frac{11}{16}$ in. by 4 ft. $8\frac{1}{2}$ ins. by 21 ft. long, sideways on	19·3
Same rolls lifting and lowering top roll: Lifting	8·5
Lowering	7·0
Forcing $\frac{11}{16}$ in. plate down	10·0
Forge fan, for 24 fires	10·5
Angle squeezer, running light	0·8
Squeezing	2·5
Riveting machine, running light	1·6
Riveting	3·0
Plate planing machine, running light	1
Reversing light	2
Cutting	5
Stern frame, boring, cutting	1·75
Angle-cutting machine, running light	1·6
Cutting angle 6 ins. by $3\frac{1}{2}$ ins. by $\frac{1}{2}$ in.	3·5
Winch, lifting 28 cwt. single purchase, 50 ft. per minute	6·5
Ending machine, running light	1·9
Cutting girder 18 ins. by 7 ins.	4 to 4·8
Cold saw, running light 24 in. saw	2·0
Cutting 4 in. by 4 in. angle	4 to 4·5
Air compressor, 10 ins. by 14 ins. cylinder, 50 lbs. pressure	15 to 23·5
Hydraulic pumps, three pumps, $3\frac{3}{4}$ ins. diameter; $4\frac{3}{4}$ ins. stroke, 60 strokes per minute; pressure, 870 lbs.	38·0
Electric winch, running light	1·2
Lifting 24 cwt. 16 ft. per minute	5·6
Vertical cylinder boring machine (marine), 14 ins. diam. bar	15·0
" " " 18 " "	20·0
Broaching machine (screw $2\frac{1}{2}$ ins. diam.)	7·5
Bevel gear planer to take wheels up to 12 ins. diam.	2·0
" " " 24 "	4·0
" " " 36 "	5·0
Sunderland gear planers to cut up to 3 D.P. (No. 6)	7·0
" " " $1\frac{1}{2}$ D.P. (No. 9)	10·0
" " " $1\frac{1}{4}$ D.P. (No. 11)	15·0

WOOD-WORKING MACHINERY.

The following allows for the motors being capable of giving 25 per cent. overload for not less than $1\frac{1}{2}$ hour, without any undue heating. The figures are based upon the average work found in the particular shops where the machines are generally used. The type of motor chosen has a certain influence upon the amount of marginal power, and this point requires careful attention.

MACHINE TYPE AND SIZE.	POWER OF MOTOR REQUIRED FOR AVERAGE WORK. H.P.	MOTOR BELTED TO THE MACHINE OR COUNTER, OR DIRECT COUPLED.	REMARKS.
24 in. and 30 in. Horiz. Log Frame Saw, 1 blade in use	10	Belted to machine	—
36 in. " "	12	" "	This machine takes between 10 and 11 h.p. to start
42 in. Horiz. Log Frame Saw, 1 blade in use	15	" "	—
48 in. " "	18 to 20	" "	—
24 in. Circular Saw Bench (hand feed)	10	" "	—
30 in. Circular Saw Bench (hand feed)	12	" "	—

Machine type and size.	Power of motor required for average work. H.P.	Motor belted to the machine or counter, or direct coupled.	Remarks.
36 in. Circular Saw Bench (hand feed)	15	Belted to machine	—
42 in. Circular Saw Bench (hand feed)	20	" "	—
24 in. Band Saw (hand feed)	1½	Belted or direct coupled	—
30 in. Band Saw (hand feed)	2	" "	—
36 in. Band Saw (hand feed)	3	" "	—
42 in. Band Saw (hand feed)	5	" "	These machines are used on heavier work than the 30 in. and 36 in. sizes
48 in. Band Saw (hand feed)	6 to 7	" "	
36 in. Band Saw (power feed)	8 to 9	" "	—
42 in. Band Saw (power feed)	11 to 12	" "	—
16 in. Double Dimension Saw (hand feed)	5	Belted to machine	—
18 in. diam. Cross-cut Machine (hand feed)	3	" "	—
24 in. diam. Cross-cut Machine (hand feed)	5	" "	—
30 in. diam. Cross-cut Machine (hand feed)	7	" "	—
7 in. by 3 in. Four-Cutter Planer (power feed)	10 to 12	Belted to counter or direct coupled to counter	—
16 in. Plain Surfacing Machine (hand feed)	3	Belted to machine	—
20 in. Plain Surfacing Machine (hand feed)	4	" "	—
24 in. Plain Surfacing Machine (hand feed)	5	" "	Machines above 20 ins. wide do not often plane the full width
Vertical Spindle Molder (light) 2½ ins. deep	2	Belted to counter or direct coupled to counter	—
Vertical Spindle Molder (heavy) 4 ins. deep	3 to 4	" "	—
Vertical Double Spindle Molder (heavy) 4 ins. deep	6 to 7½	" "	—
Vertical Double Spindle Molder (heavy) 6 ins. deep	10	" "	—
Medium type Tenoner (1 scribing spindle)	5	" "	—
20 in. Combined Surfacing and Thicknessing Machine	8	for ordinary seasoned timber	Surfacing is hand feed, thicknessing is power feed
24 in. Combined Surfacing and Thicknessing Machine	10	" "	

When motors are direct coupled it is advisable to use a flexible coupling in order to avoid shock on the motor due to fluctuating loads.

The foregoing powers assume that the cutters or saws are running at about standard speeds, and are in good working condition; also it is assumed that the rate of feeding is about the standard, and not speeded up above the makers' speeds. It is, of course, understood that any machine, particularly saw benches, can be fed until they take an abnormal amount of power, but the intelligent workman can so humour the feed without losing time that he gets almost the maximum amount of cutting done with the minimum amount of power expended.

The power absorbed per sq. ft. surface sawn per minute varies very much in some instances and it is impossible to get anything like an accurate figure, as there are so many variable factors; thus, the amount of power absorbed by the saw increases or decreases according to the amount

of set given to the teeth, and the set requires to be varied considerably according to the class of timber to be sawn: even in the same class of timber the grain in one part of the cut might be such that it closes in on the saw more than in another part of the cut.

SPEED.

The speed of the cutting edge and the angle, in knife tools, and the shape of saw tooth, speed, angle, and thickness of saw, in saws, has a great influence on the amount of power a machine will absorb when doing certain classes of work.

The following list gives the various speeds, etc., of certain machines, the information being the latest practice.

Circular Saws for ripping, soft wood	9,600 to 11,000 ft. per min. (peripheral)	
" " hard "	9,600 to 11,000 " "	
" " ordinary cross-cutting, soft	10,000 " "	
" " " " hard	8,000 to 10,000 " "	
" " (Hollow ground)	10,000 to 11,000 " "	
Ordinary plain Band Saws, soft wood	4,300 to 5,000 " "	
" " hard wood	4,300 to 5,000 " "	
Deal cutting Band Saws, hard or soft	6,500 to 7,000 "	Specially designed
Log Band Saws	6,500 to 8,000 "	machines

Reciprocating Mortising Machines:

Mortising up to $\frac{7}{8}$ in. wide,		$4\frac{1}{2}$	ins.	stroke	(not graduated),	crankshaft speed	400	r.p.m.
" "	1	"	"	"	"	"	350	"
" "	$1\frac{1}{4}$	"	6	"	(graduated)	"	270	"
" "	$2\frac{1}{2}$	"	$6\frac{3}{4}$	"	"	"	255	"
" "	3	"	7	"	(not graduated)	"	160	"

Vertical Deal Frames (Single):

To take timber 12 ins. by 6 ins.	Continuous Feed	"	300	"
" " 18 ins. by 6 ins.	"	"	250	"
" " 28 ins. by 5 ins.	"	"	200	"

Vertical Deal Frames (Double):

To take timber 20 ins. by 6 ins.	"	"	225	"
" " 24 ins. by 6 ins.	"	"	200	"

Horizontal Saw Frame (Single blade):

To take timber	24	ins.	square	2 ft. 6 ins.	stroke	(balanced)	"	320	"
" "	30	"		"	"	"	"	300	"
" "	36	"		"	"	"	"	280	"
" "	42	"		"	"	"	"	260	"
" "	48	"		"	"	"	"	230	"

Double Equilibrium Deal Sawing Frames (balance swing frames):

To cut 2 deals up to 14 inches × 5 inches, 350 revs. per min. (of crankshaft).

"	"	18	"	× 6	"	300	"	"	"
"	"	24	"	× 7	"	250	"	"	"

Bandsaws, blades running on wheels up to 3 feet diameter:

For sawing very hard ivory, etc., traverse of saw blade per minute, 750 feet.

" iron " " " " " " 250 "

Vertical Spindle Moulding Machines:

Single Spindle (light type)	5,000 to 6,000 revs. per minute.	
Double " (")	5,000 to 6,000	" "
Single " (medium type)	4,000 to 4,500	" "
Double " (")	4,000 to 4,500	" "
" " (heavy type)	3,500 to 4,000	" "

High speed Four Cutter Moulders:

Bottom spindle	4,000 revs. per minute,	6 ins. dia. cutting circle app.			
(2) Side spindles	4,000 "	"	6 "	"	"
Top spindle	5,000 "	"	6 "	"	"

The above represents the average machine up to 12 ins. by 4 ins., and gives approximate peripheral cutting speeds of :—bottom spindle, 6,300 ft. per min.; side spindle, 6,300 ft. per min.; top spindle, 7,800 ft. per min.

The extra speed given to top cutter spindle is because this cutter spindle has the heaviest work to do, having to reduce the timber to an even thickness and a given size.

Surface planing machines . . . 5,000 to 5,500 ft. per. min. of cutting circle.
Surfacing and thicknessing machines (combined) 5,000 to 5,500 „ „ „

The above instances are for average cases, and may be accepted as good standard practice at the present time.

Tenoning Machine (light) 4,500 to 5,500 ft. per min. of cutting circle.
„ „ (heavy) 3,500 to 4,500 „ „ „ „
Fret Saws . . . . 1,200 revs. per min. 1½ in. stroke.
Carving Machines (light) . . 8,000 revs. per min. of spindle.
Single Spindle Dovetailing Machines 5,000 to 6,000 revs. per min. of spindle.
Boring Augers (ordinary):

Up to 1 in. diam., 1,800 to 2,000 r.p.m. Up to 2 ins. diam., 1,000 r.p.m. Up to 3 ins. diam., 750 r.p.m.

Emery Wheels, for saw-sharpening, etc., 4,500 to 6,000 feet periphery velocity.

Force Required to Punch Holes and Cut Plates.

If W = maximum force in lbs. required to punch hole,

t = thickness of plate in ins.,

then

$$W = 190{,}000dt$$

for structural steel, soft iron, boiler and steel plate.

As W varies as dt, the same machine and power would punch a $\frac{3}{4}$-in. hole in a $\frac{3}{4}$-in. plate, as would punch a $2\frac{1}{4}$-in. hole in a $\frac{1}{4}$-in. plate, since $\frac{3}{4} \times \frac{3}{4} = 2\frac{1}{4} \times \frac{1}{4}$.

If F = force required to shear plates with straight knives inclined at an angle $\alpha°$ with each other, then

$$F = 30{,}000\, t^2 \cot \alpha°.$$

An approximate rule for finding the force required to cut metal by punching or shearing is

Force = 80 per cent. of the tensile strength of a bar having a cross-sectional area equal to the area of the metal cut.

The side thrust on a shearing blade when it is dull and the plate is tipped may be taken as equal to 1·5 W, when W is the force acting normal to the plate.

Flat punches should always be used when the thickness of the plate is greater than two-thirds the diameter of the hole. If the thickness is less than half the hole diameter, much less force is required when a shearing punch is used. Shearing punches are also recommended for cases where a large number of holes are to be punched close together, because they do not distort the plate as much as flat-ended punches.

The diameter of a punched hole is equal to the diameter of the punch, but if there is an appreciable clearance between the punch and the die, the hole will be conical.

(Lewis Jenkins, University of Cincinnati.)

Power Losses in the Machine Shop.

Engine friction, 10 per cent.; line shafting, 15 per cent.; belts and pulleys, 15 per cent.; empty machines, 15 per cent.; cutting metal, 45 per cent.; total, 100 per cent. Even this efficiency is probably only realised when everything is working at full capacity.

Of all metals cut in the machine shop, cast-iron is said to absorb least power per unit of weight removed, while the harder steels require the maximum. Thus, to remove 1 lb. of cast-iron per hour requires the continuous expenditure of from 0·15 to 0·30 h.p., or, say, an average of $\frac{1}{40}$ h.p., where steel calls for $\frac{1}{25}$ h.p.

An average lathe will cut 10 to 20 lbs. of cast-iron per hour and absorb ¼ to ½ h.p. in useful work, and similarly steel will be removed at a rate of 5 to 10 lbs. and demand ½ to ⅔ h.p. Much also depends on the condition of the machine itself, which may increase its demand for power from 0·36 to 2·3 h.p. very quickly, simply from a dry main spindle.

A lathe run without cut may absorb 0·04 to 0·16 h.p., and this will amount up to 0·3 or 0·4 simply by tightening the work in the centres. A large 84-inch wheel lathe has been found to absorb 1·75 h.p. and when cutting, 5·8 h.p. on heavy work.

Planers use ·035 h.p. per pound of cast-iron removed per hour, and 0·065 h.p. per pound of steel, and it may remove 20 lbs. of soft steel or 30 lbs. of cast-iron per hour. Ordinarily, however, the machine would not consume 1 to 1·3 h.p., as these figures would indicate, but only about ½ h.p. Adding for the machine will probably show an average power of 1·0 h.p. in ordinary work.

TOOL STEEL AND TOOLS.

Treatment of Carbon and High-Speed Steel Tools.

CARBON STEEL.

The term carbon steel has been generally applied to 'straight carbon' and 'carbon-alloy' steel. The former is obtained directly from blister steel, which is melted in crucibles, poured into a cast-iron mould and cooled. Subsequently the ingots are retreated, hammered and rolled to the desired shape and size. The carbon-alloy is obtained by adding small quantities of tungsten, chromium, silicon, etc. The latter steel has distinct advantages over the former in that it is easier to forge, is less liable to failure in hardening, and has a greater cutting capacity. The chemical composition of a representative tool of the latter class is given below.

Chemical Composition of Ordinary Carbon Tool Steel.

Carbon, 1·1 per cent.; manganese, 0·35 per cent.; silicon, 0·17 per cent.; phosphorus, 0·018 per cent.; sulphur, 0·015 per cent. The sulphur and phosphorus are undesirable quantities, the former producing brittleness when hot or 'hot shortness,' and the latter 'cold shortness.' The silicon and manganese facilitate melting and forging, and improve the quality of the finished steel. The hardness of the steel chiefly depends upon the carbon contained in it. The various tools contain different percentages of carbon to suit the work they have to do. Thus, tools subject to sharp blows should contain a low percentage of carbon, and keen-edged tools a high percentage. The carbon contents for different classes of tools are as follows:—

Tools such as snaps, hammers, drifts, stamping dies, contain about 0·65 per cent. carbon. Smiths' tools, hot and cold sets, swages, punches, shearing blades, cold chisels, miners' drills, minting dies, contain about 0·7 to 0·8 per cent. carbon.

Large milling and other circular cutters, very heavy lathe tools, drills, taps, screwing dies, reamers, contain about 1 per cent. carbon, whilst the smaller and lighter varieties of cutters, turning and similar tools, drills, and reamers, contain from 1·1 per cent. to 1·3 per cent. carbon.

HARDENING.

Hardening is effected by heating the tool to a colour between a dull and cherry-red; that is, between a temperature of 1,365° F. (740° C.) and 1,400° F. (760° C.), and immediately quenching it in water, or in a suitable bath so that the temperature is at least lowered to 390° F. (200° C.), but preferably to a normal temperature of about 60° F. Some tools, depending on their composition, will harden a few degrees below 1,365° F. and some require a slightly higher temperature. The upper limit must not be exceeded, otherwise the steel becomes coarse and weak, but providing the temperature is kept within the range the steel is apparently uninjured. The lower hardening temperature, or what is called the 'critical point,' must be reached, otherwise the steel after quenching will be too soft for cutting metal.

TEMPERING.

This is done by heating to a temperature between 390° F. (200° C.) and 600° F. (315° C.) and cooling slowly. The higher the tool is heated within this range the softer it becomes. It is of the utmost importance that the tool should be uniformly heated, and quenched, otherwise some parts will be harder than others and internal stresses set up which will produce warping and cracking. To minimise any irregularity in heating formed cutters, modern practice is to heat these in a bath of molten metallic salts which is maintained at a constant temperature by gas or oil jets.

ANNEALING.

Irregular annealing is responsible for much warping and cracking, and it is advisable that formed cutters should be carefully annealed before forging, and even again before being hardened, so as to remove any initial strains and strains produced in machining. These precautions as to annealing should be observed with all classes of steels.

* See also METALLURGY, Section XL.

Self-Hardening or Mushet Steel.

This class contains on the average about 5·5 per cent. tungsten, 2 per cent. carbon, 1·5 per cent. manganese, 1 per cent. silicon, and 0·4 per cent. chromium. As in the case of high-speed steels the proportions of the various elements have a most pronounced effect on the quality.

The self-hardening property of this steel principally depends upon a high percentage of manganese or chromium in combination with tungsten or molybdenum. Such steel is hardened by heating it to a bright red colour (about 1,500°F.) and allowing it to cool. If heated much above this temperature it will crumble in forging, and its cutting capacity is permanently impaired. When cold it is very hard, and retains its cutting edge much better than carbon steel tools.

High-Speed Steels.

High-speed steels are also self-hardening, but are usually forced in cooling by an air blast or by quenching in oil or other medium. The elements of this steel are very similar to some of the older types of self-hardening steels, but they contain much more tungsten and chromium and about one-third the quantity of carbon. Several old self-hardening steels had all the desirable elements and proportions of a high-speed steel, and it was due to the supposed 'overheating' of such steel that its latent cutting capacity was revealed, and not to any new properties or proportions.

Broadly speaking, there are two classes of this steel, viz., *ordinary* and *new* high-speed steel, and for these Mr. F. W. Taylor gives the following chemical composition :—

Chemical Composition of Ordinary and New High-Speed Steels.

Class of Steel.	Tungsten. Per cent.	Chromium. Per cent.	Carbon. Per cent.	Manganese. Per cent.	Vanadium. Per cent.	Silicon. Per cent.
Ordinary .	8·0	3·8	1·85	0·30	—	0·15
New or modern	18·91	5·47	0·67	1·1	0·29	0·043

The carbon contents in the ordinary high-speed steel is high, and the principal difference between the ordinary and new high-speed steels is that the latter contains a quantity of vanadium. The addition of vanadium raises the 'critical point' and enables the tool to withstand a higher temperature due to an increased speed of cutting. Tools of this steel are harder and retain their cutting edges longer than ordinary high-speed steel tools. They are therefore particularly suitable for cutting very hard metals, such as railway and car tyres, which have become very hard with usage, manganese steel castings, chilled rolls, etc. A tool containing 0·25 per cent. of vanadium lasted twice the time of a similar tool containing no vanadium, whilst Mr. J. M. Gledhill points out that by substituting 2 per cent. vanadium for chromium in a steel, good results were obtained when tried on medium steel, but were no better than the steel containing the much cheaper element chromium.

A steel may therefore be made of the best quality of material, but unless the elements are properly proportioned and care and skill be exercised during melting to ensure a proper mixing and avoid 'dryness' (a brittleness produced by retaining the steel in its melted state for an undue length of time), it may fall short of a first-class steel. As the percentage of the vital elements decrease, the difficulties in successful melting and mixing increase, and the most satisfactory steel to suit the various conditions prevailing in practice can only be obtained by prolonged experimenting.

To Identify High-Speed Steel Scrap.

The spark from high-speed steel ground on an emery wheel is a dark-red spark that is inclined to scatter, while the spark from ordinary carbon steel is much whiter and is not inclined to scatter. The sparks from high-speed steel are about the same colour as from cast iron, but are much finer.

Annealing.

This process is best accomplished by placing the tools in a sealed oven, furnace, or muffle, where they are uniformly heated by radiation. By excluding the air, and preventing the flame impinging on the steel, scaling is avoided and the fine cutting edges preserved. The furnace should be heated in such a manner that in about two hours' time it attains a temperature of 1,544°F. (840°C.), or very slightly higher, and should be steadily maintained at this temperature for about two hours. The heat is then cut off and both furnace and tools are allowed to cool slowly.

To release the furnace during the cooling process some smiths bury the articles in lime, powdered charcoal, soot, or coarse sawdust of some particularly hard wood. The bed in the latter case should be about 24 inches deep and contained in a box having an airtight lid. The articles when dropped on the top of the sawdust and covered with the lid work their way slowly down.

Should such a furnace not be available the tools may be placed in an iron pot or pipe and packed tight with steel cuttings or lime. The pot is then sealed with fireclay and raised to the above temperature at which it is retained for about eight hours before it is placed in charcoal ashes to cool. Good results are obtained from this process, but it is exceedingly slow, and it is doubtful if any advantage is obtained by prolonging the treatment after the annealing temperature has been attained throughout.

Forging.

If a piece has to be cut from the cold bar, this should be done before nicking the bar all round otherwise small cracks are likely to be started. Preferably the bar should be cut through at a forging heat. In forging, heat the tool slowly and uniformly (without the blast) to a dull red, then raise quickly to a bright red or yellow heat (1,800° to 1,900° F.), when it can be easily and rapidly manipulated. To prevent cracks, hammering should cease as soon as the heat has fallen to a medium red (about 1,400° F.). The steel may be rapidly reheated from this temperature for subsequent forging without any ill-effects. On account of ease and economy in forging, together with general utility and efficiency in cutting, a roughing tool is recommended. This form of tool can be quickly forged by first cutting the required length from the bar and afterwards forming the front, side and top angles by simply jumping the nose. This shape gives the minimum of waste material.

Hardening.

A temperature of 2,000° F., which is ruinous to other steels, gives this steel the property of resisting abrasion at a higher temperature. It is therefore capable of retaining its cutting edge at the high temperatures accompanying high speeds of cutting. This quality, which has been termed 'red hardness,' is not possessed by other steels. In hardening high-speed steel it is slowly pre-heated (in order to obtain a uniform heat throughout) to a medium red (about 1,550° F. then quickly raised to the following temperatures :—Ordinary high-speed and molybdenum steel from 2,000° F. to 2,200° F. (yellow or lemon heat), vanadium steel from 2,200° F. to 2,350° F. The lower figure is recommended for oil hardening and the higher for air hardening. When the heating is done in the smith's fire, obtain a good, deep bed of small coke or soft coal and pre-heat the tool by laying it in the fuel at some distance from the hottest part till the fire is thoroughly blown up. To ensure uniformity the tool during both stages of heating should be frequently turned round. Some smiths prefer to obtain final heat by placing the tool within a piece of scrap piping which is maintained at the required temperature by the surrounding fuel on the hearth. Except for the simplest form of tools such as those of the lathe type, the smith's hearth is unsuitable. Formed tools such as milling cutters, hobs, reamers, etc., are best heated in gas, oil or coke-fired muffle furnaces. Modern furnaces for this work usually have two chambers, i.e., an upper in which the cutter is pre-heated, and a lower chamber in which the cutter is raised to the final temperature. Another method of raising the cutter to its final heat is to immerse it in a mixture of molten metallic salts (barium chloride, etc.). The bath is heated and maintained at any desired temperature by passing an electric current through it. As the heat generated varies as the current passes through the circuit the arrangement is ideal for obtaining uniform conditions. The first cost, however, is large, and attempts have been made to obtain similar results from gas-heated salts in graphite crucibles. Trouble arises in the latter device through oxidisation : particles of iron oxide floating on the surface are inclined to adhere to the tool and produce soft spots.

Many have an idea that it is impossible to injure high-speed steel by overheating. This is certainly not the case. If an ordinary high-speed steel tool be heated to 2,400° F. and cooled in an air blast, its fracture will show a very coarse grain.

Once the final temperature has been attained the tool should be immediately withdrawn, as prolonging the treatment, even without raising the temperature, injures the steel and reduces its cutting capacity.

'Pack' Hardening.

It requires a man with considerable experience to harden successfully large high-speed steel milling cutters or hobs, and also a first-class furnace designed for this purpose. The disaster attending repeated attempts to do this has caused some hardeners to resort to 'pack hardening,' and possibly the desire to get away from scaling effects has also had something to do with this innovation. In this process about a 1-in. layer of charcoal ($\frac{1}{4}$-in. lumps) is made in the bottom of a cast steel box. Over the charcoal is placed a perforated plate which supports the cutters, but there is no charcoal against the cutters, etc. When several cutters are packed in the box these are separated by asbestos. When charged a lid is placed on the top of the box and sealed with clay. The box is then placed in a suitable furnace and the temperature raised to 1,250° C., and held at that for an hour or two, depending on the time required to thoroughly heat the contents. The cutters are cooled in hot oil or metallic salts in the usual way.

The object of the charcoal is to consume the free oxygen in the box before the steel gets sufficiently hot to scale. Scale is oxide of iron, and will not form if there is no free oxygen, so that however hot the steel may get, it cannot scale.

Tools or cutters which have to be rehardened after having been in use should be annealed prior to this operation, and thereby relieve the internal strains.

Before attempting to harden steel, it is advisable to remove by grinding or machining the black scale (oxide of iron) on the bar or forging, as immediately adjoining it is a thin film of decarburised material, which cannot be altogether avoided in the manufacture, and which will not be hardened.

Quenching.

Uniform cooling is just as essential to the successful treatment of steel as uniform heating, providing this is done it does not matter whether the medium is an air blast, oil, or water. To quench high-speed steel in water, however, is liable to produce cracks, and this is particularly the case in formed cutters where it is impossible to obtain uniform cooling by such a rapid process. Tools of the lathe type are usually cooled in a strong air blast, whereas formed cutters are generally quenched in a bath of salts about 500° F. or in rape or whale oil at a temperature of 250° F. and afterwards cooled in cold oil. The oil cooling gives a clean smooth surface to the cutter.

Quenching Medium.

After heating, it is usual to quench this steel in water, brine, oil, metallic salts, molten lead, etc., depending on the size of the tool or purpose for which it is intended. The function of the medium is to abstract rapidly the heat from the steel, as it passes through the critical range, and nothing more than this. Water therefore, on account of its mobility, makes an excellent cooling medium. The colder the water, and the more rapidly it is circulated, the greater the hardness will be. Cold water which has not had the chill taken off is very liable to produce cracks, whilst with boiling water it is impossible to harden at all. In the latter case a jacket of steam forms between the tool and the water, and cooling is retarded. It follows, therefore, that different degrees of hardness can be obtained by having the water at different temperatures. To raise the boiling-point, salt (as much as the water will carry) is frequently added, and for general purposes such a mixture at 40° C. (104° F.) is quite good. Keep the liquid clean and free from grease or oil. A great deal of nonsense has been spoken and written on the relative merits of quenching mediums, and to Mr. S. N. Brayshaw, M.I.Mech.E., especial credit is due for exploding many of the fantastical theories on this subject. According to Mr. Brayshaw, the ability of a liquid to bring a body immersed in it to its own temperature depends upon the following properties: (1) Specific Gravity, (2) Specific Heat, (3)Boiling-point, (4) Thermal Conductivity, and (5) Fluidity.

Specific Gravity.—The specific gravity determines the weight of liquid in contact with the tool being quenched.

Specific Heat.—The heat required to raise the temperature by a given amount, of a pound of different liquids varies enormously. If a red-hot tool weighing 1 lb. be immersed in a gallon of water weighing 10 lbs., the water quenches the tool, and its own temperature is raised about 12° C. in so doing, but if immersed in 10 lbs. of mercury, which has a very low specific heat, the temperature of the mercury rushes up more than 200° C.

Boiling-point.—When a tool is quenched in water or oil, a jacket of steam or vapour may be instantly formed, and this retards the cooling very greatly. Liquids, like solids, vary in their ability to conduct heat, but a deficiency in this respect may be compensated for by motion in the liquid.

Fluidity.—The ability of a liquid to circulate freely is one of the most important factors as regards quenching. The agitation may be caused by moving the tools about rapidly in the liquid, or by directing jets upon them.

Water.—The fact that its thermal conductivity is very low is counterbalanced by its fluidity. The higher the temperature of the quenching water, the more nearly does its effect approach that of oil, and if boiling water is used it will be even more gentle in its action than oil.

The Boiling-point of Water in relation to its Quenching Capacity.

If ordinary tool steel could be very rapidly quenched by any means to a temperature of 100° C. it would harden perfectly; but water at 100° C. (that is, boiling water) is incapable of quenching steel rapidly, and the nearer the boiling-point is approached, the worse it becomes for quenching purposes. When a red-hot tool is immersed in water, steam is generated around the tool, keeping the water away from it and interfering with the quenching. If the water is boiling, the tool is completely jacketed.

The Effect of Stirring during Quenching.—The tools cool much more quickly if they are stirred, or, what amounts to the same thing, if the water is projected in a stream upon the tools.

The Addition of Acid or Salt to the Quenching Bath.—The addition of sulphuric acid increases the conductivity of the water, but the gain from this cause is small in comparison with rapid circulation of water.

It is sometimes stated that sulphuric acid is added to prevent hardening cracks, but this is inconsistent with the other idea that it is used to obtain a greater degree of hardness. If the liability to crack is reduced, it is because the acid has made the quenching bath more gentle in its action, and thereby reduced the hardness.

Seasoned Water for Quenching.

Water ordinarily contains air dissolved in it, and this air is expelled when heat is applied. For this reason water which has recently been boiled and allowed to cool again, may be expected to quench the steel with greater rapidity and uniformity.

Quenching in Oil.—It is necessary in some cases to quench in oil, although it should be avoided wherever possible. Oil to one specification is not always the same, and even if it were, considerable changes are brought about with use, and its quenching power soon becomes different to that of new oil.

Quenching in Fusible Salts.—The ideal quenching liquid would be one which, at a temperature of say 100° C., had sufficient quenching power to cool steel so rapidly that it would harden thoroughly.

Experiments have been made by Mr. Brayshaw with chlorides, fluorides, carbonates, and other compounds, in order to find a 'melt' for quenching.

Such a 'melt' or mixture of salts differs from that used in the salt bath as ordinarily understood, in the fact that the salt bath commonly consists of a mixture of chlorides which are liquid at a red heat, and in which tools are immersed in order that they may be raised to the temperature necessary for hardening. The 'melt' now under consideration is primarily not for heating, but for cooling the tools, although it may be remarked incidentally that such a 'melt' may usefully be applied for the tempering of tools after they have been hardened.

The endeavour was to find a mixture, free from water, which would be liquid at a temperature as low as possible, have a high specific heat, and should be obtainable at reasonable prices.

A mixture known as 'Quenchite' resulted. It is liquid at 135° C. (275° F.), and at this temperature it is almost as fluid as water. It quenches steel very rapidly, has a very high boiling-point, and consequently the liquid is in direct contact with the steel almost from the moment of immersion.

The temperature of 135° C. is, however, so high that 'Quenchite' is only suitable for hardening small or thin articles. Small cutters and long reamers or taps can be quenched in it to give a glasshard surface, and with practically no warping or change of shape, but it is not sufficiently severe in its action to give the necessary hardness to a sufficient depth in the case of large tools in which there is a considerable body of steel behind the cutting edge.

Manner of Quenching.

In order to secure uniform cooling and absence of warping, all sorts of dodges are resorted to by hardeners. In the case of small articles, these are dipped bodily into the medium, so that the whole is cooled at the same instant.

Twist drills, taps, reamers, etc., are dipped vertically, and to move these sideways in the bath would distort them. To avoid this, the medium is stirred with a stick so as to give it a rotary motion, and the drill or tap is dipped in the centre of the swirling liquid, and moved vertically upwards and downwards. Blades for lawn mowers, shear blades, and similar things should be gripped in the centre with the tongs, dipped edgewise in the liquid, and moved edgewise through it, or the liquid itself may be agitated. Sometimes the blade is held between two pieces of iron slightly larger than the blade, and the whole dipped together. A hammer face, if dipped in the liquid, would cool at the edges before it did at the centre, and for this reason a sprayer or jet of water directed on the centre of the face is preferred.

Tempering.

May be accomplished by placing the cutter in a gas stove and raising the temperature to about 500° F., when the gas is shut off and the whole allowed to cool slowly.

Some prefer to first reduce the temperature of formed cutters to about 800° F. in the air blast, then to plunge the same in tallow at 200° F., when the temperature is again raised to about 500° F. and the whole allowed to cool.

Grinding.

The tool should be ground by *hand* pressure on a *wet* sandstone wheel. If it is necessary to use the emery wheel, the tool should be roughly ground to shape after forging, and finished on the wet sandstone after hardening.

Hardening High-Speed Steel Tools *

(1.)

The heat treatment usually prescribed for hardening high-speed steel tools consists in heating the nose of the tool slowly and uniformly to a red heat, then rapidly to a white or welding heat, at which the steel commences to fuse and blister, and then cooling either rapidly or slowly in air or quenching in oil.

The Department of Scientific and Industrial Research desires to direct the attention of users of high-speed steel tools to the fact that uniform and improved results may be obtained by carrying out the hardening process in two stages in accordance with the instructions given below. This treatment is the outcome of the researches of F. W. Taylor in America, and of Carpenter, Edwards, Dempster Smith, and others, in this country. It may be stated that the adoption of this process for the hardening of special tools required for a most important purpose in connection with the war has yielded consistently uniform results, which are immensely better than the best results previously attained. The procedure given below is that adopted by Mr. Dempster Smith in the researches on tool steel carried out in the Manchester School of Technology for the Manchester Association of Engineers, with the aid of a grant from the Research Department.

First or High Heat Treatment.—Thoroughly preheat the tool to about 800° C. (1,472° F.), and transfer immediately to a high temperature furnace at a temperature of not less than 1,350° C. (2,462° F.) for ordinary high-speed tools, and about 1,320° C. (2,408° F.) for super high-speed steel. Retain it in the furnace until the surface of the tool nose is covered with small blisters slightly larger in size than a pin's head, in the case of ordinary high-speed steel, or until the entire surface of the nose is in a state of incipient fusion, or in the case of super high-speed steel presents a greasy appearance. The steel in the first case will have attained a temperature of about 1,320° C. (2,408° F.), and in the second a temperature of 1,290° C. (2,354° F.). Remove the tool and cool in an air blast of large volume.

Second or Low Heat Treatment.—Reheat the tool hardened as above to 570° C. (1,058° F.), for half an hour for tools with sections of half an inch square, or longer in the case of larger sections, and cool in an air blast or in still air. If carefully handled a higher temperature of about 610° C. (1,130° F.) may be adopted if the treatment is limited to ten minutes for tools of the smaller section, and slightly lower for those of larger section. All temperatures for the first and second heat treatment should be measured by means of a pyrometer, and care should be taken to ensure that the portion of the tool under treatment is uniformly heated throughout to the required temperature.

(2.)

The prevailing method of heating tools in ordinary gas furnaces, even when the utmost care is exercised, does not yield uniform results as regards cutting efficiency. The following procedure will be found to yield uniform results :—

First or High Heating.—The preliminary heating to about 800° C. (1,472° F.) may be carried out in a gas-fired muffle or semi-muffle. Care should be taken at this stage to ensure that the whole of the nose of the tool or the part to be heated subsequently to a high temperature is brought uniformly to the required temperature. The high heating to 1,350° C. (2,462° F.) should be carried out in a gas-fired furnace with a reducing flame so disposed that the heating of the tool is effected more by radiation from the heated surfaces of the furnace than by contact with the gases. It is important that the tool shall not be brought directly in contact with the burning gases. Detailed drawings of a furnace to fulfil the above conditions will be furnished on application to the Department.

Second or Low Heating.—This may be carried out at 570° C. (1,058° F.) in a gas-fired muffle or in a lead bath.

Tool Steels.

The treatment given above may be applied to steels containing tungsten, chromium, or vanadium, and generally to ordinary brands of high-speed tool steel. It may be found that certain special steels require a modification of the above treatment, particularly of the temperature required for the second heating ; in such cases users are advised to refer to the makers of the steel, or, if possible, to investigate the matter for themselves.

Results.

When the heat treatment described above is carefully carried out, it will be found that the cutting efficiency of the tools is high and is maintained throughout the entire length of the nose, and that no further heat treatment is necessary after grinding (see page 1071). The second heat treatment toughens the steel, and its advantage is very pronounced when working tools at high speeds

* Communicated by the Department of Scientific and Industrial Research to the *Ministry of Munitions Journal*, and reproduced by permission of the Editor of that Journal.

Furnace.

A type of gas muffle recommended is shown by fig. 58.

Oil Fuel Heating Systems.

(Abstract of paper read before the Royal Society of Arts by Mr. A. F. Baillie.)

Billet-heating furnaces using the air-jet system, with air and oil preheated by the exhaust gases showed a consumption of 9 lbs. of oil per 100 lbs. of metal heated for drop stamping. In heating furnaces for nut-making, the consumption was found to be about 4 lbs. of oil per 100 lbs. of metal heated, and for bolt-making about 4½ lbs. A test on a bolt-making furnace 9 ins. by 12 ins. by 20 ins. resulted in 400 to 500 gross of bolts of from ¾ in. to 1 in. per week on a consumption of 300 gallons of oil. Rivet-heating tests show for some furnaces on this work a consumption of 10·3 lbs. of oil per 100 lbs. of metal heated. For pot furnace work a tilting crucible furnace takes about 10 lbs. of oil per 100 lbs. of gunmetal melted, an open-hearth furnace about 11·8 lbs., and a lift-out furnace about 12·5 lbs. of oil. Stock converter furnaces for melting cast iron and steel-making take about 15 lbs. of oil per 100 lbs. of metal for melting only, but the total, including lighting and heating up, runs to about 40 lbs. In glass works the tank type of furnace takes about 37 lbs. per 100 lbs. of finished glass. An Italian plant of open-hearth Siemens basic furnaces of 55 tons capacity, with an output of 150 tons of steel per 24 hours per furnace, the oil consumption ranges from 16·5 tons to 18·75 tons, or from 11 lbs. to 12·5 lbs. per 100 lbs. metal. The air pressure for atomisation was 4 kilos. per sq. centimetre.

The Burdon Oil-Gas System.

In this system the oil fuel, first very finely atomised and intimately mixed with a large volume of air at *low pressure*, is converted into gas in a small retort, and from the retort the gas is delivered into the furnace (through pipes of varying diameters), where, by the regulation of the air supply, any desired flame can be obtained. The intimate mixture of gas and air produces very high heating results, the full calorific value of the fuel being obtained.

The temperature is regulated by adjusting the valves which control the oil and air supplies.

By proper regulation of the air the fuel burns with a non-luminous Bunsen flame, and complete combustion with entire absence of smoke is easily secured.

The air and fuel can be supplied in any proportion desired, so that the flame in the furnace may be either reducing, oxidising, or neutral as desired.

A furnace 10 feet by 6 feet can be brought from cold to approximately 3,000° F. in less than three hours.

The Burdon system can be used for billet heating, slab and bloom heating, chain making, all kinds of welding and forging, case-hardening, tube heating, rivet and bolt making and heating tin smelting, tool heating, horseshoe making, etc., also for steam raising, superheating, glass melting, metal melting, assaying, insulator making, etc.

Later developments effected in this system enable any quality of liquid fuel, either heavy crude fuel oil, or such as a mixture of 50 per cent. pitch and 50 per cent. of a tar-oil, or, alternatively, any of the lighter fuel oils, such as creosote and gas oils, etc., to be used with results giving high heating efficiency and economy in oil fuel.

The furnaces, which are made in various sizes, from small portable rivet heaters up to large forging types, consist of a combined generator and heating chamber. The oil is contained in the tank, from which it flows, without pressure, to the atomiser and the hot carburettor, where it is vaporised and intimately mixed with the air supply, forming a combustible gas, which passes to the actual heating chamber. The waste gases from this chamber pass through flues to the upper chamber to maintain a sufficient heat in the carburettor, to gasify the oil fuel.

Among the advantages claimed for this system are:

Cleanliness of installation and economy of space.

The producers can be applied to existing furnaces with a minimum of alteration or reconstruction.

Remarkably small fuel consumption, and no coal or coke trimming, stoking, or removal of clinker or ashes.

The cost and time of working is less than for coal or coal gas-fired furnaces.

Compressed Gas System.

There is an optimum pressure for the various applications of gas, and in order to meet this the gas can first be compressed to (say) 280 inches head of water (about 10 lbs. per sq. inch) which can be controlled at the various furnaces, etc., to the pressure required at each place. Pressures for typical work are as follows:—For melting tin or solder 65 inches; small hardening furnaces 65 inches; for blueing steel 65 to 80 inches; for the steel tempering lead pot 150 inches; while for melting metals at 2,000° F. to 2,600° F. 200 inches should be used.

Soldering irons can be heated at 65 inches pressure, which pressure is also convenient for the lighting.

With gas at 2*s.* per 1,000 and coal at 20*s.* per ton, there is little to choose between gas and coal firing as regards fuel costs. As regards cleanliness, uniformity, wear and tear of pots, and accuracy of control, the advantages are however in favour of gas, and in many cases with properly

GAS MUFFLE FURNACE FOR HEATING HIGH-SPEED TOOLS.

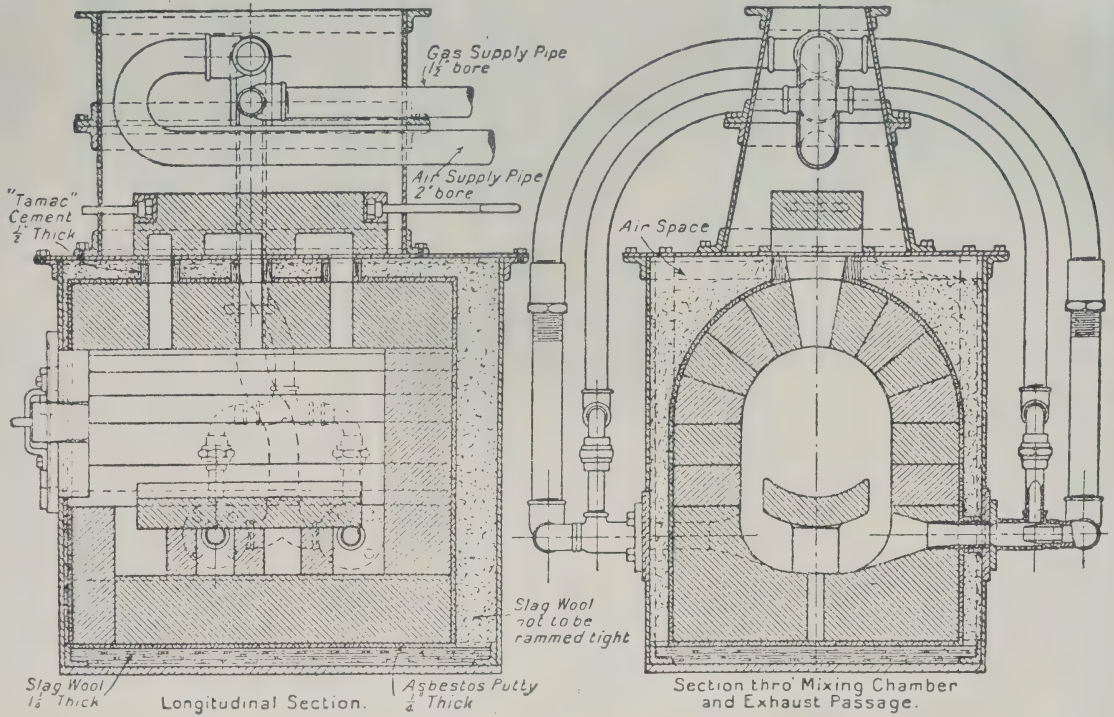


FIG. 58.

designed furnaces it is greatly superior to coal, not only from the point of view of cost, but from convenience, reliability, and ease of manipulation. There is also a large saving of time taken and space occupied by a gas furnace as compared with a coal furnace of similar power, factors of great importance when increased output is called for.

Low-pressure gas with air blast is largely adopted.

TABLE OF COMPRESSOR DATA.

Capacity of Compressor. Cub. ft.	H.P. required.	Floor Space. Ins.
100	·12	18 by 7½
200	·25	22 „ 9
500	·5	23 „ 12
1,000	1·0	27 „ 17
2,000	1·5	32 „ 19
4,000	2·5	40 „ 23
10,000	3·5	48 „ 29

Colours of Tempered Steel.

The *tempering* in this case relates to lowering the degree of hardness obtained by plunging the steel when heated to a cherry red (1,650° F., 899° C.) in cold water :—

Colour.	F.°	C.°	Colour.	F.°	C.°
Very pale yellow .	430	221·1	Dark purple . .	550	287·8
Straw yellow . .	460	237·8	Clear blue . . .	570	298·9
Brown yellow . .	500	260·0	Pale blue . . .	610	321·1
Bright purple . .	530	276·7	Blue tinged with green .	630	332·2

Composition of Quenching Baths for Tempering Cutting Tools.

Fused nitrates of potassium and of sodium are too high in temperature for certain cutting tools, as they do not permit of cooling below 328° F. (220° C.). Mixtures of nitrate of potassium and of nitrate of sodium can, however, be employed, and a series of mixtures, fusing at different temperatures, be obtained.

Temperature.		Nitrate of Potassium.	Nitrate of Sodium.	Temperatures.		Nitrate of Potassium.	Nitrate of Sodium.
F.°	C.°			F.°	C.°		
536	280	0 parts	100 parts	279	137	55 parts	45 parts
446	230	20 „	80 „	293	145	60 „	40 „
342	172	40 „	60 „	437	225	80 „	20 „
293	145	50 „	50 „	635	335	100 „	0 „

Higher temperatures than 752° F. (400° C.) cannot be obtained with these mixtures. In steels where without extreme hardness absolute absence of brittleness is necessary, 932° F. (500° C.) to 1,112° F (600° C.) are temperatures more suitable. The following bath gives, on fusion, a temperature of 932° F. (500° C.) :—

Sodium chloride, 1 part; potassium chloride, 1 part; fused calcium chloride, 2 parts; hydrated barium chloride, 1 part; hydrate strontium chloride, 3 parts.

For a bath fusing at 1,292° F. (700° C.) the following mixture may be used :—

Hydrated boric acid crystals, 1 part; silver sand, 1½ part; anhydrous potassium carbonate, 1 part; anhydrous sodium carbonate, 1 part.

When prolonged treatment is required, a little cyanide or charcoal may be added to prevent superficial decarburisation; but, in view of the strongly cementating action of cyanide, this salt must be used with caution.

QUENCHING LIQUIDS.

When hot steel is continually dumped into a tank of oil, the lighter and volatile oils are evaporated and finally only the heavier oils will be left. When the oil becomes thick, it will not take the heat away from the steel rapidly; consequently, the steel will be soft and will not

show the maximum physical properties. To obtain good results, the quenching liquid should be of a constant specific gravity and maintained at a temperature of 70° to 90°, but the exact temperature at which the bath is maintained does not matter so much if it is always the same. To ensure the best results, the bath should always be maintained at a certain specific gravity and at some certain temperature. (*'Machinery,' New York.*)

In hardening and tempering a chisel, the best way, says the *English Mechanic*, is to heat it to a very dull red for a good inch up from the edge, holding the tongs in the left hand, and in the right a rub stone, a piece of broken grindstone or emery wheel, or, failing these, a piece of emery cloth wrapped round a small file. Dip the chisel for about half-an-inch until it just turns black, then withdraw it from the water and let it rest against the side of the pail, so as to steady it; rub it sharply with the stone, so as to brighten it, and watch the colour as it runs down from the part which is still hot, and when the edge is of a deep plum colour, verging on blue, dip it right out.

Grinding.

1. The tool should be forged roughly to shape and ground to the requisite contour by means of a composition wheel (preferably dry) without any lubricant being applied. The heating of the tool during grinding at this stage is not of any importance.
2. After shaping, the tool should be subjected to the two heat treatments indicated (page 1067).
3. It is advisable to finish-grind with a copious stream of water playing on the nose of the tool. If reasonable care to avoid heating be exercised, the tool may be ground on a composition wheel running in water.

Shape of Tool for Roughing Purposes.

The durability or length of time a tool will continue to cut under constant conditions as to feed, speed, etc., increases with the nose radius (fig. 59). The tendency to chatter, however, limits

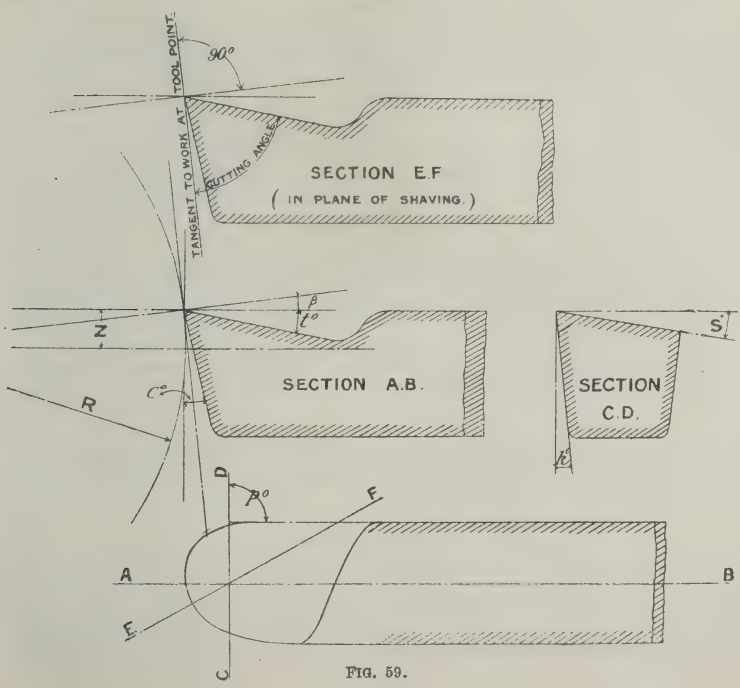


Fig. 59.

the extent to which the nose radius may be increased, and a round-nosed tool having a radius of $\frac{1}{2}$ in. to $\frac{5}{8}$ in. has been found to give the most satisfactory results for the general run of work. With work of a springy character a smaller nose radius may be required. The durability is also affected by the cutting angle, *i.e.*, the angle between the cutting face of the tool and the tangent to the work or 90° minus the top rake angle. The cutting angle is increased when the tool edge stands above the centre of the work, by an amount equal to the angle contained between the cutting edge of the tool and the axis of the bar.

The cutting angles which give the greatest durability to tools for roughing purposes are as follows :

Material.	Cutting Angle.	Material.	Cutting Angle.
Soft steel . . .	65°	Soft cast iron . .	70°
Medium steel . .	70°	Medium cast iron .	75°
Hard steel . .	80°	Hard Cast Iron . .	85°
Nickel steel . .	80°	Brass . . .	85°
Manganese steel . .	85°	Hard gunmetal . .	85°

The above angles are measured in a plane at 45° to the length of the tool which is approximately the plane of the shaving. The side and front clearance angles for tools with cutting edges on the line of centres need not exceed 6°. The shape of the tool recommended is shown below.

Front top rake ($t°$) ; height above centre (Z) ; front top rake (reduced to centre) ($\beta + t°$) ; side top rake ($S°$) ; front clearance ($C°$) ; radius of work (R) ; angle (β) ; front clearance (reduced to centre) ($C° - \beta$) ; side clearance ($k°$).

Steel Tool Shanks.

By the ordinary method of welding, or securing by other means, high-speed steel tips to ordinary steel tool shanks, the shank is weakened considerably at the point where the high-

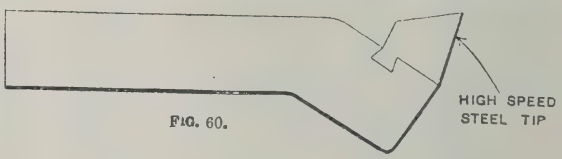


FIG. 60.

speed tip is applied, owing to the shank being cut away to accommodate the steel tip. In the J.B.R. system (the *J.B.R. Tool Co., Ltd., Longsight, Manchester*), fig. 60, the shank retains its original strength, the steel tip being held in a dovetail slot and brazed in position by a method which renders the two materials, when heat-treated, absolutely homogeneous at the joint.

The tools, which are stocked in a range of thirty-nine different shapes, each shape being manufactured in four sizes and two heights, are being largely used.

Sharpening-angles of Wood Machine Cutters.

Adzing soft wood across the grain, 30° ; planing machines, ordinary soft wood, 30° ; gouging and ploughing machines, 30° to 35° ; hard-wood tool-cutters, 35° to 40°.

Cutting Speeds for Carbon Steel Lathe Tools.

The following table gives the appropriate cutting speeds for different depths of cuts and traverses when roughing out with carbon steel tools of the lathe type, having a nose radius $\frac{1}{8}$ in., on various metals. When finishing soft, medium, and hard steel the cutting speeds which may be used are 20 ft., 15·5 ft., and 11 ft. per minute, whilst for finishing soft, medium, and hard cast iron the cutting speeds are 24 ft., 19 ft., and 14 ft. per minute in the order given. When finishing brass and copper, speeds of 80 ft. and 150 ft. per minute respectively should be adopted. The finishing cut should not exceed $\frac{1}{32}$ in. in any case. (*Dempster Smith.*)

TABLE OF CUTTING SPEEDS IN FEET PER MINUTE FOR CARBON STEEL LATHE TOOLS.

Cut.	Transverse.	Soft Steel.	Med. Steel.	Hard Steel.	Soft C. I.	Med. C. I.	Hard C. I.	Med. gunmetal	Soft Brass.
1/64	1/64	40·0	32·0	24·0	32·0	27·0	20·0	50·0	100·0
1/32	"	38·0	31·4	22·5	30·5	25·0	18·0	46·0	90·0
3/64	1/16	36·0	30·0	21·0	28·3	22·6	17·0	40·3	79·2
1/16	"	34·0	28·5	20·0	25·6	20·4	15·3	35·2	71·5
5/64	1/8	33·0	27·0	19·0	23·8	19·0	14·3	32·0	66·6
3/32	"	32·0	26·0	18·6	22·5	18·0	13·5	29·7	63·0
7/64	3/16	31·5	24·5	17·5	21·5	17·2	12·9	28·0	60·2
1/8	"	30·0	23·2	16·6	20·7	16·5	12·4	26·6	57·9
9/64	1/4	28·6	22·2	15·9	20·0	16·0	12·0	25·4	56·0
5/64	1/32	36·0	31·4	22·5	30·5	25·0	18·0	46·0	90·0
3/32	"	34·0	30·0	21·0	28·2	22·6	17·0	40·3	79·2
7/64	1/16	32·0	27·0	19·0	23·8	19·0	14·3	32·0	66·6
1/8	"	30·0	24·5	17·5	21·5	17·2	12·9	28·0	60·2
9/64	1/8	28·6	22·2	15·9	20·0	16·0	12·0	25·4	56·0
5/32	"	26·5	20·6	14·7	18·9	15·1	11·3	23·6	52·9
11/64	3/16	25·0	19·4	13·9	18·1	14·5	10·8	22·2	50·6
3/16	"	23·7	18·4	13·2	17·4	13·9	10·4	21·1	48·7
13/64	1/4	22·7	17·6	12·6	16·8	13·4	10·1	20·2	47·0
7/64	1/32	32·0	26·0	19·0	23·8	19·0	14·3	32·0	66·6
1/16	"	28·6	22·2	15·9	20·0	16·0	12·0	25·4	56·0
9/64	3/32	25·0	19·4	13·9	18·1	14·5	10·8	22·2	50·6
5/32	"	22·7	17·6	12·6	16·8	13·4	10·1	20·2	47·0
11/64	"	21·1	16·4	11·7	15·9	12·7	9·5	18·7	44·5
3/16	1/8	19·8	15·4	11·0	15·2	12·2	9·1	17·6	42·6
13/64	"	18·8	14·6	10·5	14·6	11·7	8·8	16·7	40·9
7/32	"	18·0	14·0	10·0	14·1	11·3	8·5	16·0	39·5
9/64	3/32	30·0	24·0	17·5	21·5	17·2	12·9	28·0	60·2
5/64	"	25·0	19·4	13·9	18·1	14·5	10·8	22·2	50·6
3/32	"	21·8	17·0	12·1	16·3	13·1	9·8	19·4	45·7
7/64	"	19·8	15·4	11·0	15·2	12·2	9·1	17·6	42·6
1/8	"	18·4	14·3	10·2	14·4	11·5	8·6	16·4	40·2
9/64	"	17·3	13·5	9·6	13·7	11·0	8·2	15·4	38·4
5/32	"	16·4	12·8	9·1	13·2	10·6	7·9	14·6	37·0
11/64	"	15·7	12·2	8·7	12·8	10·2	7·7	14·0	35·8
3/16	1/8	22·7	17·6	12·6	16·8	13·4	10·1		
7/64	"	19·8	15·4	11·0	15·2	12·2	9·1		
1/8	"	18·0	14·0	10·0	14·1	11·3	8·5		
9/64	"	16·7	13·0	9·3	13·4	10·7	8·0		
5/32	"	15·7	12·2	8·7	12·8	10·2	7·7		
11/64	"	14·3	11·1	7·9	11·9	9·5	7·1		
3/16	"	12·5	9·7	6·9	10·8	8·6	6·5		
1/4	"	11·3	8·8	6·3	10·0	8·0	6·0		
5/16	1/4	14·3	11·1	7·9	11·9	9·5	7·1		
3/8	"	12·5	9·7	6·9	10·8	8·6	6·5		
7/16	"	11·3	8·8	6·3	10·0	8·0	6·0		
1/2	"	10·5	8·2	5·8	9·5	7·6	5·7		
9/16	"	9·9	7·7	5·5	9·0	7·2	5·4		
5/8	"	9·0	7·0	5·0	8·4	6·7	5·0		
11/16	"	8·4	6·5	4·6	8·0	6·4	4·8		
3/4	"	7·9	6·1	4·4	7·6	6·1	4·6		

For ordinary Mushet steel the above speeds may be increased from 33 per cent. in the case of the lighter cuts to 50 per cent. for the heavy cuts.

The following table gives the revolutions per minute for different cutting speeds and diameters of work or cutters:—

RELATION BETWEEN REVOLUTIONS, CUTTING SPEEDS AND DIAMETERS OF WORK.

Cutting Speed in Ft. per Min.	Diameter of Work or Cutter, in Inches.																	
	¼	⅜	½	⅝	¾	⅞	1	1⅛	1¼	1½	1⅝	2	2½	3	3½	4	5	6
10	152	102	76	61	51	44	38	34	30	25	22	19	15	13	11	10	8	6·3
15	229	153	115	92	76	65	57	51	46	38	33	29	23	19	16	14	12	10
20	305	204	153	122	102	87	76	68	61	51	44	38	31	25	22	19	15	13
25	382	255	191	153	127	109	95	85	76	64	55	48	38	32	27	24	19	16
30	458	306	230	184	153	131	114	102	92	76	65	57	46	38	33	29	23	19
35	535	357	268	214	178	153	133	119	107	89	76	67	53	45	38	33	27	22
40	611	407	306	245	204	175	153	136	122	102	87	76	61	51	44	38	31	25
45	686	458	344	275	229	197	172	153	138	115	98	86	69	57	49	43	34	29
50	764	509	382	306	255	218	191	170	153	128	109	95	76	64	55	48	38	32
60	917	611	459	367	305	262	229	204	183	153	131	115	92	76	66	57	46	38
70	—	713	535	428	356	306	267	238	214	178	153	134	107	89	76	67	54	45
80	—	815	611	489	407	349	306	272	244	204	175	153	122	102	87	76	61	51
90	—	917	688	550	458	393	344	305	275	229	196	172	138	115	98	86	69	57
100	—	—	764	611	509	437	382	339	305	255	218	191	153	128	109	96	76	64
110	—	—	841	672	560	480	420	373	336	280	240	210	168	140	120	105	84	70
120	—	—	917	734	611	524	458	407	367	306	262	229	183	153	131	115	92	76
130	—	—	994	795	662	568	497	441	397	331	284	248	199	166	142	124	99	83
140	—	—	—	856	713	611	535	476	428	356	306	268	214	178	153	134	107	89
150	—	—	—	917	764	655	573	509	458	382	327	287	229	191	164	143	115	95

Cutting Speed in Ft. per Min.	Diameter of Work or Cutter, in Inches.																	
	7	8	9	10	12	16	20	24	30	36	42	48	60	72	84	96	108	120
10	5·4	5	4	3·8	3·2	2·4	2	1·6	1·3	1	·9	·8	·6	·5	·5	·4	·4	3
15	8	7	6	5·7	4·8	4	3	2	1·9	1·6	1·4	1·2	1	·8	·7	·6	·5	·5
20	11	10	8	7·7	6	5	4	3	2·5	2·1	1·8	1·6	1·3	1·1	·9	·8	·7	·6
25	14	12	11	10	8	6	5	4	3·2	2·6	2·3	2	1·6	1·3	1·1	1	·9	·8
30	16	14	13	11	10	7	6	5	3·8	3·2	2·7	2·4	1·9	1·6	1·4	1·2	1	1
35	19	17	15	13	11	8	7	6	4·5	3·7	3·2	2·8	2·2	1·9	1·6	1·4	1·2	1·1
40	22	19	17	15	13	10	8	6·3	5·1	4·2	3·6	3·2	2·6	2·1	1·8	1·6	1·4	1·3
45	25	21	19	17	14	11	9	7	5·7	4·8	4·1	3·6	2·9	2·3	2	1·8	1·6	1·4
50	27	24	21	19	16	12	10	8	6·5	5·2	4·6	4	3·2	2·6	2·3	2	1·8	1·6
60	33	29	25	23	19	14	11	10	8	6·4	5	4·8	3·8	3·2	2·7	2·4	2·1	1·9
70	38	33	30	27	22	17	13	11	9	7·4	6	5·6	4·5	3·7	3·2	2·8	2·5	2·2
80	44	38	34	31	25	19	15	13	10	8·5	7	6	5·1	4·3	3·7	3	2·8	2·6
90	49	43	38	34	28	21	17	14	11	9·6	8	7	5·8	4·8	4·1	3·6	3·2	2·9
100	55	48	42	38	32	24	19	16	13	10	9	8	6·4	5·3	4·6	4	3·6	3·2
110	60	53	47	42	35	26	21	18	14	12	10	9	7	5·8	5	4·4	3·9	3·5
120	65	57	51	46	38	29	23	19	15	13	11	10	7·7	6·4	5·5	4·8	4·2	3·9
130	71	62	55	50	41	31	25	21	17	14	12	10·4	8	6·8	5·9	5	4·6	4·2
140	76	67	59	53	45	33	27	22	18	15	13	11	9	7·4	6·3	5·6	5	4·5
150	82	72	64	57	48	36	29	24	19	16	14	12	10	7·9	6·9	6	5·3	4·8

CUTTING WITH ORDINARY AND HIGH-SPEED STEEL.

Steel.

Trials showed that 20 ft. per minute with a cut of $\frac{1}{16}$ in. by $\frac{1}{8}$ in. is the superior limit of speed for ordinary water-hardened steel operating upon Whitworth fluid-compressed soft steel;

whilst an ordinary Mushet tool can run at over 30 ft. per minute, with a $\frac{3}{16}$ in. by $\frac{1}{16}$ in. cut upon the same material.

Cast Iron.

The trials on *medium cast iron* showed that 25 ft. per minute is too high a speed with $\frac{1}{16}$ in. by $\frac{1}{16}$ in. cuts; 17 ft. is too high with $\frac{3}{16}$ by $\frac{1}{16}$ in. cuts; and 15 ft. is too high with $\frac{3}{16}$ in. by $\frac{1}{8}$ in. cuts for ordinary water-hardened tool steels. An ordinary Mushet ran for over 20 minutes at 23·6 ft. per minute on a $\frac{1}{16}$ in. by $\frac{1}{16}$ in. cut; whilst it failed at 19 ft. per minute on a $\frac{3}{16}$ in. by $\frac{1}{16}$ in. cut, and at 15 ft. per minute on a $\frac{3}{16}$ in. by $\frac{1}{8}$ in. cut on this material.

In the case of *soft cast iron*, 22 ft. per minute was too high a speed for water-hardened steel taking a $\frac{3}{8}$ in. cut by $\frac{1}{8}$ in. traverse; whilst an ordinary Mushet lasted over half an hour at 31·6 ft. per minute with the same cut upon the same material.

Heavy cuts are much more economical in power, for the waste work is a much larger proportion of the whole work required at light cuts than at heavy cuts. It is probably this fact which has given rise to the commonly accepted opinion that the cutting stress increases with the speed (as instanced by the objections of workmen to increase cutting speeds on account of the anticipated springing of the work), contrary to the results of these trials, which rather show the reverse to be the case. In soft steel, for example, the average horse-power required for actual cutting at $\frac{1}{16}$ in. by $\frac{1}{16}$ in. was 3, while at $\frac{3}{8}$ in. by $\frac{1}{8}$ in. it was 15. The friction horse power of lathe and counter-shaft was about $2\frac{1}{2}$ h.p. The weight removed was about 105 lbs. and 445 lbs. per hour at the light and heavy cuts respectively, so that the weight removed per gross h.p. hour was 19·1 lbs. for the light and 25·3 lbs. for the heavy cut. Neglecting frictional loss, these figures would have been 35 and 29 respectively.

In the case of cast iron, the gain in power at the heavier cuts is still more conspicuous. Here the effective h.p. was 1·7 and 5·5 at the $\frac{1}{16}$ by $\frac{1}{16}$ and $\frac{3}{8}$ by $\frac{1}{8}$ cuts respectively, the waste h.p. being about 2. The weights removed were $42\frac{1}{2}$ lbs. and 198 lbs. per hour, and the weights per h.p. hour were therefore: 11·35 lbs. at the light cut, and 26·5 lbs. at the heavy cut.

These results show that more material can be removed in a given time with a heavy cut at its proper speed than with a light cut at the highest speed which the new steels can take, and that this can be done at a smaller expenditure of gross power per ton of shavings removed.

Variation of Cutting Force with Area of Cut, and of Cutting Stress obtained in the above trials.

For steel the cutting force is simply proportional to the area of cut, and this force has the following approximate values in tons per sq. in.:—

For soft steel (fluid compressed)	. 115	For soft cast iron	51
„ medium steel „ „	. 108	„ medium iron	84
„ hard steel „ „	. 150	„ hard iron	82

Fine traverses take more power than coarse ones.

Cutting stress does not vary much with speed, but tends to diminish.

Special high-speed tool steels will cut twice as fast as ordinary Mushet steel, and more than four times as fast as ordinary water-hardened steel.

(*Proc. Manchester Assoc. of Engrs.* 1903.)

Cutting Speed for High-Speed Steel Tools when Cutting Steel.

An extensive series of experiments have been carried out by the Lathe Tool Research Committee of the Manchester Association of Engineers, when operating on steel under different conditions.* The Committee in many of its investigations selected cutting speeds which would give a twenty minutes life to the tool, but state that for practical purposes and economic considerations, a speed which will give a two hours run is desirable. A relation is, therefore, given between the cutting speed and durability of the tool, and this conforms to the expression:

$$VT^{\frac{1}{3}} = \text{constant},$$

where,

V = cutting speed in ft. per min., and T = duration in mins.

* See Report of Lathe Tool Research Committee, published by H.M. Stationery Office, 1922. Price 5*s*.

The value of the constant varies with the material operated upon and also with the dimensions of the cut. This relationship is of the utmost value, since for a given material, tool, and cut, if the cutting speed V and the life of the tool T be observed, the cutting speed V_1 for the tool to last a time T_1 can be obtained, thus:

$$V_1 = V\left(\frac{T}{T_1}\right)^{\frac{1}{8}}.$$

The change in the cutting speed with time as given by the above law may be better appreciated from its graphical representation in fig. 61, where the appropriate cutting speed is plotted on a base of tool life.

As an example, assuming a tool under test lasts 40 minutes when cutting at 110 ft. per minute,

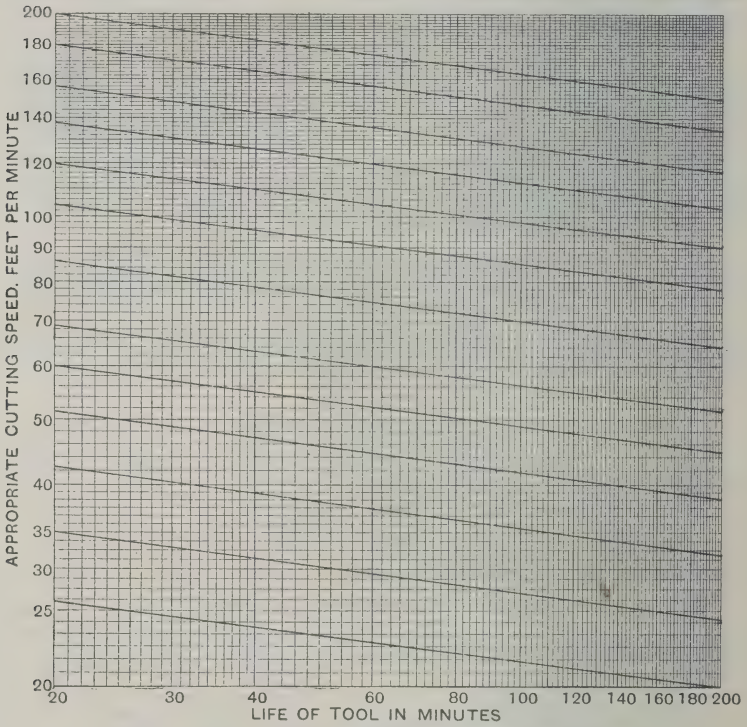


FIG. 61.—Relation between appropriate cutting speed and life of tool.

and it is required to find the speed at which the tool will last 120 minutes, other conditions remaining unaltered. Trace vertically above 40 minutes until the diagonal line, passing through the horizontal line of 110 ft. per minute, is reached. Follow this diagonal line until vertical over 120 minutes, when it intersects the horizontal line at 96 ft. per minute, which is the speed required.

CUTTING SPEED AND DIMENSIONS OF CUT (STEEL).

The value of the cutting speed varies with the dimensions of the depth of cut and traverse, shape and section of tool, as well as the material cut.

The cutting speeds for the various cuts and traverses given in fig. 62 correspond to a 2-hours' life of tool (which is considered to be the most economical life) when cutting a steel having an ultimate tensile stress of 38 tons per sq. in. and without any cooling medium.

The values were obtained when using a 1¼ in. square tool having a 60° plan angle, 70° cutting angle, and ⅛ in. nose radius.

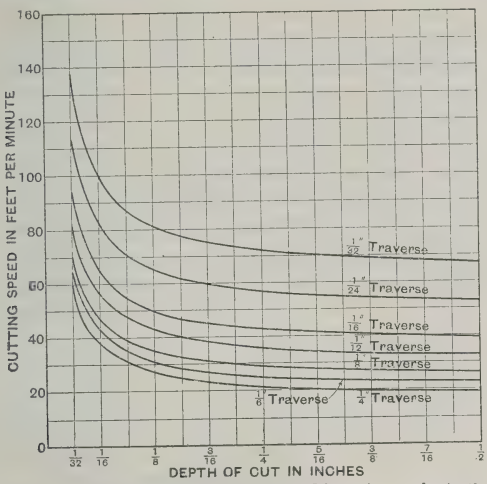


FIG. 62.—Relation between cutting speed in feet per minute (for a life of 2 hours) and depth of cut and traverse when cutting medium hard steel.

Cooling Medium.—When a liberal supply of cooling mediumi s used, the speed may be increased. For instance, with traverses of $\frac{1}{32}$ in. and ⅛ in. and various depths of cut on mild steel, it was found that the cutting speed might be increased by approximately 15 ft. per minute. The permissible increase will obviously depend upon the quality of cooling medium, dimensions of the cut, material cut, etc.

Tool Section.—Taking a 1¼-in. square tool as the standard, then for tools of smaller sections, say ½, ¾, and 1 in. square, the cutting speeds can be obtained by multiplying the value for the 1¼-in. square tool by the following factors: 0·815, 0·92, 0·97 for the sections in the order given, whilst for a 1½-in. square tool the permissible speed would only be about 2 per cent. greater than that of the 1¼-in. square tool.

Cutting Angle.—The results of the trials show that when the ratio of depth of cut to traverse is greater than unity, the 70° cutting angle tool gives the greatest durability for a given cutting speed, but when the ratio is equal to unity a cutting angle of about 80° is the most efficient and allows of an increase in the cutting speed of about 8 per cent. above that of the 70° tool. The relatively rough surface produced and the increased power consumption for the 80° cutting angle is likely to result in the 70° cutting angle being adopted for general workshop use. It may happen, however, that the power available is such as to demand a more acute cutting angle, in which case the permissible cutting speeds for tools having cutting angles of 50° and 60° are 71 per cent. and 82 per cent. respectively of that for the 70° angle tool, and these values are irrespective of the ratio of depth of cut to traverse.

Plan Angle.—Should the work and machine be sufficiently stiff or of a character such as would permit of a small plan angle tool, say 30° or 45° being used, then the cutting speed may respectively be taken as 1·55 or 1·21 times that given for the 60° plan angle tool. But should circumstances demand that a 75° or 90° plan angle tool be used then the permissible cutting speed is respectively 88 per cent. or 82 per cent. of that given for the 60° plan angle tool.

CUTTING SPEED AND DIMENSIONS OF CUT (CAST IRON).

The cutting speed when cutting soft cast iron, having a compressive strength of 29 tons per sq. in., with a 1¼ in. square tool, having a 75° cutting angle, ⅛ in. nose radius, and 60° plan angle (see fig. 59, p. 1071), are given in fig. 63. The speeds correspond to a 2-hours' life of tool (*i.e.* the length of time the tool will last before the cutting edge fails, and which is considered to give the most economical results) when cutting dry.

While the cutting speed decreases as the crushing strength of the material increases, a comparatively close relation between these cannot be given, as the cutting crumbles irregularly on the tool face.

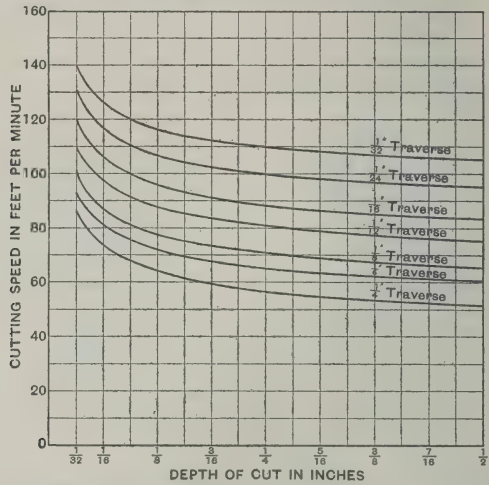


FIG. 63.—**Relation between cutting speed in feet per minute (for a life of tool of 2 hours) and depth of cut and traverse when cutting cast iron.**

The modification to be made in the cutting speed for other sections of tool, plan angle, etc., are for practical purposes similar to that given above for steel.

There is no close relation between the ultimate strength, or Brinell Hardness figure, of various metals and the cutting speed, but for ordinary straight steels (*i.e* containing little or no nickel, chromium, or tungsten), there is an approximation relationship which falls within ordinary workshop limits.

(*Proc. Manchester Assoc. of Engrs.* 1925.)

Cutting Lubricants.

The mineral oils best suited for use as cutting lubricants, either alone or mixed with animal or vegetable oil, are preferably of pale colour and low viscosity, ranging from 100 to 200 seconds Redwood at 100 deg. F., those of lower viscosity being used for high-speed conditions and those of higher viscosity for slow speeds. Tinged lard oil, containing as much as 10 or 15 per cent. of free fatty acids, is the animal oil most frequently used, either alone or in admixture : prime lard oil, which is almost free from acid, is more expensive, but is less inclined to gum under severe conditions of heavy cut and high speed. Wherever possible, a mixture of lard oil and low cold test mineral oil is to be preferred on account of greater fluidity in the cold. Cotton-seed oil oxidises more readily than rape oil, and should not be used for cutting lubricants that are to be employed in a circulation system.

Good cutting oils used straight will not cause rusting, but those containing fixed (animal or vegetable) oils, such as lard oil with a large percentage of free fatty acid, will give rise to verdigris on brass. Vegetable oils such as rape, with a small percentage of free acid, do not produce verdigris unless they are rancid. Cutting emulsions made up from cutting compounds or soluble oils and water cause rusting if they are used too weak or if they contain acid. Water containing sodium chloride is most destructive to emulsions and must not be used, nor must hard water on account of the precipitate caused by the calcium and magnesium salts in it. Emulsions of oil and water are not stable in the presence of even minute quantities of acid ; to a limited extent they can be reformed by neutralising the acids with ammonia, but excess of alkali may facilitate corrosion.

Factors in Selection.

Low speeds and shallow cuts require little cooling and lubrication. Low speeds and heavy cuts demand a lubricant of great oiliness, particularly if the material is tough. High speeds and shallow cuts demand a cutting medium with great cooling properties ; hence emulsions are

THE IRONMONGER

HARDWARE AND METAL TRADES ADVERTISER.

ESTABLISHED 1859.

PUBLISHED WEEKLY

28 ESSEX STREET, STRAND, LONDON, W.C.2.

For over seventy years The Ironmonger has been the leading newspaper published in the interests of buyers and users of hardware, metals and mill-furnishings.

To the factory owner and his buyer it is of equal interest, as in addition to voluminous information as to prices, etc., it publishes regularly reliable and informative articles on such subjects as :—

Losses in Belt Driving
Cutting Tools
Bolts and Rivets
Drills and Drill Grinding
Electric Furnaces
Heat Transmission
Boiler-House Efficiency and Economy

All this information is obtainable for the cost of a year's subscription, £1, which sum includes a copy of The Ironmonger Diary and Hardware Buyers Guide, published each December.

To anyone interested a specimen copy of The Ironmonger will be sent on application accompanied by trade card.

Rivet Your attention on

Boyer

Pneumatic Riveting Hammers

For all Classes of Riveting

CONSOLIDATED PNEUMATIC

TO reduce Costs and speed up output, you need the World's best Pneumatic Hammers, Drills and Grinders and in the BOYER LITTLE GIANT and RED GIANT you have them. Money cannot buy better.

See Editorial pages for C.P. Pneumatic and Electric Tools.

Write for particulars to :

THE CONSOLIDATED PNEUMATIC TOOL CO., LTD.

EGYPTIAN HOUSE, 170 PICCADILLY, LONDON, W.1

Telegrams : " Caulking, Piccy, London." *Telephone : Gerrard 9215*

Branches all over the World

F. 1079

favoured and should be used if the speeds are particularly high. Turpentine substitute is a satisfactory lubricant for aluminium, but being inflammable must be used with care. A mixture of paraffin oil and lard or other cutting oil for high speed work on aluminium is also dangerous and has led to several fires. Cutting emulsions which possess the necessary cooling properties, and are not inflammable, are to be preferred. High speeds and heavy cuts demand a cutting lubricant with great cooling and lubricating properties; hence heavily compounded cutting lubricants of low viscosity must be used. For tough material and heavy cuts an emulsion containing 15 to 20 per cent. of a vegetable oil has been reported to be satisfactory.

Cutting emulsions are nearly always used for brittle material, and frequently for tough material if the speeds are high and the cuts light; when the material is tough and the cut heavy it is necessary to employ cutting lubricants used straight and containing 10 to 50 per cent. of animal or vegetable oils, or consisting entirely of such oils. Emulsions, in some cases, have been found to form a deposit on the working parts of automatic screw-cutting machines; this may be avoided by using straight oils, but emulsions should be adopted wherever possible in view of the present scarcity of oil. The amount of soluble oil or soluble compound used in preparing the cutting emulsion varies from 2½ to 20 per cent.; the richer mixtures are used for severe conditions, and the weaker for light duty or for materials like brass and aluminium, where there is no danger of rusting.

The cutting lubricant may be applied by hand by a drop-feed system, or by some system employing gravity or pumping. In large machine shops the cutting oil is sometimes circulated in pipes throughout the works, returning through other pipes to a central tank; group systems with central tanks are excellent where one mixture is used for all the machines on the circuit. The return pipes should be large and arranged for easy access for cleaning; isolating valves should be employed to sectionalise the system when large; efficient strainers should be fitted on all return pipes and pump sections, and should be cleaned daily. Tanks, as a rule, should be cleaned out every four weeks, and return pipes every four months. Any scum formed should be skimmed off the tanks daily. It must be remembered that the oil or emulsion, in any system in which it is circulated over and over again, is exposed to the admixture of dust and dirt from the machine shop and to the oxidising influence of air. The suction of any pumps that are employed should always be covered to prevent air from being drawn in, since aeration of the oil or emulsion has a strong oxidising effect upon it.

In machining aluminium (drilling, planing, etc.), use a lubricant of lard oil, kerosene, or a mixture of kerosene and gasolene or lard oil, for free cutting and a good finish.

Effects on Health.

Workers should be instructed not to wipe their hands on rags before washing, and to avoid washing their hands in the cutting compounds. Ether soap, which dissolves oil, has been found useful in preventing inflammation of the hair follicles, and dusting the arms with a powder containing equal parts of starch and zinc oxide before starting work prevents the action of the oil on the skin.

Workers whose hands have become the seat of septic infection should not be allowed to work on machines, as they are liable to infect the oil with germs and so infect others. As a general rule, frequent washing with soap and hot water is sufficient to bring about rapid cure of folliculitis produced by plugging of the glands. Subsequently the skin may be dusted with zinc oxide and starch powder, and where this has been insufficient a mild antiseptic applied on lint has relieved the irritation and given good results. Septic infection of the skin due to cuts should be treated on general principles with suitable antiseptic dressings.

(*Bulletin No. 2, the Department of Scientific and Industrial Research.*)

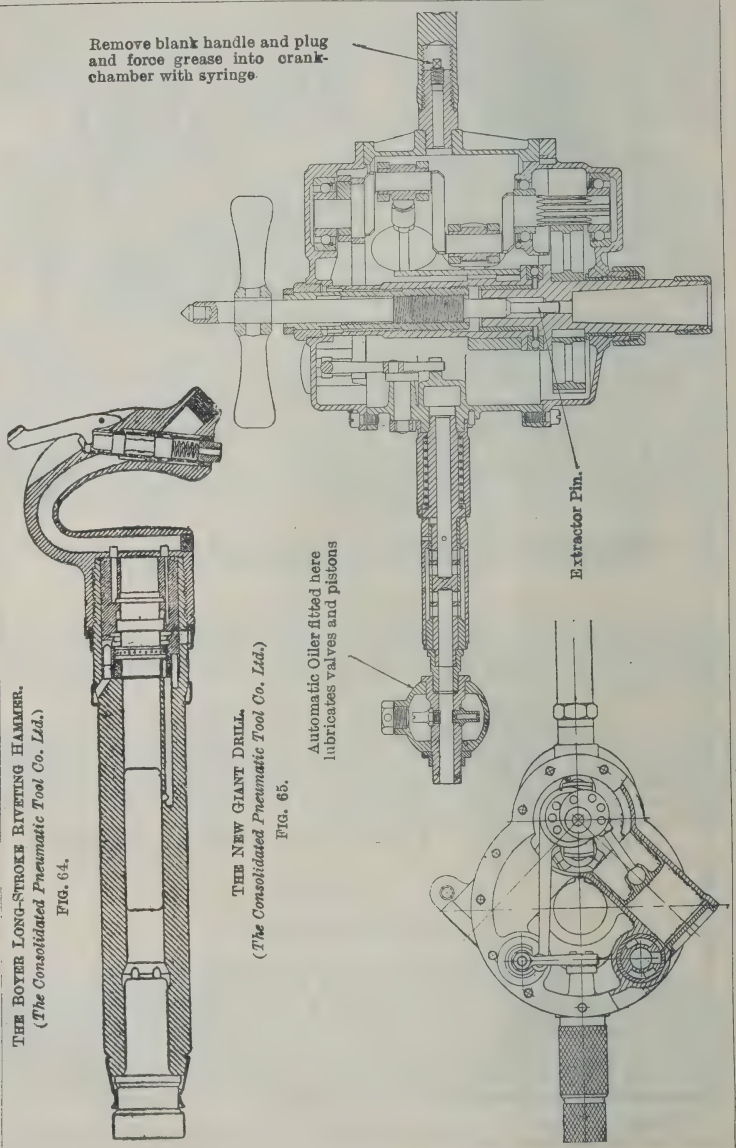
PORTABLE PNEUMATIC TOOLS.

The Boyer Long-Stroke Hammer.

(*The Consolidated Pneumatic Tool Co., Ltd.*)

This (fig. 64, p. 1080) is one of the best and most generally used of all pneumatic riveting hammers, at the present time. It has one distributing valve, which is placed at the rear end of the cylinder. The air-ports in the valve and cylinder of this hammer are so designed that the full air pressure is available to drive the piston forward on its working stroke, but the ports in valve and cylinder provided for effecting the return or idle stroke of the piston are of small area, so that the pressure used for this work is considerably reduced by wire-drawing, and the piston is returned practically without shock to the operator. The vibration of these hammers is thus reduced to a minimum.

For holding up against the riveting hammer, any of the present hand methods may be employed, and quite good work can be done either with the old-fashioned mawl, or a plain dolly bar



THE BOYER LONG-STROKE RIVETING HAMMER.
(*The Consolidated Pneumatic Tool Co. Ltd.*)
FIG. 64.

THE NEW GIANT DRILL.
(*The Consolidated Pneumatic Tool Co. Ltd.*)
FIG. 65.

consisting of a piece of wrought iron weighing about 28 lbs. Where a suitable backing can be found for it, the pneumatic piston holder-on is most useful, and for boiler work and in certain classes of heavy shipyard riveting the hammer type of holder-on is frequently used. This gives a series of blows on the head of the rivet whilst it is being formed.

Pneumatic riveting hammers are now recognised as thoroughly practical and efficient tools for dealing with any class of work which was formerly done by hand, and rivets from $\frac{1}{2}$ in. up to $1\frac{1}{2}$ in. diameter can be closed by suitable tools. It takes from 7 to 10 seconds to close a $\frac{7}{8}$ in. snap-headed rivet.

The air consumption of these tools is generally from 28 to 35 cubic feet of free air per minute, and the pressure may be from 80 to 100 lbs. per square inch, the latter figure being necessary for the heaviest class of riveting, and particularly for steam- or water-tight work.

Caulking Hammers.

The chipping or caulking hammer has now taken its place as an indispensable every-day tool, without which present rates of output could not be maintained. The average chipping hammer will remove mild steel at the rate of $3\frac{1}{2}$ to 4 oz. per minute, and the air consumption is from 17 to 20 cubic feet of free air per minute: pressure from 80 to 100 lbs. per square inch.

Drills.

A pneumatic drill of average size can be relied upon to drill in mild steel plate 1 in. diameter holes at the rate of 1 in. to $1\frac{1}{2}$ in. deep per minute, and larger and smaller holes in proportion, the air consumption varying from 30 cubic feet in the average drills to 50–60 in the larger tools. Usual air pressure 80 lbs. per square inch. Reamering and tapping are also among the varied uses to which these tools can profitably be put. Pneumatic drills are now available in a large range of sizes, from a tool weighing 8 lbs. for drilling $\frac{3}{16}$ in. holes up to a weight of 85 lbs. and a capacity for 3 in. drilling and tapping. (See figs. 35, 36, 37, 38, pages 1044, 1045, 1046, and fig. 65, page 1080).

Pneumatic Tools for Salvage Work.

The chipping hammer and drill mentioned above can be fitted for use under water, for ship repairs, dock construction, and other similar work, and are already very largely used in this connection.

Air Mains and Receivers.

Air Mains for the distribution of compressed air are generally made of wrought-iron piping of steam quality, and should be so arranged that the rate of flow will not very much exceed 50 feet per second. At intervals along this main, or wherever tools are likely to be employed, a tee piece should be inserted which will take a connection for the flexible rubber hose which joins up to the tools. An important point to take care of is the provision of adequate means for draining any moisture from the air receiver or the pipe lines themselves, and it is advisable to arrange for the discharge of the compressor to deliver the air at the top of the receiver, and to connect the air main lower down. This assists in depositing any moisture which may be in the air as delivered from the compressor.

Air Receivers should be fitted with a manhole, in order that their interior condition may be examined from time to time, as it is possible for corrosion to take place, or on the other hand, if the lubrication of the compressor is not carefully attended to, a large deposit of oil will be formed, which may result in an explosive mixture being present in the receiver.

Air should not be throttled at connections. It is a good plan to introduce water traps in suitable places in a pipe system. Strainers should be carefully watched, they are liable to get choked; they may be cleaned by blowing air through them in the reverse direction.

Tubing and Tube Couplings.

To secure good, economical service, air hose must not kink. Kinking not only shuts off the supply of air at the time, but it creates a permanent injury in the wall of the hose, which later develops into a leak. To prevent accidents of this character, the thickness and number of plies must be proportioned to the size of the hose; the rubber must be of good quality; and the inner tube, fabric, and outer cover must be properly proportioned to each other.

Sometimes customers specify wire winding as a protection against outside wear. The practice is not recommended. It makes the hose heavy and hard to handle. Once bent, the hose is hard to reshape; and a wire covering is more costly than the thick rubber cover recommended as the best protection for the outside of the hose.

A moulded hose is advisable for any length over 50 ft. This type of hose eliminates extra couplings, which retard the flow of air and decreases the pressure at the tool. This is an important feature. It has been found that a 15-lb. increase in pressure can make a 37 per cent. difference in the amount of work accomplished by the tool.

The pressure which the hose will stand is inversely proportionate to its diameter. A 1-in. 6-ply hose may have a bursting pressure as high as 800 lbs., while a 2-in. 6-ply hose made of the same materials would burst at 650 lbs. Hence, care always should be taken to specify greater number of plies for the larger sizes.

Flexible Metallic Tubing (*The United Flexible Metallic Tubing Co. Ltd.*, 139 *Queen Victoria Street, London*) is being largely used for pneumatic tools. As the tubing withstands crushing and cannot kink, it is well adapted for conveying compressed air to pneumatic drillers and other machines.

TABLE GIVING THE THEORETICAL VOLUME OF EQUIVALENT FREE AIR, IN CUBIC FEET,

that will flow per minute at various pressures through straight pipes of different diameters, each 100 ft. long, without any reduction of pressure.

Initial and Terminal Gauge Pressure, Lbs.	Nominal Diameters of Pipes in Inches.																		Initial and Terminal Gauge Pressure, Lbs.
	1	1½	2	2½	3	3½	4	5	6	7	8	10	12	15	18	20	22	24	
10	24·05	75·67	153·0	244·7	433·3	636·4	883·8	1585	2542	3701	5134	9212	14530	22530	36280	48180	61870	77880	10
20	28·48	89·54	117·9	288·8	512·9	753·0	1045	1871	3006	4381	6076	10900	17210	26640	42940	57000	73220	92150	20
30	32·35	101·6	202·0	329·1	582·5	855·0	1187	2124	3415	4975	6900	12380	19540	30270	48750	64760	83160	104630	30
40	35·82	112·5	223·5	364·2	644·7	946·4	1314	2352	3779	5507	7635	13700	21620	33500	53960	71640	92010	115800	40
50	**38·96**	**122·4**	**243·4**	**396·3**	**701·5**	**1030**	**1430**	**2558**	**4114**	**5993**	**8312**	**14910**	**23520**	**36450**	**58730**	**74300**	**100150**	**126030**	**50**
60	41·83	131·4	261·1	425·4	752·9	1105	1535	2747	4416	6433	8920	16000	25260	39120	63040	83660	107470	135240	60
70	44·53	139·9	278·0	452·9	801·8	1176	1634	2925	4701	6848	9499	17030	26900	41660	67110	89100	114420	144000	70
80	47·08	147·9	294·0	478·8	847·6	1244	1728	3091	4971	7240	10040	18000	28430	44050	70960	94180	120970	152230	80
90	49·54	155·6	309·3	503·8	891·8	1307	1817	3253	5230	7619	10560	18940	29920	46340	74660	99100	127300	160190	90
100	**51·88**	**163·0**	**324·0**	**527·5**	**933·8**	**1370**	**1904**	**3407**	**5477**	**7979**	**11050**	**19850**	**31330**	**48530**	**78190**	**103770**	**133300**	**167720**	**100**
110	54·10	169·9	337·8	550·1	973·9	1429	1985	3552	5712	8320	11530	20690	32670	50610	81540	108220	139000	174920	110
125	57·15	179·5	356·8	581·3	1028	1510	2097	3754	6034	8789	12180	21860	34520	53470	86140	114430	146850	184800	125
150	62·10	195·1	387·8	631·7	1117	1641	2280	4080	6558	9553	13240	23760	37520	58120	93610	124280	159600	—	150
175	66·71	209·6	416·5	678·4	1201	1762	2448	4382	7044	10250	14220	25520	40300	62420	100520	133460	—	—	175
200	**70·93**	**222·9**	**443·0**	**721·4**	**1277**	**1874**	**2603**	**4659**	**7489**	**10900**	**15120**	**27140**	**42850**	**66370**	**106900**	**141910**	—	—	**200**
250	78·70	247·3	491·4	800·4	1416	2079	2889	5170	8309	12100	16780	30110	47550	73640	118620	—	—	—	250
300	85·85	269·9	536·3	873·4	1546	2269	3152	5642	9063	13210	18320	32870	51890	80380	—	—	—	—	300
400	98·55	309·9	615·7	1002	1775	2606	3619	6477	10400	15160	21020	37730	59580	92270	—	—	—	—	400
500	109·7	345·1	685·3	1116	1976	2902	4031	7215	11590	16890	23420	42020	66340	102750	—	—	—	—	500
600	**119·9**	**377·3**	**749·5**	**1220**	**2160**	**3172**	**4405**	**7882**	**12670**	**18460**	**25600**	**45940**	**72520**	—	—	—	—	—	**600**
700	129·3	406·6	807·6	1315	2328	3419	4748	8496	13650	19890	27590	49500	78150	—	—	—	—	—	700
800	138·1	434·1	862·0	1404	2486	3650	5070	9072	14580	21240	29450	52840	—	—	—	—	—	—	800
900	146·3	460·0	914·4	1489	2636	3871	5373	9618	15460	22520	31230	56040	—	—	—	—	—	—	900
1000	154·3	485·1	963·5	1569	2778	4079	5664	10130	16290	23730	32920	59060	—	—	—	—	—	—	1000
1100	**161·3**	**507·1**	**1008**	**1641**	**2906**	**4264**	**5924**	**10590**	**17040**	**24820**	**34430**	—	—	—	—	—	—	—	**1100**
1200	168·6	530·3	1053	1716	3037	4457	6193	11080	17810	25940	35980	—	—	—	—	—	—	—	1200
1300	175·7	552·2	1097	1787	3163	4643	6448	11540	18550	27030	—	—	—	—	—	—	—	—	1300
1400	182·2	572·4	1137	1853	3280	4814	6686	11970	19230	28020	—	—	—	—	—	—	—	—	1400
1500	188·2	591·5	1175	1914	3389	4973	6908	12360	19870	—	—	—	—	—	—	—	—	—	1500

(The Consolidated Pneumatic Tool Co.)

TABLE SHOWING LOSS IN AIR PRESSURE, IN POUNDS, DUE TO FRICTION IN PIPES 100 FT. IN LENGTH.

Gauge Pressure at Entrance to Pipe, 100 lbs.

Pipe Diameter. Ins.	Cubic Feet Free Air delivered per Minute.																												
	25	50	75	100	125	150	200	250	300	350	400	500	600	700	800	900	1000	1200	1500	1800	2000	2500	3000	3500	4000	5000	6000	7000	8000
1	0·3	1·18	2·66	4·75	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1¼	0·08	0·33	0·73	1·3	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1½	—	0·13	0·29	0·6	0·8	1·15	2·04	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
2	—	0·03	0·07	0·12	0·18	0·26	0·47	0·73	1·05	1·43	1·87	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
2½	—	—	—	0·03	0·05	0·08	0·14	0·21	0·31	0·42	0·54	0·85	1·22	1·66	2·17	—	—	—	—	—	—	—	—	—	—	—	—	—	—
3	—	—	—	—	—	0·03	0·05	0·08	0·11	0·15	0·2	0·31	0·45	0·61	0·8	1·01	1·25	1·8	—	—	—	—	—	—	—	—	—	—	—
3½	—	—	—	—	—	—	—	—	0·05	0·07	0·09	0·14	0·21	0·28	0·37	0·46	0·57	0·82	1·28	—	—	—	—	—	—	—	—	—	—
4	—	—	—	—	—	—	—	—	—	—	0·05	0·07	0·1	0·14	0·18	0·23	0·28	0·41	0·64	0·92	1·13	1·77	—	—	—	—	—	—	—
4½	—	—	—	—	—	—	—	—	—	—	—	0·04	0·06	0·08	0·1	0·13	0·16	0·23	0·35	0·51	0·63	0·98	1·41	—	—	—	—	—	—
5	—	—	—	—	—	—	—	—	—	—	—	—	0·03	0·04	0·06	0·07	0·09	0·13	0·2	0·29	0·36	0·56	0·81	—	—	—	—	—	—
6	—	—	—	—	—	—	—	—	—	—	—	—	—	—	0·02	0·03	0·04	0·05	0·08	0·11	0·14	0·22	0·32	0·43	0·56	—	—	—	—
7	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	0·02	0·02	0·04	0·05	0·06	0·1	0·14	0·2	0·25	0·4	0·57	—	—
8	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	0·02	0·03	0·03	0·05	0·07	0·1	0·13	0·2	0·29	0·39	0·51
10	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	0·01	0·02	0·02	0·03	0·04	0·06	0·09	0·13	0·16
12	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	0·01	0·01	0·02	0·02	0·04	0·05	0·06

(The Consolidated Pneumatic Tool Co.)

Belliss & Morcom Pneumatic Hose Coupling.

(Belliss & Morcom, Ltd., Ledsam St. Works, Birmingham.)

This coupling (fig. 66), which is simple in construction and convenient in use, does not rely on springs to maintain its lock, being immediately sealed by the internal pressure, and cannot unlock of its own accord in or out of use. It is claimed to be absolutely air-tight; easily and quickly disconnected; practically unbreakable, being made from steel bar; dust-proof, and being gun-blacked, will not easily rust. Having no external projections, it can be easily drawn through close quarters or past work lying about the shop.

The rubber jointing, being held securely in place, cannot blow out, but can be easily removed by hand for replacement when required.

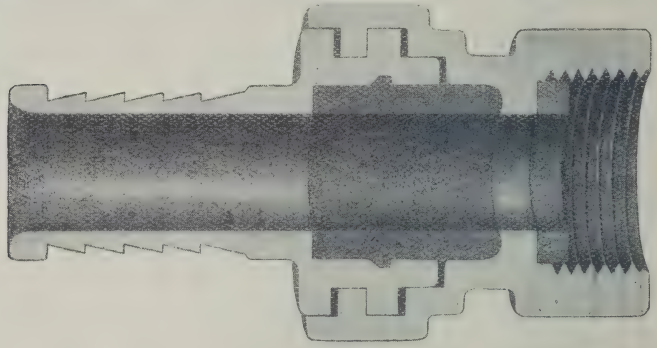


Fig. 66.

It is screwed on the shank and tapered to facilitate fitting on the hose. The ends of each hose pipe can be locked together, thus giving a neat appearance when coiled up. Additional hose for extensions can be connected up instantly. The coupling is supplied in the following sizes:—$\frac{1}{2}$ in. gas for chipping hammers, drills, riddles, etc.; $\frac{3}{4}$ in. gas for riveting hammers, etc.; 2 in. gas for coal-cutting tools and pressure mains.

The 'Flexstel' System of Ball and Socket Joints.

(W. H. Dorman & Co. Ltd., Stafford.)

This system of Flexible Pipe Joints provides complete units (fig. 67), having standard flanged or screwed ends, and may be readily interposed between suitable lengths of piping to provide for movements up to an included angle of about 40°.

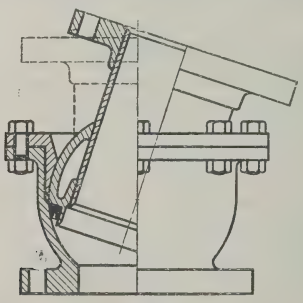


Fig. 67.

A special feature of the joint is the position of the packing ring, which is placed on a step on the leading end of the ball member, sealing the joint and insulating the spherical bearing surfaces

THE IRONMONGER

HARDWARE AND METAL TRADES ADVERTISER.

ESTABLISHED 1859.

PUBLISHED WEEKLY

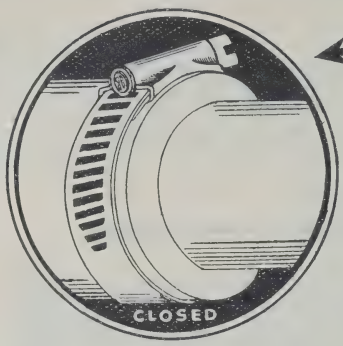
28 ESSEX STREET, STRAND, LONDON, W.C.2

For over seventy years The Ironmonger has been the leading newspaper published in the interests of buyers of hardware, metals and mill furnishings.

Its news service is invariably up-to-date and its information reliable, while its weekly record of current prices of staple commodities gives to its pages a world-wide reputation as a valuable buyers' guide for the purchasing departments of engineers, shipbuilders, collieries, etc. etc.

All this information is obtainable for the cost of a year's subscription, £1, which sum includes a copy of The Ironmonger Diary and Hardware Buyers Guide, published each December.

To anyone interested a specimen copy of The Ironmonger will be sent on application accompanied by trade card.



Tight with equal pressure all round.

THIS

'JUBILEE' Worm-Drive

(Pat.) **CLIP** British Manufacture.

Will always **stop the LEAK or DRIP,** and the type you should have fitted on your **RADIATOR JOINTS, UNIVERSAL JOINT COVERS, AIR, OIL and WATER HOSE JOINTS,** and all other purposes where clips or wire are required.

Finish now with your old-fashioned and inefficient types, **GET "JUBILEE" WORM-DRIVE CLIPS with GUARANTEED COMPLETE SATISFACTION.**

ALL IN ONE PIECE, NO PARTS TO LOSE

EASY-TO-FIT

NEVER LEAK

Manufacturers :

L. ROBINSON & CO.,

London Chambers, Gillingham, Kent.

Stocked by Factors & Merchants

SEND FOR A FREE SAMPLE CLIP FOR TEST

F. 1085

from contact with the fluent. The outer surface of the packing is so shaped to conform to the spherical surface of the cup, and being rather full in size provides for initial compression and ensures the joint being leak proof at low pressures. The joint is effectively sealed at higher pressures by the lip formation of the packing ring.

These joints are made in a range of sizes from ¼ in. to 12 ins. in bore, sizes up to 3 ins. being usually made with ends screwed to British Standard pipe threads, and sizes 2 ins. upwards have flanged ends faced and drilled to suit British Standard Tables No. 1 or No. 2. Joints with screwed ends can be had with external or internal standard screw threads and are of many types, the straight joints having both single and double ball connections, while the range of elbow joints includes 90° single and double, 135° single, and 40° double.

'Flexstel' joints are used for conveying compressed air, steam, water, oil, gas, acids, etc., a suitable packing ring being fitted for each use.

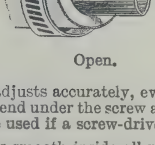
These joints can be fitted with obvious advantages to cater for the movement of machines or any moving bodies where it is desired to apply high or intermittent pressures of any of the fluents referred to above.

Jubilee Worm-Drive Hose Clips.

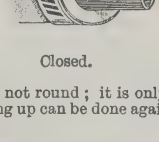
(Patent.)

(*L. Robinson & Co., Gillingham, Kent.*)

In these clips (fig. 68), a direct pull is exerted on a steel ribbon by a worm with heavy square-cut threads, which cannot be stripped. The arrangement gives an even and equal pressure round the hose, is easily fitted, and, when being removed, opens out automatically.



Open.



Closed.

Fig 68.

The clip adjusts accurately, even on hose too large or pipe not round; it is only necessary to push the free end under the screw and tighten up; the tightening up can be done again and again—a coin can be used if a screw-driver is not to hand.

The clip is smooth inside all round, and gives an even bearing; there are no sharp edges to cut, or spaces to permit leakage.

Being made of best quality steel, heavily galvanised, it is non-rustable and practically indestructible.

The whole arrangement is in one piece, so that there are no parts to lose and nothing to put together.

To fit from ⅝ in. to 3¼ in. outside diam., only five sizes are needed, viz.:—

Size.	Outside Diam. of Hose.	Size.	Outside Diam. of Hose.
No. 0	⅝ in. to ⅞ in.	No. 3	2 in. to 2⅝ in.
No. 1	⅞ in. to 1½ in.	No. 4	2⅝ in. to 3¼ in.
No. 2	1½ in. to 2⅛ in.		

Larger sizes are also supplied.

The clip will stand up to practically any pressure, and is officially approved by the British Air Ministry for use on aircraft; also by the Metropolitan Railway and the London, Midland & Scottish Railway for the automatic brakes on their trains, and by a number of fire brigades.

Care of Pneumatic Tools.

(1)

Unless pneumatic tools are kept clean, well lubricated, and limited to the uses for which they are intended, they will not give the service for which they are designed. Wherever possible, the air supply should be drawn from some point outside the building where the air is most apt to be clean, cool and dry. In any event, a strainer should be placed over the exposed end of the inlet pipe. Such a strainer can be easily made by covering the opening with wire netting over which one or two layers of muslin have been placed. A strainer should also be placed in the pipe lines leading directly to the pneumatic tools, and a strainer inserted between the pneumatic hammers and the hose nipple.

Lubrication.—Valves and pistons for both hammers and drills require a light machine oil, and as the compressed air which comes in direct contact with these parts has a tendency to drive the lubricant out through the exhaust, it is advisable to oil such parts freely and often, about once every hour while the tool is in constant use. Such portions of the tools as the gear and crank cases to which the compressed air does not have direct access are best lubricated with grease. This grease may be forced into the crank cases through the dead air handle by means of a squirt gun or syringe. A filling every 10 hours while the drills are in constant use gives the best results. It is a good plan to immerse pneumatic hammers in a bath of benzene or kerosene over-night and then blow them out under pressure the following morning, after which they should be thoroughly lubricated with a light machine oil.

(2)

The greatest damage done to the working parts of air tools may be found in the compressor, or rather in the air delivered by the compressor, for not by any means do all users of air tools appreciate the actual necessity for clean, dry air. Until the advent of the air filter, dirty, gritty, moist air had been the cause of thousands upon thousands of pounds being spent in needless repairs on air tools, and on the compressor itself, especially that of the portable type. Besides seeing that clean air is delivered by the compressor, care should also be taken to keep dirt out of the hose and leaders. Open hose ends are often allowed to lie on dusty, dirty, and even sand-covered floors, and then be hooked up to motors and guns without a thought of the necessity for blowing out before connecting.

Motor and gun valves require regular inspection in order that impending trouble may be caught in time, but with clean air and regular, systematic oiling with the right kind of oil, the time needed for inspection can be reduced greatly. Of course, the utilisation of a known, valued oil is a positive requirement. An oil of good quality will remain for some time in the valves and around the plunger of a riveting or a chipping gun, and guns with the right kind of oil have been known to work perfectly when oiled once in three hours of continual service.

(*'Compressed Air Magazine.'*)

Pneumatic Riveting.

According to information received from one of the leading East Coast shipyards, the best day's work on a shell with pneumatic riveting is 700 rivets of $\frac{7}{8}$ of an inch diameter, and for hand work, under the same conditions, 430. An average day's work, however, is considerably less than this, and may be taken as being 510 rivets with pneumatic rivets, against 264 with a hand squad, the figures being taken for work on all parts of one ship. According to the *Board of Trade Journal* the following may be taken as indicating the difference in the day's work of a hand and pneumatic riveting squad:—On the shell, with $\frac{7}{8}$ in. rivets, hand-riveting, 260 to 360 rivets; with pneumatic riveting, 500 to 550; on the deck, with $\frac{5}{8}$ in. rivets, hand-riveting 450, pneumatic 700; on deck with $\frac{3}{4}$ in. and $\frac{7}{8}$ in. rivets, hand-riveting 350, pneumatic 550. Again, the pneumatic tool requires fewer men to work it. A hand riveting squad consists of three men and one or two boys. From two of these squads three pneumatic riveting squads can be formed, effecting a saving of 50 per cent. in labour alone. A third advantage of the pneumatic tool is that it gives both the other great advantages with less exertion and consequently less fatigue on the part of the men. (*'Machinery.'*)

Cost of Compressing Air.

In a large shipyard on the north-east coast the actual cost of compressing 100,000 cu. ft. of free air to 100 lbs. per sq. in. pressure with electric current at 1*d.* per unit worked out as follows:—

	s.	*d.*
Power	20	1
Circulating water	0	3
Labour	6	0$\frac{3}{4}$

(*R. W. Wilson. Paper read before Inst. of Mech. Engs., Feb.* 8, 1924.)

Mechanical Hammers.

The Kango Electro-Mechanical Hammer.

In this hammer (fig. 69) an entirely new principle is employed, centrifugal force is utilised for converting rotary motion into progressive kinetic energy, 3,000 blows per minute being delivered at the head of the tool in use, whether cold chisel, fettling tool, keyway cutter, stone driller, stone dresser, etc.

In the upper part of the casing of the arrangement is a motor which drives through a sliding and flexible coupling a spline shaft operating the striking element, which is housed in the lower part of the casing, a special oil-retaining device being fitted to prevent the oil in the lower part of the casing reaching the motor.

The striking element consists of a special steel structure carrying a bevel pinion and two crown wheels. The pinion is centrally placed and drives the two crown wheels, which are mounted on pins projecting from the sides of the striker, in opposite directions. On one side of

each crown wheel is formed a heavy semi-circular weight, and the teeth are so engaged with those of the pinion that these weights are opposite each other when they are at the top and bottom of their revolution. The lower end of the striking element forms the hammer head, which conveys the blow to a specially formed transmitter secured in the lower end of the casing by a flexible diaphragm which renders oil-tight the lower part of the casing as the whole of the striking element works in a bath of oil. The striking element is kept towards the transmitter by a strong spiral spring which also serves to limit its oscillation.

The centrifugal force created by the weights on the crown wheels coming into juxtaposition at opposite points in their revolution, imparts a reciprocating motion to the whole striking element, the movement upwards being partially absorbed by the spring previously referred to, whilst the downward movement is free to convey the full impact to the transmitter and through this to the tool being used.

All tools can be instantly extracted or replaced.

Hammers are at present constructed suitable for 110, 200, or 220 volts; A.C. or D.C. current consumption about 300 watts only; the current supply can be taken from any lamp socket.

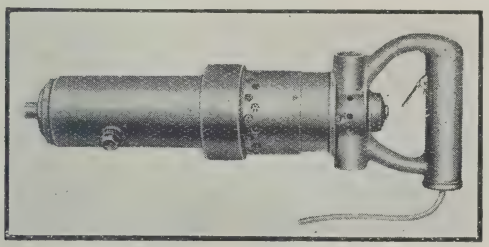


FIG. 69.

The weight complete of the hammer (which cannot be overloaded) is only 9 lbs.; no compressor is employed; all working parts of the striking mechanism work in an oil-bath and are easily accessible.

Diamond Turning Tools.

For turning materials, such as ebonite, galalith, bakelite, erinoid, and fibre, diamonds are greatly superior to any class of steel tool. More recently they have been applied to the turning of aluminium, aluminium alloys, and brass. On the Continent many of the largest makers of motor cars employ diamond turning for machining alloy pistons, and in this country the practice is gaining ground. On the Continent, also, diamonds are being used for boring gudgeon pin holes in aluminium pistons, and also as boring cutters held in bars for boring crankshaft bearing housings in aluminium crank-cases.

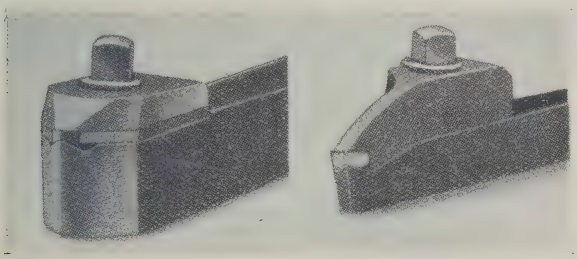


FIG. 70.

Fig. 70 shows two diamond tools specially designed for turning aluminium pistons. That on the left is for turning the cylindrical portions of the body of the piston, while that on the right is a square-nosed diamond for turning the piston ring grooves.

The diamond in the body turning tool is cut so as to give three small facets under 1 millimetre wide—a central facet and one on either side. When the tool is first used the central facet is set accurately so as to be parallel with the axis of the work.

With a feed of 0·1 millimetre the cuts taken by the diamond facet overlap, and providing the facet has been accurately set as described, the resulting finish is practically equivalent to a polished surface, free from tool marks. On an aluminium alloy piston, 3¼ ins. in diameter and 3½ ins. long, a spindle speed of 750 revolutions per minute is used, the best cutting speed being about 600 feet per minute, and the turning time about 60 seconds.

On such work each facet of the diamond lasts two months before any re-setting is required. As the remaining facets are still available, each diamond lasts about six months before it need be taken down and sent to the suppliers for re-cutting, after which it is useful for another long spell of work. It will be realised that with a very slow-wearing tool, such as a diamond, size is easily maintained, especially as little heat is developed during the cutting operation, the whole of the shaving being removed by a true cutting process without any sign of tearing.

A diamond tool appears to leave a finish greatly superior to grinding, and there is no risk of leaving abrasive particles embedded in the piston, it costs many hundreds per cent. more than a steel tool, but as its total life is about eighteen months, with very few resettings in the machine or redressings by the suppliers, it must be considered very economical, despite the apparently high first cost.

The diamond is of wedge shape, being deeper at the back than the front, while in plan the rear portion is dressed to V-form and fits a corresponding vee in the intermediate plate, seen under the upper clamp plate. The thickness of the intermediate plate is such that when the diamond is clamped in the holder the intermediate plate is gripped simultaneously, and the diamond relieved of any crushing pressure.

The diamond in the other tool is fitted without an intermediate plate, but is veed as shown on the under surface, to correspond with the lower portion of the holder. The diamond is the exact width of the slot required in the piston, but the extreme ends of the front cutting face are dressed to reduce the rake and remove the sharp corner. If these are left there is a risk of the diamond chipping at these points. These diamonds can be guaranteed to finish grooves in aluminium pistons within limits of 0·00025-in. in width.

The methods of holding these diamonds are the subjects of patents.

(' *Machinery*,' *August* 22, 1925.)

Files.

DESCRIPTION.

Single-cut.		*Double-cut.*		*Rasp.*	
Rough	Second-cut	Coarse	Smooth	Coarse	Second-cut
Coarse	Smooth	Bastard	Dead-smooth	Bastard	Smooth
Bastard		Second-cut			

The terms ' rough,' ' coarse,' ' bastard,' ' second-cut,' ' smooth,' and ' dead-smooth,' have reference only to the coarseness of the teeth, while the terms ' single-cut,' ' double-cut,' and ' rasp ' have special reference to the character of the teeth.

There are more than 50 types of files recognised in the file manufacturing trade, and each type is supplied in 24 different lengths. Each length can generally be obtained in one of five cuts of varying coarseness.

RE-SHARPENING FILES.

(1)

The sand-blast process consists of injecting fine sand against the backs of the teeth by means of a steam jet, and is applicable to all files, from the rasp to the finest Groubet files. The action of the sand is to cut away the backs of the file teeth, thus forming a straight level on the teeth back, and giving a new cutting edge, and the process occupies from 3 to 5 minutes.

(2)

Another method is first to carefully cleanse the files with hot water and soda; they are then brought into connection with the positive pole of a battery, in a bath composed of 40 parts sulphuric acid and 1,000 parts of water. The negative (zinc) pole of the battery is connected to a copper spiral surrounding the files but not touching them. This arrangement is the result of practical experience. When the files have been in the bath for 10 minutes they are taken out, washed, and dried, when the whole of the hollows will be found to have been attacked in a sensible manner; but should the effect not be sufficient, the files are replaced in the bath for the same period as before. Sometimes two operations are necessary, but seldom more. The files thus treated are to all appearance like new ones, and are good for 60 hours' work.

(3)

The following is a formula for an etching or recutting solution without electric current :—

Nordhausen (fuming) sulphuric acid	1 part.
Nitric acid	1 ,,
Concentrated sulphuric acid	½ ,,

All the above parts are given by volume.

The process is carried out by first submerging the files for cleansing for a period of 15 minutes in a solution of caustic soda in boiling water, the proportions being about 150 grains of caustic soda to 2 gallons of boiling water. With ordinary files this will suffice for about 150. After cleansing, the files are placed in a tank which is acid proof, and 13 parts by volume of water are added. The files should not overlap and should be arranged so that the liquor can obtain access to all parts to be recut.

The above mixture is then added and the files allowed to stand for 5 to 10 minutes.

To Clean Files.

Files clogged with tin or lead can be cleaned in a few seconds by strong nitric acid. For iron filings, use blue vitriol; rinse in water, and dip in nitric. For copper or brass, use nitric several times; for zinc, dilute sulphuric acid. After any of these treatments, rinse the files in water, brush vigorously, and dry in sawdust, or by burning alcohol on the file.

When files become clogged with aluminium or aluminium alloys a strong solution of potash will dissolve the aluminium and leave the file teeth unaffected.

Plating Files.

Chromium is used for the plating of files to be used for cutting soft metals. One firm which treats its files in this manner has found that the metal being filed does not clog up the file teeth as it does on untreated files; moreover, their useful life is greatly increased.

Auxiliary File Handle.

Figs. 71 and 72 show a form of auxiliary file handle, which was constructed for use in connection with an awkward filing job by Mr. H. Moore, of Hamilton, Ontario, and may be found convenient by other workers. The arrangement was devised to assist in filing out the ends of a number of channel pieces used on pump work. The position was such that only the tops of the fingers could rest on the file, and pressure could not be brought to bear on it in the usual way.

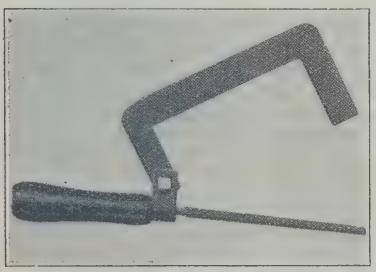


Fig. 71.

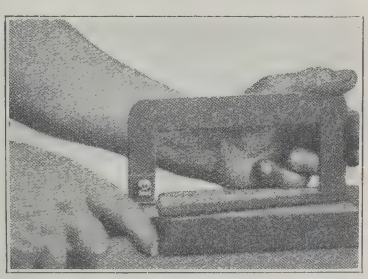


Fig. 72.

A clip of sheet metal was made to fit around the ferrule of the handle and a bridge piece, cut from No. 13 U.S. plate was pivoted to this on a bolt and nut. The bolt tightened the clip on the handle, and pressure could be brought to bear on the file tip by the bridge piece. As the bridge piece is jointed, any want of alignment between the file and handle can be accommodated. Fig. 71 shows the complete arrangement, and fig. 72 the auxiliary handle in use.

(*'The Engineer.'*)

Tube Cutters.

The particular feature in these cutters (fig. 73) consists in widely separated duplex rollers, arranged in a block on the spindle end, making it practically impossible to set the tube cutter

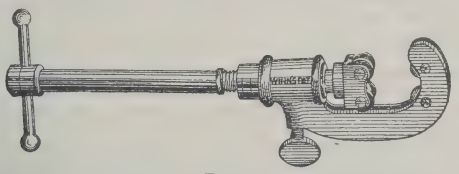


Fig. 73.

on the tube operated upon in any other way than absolutely at a right angle with it. All who have had occasion to cut a tube will realise the important gain this means, because no time is

lost in cutting screw tracks on the tube, and there is no consequent breakage of cutters. The rollers save all friction on the tube, and two cutters are provided in the hook, so that the tubes can be cut off with great ease and speed. There is a further saving of time by the use of these tools, in the quick adjustment for different sizes (¼ in. to 1½ in. for one tool).

Using Woodworking Tools on Metal.

Aluminium, Duralumin (in the annealed state or just after heat treatment), and soft brass, can all be planed with an ordinary carpenter's block or jack-plane, the same as wood.

The blade of the plane used for these materials will require more frequent sharpening, however, and it is better to grind it to a sharper angle, which gives less clearance and a larger included angle, thus making the blade stronger and better able to resist wear on the cutting edge. The ordinary wood chisel may also be used on the metals mentioned, provided it is ground with a shorter bevel. The ease with which these soft metals can be worked with hand tools is surprising. They can be sawn with a coping saw or compass saw.

Hack-Saws.

Insufficient attention is given to the lubrication of hack-saws while cutting. It is as necessary to apply ample lubricant to hack-saws as to any other cutting tool, and many power hack-saw blades merely rub their way through stock because the first few strokes have destroyed the hardness of the teeth, owing to the temperature attained in the absence of lubricant.

Hand hack-saw blades generally suffer the same careless treatment accorded to files. The proper use of these hand tools is not impressed upon the semi-skilled labour of to-day, and the hack-saw bills of many firms could be reduced considerably if more attention were given to this matter.

Experiments undertaken in France to determine the best method of sawing metal by hand indicate that it takes less time to saw off a rectangular bar of steel when the saw is moved parallel to the shorter side of the rectangle than when it moves parallel to the longer side. It was also found that the shorter the stroke of the saw, the quicker would the bar be sawed off. In sawing off a rail by a hack saw, it required nearly three times as long when the stroke was 4 ins. in length, as compared with when it was somewhat less than ½ in. in length. The work performed per minute decreases regularly for each ½-in. increase in the length of the stroke. It is also stated that by holding down the end of the saw farthest away from the workman—not by hand but by a weight—the time required for sawing through a bar is shortened, a thinner blade can be used, and a better surface on the ends of the sawed-off bar is obtained.

Cold Chisels.

A large number of the chisels now in use are not forged, shaped, and hardened in the best way to give long life and adequate service. Considering the making of a chisel from ⅝ in. octagon tool steel, the following method has been found to give good results:—

Forging.—Cut off a length of steel, depending on the length of chisel desired, and heat up the end for 2 ins. to a bright cherry-red. Trim off two opposite sides so as to form a blunt-nosed tool and eliminate the danger of lapping when drawing out. The horn of the anvil should be used in drawing out, inasmuch as this will have the least possible tendency to widen the piece, and therefore the minimum amount of 'edging-in' or hammering on the edge will be necessary. A chisel should not be hammered on the edge, because the grain of the steel will thereby be distorted or 'crumpled up,' and this always has a tendency to weaken any metal. If in the final forging operation the chisel gets a little too wide, it can be trimmed off on the emery-wheel during the grinding operation.

Draw the chisel out so that it will be about ¼ in. thick at the end and about ⅝ in. thick and 1½ in. back from the end. The forging should be finished with light blows until the steel has almost lost colour, but it absolutely must not be struck after the colour has disappeared. It is good practice to reheat the steel to a dull red without using any blast, and give it a second hammering with light blows until the colour has again almost disappeared.

The four important things to remember in forging a chisel are, therefore: Draw out at a good cherry-red heat: finish with light blows at a dull-red heat; do not hammer after the colour has disappeared; hammer as little as possible on the edge, and then only when the steel is fairly hot.

Grinding.—Grind the chisel before it is hardened, as it can be ground faster in this way without the danger of drawing the temper. The shape of the edge of the chisel is very important, although this fact is often overlooked. Fig. 74 shows a chisel ground with a concave edge. If this is driven down on to a flat surface, it is obvious that a great strain will be put on the corners, and they are almost sure to break off. Fig. 75 shows a chisel ground with a perfectly flat edge. If this chisel is driven down on to a flat surface and held perfectly straight the cutting strain will be distributed evenly over the entire edge, and the chisel will be satisfactory. It must be remembered, however, that it is almost impossible to hold a chisel absolutely straight, and,

therefore, either one corner or the other will be severely stressed by the chisel being tilted over. Fig. 76 shows a chisel ground with a slightly convex edge, which is by far the best for ordinary work. The corners of a chisel are always the danger point, and with the convex edge these corners are protected even when the chisel is tilted over considerably to one side or the other.

FIG. 74.

FIG. 75.

FIG. 76.

Hardening and Tempering.—After the chisel has been ground to the desired shape, heat it to a dull cherry-red colour for about 2½ ins. from the end, and quench it vertically in cold water to a depth of 1¼ ins., moving it up and down until no red colour is left in any part of the steel. The part which has been drawn out should now be polished with emery cloth and the temper drawn to a dark purple or a blue by holding the chisel over the fire or in a furnace. Always draw a chisel a little more in winter than in summer. It has been recommended to harden the chisel back much farther than usual, because a chisel so made can be ground a great many times without redressing. (*Joseph T. Byerson and Son, Chicago.*)

To Reduce the Internal Diameter of a Steel Bush.

A bush was required to be a force fit on a 6-in. shaft: and should have been bored 5·990 ins. ± 0·002 in., but was bored 0·015 too large.

The bush was heated to just a red, about 2 ins. up at one end, and just 1 in. of the 2 ins. dipped in cold water. It was next heated another inch or two up, keeping the first cooled inch as cool as possible, and then dipped a further inch into water. This was continued inch by inch, each cooled inch apparently pulling in the inch in the length adjacent to it until the whole of the bush (or sleeve) was pulled in quite twenty thousandths, and required to be skimmed out 0·005 in. to force it on to its shaft. (*J. T. T.*, '*Machinery.*')

Gripping Pieces of Uneven Width in the Machine Vice.

If two or more pieces have to be held in the machine vice for drilling or other operation, and their widths differ slightly, a strip of lead, *s*, placed as shown in fig. 77 will compensate for the

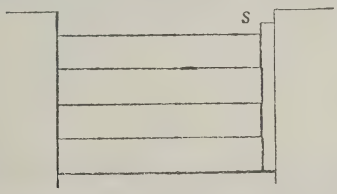


FIG. 77.

irregularity and afford a secure grip. Or a piece of belting, cloth, canvas, wire meshing, etc., will serve the purpose.

Block for Holding Tapered Work in the Vice.

Holding irregular or wedge-shaped parts in the vice while chipping, sawing, or performing other manual operations is awkward unless the parts are supported by blocking. It is possible,

however, to hold tapered work quite securely by using a simple block, such as shown at A in fig. 78, in conjunction with the regular vice jaws. This special block should be about the same

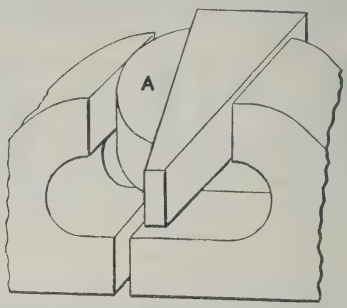


FIG. 78.

width as the jaws, the convex side being made semi-circular. It is preferable that the flat side of the block be checked or grooved in order to give it a better grip on the work.

(*G. A. L.*, '*Machinery*,' *June* 5, 1924.)

Pipe Bending.

Figs. 79 and 80 show two inexpensive methods in common use by practical men doing pipe bending work.

The first method, fig. 79, is to use a piece of hard wood with one end rounded, using a radius equal to half the width, then groove this rounded end to a depth equal to half the outside diameter

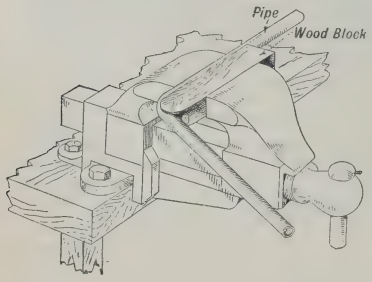


FIG. 79.

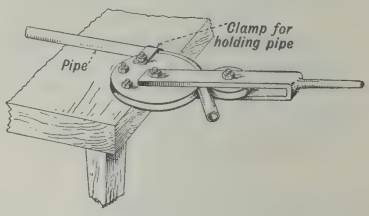


FIG. 80.

of the tubing. Clamp the block and the tubing in an ordinary bench vice as shown, and then pull the tubing around the block to make the desired bend.

In the second method, fig. 80, two iron grooved wheels of any desired radius are used ; the grooves on the outside diameter of the wheels should be made on a radius to fit the tubing to be bent. Mount these two wheels between two pieces of flat iron or wood fashioned at one end to form a handle. One wheel is attached to a bench as shown, or held in a vice.

Either of the two figures shows implements that do not cost much to make and are used extensively by practical men working tubing or pipe. Tubing or pipe bent this way does not require filling with sand, rosin, or any other material ; neither is it necessary to apply heat.

(*B. M. Adams*, '*Power*,' *January* 3, 1928.)

Plate-Bending Tool.

This tool is useful for the purpose of bending plates to a right angle. Fig. 81 shows a $\frac{5}{8}$-in. thick plate bent by means of the arrangement.

For the particular plate being bent, the tool consisted of a link, as shown in the figure, measuring inside about 15 ins. long and 2 ins. wide, and made of $1\frac{1}{2}$-in. diameter iron. It was laid across the anvil of a steam hammer, and the plate, heated at the middle, laid on top of the tool. A

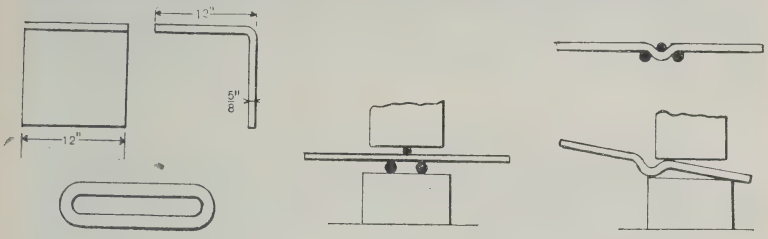


FIG. 81.

piece of $\frac{7}{8}$-ins. diameter iron, fully 12 ins. long, was laid across the top of the plate at the place where the bend was required, and directly above the inside space of the link underneath, as shown in section in the figure. One sharp blow of the steam hammer drove the piece of iron down and bent the plate; the plate then had three distinct bends. The tool was now removed and the two halves of the plate straightened, first one side and then the other, by a few blows, as shown in the figure, leaving a perfect right angle.

(*J. A.*, *Mechanical World*, *June* 15, 1928.)

Hand-Operated Tube Former.

The device shown in fig. 82 was recently made for use in forming a large number of small sheet-metal cylindrical tubes with lap joints. The tubes were for the purpose of inlet ventilators

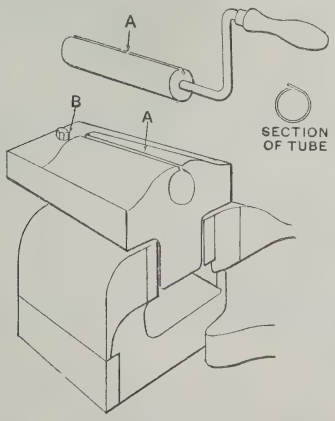


FIG. 82.

fixed in railroad passenger car floors, being used in connection with the steam heating apparatus. The stock used for the cylinders was No. 26 (0·0181-in. thick) lead-coated sheet steel, cut into pieces $2\frac{1}{4}$ by $4\frac{1}{4}$ ins.; the formed tubes being 3 ins. diameter by $4\frac{1}{4}$ ins. long.

The tube-forming device is clamped in a bench vice, and consists of a cast-iron body and a steel roller with a handle attached. The steel roller is $\frac{3}{8}$-in. in diamater, and has a narrow groove cut along one side, as shown at *A*. The metal part of the handle is oxy-acetylene welded to the roller and provided with an ordinary wooden handle. The stud *B* serves as a stop for locating the roller.

The operation of forming the sheet metal into a tube consists of placing the sheet in the narrow groove, holding it evenly in place with the left hand, and giving the handle a quick turn with the right hand; this forms the sheet metal around the tube, after which the roller is withdrawn and the tube slipped off. The tubes are formed with this simple device at the rate of from 600 to 700 pieces per hour. The lap joint is soldered, and a pressed washer is later secured to one end by soldering. The washer is used for securing the tube to the floor of the car with screws or nails.

(A.E., *Machinery*, *June* 28, 1928.)

Machining Aluminium.

In machining aluminium the tools should have more side and top rake than tools for cutting steel. The front clearance of the tool should be about 6°, the top rake from 30° to 50°, making the total angle of the cutting edge of the tool from 35° to 55°, and a side rake of from 10° to 20° aids the cutting action. The size of the angles used varies to some extent with the nature of the alloy and the work being done, the type of tool, and the conditions of use. It is very important to have keen tool edges, for on this factor the success of machining aluminium and its alloys largely depends. Hand stoning on a fine oil stone is recommended after a finish grinding in the preparation of edges.

Parting tools should have a top rake of from 12° to 20° and the front clearance angle should be decreased to about 3° or 4°. Milling cutters give the best results if they are of the coarse tooth spiral type and have a considerable amount of top rake on their cutting edges. Milling cutters with nicked teeth in some cases assist in decreasing the chip size. Face-milling cutters with inserted teeth, helical milling cutters and staggered tooth milling cutters can be made to work well with aluminium if the cutting edges are provided with suitable top rake. Also with considerable top and side rake good threads can be cut in aluminium of the softest quality in an engine lathe using a single-pointed threading tool. It has been found that very satisfactory taps for use in aluminium can be made by chasing threads on annealed high-carbon twist-drills followed by re-tempering and finishing.

Because of its lower modulus of elasticity aluminium has more recoverance than steel after a tapping operation, and the expedient of using slightly over-sized taps is therefore often resorted to in maintaining holes of required dimensions.

While the single-fluted twist drills used in drilling hard wood have been found in some cases to give better results than the usual form of drill, the best type for use with aluminium is that in which the flutes have a greater spiral angle, that is, more twists per inch; straight-fluted twist drills which are often used in drilling brass are quite unsuitable.

The most suitable saws for cutting aluminium are of the coarse tooth type, and hand hack-saw blades of the 'wavy-set' kind, work very satisfactorily.

Aluminium is machined to best advantage by using comparatively high speeds and fine to medium feeds. With ordinary carbon-steel tools speeds between 500 and 800 feet per minute are possible under some conditions and higher speeds are obtainable with high-speed steel tools. The feed may vary from $\frac{1}{4}$-inch for roughing cuts to a few thousandths for finishing cuts. Usually the finer the feed the higher the speed, and the more curl there is to the chip the slower the speed. When machining with coarse feeds it is necessary to remember that aluminium, like other metals, shows a rise in temperature and, moreover, its coefficient of expansion is greater than that of steel and, therefore, due dimensional allowances should be made.

A satisfactory lubricant or cutting compound for general use contains equal parts of kerosene and lard oil, while for heavy cuts and slow feeds, such as in roughing work or tapping, pure lard oil gives good results. In milling, sawing, and drilling, soluble cutting oils are satisfactory and more economical than kerosene and lard oil lubricant. Paraffin-base compounds such as those used in machining brass are quite unsatisfactory for use with aluminium. Alloys of the duralumin type, heat treated, possess the best machining properties, and aluminium-silicon alloys are the most difficult to machine.

(M.C., *Machinery*, Oct. 10, 1927.)

For grinding aluminium a soft or medium soft wheel should be used at speeds up to 10,000 feet per minute for carborundum wheels, and up to 6,000 feet per minute for wheels with shellac or other soft binders. Coarse wheels may be prevented from clogging by rubbing them with paraffin wax.

Single cut files should invariably be used for aluminium. Clogged files may be readily cleaned by dipping them in strong caustic soda solution, after which the metal particles are easily wiped off. This process should be followed by washing and drying in sawdust.

Machining Ebonite and Fibre.

Ebonite.

Ebonite should be run at the highest possible speed obtainable on the machine, and a mixture of 1 part of paraffin to 4 of lard oil used as coolant.

Turning tools, drills, reamers, and taps for ebonite should be of high-speed steel, glass hard, except where either diamond* or sapphire tools can be employed.

Drills should be ground away in the flute to allow of easy chip removal, and also to reduce the amount of surface of the tool in contact with the ebonite to a minimum. This principle of reducing contact to the lowest degree should be observed in all tools used on ebonite, since friction (together with the action of the sulphur) is a large factor in producing a temperature which quickly destroys the cutting edges.

Whenever possible, diamond tools* should be used for boring holes which are sufficiently large in diameter to admit a substantially mounted stone and possess a comparatively regular bore. For reaming, use plain flat reamers or bits, having two acute lips or cutting edges, and a land not exceeding $\frac{1}{64}$ in. in width or, as an alternative, a small bar with a bit or cutter of high-speed steel inserted in the end, this being simple and easily removed.

Tap and dies for ebonite should have narrow lands, and, wherever possible, should be run at a lower speed (if the machine is capable of it) than that employed for the rest of the operations on the component; 60 ft. per minute is a satisfactory speed, and there will be little danger of tearing-up the thread.

Feeds for turning, drilling, and reaming operations on ebonite should always be high, and about three times those used on brass. This remark does not apply so much to forming operations, when the feeds should be a little less than for brass on account of the fragile nature of the material.

Fibre.

Red and black fibre can be worked in much the same manner as ebonite as regards feeds, speeds, and the type of tools employed, but diamond tools are not so satisfactory as on ebonite on account of the irregular quality of these materials, for when the stone strikes a bad patch in the rod or tube there is a tendency to dislodge the diamond or sapphire. Drills, which should be of high-speed steel, should be fed at a higher rate than for ebonite, as should also reamers. The 1 to 4 solution of paraffin and lard oil will be found a good lubricant.

Erinoid, galalith, and other lactic base compounds are best machined when run at a high speed similar to that for ebonite, but, owing to the fact that these materials do not generate heat to the same degree as ebonite or fibre, they do not require any lubricant, and should be run dry. High-speed steel tools should be used, and wherever it is possible to employ a diamond tool* for either boring, turning, or forming, it will be found an advantage to do so. The same points in regard to the type of tools apply to erinoid and galalith as in the case of ebonite, whilst the feeds are approximately the same. (*'Machinery,' May* 24, 1923.)

Tool-Steel for Cutting Ebonite and Fibre.

High-speed steel frequently fails to give satisfactory results when machining ebonite and similar materials. Certain of the leading steel makers produce steels specially for work of this character. Edgar Allen & Co., Ltd., Sheffield, for example, make such a steel that requires to be quenched out from a temperature of 780° C., after which it is let down to a straw colour.

Saving Worn Parts by Electro-deposition.†

This process (developed at the Westinghouse Research Laboratory) has been used to build up shafts, pins, bolts, gear centres, and similar parts, and can be employed to build up the worn surfaces inside automobile engine cylinders, also for building up parts accidentally machined under size.

A plating bath is made up of 2·5 lbs. of commercial ferrous ammonium sulphate per gallon of water plus a small amount of ferrous carbonate (freshly precipitated and kept under water in order to keep the solution practically neutral) plus a small amount of powdered charcoal, which helps to prevent pitting. The equivalent amounts of commercial ferrous sulphate and ammonium sulphate may be substituted for the ferrous ammonium sulphate. If the solution is made up from the ferrous sulphate and ammonium sulphate, the granular or 'crystal meal' ferrous sulphate should be used. The ammonium sulphate must prove to be free from sulphocyanate when tested with ferric-sulphate or ferric-chloride.

All small pieces are plated in earthenware jars, and large pieces in lead-lined wooden tanks or stoneware tanks. Waterproof cement tanks can also be used. The anodes are made of

* See p. 1087.

† *Machinery*, Nov. 8, 1923 (*Abstract*).

'Armco' iron, cylindrical in shape. Micarta discs with a hole cut in the centre are fitted in each end of the anodes. The pieces to be plated are made the cathode and held stationary so that they extend into the plating solution through the holes in the micarta discs. The anodes are attached to a device which moves them up and down, thus keeping the plating solution stirred and the ferrous carbonate and charcoal in suspension. The temperature is held at approximately 60° C., a few degrees variation either way not doing any harm. A current density of 0·43 ampere per sq. in. is used. This deposits metal at a rate that increases the diameter 0·006 in. per hour.

This method of procedure applies only to pieces such as shafts, bolts, pins, etc. For pieces such as gear centres, automobile cylinders, etc., where the inside diameter needs to be reduced, the anode is made to go down through the centre, and micarta rings are fastened to the anode to obtain the required stirring. The current density could be raised considerably, but as the density specified adds the deposited material at the rate of 0·003 in. on a side, or increases the diameter at the rate of 0·006 in. per hour, it is evident that the ordinary repair job requiring an increase in diameter of approximately 0·010 in. to 0·020 in. can be quickly completed.

To clean the work, the piece to be plated is made the cathode in an alkaline solution consisting of 6 to 10 per cent. of caustic soda and 5 to 10 per cent. of commercial sodium carbonate, using a current density of 3 amperes per sq. in. for a period of three minutes. It is then washed in running water, transferred to a 30 per cent. commercial sulphuric acid solution, and given a high enough current density to make it passive for a period of three minutes or longer, according to the condition of the piece. If it is free from rust to begin with, three minutes is long enough, but if especially rusty it requires a longer time.

The piece being cleaned should come out of the acid cleaning solution bright and silvery, except in the case of cast iron, which will be black due to the graphite content. When taken from the acid cleaning solution, the piece is washed in running water and immediately transferred to the plating bath without giving it a chance to dry and form a thin film of rust. It is essential for the current density to be kept high enough (at least 3 amperes per sq. in.) in the acid solution so that the piece being cleaned is kept passive. When passive, there is a rapid evolution of gas, and the piece is bright and silvery in appearance when it is taken out, but if allowed to become active, it has a dull black appearance. When a piece is covered with oil or grease it is cleaned with petrol or benzol before being cleaned electrolytically.

If the method of cleaning and plating is properly followed, the deposited metal will be free from pits, bright and velvety in appearance, and will adhere perfectly to the parent metal.

As it is impossible to build up a piece accurately to size, it is necessary to build the work up a few thousandths over-size and then turn or grind it to the finish size. This method of recovery can be used with steel, cast iron, or case-hardened material, and the deposited material can be case-hardened when necessary.

If there are portions of the work that are not to be plated or built up, for protection these parts are coated with a mixture of 90 per cent. hydroline and 10 per cent. paraffin. The hydroline must have a high enough melting point so that it will not soften and run at 60° C. This protective coat is applied by melting the mixture and then dipping the piece to be coated, or by painting the mixture on with a brush. After the piece is plated, the protective coat can be readily removed by washing with benzol.

The following equipment would be required for a small plant: One 30-volt motor-generator set, 500 to 1,000 amperes capacity, according to capacity needed for cleaning; one 8 to 10-volt motor-generator set, 200 to 400 amperes capacity, or some other source of direct current such as storage batteries for plating; one or more ½-horse-power alternating-current motors for the stirring device; one or more reduction gears for the stirring device; and the necessary ammeters, voltmeters, crocks, heating coils, rheostats, switches, wire, tools, etc.

Metal Spraying.*

The unit of all the metal-spraying processes is the wire spraying pistol, a small and handy apparatus (weighing about 3½ lbs.), similar in appearance to a paint pistol, but differing in its operation and design. The wire (of the metal to be sprayed) (about 1 mm. in diameter) is pulled into the pistol and pushed through the nozzle (at a rate of about 12 ft. per min.) by means of rollers actuated by a small air turbine in the case. The pistol is fed with oxygen and some combustible gas under pressure, and also with compressed air, which projects minute particles of metal from the ever-melting end of the wire, the spray covering a surface about 2 ins. in diameter at a distance of 5 ins. from the nozzle.

The metal-spraying pistol will spray all metals (magnesium excepted) which can be obtained in wire form and which will fuse in the oxy-hydrogen flame.

* Abstract of paper on 'Metal Spraying and Sprayed Metal,' by T. Henry Turner and W. E. Ballard, read before the Institute of Metals, Sept. 1924.

For small articles, such as may be treated in rumbling barrels, a mechanical device has been introduced. Similarly, a machine is being operated which uniformly sprays with lead the outside of long metal pipes for use in the chemical industry. The interior of tubes may be sprayed by attaching to the ordinary pistol a long adapter, at the end of which a rotating nozzle is fixed. This produces a fan-shaped spray, and is driven by the air-blast used for projecting the metal. Thus, with the same unit, or pistol, tubes up to 12 ft. or so in length may be coated internally, so long as they are of sufficiently large bore to allow the introduction of the rotating end—that is to say, 1½ in. or more in internal diameter.

The preparatory sand-blasting of tubes internally presents some difficulty. The present-day practice is to sand-blast almost every article which is to be metal sprayed, and it is strongly advocated that where possible the article to be sprayed be heated. Devices are in use with which heating, sand-blasting, and spraying are carried out mechanically in one and the same machine, and the production of other such special machines is but a matter of time and of demand. For metal work sand-blast pressures of about 30 lbs. per sq. in. and hard black flint sands of about 20-mesh grade are recommended. After sand-blasting the surface should be sprayed at once, to avoid the possibility of contamination with grease and dirt.

Sprayed metal is being used successfully as a coating on ships' propellers. Composite coatings of more than one metal have given high resistance to corrosion. For example, zinc followed by aluminium is very suitable in cases of attack by atmospheres containing SO_2. It has been shown that the successful composite coatings are those of metals nearly related in the electro-chemical series. Very good results have been obtained by spraying zinc upon iron and steel objects—this has the advantage over practically all other methods of corrosion resistance, that it can be applied when and where desired over the top of rivets, cold-worked portions, and in general upon part or all of articles which are fully built up.

Building up Solid Articles by Spraying.

In a steel mould copper, zinc, tin, aluminium, and iron bars have been produced by spraying longitudinally for one set of tests, and transversely for a second set of tests. It was found that the bars so produced could all be sawn, filed, planed, turned, and machined much in the same way as solid metal. The bars had a clear metallic ring when struck.

Tested by scratch or by scleroscope, the sprayed metal is decidedly harder than similar metal when cast. However, when tested by the ball indentation method there seems little difference between the two, although the sprayed metal is in some cases slightly softer than the cast.

The oxygen consumption with hydrogen as the combustible gas (in cu. ft. per lb. of metal sprayed) is as follows:—lead, 2·64; tin, 2·86; zinc, 5·32; copper, 10·82; nickel, 14·6; iron, 17·4; aluminium, 22·7.

The approximate consumption of oxygen with other gases is given by multiplying the foregoing quantities by 1·7 for coal-gas, and 0·8 for acetylene.

The amount of 'metal (in lbs.) sprayed per hour,' using hydrogen, is as follows:—lead, 16·0; tin, 7·5; zinc, 5·5; aluminium, 1·5; copper, 1·25; bronze, 1·25; brass, 1·375. The metal 'deposited' is rather less than this.

The air pressure used in England is usually about 45 lbs. per sq. in., whereas on the Continent it is usual to employ a pressure of 35 lbs. The higher pressure used in England has been found to be an undoubted advantage when spraying lead, tin, and zinc, but it would be better if the air pressure were varied with the metals being sprayed. The higher pressure of 45 lbs. per sq. in. should be used with tin, and the lower pressure of 35 lbs. per sq. in. used when such metals as copper are sprayed.

The process is at the present time used in France and Germany to a considerable extent on shipping work, and for the protection of overhead transmission line poles and wireless masts for big stations. It is being increasingly used in this country for the protection of certain portions of ships and for propellers. Both wooden and steel bulkheads are being coated with zinc in many shipyards. It has also been used very successfully, in Germany, to replace the usual copper cable rail connectors on electric railways, so that at present on the Berlin and Hamburg underground railways there are miles of rail without a copper connector. During the war, when copper was scarce in Germany, the copper cables were removed from the electric railways in Berlin and electric connection between the two rails secured by sand-blasting *in situ* the rail ends and the inside surfaces of the fish-plates. These surfaces were then immediately sprayed with zinc, and when bolted together again gave excellent electrical contact, and apparently they have been satisfactory ever since. The electrical resistance of this type of joint has been found to be even less than with the best copper cable joints. It has become the practice of the Siemens-Schuckert works to install this type of joint in place of the customary one.

See also 'Metal Spraying,' by T. H. Turner and N. F. Budgen. 15*s*. Charles Griffin & Co., Ltd., London.

Hardness of Materials.

Different Methods of Determining Hardness.

The test to which a material is generally subjected in order to obtain a measure of its hardness may be of two classes, viz. :—

(1) The resistance offered by the material under investigation, (2) the resistance offered by the material to abrasion or cutting.

Of these, the first finds most extensive favour in practice, and the indenting medium may be a chisel, cone, or sphere. Repeated use on hard materials is inclined to blunt the chisel edge or point of the cone, and the sphere is the agent commonly adopted.

In 1900 Brinell introduced a method of ascertaining the hardness of a material by indenting it with a hardened steel sphere subjected to a predetermined load. The ratio of the load on the sphere to the area of indentation produced gives the hardness figure. The ball commonly used is 10 mm. diameter. The load for the softer materials is usually 500 kilograms, and that for the harder metals 3,000 kilograms.

The load should be applied for at least 30 seconds in order to secure equilibrium between it and the resistance of the material. The diameter of indentation is observed by means of a special microscope fitted with a scale, and from this the depth of indentation can be calculated since :—

$$\triangle = \tfrac{1}{2}\,(D - \sqrt{D^2 - d^2}),$$

where,

$\triangle$ = depth of indentation; d = diameter of indentation; D = diameter of sphere.

The hardness figure or number N for a load P is given by the expression:—

$$N = \frac{P}{\frac{\pi}{2} D \left(D - \sqrt{D^2 - d^2}\right)} = \frac{P}{\pi D \triangle}.$$

The hardness number is therefore inversely proportional to the diameter of ball and depth of indentation, and for a ball of constant diameter is directly proportional to the load. The hardness numbers for diameters of impressions made by a ball 10 mm. diameter under a load of 3,000 kilograms are given in the table, page 1223.

The depth of indentation may be obtained directly by means of a micrometer microscope by observing the difference between the readings of the datum line on the piston when the parts are brought together and after the load has been applied and removed. This procedure precludes the possibility of including the compression of the piston, sphere, and other parts of the machine beyond the datum line.

As it is difficult to bring the parts together without indenting the specimen to some extent, a small initial load may be applied. Observations of the datum line are then made when the initial load is applied, and also after the full load has been applied, and when the load has been brought back again to its initial value.

By measuring the actual depth of penetration, the error due to the elastic deformation of the sphere (which is included when the diameter is observed) is avoided. This elastic deformation is relatively greatest when the load, and consequently the indentation, is least. As the indentation increases so does the lateral component of the resisting force, and this tends to keep the sphere to its true form.

In testing some cast irons the metal flows from beneath the ball and causes a ridge to be formed which stands proud of the original surface, whilst with bronze and some of the copper alloys the metal is drawn in. In the case of the former the diameter observed is greater than the true value corresponding to the depth of indentation, whilst in the latter case the diameter is less than its true value.

The Shore Scleroscope.

This instrument consists of a small hammer, weighing about $\frac{1}{\sqrt{5}}$ of an ounce and fitted with a diamond which is somewhat spherical in shape at the striking end. The hammer is suspended on a stirrup piece fitted to a head on a vertical stand and upon which is also mounted a glass tube about 10 inches long and a graduated scale subdivided into 140 parts. The hammer is released and raised by a pneumatic device and moves within the tube. As the end of the glass tube is made to rest upon the material under investigation, the hammer always falls through a definite height and the measure of hardness is obtained by observing its rebound on the scale. The hammer on striking makes a permanent impression in the material, but as this is very small and irregular in outline it is impossible to measure the depth or diameter with any approach to accuracy.

Abrasion Test.

An early conception was to scratch a smooth surface of the material under investigation with a sharp point of a mineral, a number of minerals being selected to constitute a scale of hardness. In their order of hardness these were :—(1) talc, (2) gypsum, (3) calcite, (4) fluor, (5) apatite, (6) felspar, (7) quartz, (8) topaz, (9) sapphire, (10) diamond. The method is only suitable for materials which differ widely in hardness.

The Sclerometer.

This instrument was developed by Prof. Turner to obtain a measure of hardness of materials which differ but slightly in value. It consists of a diamond carried at one end of a balance lever in a horizontal plane and which rests upon a knife edge fitted into a vertical pillar. The pillar is free to turn by hand, so that the diamond is made to travel across the smooth face of the metal being investigated. The end of the lever carrying the diamond is loaded, and the weight in grams necessary to produce a scratch of standard depth gives a measure of the hardness.

With apparatus similar to the above, some have taken the width of the scratch produced by a diamond under a constant load as the measure of hardness. As the scratch is very minute, it has to be measured by means of a microscope, and the surface must be smooth and polished in order to give a sharp definition. The test is therefore more adaptable to the laboratory than to the workshop.

Drilling Test.

This test can be made with an ordinary sensitive drilling machine having a vertically free spindle. A constant load is applied to the spindle, so that the drill is made to penetrate the material under observation. The revolutions made in drilling a hole of predetermined depth gives a measure of the hardness or tenacity.

With the object of observing the variation in the hardness as the drill proceeds, the machine is fitted with a drum which rotates with the descent of the spindle. A screw, attached to the drum support, carries a nut in which is fixed a pencil. The screw is driven from the spindle, and as the nut cannot rotate the pencil moves in the plane parallel to the drum axis. The combined motion causes a diagonal line to be traced on a piece of paper around the drum, the height and horizontal length of the line being proportional to the depth drilled and revolutions respectively. The slope of the line varies with the hardness, and a soft or hard patch in the material under test is shown on the graph by the line becoming steeper or flatter in the order mentioned at that part. The drill is periodically tested for sharpness on a piece of material of standard hardness. (*Professor Dempster Smith.*)

BRINELL HARDNESS NUMBERS (for load (P) of 3,000 Kg.).

DIAMETER OF BALL (D) = 10 mm; $H = \frac{P}{D^2}\left(\frac{2/\pi}{1-\sqrt{1-(d/D)^2}}\right)$

Diam. of Impression. mm. d	Hardness Number. H	Diam. of Impression. mm. d	Hardness Number. H	Diam. of Impression. mm. d	Hardness Number. H	Diam. of Impression. mm. d	Hardness Number. H
mm.		mm.		mm.		mm.	
2·0	946	3·25	351	4·5	179	5·75	105
2·05	898	3·3	340	4·55	174	5·8	103
2·1	857	3·35	332	4·6	170	5·85	101
2·15	817	3·4	321	4·65	166	5·9	99
2·2	782	3·45	311	4·7	163	5·95	97
2·25	744	3·5	302	4·75	159	6·0	95
2·3	713	3·55	293	4·8	156	6·05	94
2·35	683	3·6	286	4·85	153	6·1	92
2·4	652	3·65	277	4·9	149	6·15	90
2·45	627	3·7	269	4·95	146	6·2	89
2·5	600	3·75	262	5·0	143	6·25	87
2·55	578	3·8	255	5·05	140	6·3	86
2·6	555	3·85	248	5·1	137	6·35	84
2·65	532	3·9	241	5·15	134	6·4	82
2·7	512	3·95	235	5·2	131	6·45	81
2·75	495	4·0	228	5·25	128	6·5	80
2·8	477	4·05	223	5·3	126	6·55	79
2·85	460	4·1	217	5·35	124	6·6	77
2·9	444	4·15	212	5·4	121	6·65	76
2·95	430	4·2	207	5·45	118	6·7	74
3·0	418	4·25	202	5·5	116	6·75	73
3·05	402	4·3	196	5·55	114	6·8	71·5
3·1	387	4·35	192	5·6	112	6·85	70
3·15	375	4·4	187	5·65	109	6·9	69
3·2	364	4·45	183	5·7	107	6·95	68

For other test loads the hardness numbers are proportional to those in the table.

British Standard Tables of Brinell Hardness Numbers.*

(No. 240—1926.) ***(Abstract.)***

Definition.

1. The Brinell hardness number is the quotient of the applied load divided by the spherical area of the impression. The Brinell hardness number is given by the following formula :—

$$H = \frac{P}{\frac{\pi D}{2}\left(D - \sqrt{D^2 - d^2}\right)} = \left(\frac{P}{D^2}\right)\left(\frac{2\pi}{1 - \sqrt{1 - (d/D)^2}}\right)$$

where,

P = Load (in kilogrammes); d = Diameter of impression (in millimetres);
D = Diameter of ball (in millimetres); H = Brinell hardness number.

Note.—The spherical area must be calculated from the average diameter of the impression obtained by taking two readings at right angles, and not from the depth of the impression.

Apparatus.

2. (*a*) The micrometer microscope or other measuring device used shall be capable of measuring the diameter of the impression to $\pm$ 0·5 per cent.

Note.—An accuracy of measurement of $\pm$ 0·05 mm. (0·002 in.) may be accepted for impressions made with a ball of 10 mm. diameter.

(*b*) The balls used in Brinell hardness testing shall be of hardened steel or of some harder material.

Note.—When ordinary hardened commercial steel balls are used, Brinell hardness numbers exceeding 500 may be subject to error owing to permanent distortion of the ball. The error increases with increasing hardness of the material tested. With hardness numbers above 500 care should be taken to see that the balls are considerably harder than the material to be tested.

(*c*) The standard balls and loads used for Brinell hardness testing shall be as follows :—

Diameter of Ball.	Load.			
	$\frac{P}{D^2} = 1$	$\frac{P}{D^2} = 5$	$\frac{P}{D^2} = 10$	$\frac{P}{D^2} = 30$
mm.	Kg.	Kg.	Kg.	Kg.
1	1	5	10	30
2	4	20	40	120
5	25	125	250	750
10	100	500	1000	3000

The limit on the diameter of the balls shall be $\pm$ 0·0025 mm. (0·0001 in.).

Note —The same Brinell hardness number is given by tests on the same uniform material with balls of different diameters when the same value of the ratio $\frac{P}{D^2}$ is used.

Test Specimens.

3. (*a*) The centre of the impression shall be not less than two and a half times the diameter of the impression from any edge of the test specimen.

(*b*) The thickness of the test specimen shall be at least seven times the depth (t) of the impression as given by the formula $t = \frac{P}{\pi D H}$ or, alternatively, shall be such that no bulge or other marking showing the effect of the load appears on the side of the piece opposite the impression.

When tests are made on thin specimens, care should be taken to ensure that the under surface of the specimen is smooth and in good contact with a smooth supporting surface of hardened steel.

(*c*) The surface on which the impression is to be made shall be polished if the diameter of ball used is 1 mm. or 2 mm. If the diameter of ball used is 5 mm. or 10 mm., it is still advantageous to use a polished surface, but, if a high order of accuracy is not required, the surface may alternatively be filed, ground, or smoothly machined.

* By permission of the British Engineering Standards Association.

4. (*a*) The value of the ratio $\frac{P}{D^2}$ shall be as follows:—

For steel and materials of similar hardness	30
For copper alloys and materials of similar hardness	10
For copper	5
For lead and tin and materials of similar hardness	1

It is advisable that the ratio of the diameter of the impression to the diameter of the ball shall not exceed 0·6.

(*b*) The load shall be applied slowly and progressively to the test specimen.

(*c*) The full load shall be maintained for at least 15 seconds when the ratio $\frac{P}{D^2} = 30$ and for 30 seconds when the ratio $\frac{P}{D^2} = 10$, 5 or 1.

Note.—For steel, experiment has shown that an approximate indication of the tensile strength in tons per sq. in. can be obtained by multiplying the Brinell hardness numbers by 0·22. In cold-worked material and in exceptional cases the ratio may differ widely from this figure.

The Specification gives twelve tables with diameters (D) of balls varying from 10 mm. to 2 mm. and loads (P) varying from 1000 kg. to 4 kg.

Balls for Brinell Tests.

At a meeting (1927) of the American Society for Steel Treating, a paper was read on the use of an iron-carbon-vanadium alloy for balls for making Brinell tests. When Brinelling steels of such hardness as causes ordinary Brinell balls to deform, the alloy recommended for use in testing consists of 2·9 per cent. carbon and 13 per cent. vanadium, the remainder being iron. Balls made from this material, and heat-treated, are found, when tested on steels of approximately 700 Brinell hardness, to flatten only one-fifth as much as the ordinary balls used and only one-half as much as certain special balls.

The Herbert Pendulum Hardness Tester.

This instrument consists of an arched weight of 4 kg. pivoted on a fixed ball of 1 mm. diameter, the ball being of hardened steel, ruby, or diamond. The position of the centre of gravity of the instrument can be varied, but for all standard tests the centre of gravity is 0·1 mm. below the centre of the ball, and the instrument constitutes a very short compound pendulum. A curved spirit tube with bubble and scale graduated from 0 to 100 serves to measure the amplitude of the oscillations. On rocking the instrument the specimen is slightly indented, but the degree of hardness is not measured by the indentation but by the swing of the instrument; the readings are, therefore, expeditiously made.

Time Test.

This is the number of seconds taken by 10 single swings of the pendulum when placed gently on the specimen and caused to oscillate through a very small arc. The 'time hardness numbers' (seconds for 10 swings) vary from 3 seconds on lead to 20 seconds on mild steel, 85 on very hard steel, and 100 on glass.

The pendulum time test, like the Brinell test, measures 'indentation hardness' or resistance to indentation.

Work-hardening Test.

Another test is that which measures the 'work-hardening capacity' of metals, or the increase of hardness which can be induced in them by working them cold. In making this test the Pendulum is placed on the specimen in a tilted position with the bubble at 0 on the scale and is then released. The pendulum swings through a certain arc which depends upon the hardness of the metal, and the position of the bubble on the scale at the end of the first swing measures the 'work hardness' or resistance to working by rolling.

The next operation is to tilt the pendulum and complete the swing by hand. The bubble is then at 100 on the scale and is again released. The effect of tilting the pendulum is to elongate the impression and to harden the rolled surface by cold work. When the pendulum is released the ball rolls back along the hardened surface, and the position of the bubble on the scale at the end of the swing measures the hardness of the work-hardened surface. This process is repeated, the pendulum being tilted alternately to right and to left, and the hardness measured after each successive rolling by the ball until a maximum is reached. The difference between the first reading and the maximum reading measures the work-hardening capacity of the metal. Glass

has zero work-hardening capacity. The first reading and the subsequent readings are the same. Manganese steel is shown to be soft by the Brinell test, soft by the pendulum time test, soft by the first reading of the work-hardening test, but after being rolled by the ball it immediately becomes as hard as hardened tool steel, which explains why it cannot be worked.

The time test and the work-hardening test have important applications owing to the ease with which they can be applied to heated specimens. The pendulum equipped with a diamond ball measures the hardness of hot specimens as easily as cold, and the tests are conveniently made while the specimen is actually in the electric furnace.

SECTION XXI

PART II

GRINDING, ABRASIVES, AND POLISHING APPLIANCES.

ABRASIVE MATERIALS — NATURAL GRINDSTONES — ABRASIVE WHEELS AND BLOCKS — LUBRICATION — DRY VERSUS WET GRINDING — SELECTION OF WHEELS — SPEEDS OF WHEELS — MOUNTING ABRASIVE WHEELS — GRINDING MACHINES — WHEEL GUARDS — GRINDING PROCEDURE — CYLINDRICAL GRINDING — EXTERNAL GRINDING — CENTRELESS GRINDING — INTERNAL GRINDING — SURFACE GRINDING — GRINDING ALLOWANCES — POLISHING — SAND BLASTING.

Revised and amplified by Alfred B. Searle
(Consultant and Advisor on Abrasive Materials)

GRINDING, ABRASIVES, AND POLISHING APPLIANCES.

The appliances used for grinding and polishing may be divided into two chief groups :—

(*a*) The *abrasive materials*, which may be in the form of powders, liquids, wheels, blocks, cups, etc.

(*b*) The *machines* used in the application of such materials, and commonly known as ' Grinding and Polishing Machines.'

Abrasive Materials.*

The chief *natural* abrasive and polishing materials are sandstone, sand, emery, corundum, diamond dust, pumice, rotten stone, tripoli powder, rouge, lime, whiting, and chalk. The chief *artificial* abrasives are : (i) carbon silicide, which is known by various trade names, such as carborundum ; and (ii) fused or crystalline alumina or artificial corundum, which is also known under a variety of trade names.

These materials are ground and then sifted so as to ensure the particles being of sizes suitable to the particular purpose for which they are to be used ; this being a matter of great importance.

* For further details see *Abrasives*, by A. B. Searle (Pitman).

These abrasive materials may be used in the form of powders or pastes, or suspended in water, oil, or other convenient liquids (as liquid polishes), or they may be made into grindstones, wheels, blocks, etc.

Natural Grindstones.

Natural grindstones are made by cutting blocks of suitable stone to the desired size and shape. The stones used for this purpose are shown by the following tables :—

FOR GRINDING MACHINISTS' TOOLS.

Name of Stone.	Kind of Grit.	Texture of Stone.	Colour of Stone.
Nova Scotia	All kinds, from finest to coarsest	All kinds, from hardest to softest	Blue, or yellowish-grey
Bay Chaleur (New Brunswick)	Medium to finest	Soft and sharp	Uniformly light blue
Liverpool or Meling	Medium to fine	Soft, with sharp grit	Reddish
Yorkshire and Derbyshire grit stone	Medium to fine	Hard to soft, as selected	Biscuit to grey

FOR WOOD-WORKING TOOLS.

Name of Stone.	Kind of Grit.	Texture of Stone.	Colour of Stone.
Wickerolz	Medium to fine	Very soft	Greyish-yellow
Liverpool or Meling	Medium to fine	Soft, with sharp grit	Reddish
Bay Chaleur (New Brunswick)	Medium to finest	Soft and sharp	Uniformly light blue
Huron, Michigan	Fine	Soft and sharp	Uniformly light blue

FOR GRINDING BROAD SURFACES, AS SAWS OR IRON PLATES.

Name of Stone.	Kind of Grit.	Texture of Stone.	Colour of Stone.
Newcastle	Coarse to medium	Hard	Yellow
Independence	Coarse	Hard to medium	Greyish white
Massillon	Coarse	Hard to medium	Yellowish white

FOR PULPING WOOD IN PAPER MANUFACTURE.

Name of Stone.	Kind of Grit.	Texture of Stone.	Colour of Stone.
Grindleford Grit	Course	Medium	Biscuit

Maximum Speed of Natural Grindstones.

Except under expert advice, grindstones should not be run at a higher peripheral speed than 3,400 ft. per min. This corresponds to the following speeds of rotation :—

Diameter of Stone.	Revolutions per Minute.	Diameter of Stone.	Revolutions per Minute.	Diameter of Stone.	Revolutions per Minute.
Ft. Ins.		Ft. Ins.		Ft. Ins.	
8 0	135	6 0	180	4 0	270
7 6	144	5 6	196	3 6	308
7 0	154	5 0	216	3 0	360
6 6	166	4 6	240		

To True Grindstones.

Drive at a moderate speed and true up with a piece of iron tube. Keep turning the tube, which should be held at right angles, and turn it as the edge grinds away. The stone should be *dry*, not wet.

Abrasive Wheels and Blocks.

The particles of abrasive material are held together by clay or by a mixture of materials in which clay is an important ingredient, the wheels being moulded from a plastic state and afterwards burned in kilns at a temperature of about 1,400° C. In other abrasive wheels the binding materials are shellac, rubber, or silicate of soda.

Abrasive wheels, blocks, etc., are classified according to the nature of the abrasive (corundum, carborundum, emery, etc.), the size of the abrasive particles and the hardness or tenacity of the bond.

Much economy, speed, and quality can be obtained by using abrasive wheels in the place of files, especially when the wheels are used on the varied machines to which they are applied. The great profit to be derived from the general introduction of abrasive wheels is due to saving in wages, files, increased output, and improved work.

It should no longer be assumed that a grinding wheel cannot remove metal and shape work as quickly as a steel tool, but the latter should be used when it can do the work more cheaply, and the grinding wheel when it excels. Poor wheels, or good wheels wrongly used, will still remove metal slowly, but a modern grinding wheel used in a modern machine is a milling cutter just as truly as if made of steel, and is to be regarded as a cutter with millions of teeth, which, while they may not cut so deeply, can cut more rapidly—as, for example, at speeds of 6,000 or more surface feet per minute. Cams and crankshafts can be produced more cheaply by grinding than by turning. With automatic grinders it is possible to turn out work even more rapidly. Where 300 pieces per day were ground by piecework, 2,300 are now produced.

The *grade* of a wheel is represented by a letter, being the hardest and the softest. The term 'grade' is confined to the relative hardness of the wheel or block as a whole, and does not refer to the hardness of the abrasive particles. This distinction is important, because the cutting or abrasive power depends quite as much on the bonding material as on the abrasive particles.

No wholly satisfactory method of grading abrasive wheels, etc., has yet been devised, various mechanical devices of testing having proved unsatisfactory. The method adopted by manufacturers consists in gouging the wheel with a tool resembling a screw-driver, the 'feel' indicating the grade with fair accuracy to an experienced tester. It is, however, almost impossible to compare accurately the grades of abrasive wheels made with different bonds, as they do not work equally satisfactorily with any material.

The term '*grit*' is used to indicate the sizes of the abrasive particles present, and it is usually expressed by figures indicating the number of meshes per linear inch in a sieve through which the abrasive particles will pass completely. Thus, in a 60-grit wheel or block the abrasive particles will pass completely through a sieve having 60 holes per linear inch. It should be observed that the grit only indicates the largest particles, and has no reference to their average diameter nor to the proportion of much smaller particles present. It is understood, however, that there is only a small proportion of particles sufficiently fine to pass through a sieve which is represented by the next grit figure.

Grit Sizes of Grinding Wheels.

Class of Work.	Emery.	Alundum.	Corundum.	Carborundum.
Glass and marble polishing	150	200	240	—
Fine hardened steel work	40–80	46–100	50–120	60–150
Sharpening milling cutters	40–50	46–50	46–60	50–80
General machine shop use	20–24	20–30	20–36	24–40
Brass castings	16–20	20–30	24–36	30–40
Agricultural implements	14–20	16–24	16–30	16–36
Rough castings	10–16	12–16	12–20	14–36
Marble	—	—	—	2–4

(Engineering News.)

WHEELS FOR DIFFERENT MATERIALS AND OPERATIONS.

Material.	Operation.	Classification.
	External Work.	
Soft Steel	General	24 Comb. L.M. Alundum
,,	Fine finish	60 L. ,,
Hard Steel	General	60 J.K. ,,
,,	Fine finish	80 K. ,,
Cast Iron	General	24 K. Crystolon
,,	Chilled Rolls (Roughing)	36 K. ,,
,,	,, (Finishing)	60 J. ,,
	Internal Work.	
Soft Steel	—	46 K. Alundum
Hard Steel	—	60 J.K. ,,
,, Fine finish	Fine finish	80 K. ,,
Cast Iron	—	36 J. Crystolon
,,	—	38–46 J. Alundum

(*W. D. Adamson.*)

Material.	Wheel Maker.	Bond.	Classification.
Copper (Roughing)	Norton Co.	Elastic	36–50 Crystolon
,, (Finishing)	,,	Vitrified	80 L ,,
Aluminium	,,	Elastic	24–46 Soft to Med. Crystolon
,,	Vitrified Wheel Co..	Elastic	24–46 Medium Carbolon

The *suitability* of an abrasive wheel or block for any given purpose depends on the grade or hardness of the wheel as a whole, on the grit or size of the abrasive particles, and on the *bond* which unites them together.

For grinding steels and the harder materials, *i.e.* materials with a high tensile strength, wheels or blocks composed of an aluminous abrasive (corundum emery) are preferable. For cast iron, chilled iron, brass, soft bronzes, aluminium, copper, granite, marble, leather, wood, and other materials of lower tensile strength, wheels or blocks composed of a carbide abrasive (carborundum, crystolon, etc.) should be used. The binding agent which unites the particles of abrasive also determines the suitability of the wheel or block for any given purpose. Thus—

(i) Rubber-bonded abrasives are very tough and resistant, and so are specially suitable where a bond of exceptional strength is required in conjunction with a thin wheel or ring, as in the case with various cutting operations, such as slotting iron plates, sawing stone, cutting glass and saw teeth, or sharpening milling cutters of awkward shape, as well as for cutting metals, grinding chilled iron rolls, and for other work where a fine finish is required. Shellac-bonded wheels are too soft for this purpose. They are also used where it is necessary that the shape of the abrasive article should be retained as long as possible. This is particularly the case with the thin cutting discs used by watchmakers, dentists, etc.

(ii) Shellac-bonded wheels are used for cutter and reamer grinding, pointing drills, sharpening knives, saws, and other edge tools, and for similar purposes as rubber-bonded wheels, but as they are more elastic they can be used to give a finer finish. They will not stand high temperatures, but can be used for wet or dry grinding. They are less porous and less free cutting than wheels with a vitrified bond. They are singularly free from ' glazing,' as the baked shellac has lost the stickiness characteristic of the natural material.

(iii) Silicate wheels are very close in texture, and as they cut freely they are very useful for grinding tools and cutlery.

(iv) Abrasive articles made with a vitrified bond are characterised by a high porosity, a great resistance to heat and acids, and great freedom in cutting. They may be used for almost any purpose if care is taken to select them of suitable grit and grade, though when very thin cutting wheels are required it is often preferable to use a shellac—or rubber-bond may be

preferable. If too much bond is present the abrasive articles tend to 'glaze' badly, owing to the bond not falling away with sufficient rapidity from the cutting points. With too little bond, on the contrary, the articles will be weak and, in the case of wheels, cylinders, or rotating rings, they may be dangerous on account of their liability to break when in use.

Lubrication.

The rate, effect, and efficiency of the grinding are also affected by the water or other lubricant applied to the wheel when in use. The function of the water is to carry away the heat resulting from the friction between the abrasive and the article being ground, and to wash away any loose particles dislodged by the grinding.

The water supply for all external grinding operations should be ample in volume without force. Using, say, a 14 in. diameter wheel, not less than 5 gallons and up to 10 gallons per minute should be used; on larger wheels 20 to 50 gallons per minute.

For many purposes, solutions of soap or emulsions of oil and water are preferable to plain water. A solution of carbonate of soda containing 2 oz. per gallon has the advantage of not producing rust, especially if ¼ or ½ pint of lard oil is mixed with it.

In grinding aluminium it is advisable to use a lubricant of lard, oil, kerosene, or a mixture of kerosene and gasolene or lard, or oil and gasolene.

Dry *versus* Wet Grinding.

In certain circumstances water may—or must—be dispensed with. Whenever it is necessary to see the edge of the wheel distinctly, to be sure of reaching some particular spot, water cannot be used, as it obstructs the view. Work revolving in the chuck or on the faceplate is also generally ground dry, as water is awkward to deal with on ordinary machines under such conditions. Cutters are also usually ground dry, although special care has to be taken to prevent 'drawing the temper' of the cutting edges. The amount to be removed is, in most cases, small, so that the sacrifice of time is balanced by the greater convenience of dry grinding.

More material can be ground away by dry internal grinding than by wet, and it is fairly common practice to run water on the work externally to dissipate as much heat as possible whilst grinding dry internally. On such classes of work as aeroplane cylinder grinding, or ring gauges, where a high degree of accuracy and finish is essential, water must be used, but it should be in such volume as to prevent as far as possible the chips being carried round to the wheel and interfering with the cut.

In *wet grinding*, care should be taken that the water supply is steady and adequate, and particular attention must be paid to this in the case of tool grinders, where it is of the utmost importance that the tool should be effectively covered with water, to obviate any loss of temper in the steel during the process of grinding. Some grinding machines provide for this; *e.g.* in one such patent machine the water is forced by a centrifugal pump through a series of small holes in the tool rest itself in such a manner as to completely envelop the tool being ground. A noticeable feature of this grinder is that there is total absence of splashing.

Selection of an Abrasive Wheel

In the the most important points to be kept in mind are *suitability for the particular class of work*, and *safety from accident*.

It is obvious to everyone that a wheel specially manufactured for fettling iron castings is not the one to be used for needle pointing, but few people, except those connected with the trade, have any conception of the number of kinds or classes into which such wheels have been divided and subdivided, each main group (designated by symbols, B, S, SS, G, R, M, etc.) being varied in hundreds of ways to suit the work.

The general practice is to state the purpose for which the wheels are required and leave the selection to the judgment of the makers; it is to their interest to give satisfaction. The other particulars to be given when ordering are—quantity required, diameter, thickness, and size of centre hole.

Grindstones are softer when wet than dry, and should not be left partly immersed in water, or they will wear unevenly. Stones made with sorel cement (basic magnesium chloride) as a bond should not be used for wet grinding as the cement is disintegrated by water.

Speed of Wheels.*

This varies somewhat according to the make and the purpose for which they are used, but no abrasive wheels should be operated at a speed exceeding that recommended by the manufacturer thereof. The following speeds are equivalent to a peripheral speed of 5,000 ft. per minute :—

SPEED TABLE.

Diam. of Wheel in Ins.	Speed. Revs. per Min.	Diam. of Wheel in Ins.	Speed. Revs. per Min.	Diam. of Wheel in Ins.	Speed. Revs. per Min.
1	19,099	18	1,061	42	455
2	9,549	20	955	44	434
3	6,366	22	868	46	415
4	4,775	24	796	48	397
5	3,820	26	733	50	383
6	3,183	28	683	52	369
7	2,728	30	637	54	354
8	2,387	32	596	56	341
10	1,910	34	561	58	330
12	1,592	36	531	60	319
14	1,364	38	503		
16	1,194	40	478		

The peripheral speed of a grinding wheel—which is a function of its diameter and the number of revolutions per minute which the wheel makes—has a marked effect on the apparent hardness of the wheel. Hence, the hardness or grade of an abrasive wheel or cylinder appears to vary with the speed at which it rotates. Thus, by running a soft wheel above its normal speed it appears to become harder, and by reducing the speed of a hard wheel it appears to become softer. Hence, by varying the speed within reasonable limits a wheel which is normally too soft or too hard may be used satisfactorily.

As a wheel wears, its peripheral speed is reduced and the wheel appears to soften, but by increasing the number of revolutions per minute so that the original peripheral speed is obtained the wheel appears to regain its former hardness. Many workers suppose that the faster the work revolves the greater will be the output, and so they use very hard wheels which require maximum driving power and cause a maximum wear of the machine. Yet, in most cases, they produce work of inferior quality which requires a longer time and greater effort to produce than if softer wheels and slower work speeds were used.

The following speeds are typical, but should be modified to suit particular cases :—

Application.	*Peripheral Speed in Feet per Min.*
Cylindrical grinding	5,500 to 6,500
Snagging and general off-hand grinding on bench and floor stands	5,000 ,, 6,000
Surface grinding	4,000 ,, 5,000
Knife grinding	2,000 ,, 3,000
Hemming cylinders	2,100 ,, 2,400
Tool grinding	3,000 ,, 4,000†
Vertical surface grinding machines	4,000 ,, 4,500
Elastic and rubber cut-off wheels	9,000 ,, 12,000
Polishing wheels	about 7,000

* Some of the following tables and recommendations are taken from a booklet issued by the National 'Safety First' Association, 119 Victoria Street, London, S.W. 1.

† Some large users of grinding wheels regard as maximum peripheral speeds for machinist's tools 0–1,000 ft. per min., and for carpenter's tools 550–600 ft. per min.

For speeds higher than the above, slightly softer grades should be used to off-set the harder cutting action ; for lower speeds slightly harder grades than would be ordinarily supplied should be used.

It is impossible to state exactly the speed at which any article or part to be ground should rotate. It is largely a matter of experiment, as the speed should be suited to the abrasive and to the nature of the material to be ground. With Norton cylindrical grinders a peripheral work-speed of 60 to 80 ft. per minute is often used for roughing, and 30 to 40 ft. per minute for finishing. It is customary to rough-grind at a higher surface speed of work than on finish-grinding.

In grinding cylinders the speed at which the article to be ground rotates is of great importance, as by running the grinding wheels at a constant speed and varying the speed of rotation of the article to be ground it is easy to secure uniform results from wheels having a variation in speed of 2,000 ft. per minute, and a variation from true grade within practical wheel-making limits. Thus, by varying the speed of the article to be ground, the necessity for variable revolutions for the wheel, or close limits of speed and grade for the wheel, are made unnecessary, and variable speed for the work accomplishes all that variable wheel spindles can accomplish, and much more, in a more simple way, at less expense.

The following grinding-wheel speeds are recommended for different types of grinding, in a recent number of *Grits and Grinds*, published by the Norton Co. of America :—

Type of Grinding.	Recommended peripheral speed (feet per minute).	Maximum peripheral speed (feet per minute).
Cylindrical grinding	5,500	6,500
Internal grinding	5,000	6,000
Snagging and general off-hand grinding on bench and floor stands	5,000	6,000
Surface grinding	4,000	5,000
Knife grinding	3,500	4,000
Hemming cylinders	2,100	2,400
Wet tool grinding	5,000	6,000
Cutlery grinding	4,000	5,000
Shellac and rubber cutting-off wheels	9,000	12,000

Every care must be taken to guard against the possibility of wheels being run at too high a speed. Thus, in the case of variable speed motor drive the speed control should be enclosed in a locked case, and cone pulleys should not be employed for speed regulation unless belt locking devices are provided.

New wheels should be run at working speed for 1 minute before applying the wheel to the work, the operator standing to one side. A heavy cut should not be taken with a cold wheel, but the wheel should be allowed to warm up slowly.

Storage of Wheels.

Extreme care should be exercised in the storage of wheels. They should be stored in a dry place and supported on edge in racks, except straight-sided shellac and rubber-bonded wheels of $\frac{1}{4}$ in. or less in thickness, which should be laid flat on a straight surface to prevent warping.

Wheels used for wet grinding should not be allowed to stand partly immersed in water, as this would tend to throw the wheel badly out of balance.

Mounting Abrasive Wheels.

If there is a large area of contact the method of mounting and safeguarding abrasive wheels and the condition of the grinding machine are of great importance.

Size of Spindle.

The minimum sizes of spindles in inches (at that part where the wheel is located) for the various diameters and thicknesses of grinding wheels should not be less than the following table :—

THICKNESS OF WHEEL IN INCHES.

Diam. of Wheel.	¼	⅜	½	⅝	¾	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4	4½	5
Ins.																			
6	¼	⅜	½	⅝	¾	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4	4½	5
7	¼	⅜	½	⅝	¾	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4	4½	5
8	¼	⅜	½	⅝	¾	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4	4½	5
9	¼	⅜	½	⅝	¾	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4	4½	5
10	¼	⅜	½	⅝	¾	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4	4½	5
12	¼	⅜	½	⅝	¾	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4	4½	5
14	¼	⅜	½	⅝	¾	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4	4½	5
16	¼	⅜	½	⅝	¾	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4	4½	5
18	¼	⅜	½	⅝	¾	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4	4½	5
20	¼	⅜	½	⅝	¾	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4	4½	5
24	¼	⅜	½	⅝	¾	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4	4½	5
26	¼	⅜	½	⅝	¾	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4	4½	5
30	¼	⅜	½	⅝	¾	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4	4½	5
36	¼	⅜	½	⅝	¾	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4	4½	5

The spindle should be threaded left or right hand as required, so that the wheel clamping nuts will tend to tighten as the spindle revolves, and the threaded portion should be long enough to provide a bearing for the entire nuts when the wheel and flanges are in place.

The diameters of the spindle bearings for machines with overhanging wheels should not be less than the wheel-location diameters in the above table, and for single wheel machines (where the wheel is mounted between two bearings) should not be less than 80 per cent. of the diameters in the above table.

The wheels should be an easy fit on their spindles or locating spigots, and the soft metal bushing should not extend beyond the sides of the wheel.

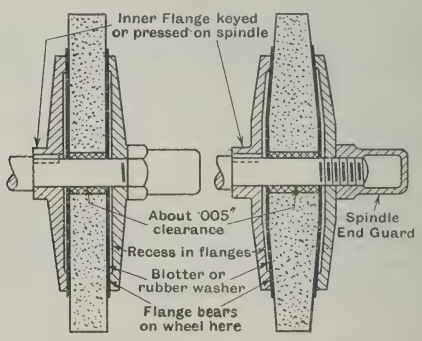


FIG. 1.

A grinding wheel should be fixed carefully and securely between recessed steel or iron plates or flanges about half the diameter of the wheel, with a washer of compressible material such as a sheet of thin rubber or one or two thicknesses of brown paper, on each side between the wheel and the flanges (fig. 1). These washers should be an easy fit on the spindle and not larger in diameter than the flange. If flanges with Babbit or lead facings are used, the thickness of the Babbit or lead should not exceed ⅛ in.

Spindle end-nuts should be tightened only sufficiently to hold the wheel firm, otherwise undue strain may be set up in the wheel.

Care should be taken in screwing up very thin wheels that too much force is not applied.

Flanges.

No abrasive wheel used on the periphery should be mounted except with suitable flanges. When wheel guards are fixed, straight, dovetail, or taper flanges should be employed. Where guards cannot be used, and also on swing frame and portable grinders, taper flanges should be used with taper wheels.

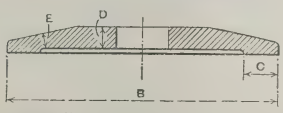
Flanges over 6 ins. diameter should be of steel of not less than 30 tons tensile. Smaller flanges may be made of cast iron. All flanges should be machined on the bearing surface and balanced. Every flange should be recessed at the centre at least $\frac{1}{16}$ in. on the side next to the wheel for a distance specified in the tables. The inner flanges should be keyed, shrunk, or pressed on to the spindle, and the bearing surface should run true with the spindle.

The minimum taper flange dimensions for taper wheels are given in the table, p. 1112. Such flanges should have a taper of $\frac{3}{4}$ in. per ft. No taper wheel over 24 ins. in diameter should be run with the wheel projecting from the flanges more than 4 ins. In the case of wheels 24 ins. diameter and under, the projection should not exceed 3 ins.

Dovetail flanges of the raised or countersunk type should be at least equal in diameter and thickness to the taper flanges. The taper provided for gripping the wheels should not be less than $\frac{1}{4}$ in. per ft. The recess should have ample clearance and be well radiused.

Cup, cylindrical, and sectional ring wheels should either be provided with guards or enclosed in protection chucks. Not more than a quarter of the height of the wheel should protrude beyond the guard or chuck. Where the thickness of the rim is less than 2 ins. the maximum distance that the wheel should project is 1 in. If the thickness of the rim is 2 ins. or more, a maximum projection of 2 ins. is allowable.

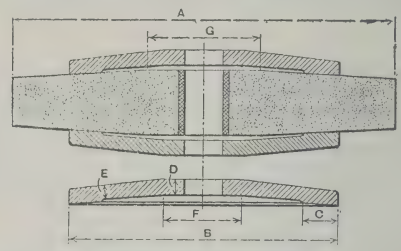
Straight Flange Dimensions.



(All Dimensions in inches.)

A. Diameter of Wheels.	B. Minimum Outside Diam. of Plates.	C. Radial Width of Bearing Surface. Minimum.	C. Radial Width of Bearing Surface. Maximum.	D. Minimum Thickness of Plate at Bore.	E. Minimum Thickness at Edge of Recess.
1	$\frac{1}{2}$	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{1}{16}$	$\frac{1}{16}$
2	1	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{3}{16}$
3	$1\frac{1}{2}$	$\frac{3}{16}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{4}$
4	2	$\frac{1}{4}$	$\frac{1}{2}$	$\frac{5}{16}$	$\frac{5}{16}$
6	3	$\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{3}{16}$
8	4	$\frac{5}{16}$	$\frac{5}{8}$	$\frac{3}{8}$	$\frac{5}{16}$
10	5	$\frac{1}{2}$	1	$\frac{7}{16}$	$\frac{5}{16}$
12	6	$\frac{1}{2}$	1	$\frac{7}{16}$	$\frac{3}{8}$
14	7	$\frac{5}{8}$	$1\frac{1}{4}$	$\frac{7}{16}$	$\frac{3}{8}$
16	8	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{7}{16}$	$\frac{7}{16}$
18	9	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{7}{16}$	$\frac{7}{16}$
20	10	$\frac{7}{8}$	$1\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$
22	11	$\frac{7}{8}$	$1\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$
24	12	1	2	$\frac{1}{2}$	$\frac{1}{2}$
26	13	$1\frac{1}{8}$	$2\frac{1}{4}$	$\frac{9}{16}$	$\frac{1}{2}$
28	14	$1\frac{1}{4}$	$2\frac{1}{2}$	1	$\frac{7}{16}$
30	15	$1\frac{1}{2}$	3	1	$\frac{7}{8}$
36	18	2	$3\frac{1}{2}$	$1\frac{1}{8}$	1
40	20	3	4	$1\frac{1}{4}$	$1\frac{1}{8}$

TAPER FLANGE DIMENSIONS.



(All Dimensions in Inches.)

A. Diam. of Wheels.	B. Min. Outside Diam. of Plates.	C. Radial Width of Bearing Surface. Min.	C. Radial Width of Bearing Surface. Max.	D. Min. Thickness of Plate at Bore. For Double Taper Wheel.	D. Min. Thickness of Plate at Bore. For Single Taper Wheel.	E. Min. Thickness at Edge of Recess. For Double Taper Wheel.	E. Min. Thickness at Edge of Recess. For Single Taper Wheel.	F. Max. Flat Spot at Centre of Plate.	G. Approx. Diam. of Flat at Centre of Wheel.
8	4	$\frac{5}{16}$	$\frac{5}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{4}$	0	1
12	6	$\frac{1}{2}$	1	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{16}$	$\frac{5}{16}$	4	$4\frac{1}{2}$
16	10	$\frac{3}{4}$	$1\frac{1}{2}$	$\frac{5}{8}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{7}{16}$	4	6
20	14	$1\frac{1}{4}$	$2\frac{1}{2}$	$\frac{3}{4}$	1	$\frac{1}{2}$	$\frac{9}{16}$	4	6
24	18	$1\frac{1}{2}$	3	$\frac{3}{4}$	$1\frac{1}{8}$	$\frac{9}{16}$	$\frac{5}{8}$	4	6
30	24	2	4	$\frac{7}{8}$	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	4	6
36	30	2	4	1	$1\frac{3}{8}$	$\frac{7}{8}$	1	4	6
42	36	3	6	$1\frac{1}{4}$	$1\frac{3}{4}$	1	$1\frac{1}{4}$		
50	42	3	6	$1\frac{1}{2}$	2	$1\frac{1}{4}$	$1\frac{1}{2}$		

Both flanges should be of equal diameter, to avoid cross-straining of the wheel. The inner flange should be keyed to the shaft and abutting against a shoulder on the same. Various firms have special mountings which have proved satisfactory.

Wheel Guards.

All abrasive wheels should have a guard or protection hood made of rolled steel. The hood should cover as much of the wheel as possible and also the spindle rod. The hood should be constructed to act as a dust hood which, when connected with an exhaust system, will eliminate the injurious effects of wheel dust.

To meet the requirements of factory inspectors, a number of wheel guards, good, bad, and indifferent, have been introduced by manufacturers. The features to be looked for in an appliance of this kind are:

(1) That it is of ample strength; (2) that it can be readily adjusted to follow the wear of the wheel. (3) The maximum angular exposure for guards used on bench and floor grinders should not exceed 90°, the exposure beginning at a point not more than 65° above the horizontal plane of the wheel-spindle. When the work demands contact with the wheel below the horizontal plane of the spindle the exposure should begin not more than 65° above, and extend not more than 60° below, the horizontal spindle plane. (4) On cylindrical grinders the maximum wheel exposure should not exceed 180°, beginning at a point not more than 65° above the horizontal spindle plane. (5) The maximum wheel exposure on surface grinders should not exceed 150°, and on swing frame and portable grinders 180°, the top half of the wheel being protected at all times. (6) Guards should be so constructed that the peripheral member may be adjusted to preserve the maximum wheel exposure. The distance between the wheel and the peripheral member should not exceed $\frac{1}{4}$ in. The connection between the peripheral and side members or between the parts of a section guard should have a strength in a radial direction at least equal to the strength of the material of which the guards are constructed.

It is essential that the guard should be kept close to the periphery of the wheel in order to limit the flight of any portion which may break away.

Face or Cup Wheels ought to be protected by being well and properly secured inside a Safety Adjustable Chuck, designed to allow only a small portion of the wheel to project whilst at work, and constructed so as to secure and support the wheel as close to the grinding face as possible.

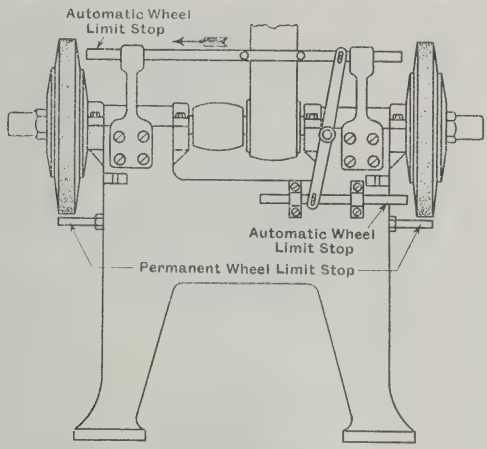


FIG. 2.

Machines fitted with two or more speeds should be provided with safety devices to prevent the maximum speed of the wheel being exceeded. In fig. 2 is shown a two-speed machine fitted with stops to limit the diameter and to prevent the belt being shifted to the high-speed pulley when the wheel exceeds a predetermined diameter.

Wheel Dressers.

Before using, the wheels should be trued up on their own spindles with a truing tool, and they should also be turned up when they get out of truth through wear.

Wheel dressers are used for 'truing' worn wheels; they should be equipped with rigid guards of sheet or cast metal over the tops of the cutters, to prevent flying pieces of broken cutters or wheel-particles from injuring the operator.

Dust Collection.

Dust generated by grinding should be suppressed or removed; preferably by localised exhaust ventilation. The hoods may be combined with the wheel guards or quite separate. They should be adjustable and should enclose the wheels as far as practicable. Hoods and ducts should be of substantial construction.

Except for very light dusts a pressure type of fan is a necessity, and special attention should be given to (*a*) the fan capacity; (*b*) the correct proportions and shapes of the ducts and branches; and (*c*) the means of collecting the dust after it has passed the fan. The efficiency of an exhaust installation is greatly increased by separating by gravity the heavy dust, burrs, fins, and miscellaneous material from the light dust, so that only the latter is directly dealt with by the fan. For this purpose some form of dust-separator (usually a cone) should be inserted in the duct between the hoods and the fan.

Rests.

Grinding wheels should revolve towards the rests, and the rests should be moved close to the wheels so as to prevent work being jammed between the rest and the wheel. The rest must never be moved while the wheel is in motion.

GRINDING MACHINES.

The modern grinding machine is practically a finishing machine in which the work produced needs no further correction. The sliding surfaces of the machine must be perfectly true planes, and there must not be rigidity, particularly in the grinding-wheel head. It must be heavy enough

to carry the largest and broadest wheels with which the machine may be possibly fitted, and also to overcome any possible want of balance in the wheel itself. The spindle and its bearings should be of ample proportions. The machine should be easily operated and controlled in its movements, so that a maximum amount of work can be obtained from it.

It is imperative that grinding machines be specially designed for the purpose intended, made of the best quality and workmanship, and that great care be taken in securing a suitable foundation as free from jar and shake as possible. The stands or bodies of the machines ought to be of a sufficient strength and massiveness to correspond with the size of wheel used. The same remark applies to the spindle, which must run in long and accurately bored bearings.

If the spindle is loose and cannot be put in good condition, a harder grade of wheel must be used than would ordinarily be recommended, so as to overcome the tendency to pound the wheel face to pieces. Light, flimsy machines, and those improperly secured to the foundation, also require harder grades than would ordinarily be used. Machines are frequently placed in the middle of a wooden floor which vibrates badly, and in this case harder wheels must be used than for a machine on a firm, solid foundation. A machine which is too light or imperfectly made is always a nuisance and is liable to prove very expensive. It is only too easy for good work to be spoiled by such machines, and they add unnecessarily to the cost of grinding, because they cannot be driven at the most appropriate speed for the work and because of various 'dodges' which must be used to obtain satisfactory results.

Rotary abrasive appliances may be mounted in a fixed frame—in which case the article to b ground, cut, or polished is moved gradually forward—or the work may be fixed and the abrasive mounted on a moving frame, so that it travels both forward and from side to side. There is an increasing feeling that this principle is wrong even in light machines ; it is impracticable in heavy ones, where a wheel-head may weigh between 1 and 2 tons. When a heavy wheel is grinding to within $\frac{3}{1000}$th part of an inch, revolving at a speed of 6,000 ft. per minute at the surface, and operating across a cutting face from 2 ins. to 4 ins. wide, it is clearly not a good mechanical proceeding to traverse such a mass under such conditions. The present practice, therefore, is to put plenty of mass into the wheel-head and to support it rigidly directly from the floor. Then the only movement which it receives is that of feeding inwards to the work. Incidentally, also, the necessary travel of the table requires a long table, which, again, avoids excessive wear. A slow table or traverse speed should be used when taking heavy cuts, but for finishing a high traverse speed is preferable.

The belt-drives from cumbrous overhead shafts have disappeared from all except the smaller and medium-sized machines, as electric drives have largely displaced this clumsy method and permit overhead-travelling cranes, or overhead tracks, to have free range. The large cylindrical grinding machines of to-day, therefore, have no belt-drive from above. A motor is installed somewhere alongside the bed, and the grinding wheel is belt-driven from that ; the work speeds and the wheel-feeds are arranged through gears enclosed in boxes protected from grit. The convenience of this arrangement, which is only rendered possible by the electric drive, is inducing manufacturers to adopt similar arrangements for the smaller machines.

In the best machines no mild steel and cast iron is used for shafts and gears. High carbon or alloy steels alone are used for these. Nickel, nickel-chrome, or chrome-vanadium steels ensure strength and durability, without which the small shafts and gears of the pitches would soon show serious wear. Friction is reduced by the universal fitting of ball-bearings for taking circular and end thrust. Lubrication is provided in various ways, ring-oiling to the main bearings, forced water-supply to the work and the wheel, and a large supply-tank or tanks with a circulating pump. New forms of steadies have been evolved, and more are used, besides which steadies for some duties are specially designed. The necessity for supporting work almost continuously receives more recognition than it formerly did, and the work is better supported on the bottom and in opposition to the wheel.

As good grinding machines are costly, concessions to the requirements of the smaller firms, in the form of various attachments, are inevitable, but the tendency is towards using plain grinding machines, each limited to one set of functions, every detail being considered and worked out for that duty alone.

Van Dorn Tool Post Grinder.

This machine, fig. 3 (made by the Van Dorn & Dutton Co., Cleveland, Ohio), is furnished with an internal attachment as shown. It may be used for internal and external grinding on the lathe, tool grinding on the milling machine, and surface grinding on the shaper or planer. The spindle makes 4,500 revolutions per minute, and the weight of the machine is about 24 lbs. These machines are extremely useful on a wide variety of work in all engineering shops.

All parts of these machines are made accurately to gauge, ensuring perfect interchangeability. All adjustments may be easily made without removing any part of the machine. The taper bearing at the wheel end ensures the correct bearing adjustment being maintained. The bearing for taking the end thrust may be adjusted to exceptionally fine limits. The bearings are bronze and the lubricating system is very efficient. The motor is air cooled. The armature is form wound, a commutator of special design giving great rigidity and perfect insulation. The windings are insulated by a special process. The brushes are easily accessible. Adjustable rear handles are on hand types.

Bearings.—The design of the bearings is a special feature of Van Dorn grinders. The bearing for the wheel end is so constructed that adjustment for compressing the bearing can be effected

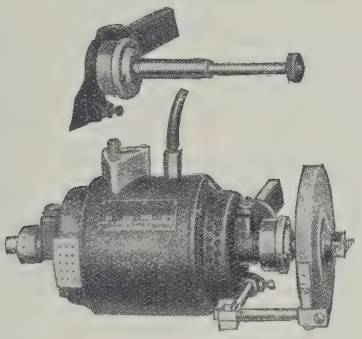


FIG. 3.

in addition to expanding; thus a perfect running fit can be obtained, whilst efficient locking arrangements ensure that the adjustment cannot be disturbed.

Longitudinal adjustment is accomplished by means of a double thrust bearing provided with a dial cap. By depressing a locking pin the dial may be moved in either direction, enabling the adjustment to be made within very fine limits.

PORTABLE PRECISION GRINDERS.

This grinder, fig. 4 (*The Consolidated Pneumatic Tool Co., Ltd.*), is specially adapted for fettling castings and grinding out cylinders; for this latter purpose it is fitted with a shaft

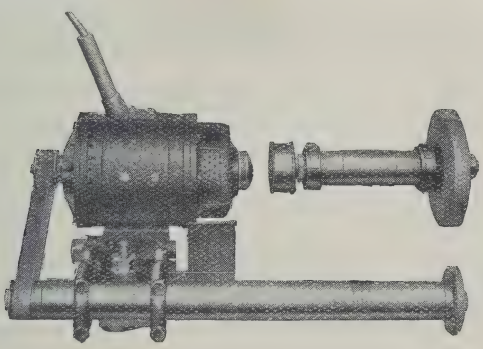


FIG. 4.

extension. The arrangement consists of a high-speed motor mounted in a special arbor, and belted to a shaft which carries an emery wheel. Two or more replaceable shafts are provided, one taking a large, and the other a very small wheel, so that a wide variety of grinding work, from motor cranks and cylinders to valves, jigs, and small tools, can be done.

Grinding Procedure.*

Cylindrical Grinding—External.

The best average *surface speed* of modern grinding wheels, as made to-day from artificial abrasives, is about 6,000 ft. per minute for external cylindrical grinding, and the useful speed range is from 6,500 to 5,500 ft. per minute; below this speed excessive wheel wear is liable and very probable, and grinding machines should be arranged so that the effective life of the wheel falls within this range. It is of the utmost importance that the speed of the wheel be maintained during the cutting operation, no matter what the speed may be, and the drive should be sufficiently powerful to prevent slowing down during momentary heavy cutting. Not only is the wheel wasted by being allowed to slow down, but the wheel face is destroyed and more frequent truing up is necessary. It is more economical, from the point of view of wheel wear to metal removed, to keep the speed up so that softer and freer cutting wheels can be used, the speed enabling them to stand up to heavy cutting without undue wear, and, further, such wheels have very little tendency to glazing.

The *grade* of the wheel is determined by the material to be ground and the rigidity or otherwise of the work, and whether it is of large or small diameter. No definite rule can be laid down for the adoption of any particular grade, but it is unusual in general machine work to require more than two wheels to a machine, one to grind all classes of steel work and the other to deal with all cast metals. Closer grading can be done where the work is all of one class, as, for instance, a machine may be employed wholly on cast-iron pistons, crankshafts, or piston-rods, etc., in which cases particular grades of wheels can be employed.

The *diameter* of the wheels should be as large as possible.

The direct result of using large diameter wheels is economy of operation, the cost of wheel per cubic inch of material removed being less than in the case of the smaller wheel.

The *wheel width* has a direct result on the output, providing it is made use of, and to utilise a wide wheel to its full extent the table-travel must correspond.

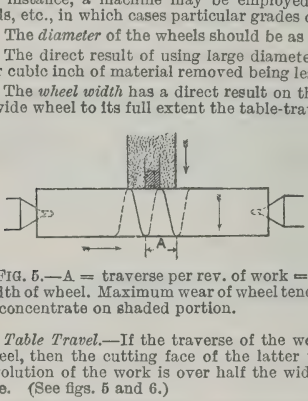


FIG. 5.—A = traverse per rev. of work = $\frac{3}{4}$ width of wheel. Maximum wear of wheel tends to concentrate on shaded portion.

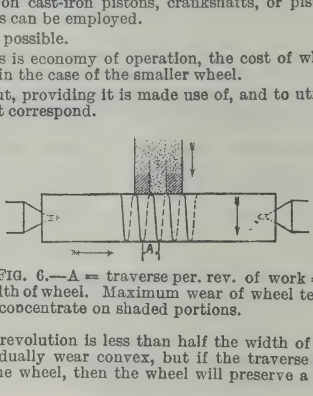


FIG. 6.—A = traverse per. rev. of work = $\frac{1}{4}$ width of wheel. Maximum wear of wheel tends to concentrate on shaded portions.

Table Travel.—If the traverse of the work per revolution is less than half the width of the wheel, then the cutting face of the latter will gradually wear convex, but if the traverse per revolution of the work is over half the width of the wheel, then the wheel will preserve a flat face. (See figs. 5 and 6.)

The ideal traverse per revolution of the work is about two-thirds the width of the wheel, but it should not, except for finishing, be less than half the width.

The table should travel as rapidly as can be used. (See '*cross feed*,' page 1117.)

The main factor governing production on external cylindrical grinding machines is the combination of wide wheels with fast table speeds, as, other things being equal, the machine which possesses these advantages is the most efficient tool.

The *work speed* has only an indirect effect on the output. With a good wheel it makes little difference to the finish obtained whether the work surface speed be, say, 30 or 60 ft. per minute, except that, if the lower speed is persisted in, it limits the table travel, and so in turn limits the output. For external cylindrical grinding the practice is to work on an average surface speed of 60 ft. per minute as a basis. Low work speed is directly conducive to distortion and inaccuracy. The heat created by the cutting action of the wheel must be dissipated or distributed as quickly as possible, and can only be done by a higher work speed and table travel aided by an ample flow of water.

Vibrations in the work itself and disintegration of the grinding wheel are more liable to occur in large diameter and heavy work than in small work. It is much easier to control vibration in small diameter work, and it would be almost impossible to rotate it at such a high speed as to disintegrate the surface of the wheel, but the tendency would be to cause unnecessary wheel wear, and if the table speed was such that the traverse was less than half the width of the wheel the later would wear more at the corners than in the centre.

Vibrations on heavy work are much more difficult to control, and much depends on the

* Abstract of Paper read by H. H. Asbridge before the Manchester Association of Engineers.

machine and method of supporting the work. No definite rule can be laid down, as so much depends on circumstances. For instance, it would be both permissible and advisable to run an 8-in. tube at a higher surface speed than an 8-in. solid shaft, the limiting factor in the tube being a surface speed such as would cause disintegration of the wheel face, there being no heavy body of metal to set up vibration, whereas in the shaft the limiting factor would be a surface speed sufficient to set up an oscillatory motion difficult to control, and which would prevent the shaft from being ground to a perfect cylinder. This same defect could also be obtained by using a too slow surface speed, due to insufficient dissipation of the heat created by the grinding wheel. Between these two extremes there is a wide margin of surface speed at which grinding can be sucessfully carried on.

A good rule is to get a wheel suitably graded for the work, and when once a suitable grade is found it should be recorded, as wheels such as are made from artificial abrasives can be duplicated within very fine limits.

The *cross feed* to the grinding wheel should operate at each reversal of the table, that is, at the end of each stroke; this is necessary to distribute the work over the whole width of the grinding wheel. Production is not governed so much by the amount of cross feed as by other conditions. There should be as much cross feed as the work, the machine, or the grinding wheel will stand; this will be from ·0005 in. to ·0015 in. on the work diameter at each reversal of table. Under these conditions, a machine with a fast table travel will give maximum output with a comparatively light cross feed, as compared with a machine having only a slow table speed, and will have the advantage also of not stressing the work to the same extent. The wheel will remove more material for a given wheel wear, and the machine will be more economical in power.

Over-run of work should be avoided (see p. 1119).

The *time required* in external grinding cannot be calculated with any approach to accuracy on the usual basis of cutting speed and feed, etc.

The finish grinding of a shaft of a given diameter and length is not a constant operation with an abrupt termination, as in other methods of machining, but is an operation starting at zero, rising quickly to the point at which possibly the maximum cutting capacity of the wheel is demanded, and then falling gradually to zero or finished size, and the finer the limit fixed for the finished diameter, the more gradual will be the closing stages of the operation. Grinding times must, therefore, so far as the wheel is concerned, be based on a mean of its cutting capacity, much lower than its maximum.

The formula, as deduced from the above method, is as follows:

Diameter of work in inches × length in feet × constant = grinding time in minutes for the removal of $\frac{1}{32}$ in. in diameter and finishing to commercial limit.

TABLE OF CONSTANTS.

Diameter of Work.	Size of Wheel.	Constant.
4-in. shaft and upwards.	26 ins. by 3 ins.	1·3
3-in. ,,	,,	1·4
2-in. ,,	,,	1·8
1½-in. ,,	,,	2·2
1-in. ,,	,,	3
3-in. shaft and upwards.	14 ins. by 2 ins.	2·2
2-in. ,,	,,	3
1½-in. ,,	,,	3·7
1-in. ,,	,,	5
3-in. shaft and upwards.	12 ins. by 1 in.	3
2-in. ,,	,,	3·8
1½-in. ,,	,,	4·5
1-in. ,,	,,	6·3

For the removal of $\frac{1}{32}$ in. in diameter allow $\frac{2}{3}$ of time obtained. For work below 1 in. diameter the grinding time tends to increase, depending entirely on rigidity of support afforded to same. Extra time should be allowed to special limits, such as may be required for drive fits, etc., or for special finished surfaces, such as are necessary for spindles, gauges, etc.

Centreless Grinding *

The objections to the use of centres to support the work during grinding, and the difficulty of accurately grinding gudgeon pins, crank pins, and other relatively long pieces of small diameter

* Particulars supplied by Dowdings' Machine Tool Co. Ltd., 38, Albert Embankment, S.E. 11.

can be avoided, and any irregularity caused by warpage in hardening removed by supporting the work in a suitable guide block. Two patterns (fig. 7) of guide blocks are used—one for rough grinding, and the other for finishing—the difference being in the relative positions of the three guide blades.

For rough grinding the near blade is adjusted so that the work projects beyond the lower front blade C, slightly more than the thickness of the material to be removed, whilst the upper blade, A, is given a clearance of 0·005–0·020 ins. In finishing, the upper and lower blades are more nearly vertical. The work is finished after it has been passed several times through the block.

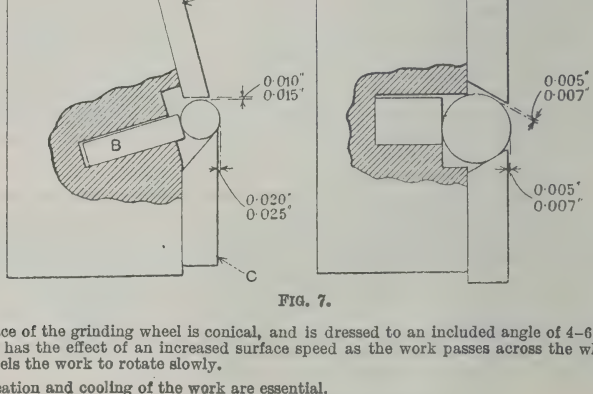


FIG. 7.

The face of the grinding wheel is conical, and is dressed to an included angle of 4–6 degrees, so that it has the effect of an increased surface speed as the work passes across the wheel face and compels the work to rotate slowly.

Lubrication and cooling of the work are essential.

The grinding is effected more rapidly than when centres are used.

Internal Grinding.

The idea that internal grinding cannot be satisfactorily accomplished at spindle speeds less than 30,000 to 50,000 r.p.m. is erroneous. Provided that there is rigidity in the spindle employed, the grinding-wheel diameter has little effect on production. The effective speed range of a good grinding wheel is very much greater than in any other form of grinding, and ranges from 1,000 to 4,000 ft. per minute, and much of the most successful grinding is done at the surface speed of from 1,500 to 2,500 ft., but a very rigid spindle is necessary.

There are two types of internal spindles, the ' tube ' type having a bearing immediately behind the wheel (this type is not to be recommended for holes, say, less than 6 ins. in length, as holes of this length, and longer, are generally insufficiently large in bore to admit of a substantial spindle), and the ' adaptor ' type. The construction of the latter type is such that the spindle bearings never enter the hole being ground, the wheel being carried on an adaptor fitted into the main spindle and held by a draw bolt, or, if the work to be ground is of limited range, the adaptor portion is formed integral with the main spindle. For general commercial work the loose adaptor is preferable, as long or short, heavy or light adaptors can be used suitable for the particular work in hand. The fact that the main spindle in this instance never enters the hole permits of it being made of liberal dimensions. The Churchill Machine Tool Co. fix the higher limit of such spindles at 1,500 r.p.m., and so long as the rigidity is there behind the adaptor no higher speed is necessary for holes from $\frac{1}{4}$ in. upwards.

The *wheel width* has a direct result on the output.

Given a spindle with the necessary rigidity, the use of wide wheels and a high table speed are the governing factors in obtaining maximum production. The standard practice of the above firm is to use all wheels from $\frac{1}{8}$ in. to 3 ins. diameter $\frac{3}{4}$ in. wide, above 3 ins. diameter 1 in. wide. Such wheels are of no advantage unless combined with high traverse speeds, in order to utilise as much of the wheel width as possible, and also to dissipate the heat generated evenly.

Table traverse speeds up to 9 ft. and 10 ft. per min. are commonly applied to internal grinding machines.

The *work speed* in internal grinding can be varied considerably above or below the speed which has been previously determined to be the best for the particular work in hand. Such variation would have very little effect on production, but might affect the finish obtained. In standard practice a work surface speed of from 100 to 120 ft. per minute is used as a basis or starting point, and varied according to circumstances.

The *cross feed* requires special care in internal grinding.

Whatever type of internal grinding spindle is used (where the work to be ground is carried on a rotating spindle) there is bound to be overhang from its supporting bracket, with more or less unavoidable spring, and whilst the slide carrying the spindle can be set to a predetermined position with great accuracy, it does not follow that the hole being ground will have reached the desired size, owing to the spring in the spindle. This fact makes the use of automatic cross feed almost impossible of accomplishment if the amount of the feed has to equal the amount of cut which the wheel will take at each traverse of the table. Hand-operated cross feed fitted with an adjustment dead stop is therefore to be preferred.

In another type of internal grinding, the work is stationary and the grinding wheel has imparted to it a planetary motion whilst rotating at a high speed on its own axis, the planetary spindle being eccentrically adjustable whilst grinding is proceeding. This type of machine has been employed for grinding motor-car cylinders when cast *en bloc*, and the possibilities of this type of machine for finishing holes are great.

Over-run of work should be avoided; that is, the wheel should never be allowed to leave the work, the traverse being only just sufficient to allow of not more than one-third of the width of the wheel projecting beyond the end of the work. If the wheel is allowed to run off, the result in external grinding will be to have the face of the wheel sheared away by the sharp ends of the work; and in internal grinding the hole will be bell-mouthed.

Surface Grinding.

There are two methods of surface grinding—one using the side of the grinding wheel,* and the other the edge of the wheel, carried on either a vertical or horizontal spindle, the more widely used being the vertical spindle type. The two methods are further subdivided according to the manner in which the work is moved, machines being made with the work carried on a reciprocating table, or on a circular revolving table, briefly classified as follows:—

1. Horizontal spindle surface grinding machine, where the work is reciprocated under the grinding wheel.

2. Horizontal spindle ring and surface grinding machine. In this machine the work is carried on a circular revolving table, generally made in the form of a magnetic chuck, and rotated under the grinding wheel.

3. Vertical spindle ring and surface grinding machine, where the grinding wheel is of cup form, carried on a vertical spindle, and the work is carried on a circular rotating table, generally in the form of a magnetic chuck and rotated under the wheel.

4. Vertical spindle surface grinding machine, where the wheel is of cup form carried on a vertical spindle, and the work is carried on a table reciprocating under the wheel.

The first and fourth types are the most commonly used, and intended for the grinding of flat surfaces of all kinds and in all materials.

The second and third types are widely used for grinding the sides of disc work, such as circular knives and saws, piston rings, disc valves, etc., using the rotary table. The second type of machine with the horizontal spindle leaves the work with a concentric finish, whereas the third type of machine with the cup wheel leaves the work with a radial finish, and which in some classes of work is objectionable, particularly, for instance, in piston rings, where the concentric finish is preferred. So far as production is concerned the vertical spindle ring and surface grinding machine will give a greater output than the horizontal spindle ring and surface grinding machine. Both machines can be arranged to grind flat, concave, or convex surfaces.

The most largely used machine for regular commercial purposes is the machine with the vertical spindle, owing to its greater productive capacity.

The best *wheel speed* for all-round surface grinding is about 4,000 ft. per minute, with very little margin of speed—that is, if a wheel is working successfully at 4,000 ft. per minute, it cannot be varied up or down from this speed without affecting production, consequently the vertical spindle machine with the cup wheel has a great advantage in this respect, due, of course, to the wheel retaining its diameter as it wears.

The grade of the wheel for surface grinding demands far closer attention than in any other type of grinding. The only remedy for a wheel that is too hard or too soft is to change it for a wheel of the correct grade.

* Grinding on the flat sides of straight wheels is often hazardous, and should not be allowed. Cup or ring wheels should be used.

A high *table speed* is the main factor governing output, using suitable grinding wheels, but the machine which will so work that the temperature is even throughout the whole ground surface will produce the most accurate results.

The vertical surface grinding machine with the cup wheel, which covers the full width of work at each stroke of the table, has a distinct advantage over the horizontal spindle machine using the periphery of the wheel, and which would have to be traversed transversely in order to cover a work surface wider than the width of wheel.

Quite apart, however, from the question of heat distribution, the vertical spindle machine has the greater productive capacity, and for grinding wide surfaces or groups of small articles this type of machine has the advantage of the horizontal spindle machine. The vertical surface grinding machine is now being adopted in all classes of engineering works, and wherever flat surfaces have to be produced, such as in machine-tool work, there is no method that can be compared with it for cheapness of production.

A machine of this type added to the equipment of any works, and used for the production of flat surfaces has, more than any other type of grinding machine, a beneficial effect throughout the works, and in this respect raises automatically the standard of work produced. In this way, given an accurate machine, it is so very easy to true up straight edges and similar tools and keep them all in a constant high state of perfection; and by having a machine of this type always in readiness to produce flat surfaces the result is that more accurate work is automatically produced throughout the works.

The *time required* for grinding a given surface can be ascertained with fair accuracy once a basis is found, based on area ground and amount of metal removed. Under normal working conditions, a grinding wheel has a certain capacity for the removal of metal, based on its cutting area or its diameter and width.

Wear of Wheels in Grinding.

The following examples will serve to show what may be expected from wheels which are suited to the conditions and work in hand :—

A batch of 72 soft steel spindles of diameters ranging from 1½ inch down to $\frac{9}{16}$ inch by 8 inches long, and having from ·010 to ·015 inch left for grinding, were finished in an average time of 4½ minutes each. The wheel was Norton 'Alundum' 60.M, running at a speed of 6,500 ft. per minute, and it lost ·0005 inch diameter for every six spindles ground. The work speed was 30 ft. for $\frac{9}{16}$ diameter, and 60 ft. for 1½ inch diameter, and the grinding limit was ·00025 inch.

Cast-iron spindle tubes 2½ inches diameter by 12 inches long took an average time of 10 minutes each to finish. The grinding limit was ·00025 inch, and the amount removed ·010 to ·015 inch. The wheel was Norton 'Alundum' 46.H, running at 5,500 ft. per minute, and it lost ·001 inch in diameter on every tube. The speed of the work was 30 ft. per minute.

These examples are given as indicating an amount of reduction in wheel size which should not be exceeded under ordinary conditions. (*Churchill Machine Tool Co., Ltd.*)

Grinding Allowances.

(For Norton and Heald Grinding Machines.)

Frequent inquiries as to the amount of stock which should be left on turned and bored work to be ultimately finished to size by grinding had led to the answer that this depends on several factors which must be ascertained before a correct allowance can be determined. The object aimed at should be to so proportion the work between the lathe and the grinding machine that the total time taken by both in producing the desired degree of finish and accuracy is reduced to the minimum.

In a desire to turn out a 'good job' the turner sometimes overlooks the fact that a smooth piece of work having only a small allowance may take as long to finish in the grinding machine as a rough-turned job on which more metal is left.

The chart, fig. 8, shows the grinding allowances which are recommended for turned or bored work up to 12 ins. diameter. Although the amounts specified may, to some, appear excessive, they may be allowed with confidence in establishments possessing a grinding department run according to modern ideas of efficiency and lathes sufficiently powerful to take the coarse feeds recommended.

Curves A and A¹ show maximum and minimum amounts to be left on the diameter of rough turned work over 12 diameters long.

Curves B and B¹ show maximum and minimum amounts to be left on the diameter of rough turned work under 12 diameters long.

Curves C and C¹ show maximum and minimum amounts to be left on diameter of smooth turned and bar-lathe work over 12 diameters long.

Curves D and D¹ show maximum and minimum amounts to be left on diameters of smooth turned and bar-lathe work under 12 diameters long.

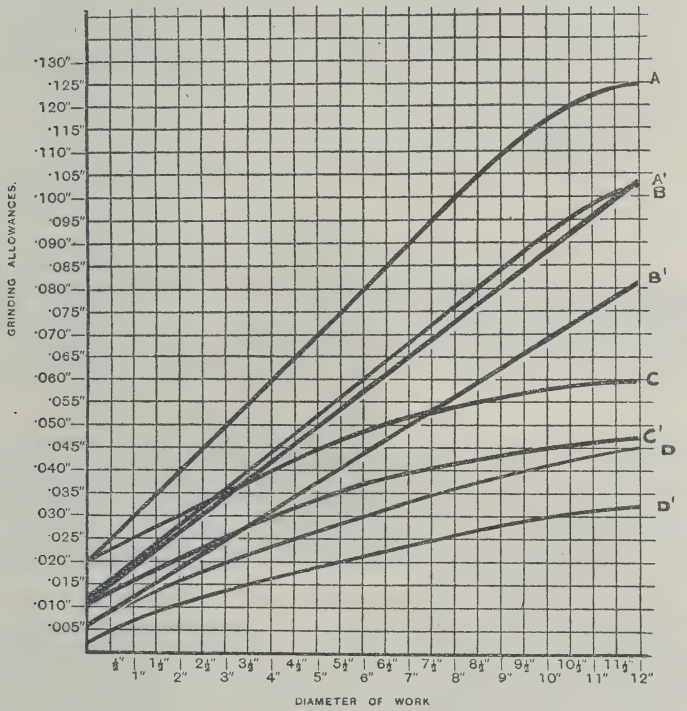


FIG. 8.—Grinding Allowances for Norton Cylindrical Grinding Machines.

NOTE.—By 'rough turned,' is meant work having coarse traverse marks, the measurements being taken on the top of the ridges. The following are the finest feeds which should be employed :—

For work up to 2 ins. diameter	30 per inch
For work from 2 ins. to 4 ins. diameter	20 ,,
For work from 4 ins. to 8 ins. diameter	16 ,,
For work from 8 ins. to 12 ins. diameter	8 ,,

Case-hardened work is an exception, because the finish must be fairly smooth to allow regular penetration of the casing compound. Use curves C and C¹ for all such work.

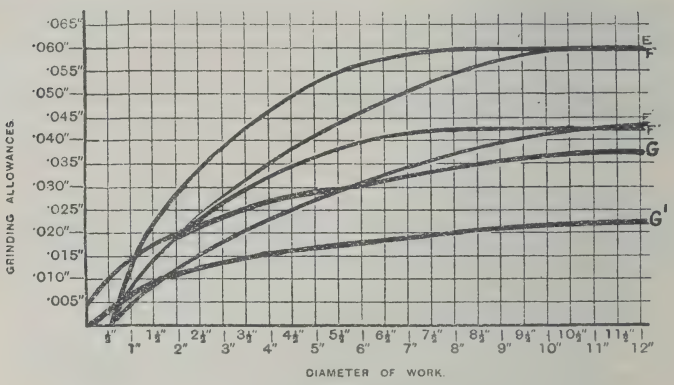


FIG. 9.—Grinding Allowances for Heald Internal Grinding Machines.

Curves E and E′ (fig. 9) show maximum and minimum amounts to be left in the bore of mild steel work that is to be case-hardened before grinding.

Curves F and F¹ show maximum and minimum amounts to be left in the bore of carbon or high-speed steel work that is to be hardened before grinding.

Curves G and G¹ show maximum and minimum amounts to be left in the bore of soft work, whether steel, cast iron, or brass.

NOTES.

The following explanatory notes will serve to show the theory followed out, and will assist in determining allowances for the exceptional jobs which will occasionally present themselves:—

Taking first *external work*, as being the more common, it will be noted that a larger allowance is made on articles which exceed twelve diameters length than in the case of shorter jobs. This is to save lathe work.

A job over twelve diameters long will spring considerably under a good cut, and there is a possibility of bends making their appearance. The largest possible allowance is therefore made on this work, so that the turner need not waste the time which would be taken in producing articles more nearly parallel and straight.

Then, again, a distinction is made between rough-turned work and that which is produced by a fine feed or made on turret lathes with a box tool.

For all work turned between centres, with the exception of that which is to be carbonised for case-hardening, the use of a coarse traverse is advocated, the finest feed per inch that should be used for various diameters being specified on the chart. Work which is to be carbonised must be turned with a fairly smooth finish, so as to allow regular penetration of the casing compound; therefore the allowance specified for smooth work over twelve diameters long should be used for case-hardened work whatever its length. If such work is hollow it may distort after quenching, therefore great care must be exercised in hardening, as the allowance above specified cannot be exceeded or the hard skin would be entirely removed in grinding to size.

The allowance for carbon and high-speed steel work which is hardened will be the same as that provided for soft work, as any distortion which occurs must be corrected in the straightening press before grinding.

Internal work, which is dealt with by the lower portion of the chart, is divided into three classes as follows:—

1. Mild steel case-hardened.
2. Carbon or high-speed steel hardened.
3. Soft work, whether steel, cast iron, or bronze.

Hardened and case-hardened work in which the hole is less than half an inch in diameter should, as shown in the chart, be bored or reamed to standard size, as there will be a sufficient

shrinking when the quenching takes place to provide a grinding allowance. It will be noted that the amount allowed in case-hardened work increases very rapidly up to 6 ins. diameter; this is to allow for distortion in hardening. Above this size the increase is slow, and care should be taken to avoid excessive distortion, as no larger allowance can be made, otherwise the hard skin would be removed by the grinding operation. In determining the allowance for soft work it should be remembered that the class of finish given to the hole will decide whether the high or low limit should be worked to. Thus, in holes which are rough drilled or bored, the allowance should approach the maximum, whilst in reamed holes the minimum allowance should be left.

(*Alfred Herbert Ltd., Coventry.*)

Polishing.

For grinding and polishing *marble, stones*, and *glass*, crushed steel has proved very satisfactory, on account of its hardness, sharpness and toughness. It is, however, but little used in this country, sand, carborundum or aluminous abrasives being generally employed. For cutting stone, marble, and glass, a metal wheel with loose sand and water is largely employed, but grinding wheels of carborundum and aluminous abrasives of medium grade and 36–50 grit are being increasingly employed. For producing a rough surface on glass, the sand-blast process is largely used.

For *smoothing wood* and similar materials, garnet-emery and sand, mounted on paper or cloth, are largely employed, garnet being especially suitable for hard woods. For finishing wooden handles, sand mounted in endless belts is largely used.

In the leather industries, and especially in the manufacture of boots and shoes, corundum and carborundum cloths are largely used for heel scouring and breasting and for buffing.

For work in which an *exceptionally hard abrasive* is required, including the truing of abrasive wheels, diamond bort or carbonado are employed.

For polishing steel articles, blocks of carborundum or aluminous abrasive or cloth or paper covered with emery or corundum is largely employed.

For polishing other metals, glass, etc., loose powders are employed. Such powders may be used in the dry or wet state, and may be rubbed on to the material to be polished, or they may be applied by means of a wheel (or 'dolly') composed of discs of cloth.

For polishing brass and other softer metals, very fine silica, quartz flour, emery carbide or aluminous abrasives, crocus, rouge, or whiting are employed. For silver and gold, well-slaked lime is largely used. The polishing powder is sometimes treated with rouge to disguise its nature and to give white metals a slightly ruddy tint, which improves their appearance.

In the so-called liquid polishes, a little oxalic acid or other chemical detergent is sometimes added to the abrasive powder, or to the liquid in which the latter is suspended.

Power required for Emery Polishing.

Using a calico mop, 12 ins. diameter by 2 ins. wide, it is usual to run these at a speed of 10,000 feet per minute. The average resistance at the circumference of a well-loaded wheel is about 10 lbs., and this would give a h.p. for the mop equal to 3. To this has to be added the friction of the spindle in its journals, and this will depend on the nature of the journal (whether plain or ball), and also the lubrication. The friction is principally due to the pull of the strap, and if its spindle is taken to be 1½ in. diameter, and the value $\mu = \cdot 075$, the frictional h.p. would be about ·25, and if a further ·25 be allowed for acceleration, the h.p. to drive the machine would be 3½. For a 10 in. diameter buff the h.p. would be proportionately less—that is, ab out 3.

Sand Blasting.

The air pressures used for different materials are, approximately, as follows:—Steel castings or forgings, 80 to 100 lbs.; malleable iron, 70 to 85 lbs.; cast iron, 60 to 70 lbs.; and brass and aluminium, 35 to 50 lbs.

For some classes of work a small nozzle opening, that is, a fine strong jet, may be desirable; for other work a broader stream, covering a larger surface but working at a lower pressure may be best. Table I. gives the relation between air flow in cubic feet of free air per minute at varying pressures, and the horse-power required to produce them by single-stage compression, for nozzles ranging in diameter from ⅛ to ⅜ in. inclusive.

Typical cylinder dimensions, together with the displacement and the horse-power required to develop various amounts of air pressure, are given in Table II.

In cases where there is moisture in the air, it is well to make provision for the elimination of the moisture by installing a moisture separator or a suitable heater.

Sand is the most commonly used abrasive on account of its relatively low price. Ordinary sand is inferior to sharp sand and crushed silica, as the two latter possess greater cutting power and are therefore more effective. Abrasives such as steel grit and shot are used to a certain extent, and the use of these more expensive abrasives is warranted under certain conditions. For classes of work such as electro-plating or galvanising, the metallic dust adhering to the work prohibits the use of carborundum, because it prevents perfect galvanising, although no difficulty is experienced in this respect with sand.

There is no one abrasive that is best adapted to all classes of work. A selection must be made with due regard to reclaiming means, to greatest economy in operation, and to maximum production. By making tests with various materials, and keeping accurate records of quality of work, output, life of the abrasive, and so on, the one best suited for the individual application is readily determined. Such tests have shown metal abrasives to have about sixty times the life of sand. This apparently large percentage may, however, be offset to a great extent by losses of abrasive in operation or in handling. In good steel foundry practice these losses may be kept as low as 10 per cent., while the average loss due to disintegration, for different grades of sand, amounts to about 20 per cent. per month. The reduction in cleaning time resulting from using metal abrasives instead of sand varies, of course, but a gain of about 15 to 20 per cent. in time appears to be in line with actual practical experience. Metal abrasives do not require large sand-storage bins and have the added advantage of producing less dust, thereby improving the general working conditions.

TABLE I.—FLOW OF FREE AIR FOR DIFFERENT SIZES OF NOZZLES.

Diameter of Nozzle. Ins.	Pressure (cub. ft. per min.) and Corresponding Horse-power Required.							
	20 lbs.	H.P.	30 lbs.	H.P.	40 lbs.	H.P.	50 lbs.	H.P.
$\frac{1}{8}$	7·70	0·63	10·00	1·03	12·30	1·50	14·50	1·99
$\frac{3}{16}$	17·10	1·40	22·50	2·32	27·50	3·26	32·80	4·49
$\frac{1}{4}$	30·80	2·53	40·00	4·12	49·10	5·99	58·20	7·97
$\frac{5}{16}$	48·17	3·95	62·89	6·48	76·60	9·36	90·70	12·43
$\frac{3}{8}$	69·00	5·66	90·00	9·27	110·00	13·42	130·00	17·81
Diameter of Nozzle. Ins.	60 lbs.	H.P.	70 lbs.	HP.	80 lbs.	H.P.	100 lbs.	H.P.
$\frac{1}{8}$	16·80	2·57	19·00	3·19	21·20	3·86	25·73	5·33
$\frac{3}{16}$	37·50	5·74	43·00	7·22	47·50	8·65	57·88	11·98
$\frac{1}{4}$	67·00	10·25	76·00	12·77	85·00	15·47	103·00	21·32
$\frac{5}{16}$	105·00	16·07	119·00	20·00	133·00	24·10	161·00	33·82
$\frac{3}{8}$	151·00	23·10	171·00	28·73	191·00	34·76	232·00	47·90

TABLE II.—CYLINDER DISPLACEMENTS AND HORSE-POWER REQUIREMENTS OF AIR COMPRESSORS.

Cylinder.		Revs. per min.	Piston Displacement. Cub. ft. per min.	Horse-power Requirements. Pressure (cub. ft. per min).					
Diam. Ins.	Stroke. Ins.			50	60	70	80	90	100
6	6	300	57	7	7·5	8·5	9·0	9·5	10
8	8	260	118	14	15·5	16·8	18·0	19·0	20
10	10	235	210	24	27·0	29·0	31·0	33·0	35
12	12	220	340	45	50·0	54·0	58·0	62·0	65

All abrasives should be screened each time before using, to remove particles large enough to clog the nozzle, and also to eliminate fine particles which only produce dust and have no abrasive quality, but which consume some of the pressure. Screen separators, frequently operated by compressed air, may be used for this purpose. (*Abstract, C.M.S., 'Machinery,' Jan.* 13, 1921.)

SECTION XXI

PART III

STEAM-HAMMERS.

(Messrs. B. & S. Massey, Ltd., Openshaw.)

Three standard types of steam-hammer are made by B. & S. Massey, Ltd.—the 'Overhanging Form with Slides,' the 'Arch Form with Slides,' and the 'Rigby type without Slides.' All these hammers can be made suitable for driving by compressed air instead of steam.

Overhanging Form with Slides.

These hammers (fig. 1) are usually preferred for general smith-work. The anvils are accessible to the workman from three sides, and the tups are supported and accurately guided by the slides, making it possible to do accurate stamping in dies.

Each hammer has two separate standards, with a space between them, through which long bars can be passed. All the hammers are double-acting and are fitted with equilibrium working valves.

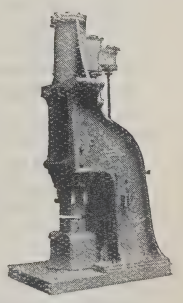


Fig. 1.

Up to and including the 12-cwt. size, the hammers are fitted with a combined self-acting and handworked valve gear; the 15-cwt. and larger sizes are ordinarily fitted with Massey's Handworked Expansion Valve Gear, but a combined self-acting and handworked gear can be fitted if desired.

The pistons, which are forged solid with the piston-rods, are fitted with steel piston-rings. The palletts are made of tough forged steel with planed and hardened faces.

The anvil blocks are about 8 times the weight of the falling mass, and in the 12 cwt. and smaller sizes they are turned to fit the bored baseplates; this greatly increases the efficiency of the hammers when it is desired to use them for stamping in dies.

DIMENSIONS, WEIGHTS, AND MEMORANDA OF OVERHANGING FORM STEAM-HAMMERS WITH SLIDES.

Standard sizes	½ cwt.	1 cwt.	1½ cwt.		3 cwt.
Intermediate size				2 cwt.	
Diameter of cylinder	4½″	5½″	6″	6″	7½″
Longest stroke	11″	12″	13″	13″	17″
Clear space between slides	5½″	6½″	6″	6″	8¼″
Overhang to pallett centres	7⅝″	9″	10″	10″	11″
Total height from floor	5′ 5″	5′ 8″	6′ 9″	6′ 10″	8′ 1″
Steam and exhaust pipes	1″ & 1¼″	1″ & 1¼″	1¼″ & 1½″	1¼″ & 1½″	2″ & 2½″
Approximate weight	1 ton	1½ tons	2 tons	2½ tons	3 tons
Without anvil block	¾ ton	⅞ ton	1½ tons	1½ tons	2 tons
Without anvil block and baseplate					1½ tons
*Diameter of bar worked efficiently	1½″	2″	2½″	3″	4″
†Approximate steam consumption (lbs. per hour)—					
(*a*) On hard blows	115 lbs.	215 lbs.	315 lbs.	410 lbs.	420 lbs.
(*b*) Average on mixed blows	19 lbs.	35 lbs.	50 lbs.	64 lbs.	73 lbs.

Standard sizes	4 cwt.	5 cwt.		7 cwt.	
Intermediate size			6 cwt.		8 cwt.
Diameter of cylinder	8″	9″	9″	10″	10″
Longest stroke	19″	21″	21″	24″	24″
Clear space between slides	10¾″	10¾″	10¾″	11¾″	11¾″
Overhang to pallett centres	11″	13″	13″	14″	14″
Total height from floor	8′ 9′	9′ 0″	9′ 0″	9′ 7″	9′ 7″
Steam and exhaust pipes	2″ & 2½″	2½″ & 3″	2½″ & 3″	2½″ & 3″	2½″ & 3″
Approximate weight	3½ tons	5 tons	5½ tons	6½ tons	7 tons
Without anvil block	2½ tons	3 tons	3 tons	3½ tons	3½ tons
Without anvil block and baseplate	1½ tons	2¼ tons	2¼ tons	2¾ tons	2¾ tons
*Diameter of bar worked efficiently	4⅜″	6″	6½″	7″	7½″
†Approximate steam consumption (lbs. per hour)—					
(*a*) On hard blows	520 lbs.	620 lbs.	710 lbs.	800 lbs.	880 lbs.
(*b*) Average on mixed blows	89 lbs.	105 lbs.	119 lbs.	134 lbs.	147 lbs.

Standard sizes	10 cwt.		15 cwt.	20 cwt.	25 cwt.
Intermediate size		12 cwt.			
Diameter of cylinder	12″	12″	13⅜″	14½″	15⅜″
Longest stroke	27″	27″	30″	33″	36″
Clear space between slides	13″	13″	14½″	16″	17″
Overhang to pallett centres	16″	16″	18″	20″	22″
Total height from floor	11′ 0″	11′ 0″	12′ 2″	13′ 0″	13′ 8″
Steam and exhaust pipes	3″ & 3½″	3″ & 3½″	3″ & 4″	3½″ & 4½″	3½″ & 4½″
Approximate weight	9½ tons	10½ tons	13½ tons	18½ tons	22¾ tons
Without anvil block	5½ tons	5¾ tons	7½ tons	10½ tons	13 tons
Without anvil block and baseplate	4 tons	4½ tons	5½ tons	7½ tons	9½ tons
*Diameter of bar worked efficiently	8″	8½″	10″	12″	13½″
†Approximate steam consumption (lbs. per hour)—					
(*a*) On hard blows	1040 lbs.	1190 lbs.	1420 lbs.	1750 lbs.	2100 lbs.
(*b*) Average on mixed blows	170 lbs.	195 lbs.	228 lbs.	275 lbs.	330 lbs.

* All these figures are necessarily rough approximations.

They are meant to apply to average engineers' smithwork, and are for efficient results on this class of work, with a steam pressure of about 60 lbs. per sq. in.

Even for this they may for occasional jobs be considerably exceeded.

There are many classes of work which require much larger hammers, and some classes which require smaller hammers than those indicated, owing chiefly to the varying amount which has to be done at a heat.

† Steam Consumption.

A steam pressure of about 60 lbs. per sq. in. is assumed.

During ordinary forging there are many blows of less than full force, and there are also necessary pauses (holding up) for adjusting work or tools during a heat.

While line (*a*), page 1126, gives the consumption on continuous hard blows only, line (*b*) gives the consumption as ordinarily reduced by these circumstances. Further, under average smithy conditions, a hammer may be assumed to be actually in motion about 15 minutes in every hour and the figures given in line (*b*) are based on this assumption.

In cases in which one boiler is put down solely to drive one hammer, the capacity of such boiler should be fixed according to line (*a*), page 1126, because otherwise the pressure would fall too low during a normal run of hard blows.

Arch Form with Slides.

These hammers, fig. 2, are made in sizes ranging from 3 cwt. to 10 tons and over, and the design provides a maximum of strength and rigidity. The 15 cwt. and larger sizes are fitted with Massey's Handworked Expansion Valve Gear.

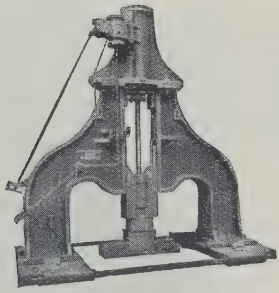


Fig. 2.—Arch Form.

Rigby Type without Slides.

These hammers, fig. 3, which are made in sizes from 3 cwt. to 3 tons, are particularly suitable where forgings of large size and awkward shape have to be dealt with. The strokes are longer

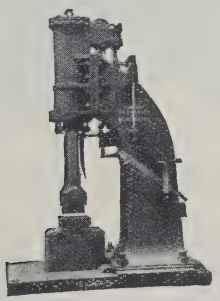


Fig. 3.—Rigby Type.

and the cylinders larger in diameter than in the other forms described. All sizes are fitted with Massey's Handworked Expansion Valve Gear.

Energy and Velocity of Blow.

If

a = area of piston, in square inches : p = average pressure of steam on piston during downward stroke, in pounds per square inch : S = stroke of piston, in feet : W = falling weight in pounds : E = energy of blow after full stroke and before striking, in foot-pounds, friction being neglected ;

$$E = (ap + W)S.$$

If

P = total pressure on piston = pa : F = total force causing downward acceleration = $P + W = pa + W$: g = acceleration due to gravity = 32·2 : V = velocity after full stroke and before striking, in feet per second, friction again being neglected ;

$$V^2 = \frac{2F\mathcal{g}S}{W}.$$

No commercial use is made of these calculations, either for the naming of sizes of hammers or for any other purpose.

The question is sometimes asked : *What weight of blow does the hammer strike?*

The force of a blow cannot be stated in terms of weight at all, because the pressure of a weight is continuous, whereas the force of a blow is expended in a moment. It has, however, been ascertained, by careful experiments, that the maximum blow of a 5-cwt. double-acting steam-hammer, with moderate steam pressure, produces a crushing effect upon a piece of hot iron as great as that produced by a load of about 30 tons, and a ½-cwt. double-acting steam-hammer a crushing effect equal to that produced by a load of about 2½ tons.

Naming Sizes of Steam-hammers.

In naming the sizes of steam-hammers, no account is taken of the pressure of steam upon the piston. Thus, a 5-cwt. hammer is one in which the falling mass weighs at least 5 cwt., but of course the force of the blow is greatly increased by the pressure of steam above the piston accelerating the fall.

Steam Pipes for Steam Hammers.

See 'Steam Pipe Bends,' Section XXX, Part II.

Foundations.

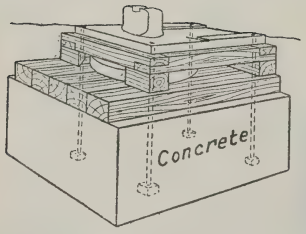


FIG. 4.—Typical Foundation for Steam-Hammers of the Overhanging Form.

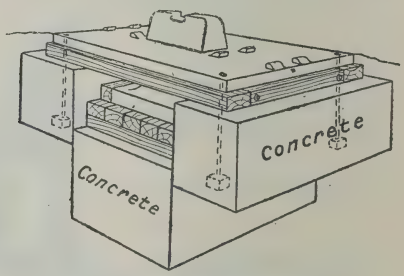


FIG. 5.—Typical Foundation for Steam-Hammers of the Arch Form.

(*B. & S. Massey, Ltd.*)

The Steam Hammers of the Overhanging Form, the Arch Form, and the Rigby Type per pp. 1125–1127 can all be driven equally well by Compressed Air, the clearances being suitably reduced.

PNEUMATIC POWER HAMMERS.

Massey Patent Pneumatic Power Hammers, Overhanging Form with Slides.

These hammers (fig. 6) are arranged for belt or for direct electric drive. They are similar to steam-hammers in many respects, and are about equal to them in working capacity, assuming the steam-hammers to be working under ordinary conditions of pressure, etc.

It is very important to note that the size name of each hammer indicates the true weight of its falling parts.

The anvil block is set diagonally, to enable long bars to be worked in either direction.

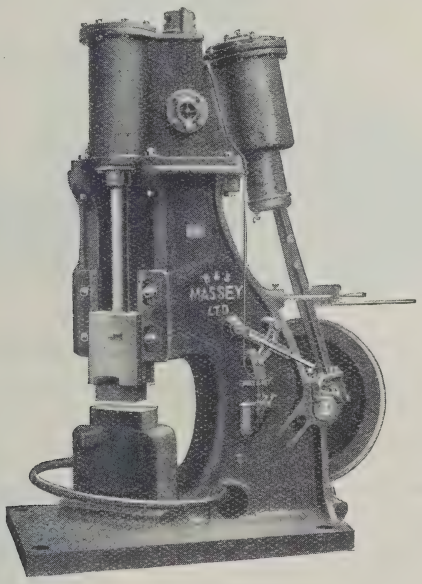


Fig. 6.

The essential feature of the hammers lies in the connecting together of two cylinders—the hammer cylinder and the pump cylinder—by air passages which are governed by a valve under the control of the operator. See fig. 7, p. 1130.

When the piston reciprocates with the valve in the 'not working' position (as shown) air flows freely through the valve from one side of the piston to the other. The tup therefore remains stationary on the anvil.

As the valve is moved upwards the by-pass through the valve is gradually closed, so that when position 4 is reached, all the air is forced from the top of the pump to the top of the hammer, and from the bottom of the pump to the bottom of the hammer, thus producing full-force automatic blows. The positions intermediate between 3 and 4 provide a wide variation in the character of blow and the length of stroke.

With the valve in position 1, the reservoir is charged from the top and bottom of the pump and is, at the same time, in communication with the bottom of the hammer. This gives a perfectly smooth rise and a steady 'Hold-up.'

When the valve is moved to position 2, the bottom of the hammer is opened to atmosphere, and compressed air from the reservoir passes into the top, thus throwing down the tup and striking a single blow.

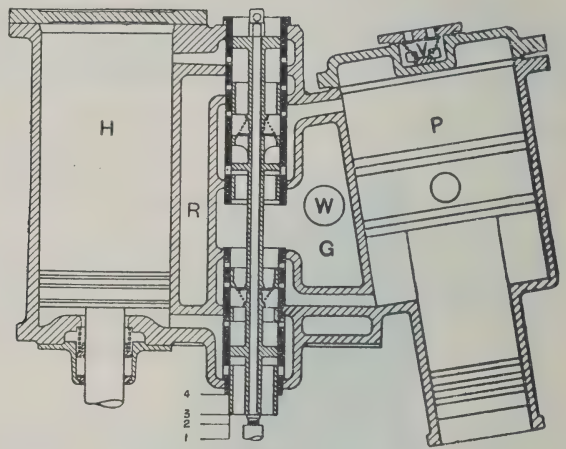


FIG. 7.

While the valve is kept in this position, the tup is held down with full pressure above the piston.

Inlet valves V and W supply air to the top and bottom of the system.

DIMENSIONS, WEIGHTS, AND MEMORANDA OF PNEUMATIC POWER-HAMMERS.

OVERHANGING FORM WITH SLIDES.

Standard sizes	½ cwt.	1 cwt.	2 cwt.	3 cwt.	5 cwt.	7 cwt.	10 cwt.	15 cwt.	20 cwt.
Longest stroke (inches)	11	12	14	17	21	24	27	30	33
Driving pulley diam. and width of belt (inches)	21×2½	24×3	30×3	33×4	39×5	45×7	48×10	27×6	30×9
Blows per min.	280	240	200	160	140	125	110	100	85
Size of bar worked efficiently (inches square)	1½	2	3	4	6	7	8	10	12
Maximum power (full blow) B.H.P.	4	7	10	16	21	25	35	48	60
Mean consumption in B.T.U. (kwt. hours) approximate	·6	·9	1·5	1·9	2·7	3·3	4·2	5·5	6·6
Size of motor (continuous rating)	3	5	7½	12	15	20	27	35	40
Approximate weight of hammer (tons)	1¼	1⅞	3	4½	6½	9½	12½	18⅞	23⅛
Ditto, without anvil block (tons)	—	—	2¼	3	4½	6⅞	8½	13	15½

(*B. & S. Massey, Ltd.*)

Massey Patent Pneumatic Power Hammers.

Arch Form with Slides.

These hammers, fig. 8, are made in the 30 cwt. and 40 cwt. sizes, and are arranged for 'direct electric drive.' The action and control of the hammers is identical with that of the overhanging

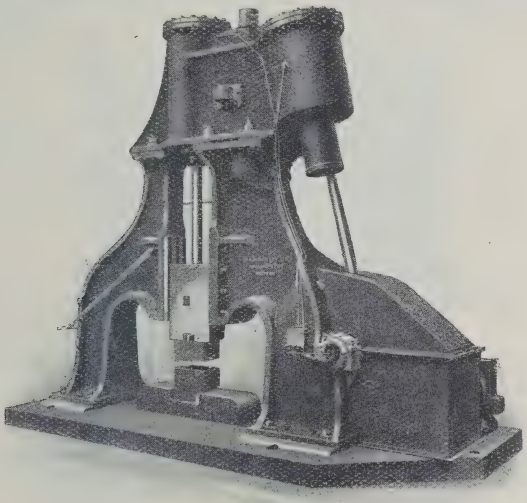


Fig. 8.

form with slides (pp. 1129-30), *i.e.* 'Single Blows' may be struck from any point in the stroke. each requiring a separate movement of the hand lever, or a series of automatic blows may be obtained of any desired force and with long or short strokes.

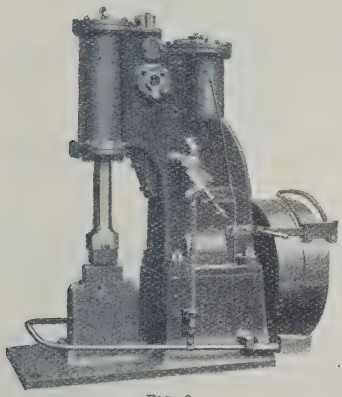


Fig. 9.

'Clear Space Type without Slides.'

These hammers also are identical in action and control with the hammers of the overhanging form with slides (pp. 1129–30). As will be seen from fig. 9 ample clear space is provided for the manipulation of large and awkward forgings, by dispensing with Slides and adopting a special Stuffing Box to guide the rod.

The hammers can be belt or motor driven, and provided with a foot lever in addition to the hand lever, if desired.

DROP STAMPS.

The Massey Friction Drop Stamp.

The practice of Stamping, in Dies, parts which have to be made in large numbers, such as motor-car parts, railway wagon parts, etc., etc., has become very general, and the tool usually adopted

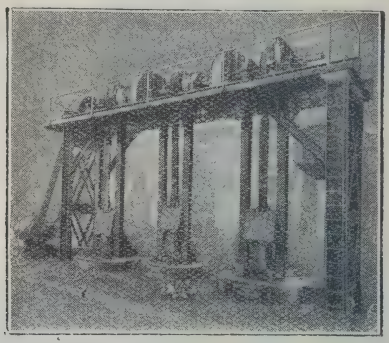


Fig. 10.—Battery of 30-15-20 cwt. Friction Drop Stamps.

for this purpose in this country is the 'drop stamp.' Latterly, the friction lifted type of drop stamp (fig. 10) has tended to supersede the steam lifted type, on account of the improved control, the greater efficiency and the smaller cost of up-keep.

Battery Form.

The usual plan is to have two, three, or four drop stamps of different falling weights arranged in a battery. The lifting mechanism in this case is mounted on girder work supported on stanchions, while the falling weight or tup is guided by vertical rolled steel guides resting in recesses in the anvil block, and supported at their upper ends by the girder work in such a way as not to transmit shock to the lifting mechanism.

Rigid Guides.

Where multiple impression dies are used, or specially accurate die alignment is required, rigid cast-steel guides can be substituted for the ordinary guide bars.

Self-contained Form.

As an alternative to the battery form, 40 cwt. and smaller stamps can be self-contained, with cast-iron guides secured to the anvil block carrying a cast-iron 'cradle' in which the lifting mechanism is arranged (see fig. 11).

Details of Friction Lifters.

The lifters themselves resemble a band brake in construction. A steel brake band lined with suitable material, and connected with the lifting arm, is made to grip a constantly revolving drum by means of a lever actuated by hand through a pulling cord, and the tup is thereby raised. As soon as the tension on the cord is released, a spring disengages the band, and the tup falls under the influence of gravity.

A special arrangement is adopted by means of which the grip of the friction band is automatically reduced as the tup rises, thereby making it necessary to maintain a constant pull on the cord in order to prolong the rise of the tup; when, moreover, the cord is held stationary, the tup remains suspended at the point to which it may have been raised, and as stated above, the tup falls when the tension on the cord is released.

It will therefore be seen that a very intimate relation exists between the movements of the tup and those of the pulling cord.

The movement of the hand is only a short one, and the pull required is very slight, but every movement of the hand is reproduced on a larger scale by the tup. Those who are familiar with the old-fashioned kick stamps in which a heavy pull was required the whole length of the stroke, and which sometimes required two or three men to act as pullers, appreciate very highly the action of friction stamps in which a control as excellent as in the kick stamp is secured with so little labour on the part of the operator.

Method of Driving.

The friction lifters are generally mounted on shafts driven by spur gearing from a quick-running back shaft. A flywheel which also serves as a driving pulley is mounted on one end of the back shaft and supplies the reserve of power needed at the moment of picking up the tup.

The drive is usually taken direct from an electric motor placed either on one of the stanchions or in the larger sizes on the floor, or in some other convenient position.

Simplex Automatic Gear.

For certain classes of work, especially stampings made straight from the bar, it is an advantage for the stamper to be able to control the stamp himself, thus saving the cost of a driver. The Massey patent 'Simplex' automatic gear achieves this by combining a foot-operated hold-up gear with a reversing gear which causes the tup to rise and fall automatically.

When the foot lever is held down the tup falls, and continues to rise and fall until the foot lever is released, when it rises and is held up without consuming any power.

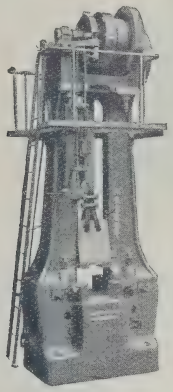


FIG. 11.

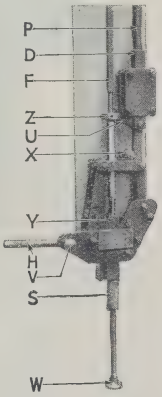


FIG. 11A.

The automatic action is obtained by means of a trip rod (F) which reproduces the movements of the tup on a small scale. The length of stroke can be adjusted by the sleeve (S) and the 'pick-up' by the hand-wheel (W), these adjustments deciding the points at which the toggle-mechanism is brought into and thrown out of action.

When the toggle levers are as shown in the illustration the spring (P) is compressed. As the collar (Z) strikes the lip (U) the toggle levers are thrown across and the spring comes into action, tightening the pulling cord and causing the tup to rise. The trip-rod then also rises, the cam (Y) eventually comes into contact with the roller on the lower toggle lever and the spring is again compressed, allowing the tup to fall.

By means of the hand lever (H) the automatic gear is thrown out of action and the stamp can then be worked by hand in the ordinary way.

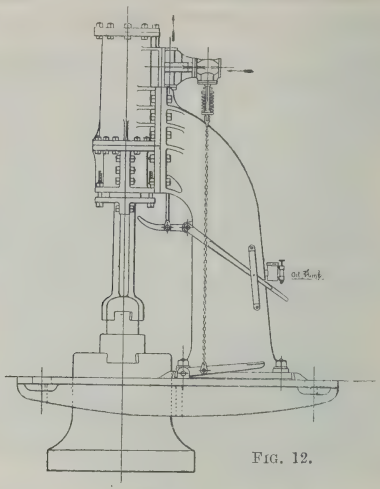


FIG. 12.

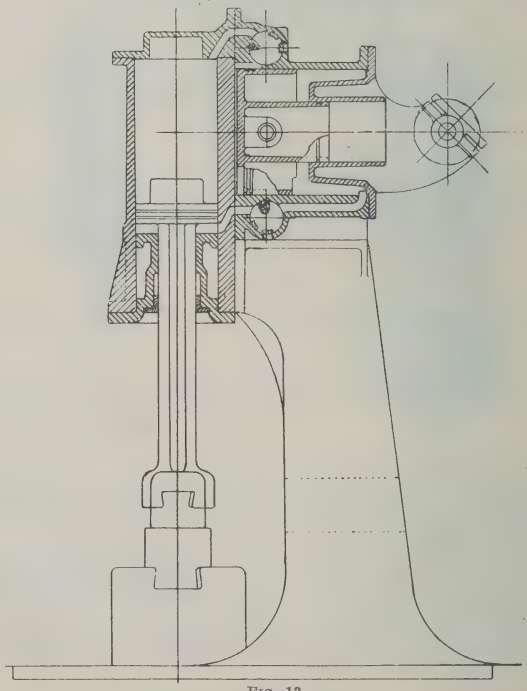


FIG. 13.

'Rigby's Patent' Hammers.

(R. G. Ross & Son, Ltd., Greenhead, Glasgow.)

The 'Rigby's Patent' Steam and Compressed Air Hammer, made by R. G. Ross & Son, Ltd., for very many years, is so well known that it hardly calls for description. In the single column form, illustrated by fig. 12, it is completely open on three sides and is quite without external guides. Owing to this open construction, it is much employed for general smith's work. The cylinder unit is of exceptionally strong construction, having a heavily flanged external mouthpiece or stuffing box which is bolted direct to the column flange, thereby taking all the working stresses and strains off the cylinder proper, and greatly increasing the life of the hammer. The ingot steel piston rod, forged in one piece, is machined, highly polished, provided with two flats and fitted at the lower end with a hammer face of high tensile steel. The mouthpiece and gland together provide a long guide for this piston rod, corresponding flats being machined in both. The gland, which is of steel, is lined with white metal to ensure a perfectly smooth bearing for

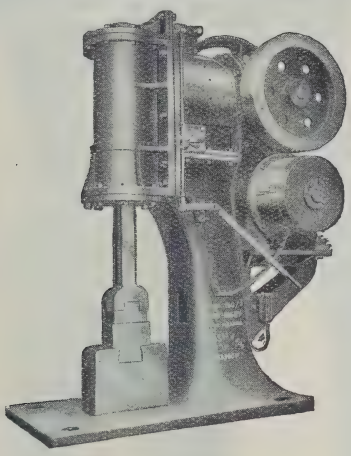


Fig. 14.

the piston rod, and flats are also machined on the outside of this gland to take all twisting strains off the adjusting gland bolts.

A double column form, also without external guides, is made by the firm, the cylinder being supported on heavy built-up girder work.

The 'Ross-Rigby' motor-driven pneumatic power hammers are constructed on the same principle, as may be readily gathered from fig. 13. The main parts of this hammer comprise a horizontal compressor with a piston of trunk type, a driving motor geared to a crankshaft, which drives the compressor, and, finally, the working cylinder and its various parts. Two interconnected rotary valves control the movement of the hammer piston, whilst for 'holding-up' and 'holding-down' springless non-return valves are employed. These hammers are made in sizes from 2 cwt. up to 10 cwt. falling weight.

Drop-Forging Equipment.

(Brett's Patent Lifter Co., Ltd., Coventry.)

Drop-forgings now form an essential feature of most engineering propositions and are, in fact, indispensable in the economical construction of automobiles, aeroplanes, agricultural machinery, railway rolling stock, locomotives, carriages, and waggons.

Types of Hammers.—Drop-hammers are generally constructed in battery form containing three or more hammers of varying sizes, or, alternatively, each hammer can be constructed as a single self-contained unit, as shown in fig. 15. The lifter or apparatus for controlling the tup is a very important feature of a drop-hammer, and two main types are in use, *i.e.* suitable for

operating with steam or compressed air, or, alternatively, what is known as the friction type, which is driven by electric motor, or belt power. The question of installing one or the other depends entirely upon convenience of power. Large drop-hammers of 5 tons and upwards are

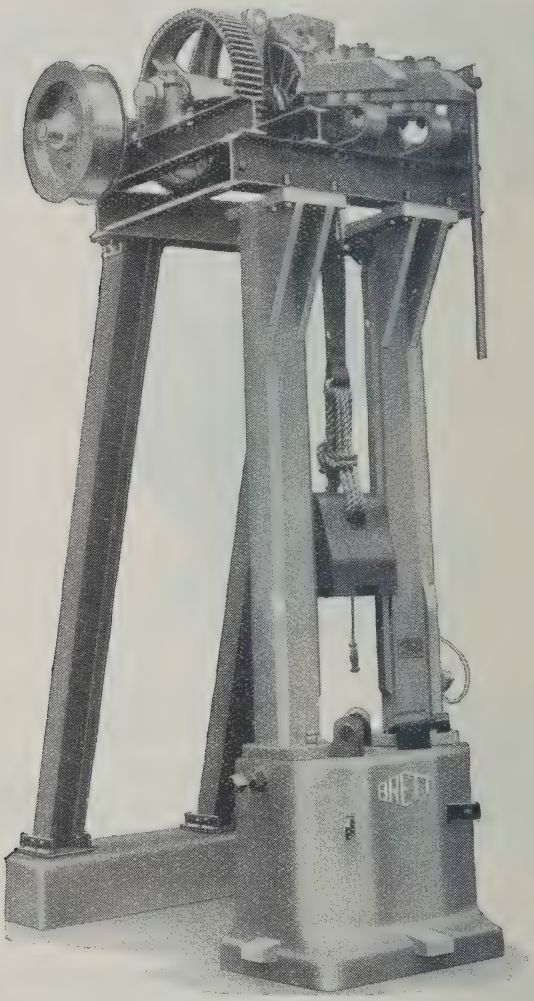


FIG. 15.

economically driven by means of a tubular boiler fired by the waste heat from the furnaces. Small hammers are frequently arranged in self-contained form, the usual standards being dispensed with.

SECTION XXII

SCREW THREADS AND SCREWS — BOLTS AND NUTS — LOCK NUTS—SPANNERS—PIPE FLANGES—KEYS—SCREW TAPS—COUPLINGS—PULLEY ARMS AND BOSSES

(pp. 1139–1177)

(Revised by Professor Dempster Smith, M.B.E., M.Sc.Tech., M.I.Mech.E.)

SECTION XXII

SCREW THREADS AND SCREWS—BOLTS AND NUTS—LOCK NUTS—SPANNERS—PIPE FLANGES—KEYS—SCREW TAPS—COUPLINGS—PULLEY ARMS AND BOSSES.

(Revised by Professor Dempster Smith, M.B.E., M.Sc.Tech., M.I.Mech.E.)

SCREW THREADS AND SCREWS.

Forms of Screw Threads.

Figs. 1, 2, 3, and 4 show four usual forms of thread, each of 1 inch pitch.

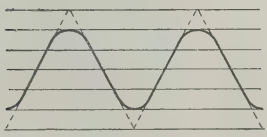


FIG. 1. Whitworth Thread-form.

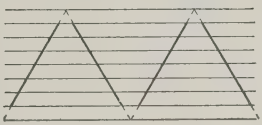


FIG. 2. American Thread-form.

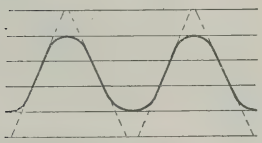


FIG. 3. Swiss Thread-form.

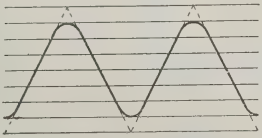


FIG. 4. Steinlen Thread-form.

In the *Whitworth* or *English Standard* thread (fig. 1) the angle made by the two sides of the thread is 55° for all sizes of thread. One-sixth of the depth of the triangle is rounded off the top and bottom, so that the angle of 55° gives for the actual depth rather more than three-fifths and less than two-thirds of the pitch.

In the *American (Sellers')* thread (fig. 2) the angle made by the two sides of the triangle is 60°, and one-eighth of the depth is cut off the top and bottom of the triangle, leaving a flat one-eighth of the pitch in width.

In the *Swiss* or *Thury* thread (fig. 3) the angle made by the two sides of the thread is 47½° and one-fifth of the depth is rounded off the top and bottom of the triangle.

For screw-plate Swiss threads the depth of the thread is three-fifths the length of the pitch.

The top of the ridge is formed by a circular arc, the radius of which equals one-sixth of the pitch. The space between two consecutive ridges is formed at the side next the core by a circular arc, the radius of which equals one-fifth of the pitch.

The Swiss thread for screws made in the lathe is the *Steinlen* thread (fig. 4). The angle made by the two sides of the thread is 58° 8′, and the radius of the connecting arcs is ·1011 of the pitch. The generating figure is an isosceles triangle, of which the base and the height are equal to the pitch. One-eighth of the depth is rounded off the top and bottom of the triangle.

The *British Association* thread is similar to that of Thury, except that the top and bottom of the threads are both rounded to the same radius, namely, two-elevenths of the pitch.

Screw Threads

Whitworth's Standard Threads.

Dimensions in Inches.				Threads per Inch.	Pitch in Millimetres.	Dimensions in Inches.				Threads per Inch.	Pitch in Millimetres.
Outside Diameter.	Diameter at Bottom of Thread.	Size for Tapping.	Pitch.			Outside Diameter.	Diameter at Bottom of Thread.	Size for Tapping.	Pitch.		
$\frac{1}{8}$	·0293	—	·0066	150	·169	$2\frac{3}{8}$	2·054	$2\frac{1}{16}$	·25	4	6·35
$\frac{1}{16}$	·0413	$\frac{3}{64}$	·0166	60	·4233	$2\frac{1}{2}$	2·18	$2\frac{3}{16}$	·25	4	6·35
$\frac{3}{32}$	·0671	$\frac{5}{64}$	·0208	48	·529	$2\frac{5}{8}$	2·304	$2\frac{5}{16}$	·25	4	6·35
$\frac{1}{8}$	·093	$\frac{9}{32}$	·025	40	·635	$2\frac{3}{4}$	2·384	$2\frac{3}{8}$	·2857	$3\frac{1}{2}$	7·257
$\frac{5}{32}$	·112	$\frac{1}{8}$	·0312	32	·793	$2\frac{7}{8}$	2·509	$2\frac{7}{16}$	·2857	$3\frac{1}{2}$	7·257
$\frac{3}{16}$	·134	$\frac{9}{64}$	·0416	24	1·06	3	2·634	$2\frac{1}{2}$	·2857	$3\frac{1}{2}$	7·257
$\frac{7}{32}$	·165	$\frac{11}{64}$	·0416	24	1·06	$3\frac{1}{8}$	2·759	$2\frac{9}{16}$	·2857	$3\frac{1}{2}$	7·257
$\frac{1}{4}$	·186	$\frac{5}{16}$	·05	20	1·27	$3\frac{1}{4}$	2·856	$2\frac{5}{8}$	·3077	$3\frac{1}{4}$	7·815
$\frac{5}{16}$	·241	$\frac{1}{4}$	·0555	18	1·41	$3\frac{3}{8}$	2·98	$2\frac{3}{4}$	·3077	$3\frac{1}{4}$	7·815
$\frac{3}{8}$	·295	$\frac{19}{64}$	·0625	16	1·587	$3\frac{1}{2}$	3·105	$2\frac{7}{8}$	·3077	$3\frac{1}{4}$	7·815
$\frac{7}{16}$	·346	$\frac{23}{64}$	·0714	14	1·814	$3\frac{5}{8}$	3·23	$3\frac{1}{16}$	·3077	$3\frac{1}{4}$	7·815
$\frac{1}{2}$	·393	$\frac{13}{32}$	·0833	12	2·116	$3\frac{3}{4}$	3·32	$2\frac{11}{16}$	·3333	3	8·466
$\frac{9}{16}$	·455	$\frac{15}{32}$	·0833	12	2·116	$3\frac{7}{8}$	3·448	$3\frac{1}{16}$	·3333	3	8·466
$\frac{5}{8}$	·508	$\frac{33}{64}$	·0909	11	2·39	4	3·573	$3\frac{3}{16}$	·3333	3	8·466
$\frac{11}{16}$	·571	$\frac{37}{64}$	·0909	11	2·39	$4\frac{1}{8}$	3·698	$3\frac{5}{16}$	·3333	3	8·466
$\frac{3}{4}$	·622	$\frac{5}{8}$	·1	10	2·54	$4\frac{1}{4}$	3·804	$3\frac{7}{16}$	·3478	$2\frac{7}{8}$	8·834
$\frac{13}{16}$	·684	$\frac{41}{64}$	·1	10	2·54	$4\frac{3}{8}$	3·93	$3\frac{9}{16}$	·3478	$2\frac{7}{8}$	8·834
$\frac{7}{8}$	·732	$\frac{47}{64}$	·111	9	2·822	$4\frac{1}{2}$	4·054	$4\frac{1}{16}$	·3478	$2\frac{7}{8}$	8·834
$\frac{15}{16}$	·795	$\frac{51}{64}$	·111	9	2·822	$4\frac{5}{8}$	4·18	$4\frac{3}{16}$	·3478	$2\frac{7}{8}$	8·834
1	·840	$\frac{7}{8}$	·125	8	3·175	$4\frac{3}{4}$	4·284	$4\frac{1}{4}$	·3636	$2\frac{3}{4}$	9·236
$1\frac{1}{8}$	·942	$\frac{61}{64}$	·1428	7	3·628	$4\frac{7}{8}$	4·409	$4\frac{5}{16}$	·3636	$2\frac{3}{4}$	9·236
$1\frac{1}{4}$	1·067	$1\frac{1}{64}$	·1428	7	3·628	5	4·534	$4\frac{7}{16}$	·3636	$2\frac{3}{4}$	9·236
$1\frac{3}{8}$	1·161	$1\frac{1}{16}$	·1666	6	4·233	$5\frac{1}{8}$	4·66	$4\frac{9}{16}$	·3636	$2\frac{3}{4}$	9·236
$1\frac{1}{2}$	1·286	$1\frac{3}{64}$	·1666	6	4·233	$5\frac{1}{4}$	4·762	$4\frac{11}{16}$	·3809	$2\frac{5}{8}$	9·676
$1\frac{5}{8}$	1·368	$1\frac{1}{8}$	·2	5	5·08	$5\frac{3}{8}$	4·887	$4\frac{13}{16}$	·3809	$2\frac{5}{8}$	9·676
$1\frac{3}{4}$	1·494	$1\frac{1}{8}$	·2	5	5·08	$5\frac{1}{2}$	5·012	$5\frac{1}{16}$	·3809	$2\frac{5}{8}$	9·676
$1\frac{7}{8}$	1·59	$1\frac{1}{2}\frac{5}{3}$	·2222	$4\frac{1}{2}$	5·644	$5\frac{5}{8}$	5·137	$5\frac{9}{16}$	·3809	$2\frac{5}{8}$	9·676
2	1·715	$1\frac{2}{3}\frac{5}{2}$	·2222	$4\frac{1}{2}$	5·644	$5\frac{3}{4}$	5·24	$5\frac{1}{4}$	·4	$2\frac{1}{2}$	10·16
$2\frac{1}{8}$	1·84	$1\frac{2}{3}\frac{7}{2}$	·2222	$4\frac{1}{2}$	5·644	$5\frac{7}{8}$	5·362	$5\frac{3}{8}$	·4	$2\frac{1}{2}$	10·16
$2\frac{1}{4}$	1·93	$1\frac{1}{16}$	·25	4	6·35	6	5·487	$5\frac{1}{2}$	·4	$2\frac{1}{2}$	10·16

British Standard Whitworth Screw Threads.*

(No. 92—1919.) (*Abstract.*)

1	2	3	4	5	6	7	8	9	10	11
Nominal Dia. of Screw.	No. of Threads per in.	Pitch.	Depth of Thread.	Full Diameter.		Effective Diameter.		Core Diameter.		Cross Sectional Area at Bottom of Thread of Bolt.
				Bolt.	Nut.	Bolt.	Nut.	Bolt.	Nut.	
In.		In.	In.	In.	In.	In.	In.	In.	In.	Sq. In.
$\frac{1}{4}$	20	·05000	·03200	·2500	·2520	·2180	·2200	·1860	·1880	·0272
$\frac{5}{16}$	18	·05556	·03555	·3125	·3145	·2769	·2789	·2414	·2434	·0458
$\frac{3}{8}$	16	·06250	·04000	·3750	·3770	·3350	·3370	·2950	·2970	·0683
$\frac{7}{16}$	14	·07143	·04575	·4375	·4395	·3918	·3938	·3460	·3480	·0940
$\frac{1}{2}$	12	·08333	·05335	·5000	·5020	·4466	·4486	·3933	·3953	·1215
$\frac{9}{16}$	12	·08333	·05335	·5625	·5645	·5091	·5111	·4558	·4578	·1632
$\frac{5}{8}$	11	·09091	·05820	·6250	·6270	·5668	·5688	·5086	·5106	·2032
$\frac{3}{4}$	10	·10000	·06405	·7500	·7520	·6860	·6880	·6219	·6239	·3038
$\frac{7}{8}$	9	·11111	·07115	·8750	·8770	·8039	·8059	·7327	·7347	·4216
1	8	·12500	·08005	1·0000	1·0020	·9220	·9240	·8399	·8419	·5540
$1\frac{1}{8}$	7	·14286	·09150	1·1250	1·1270	1·0335	1·0355	·9420	·9440	·6969
$1\frac{1}{4}$	7	·14286	·09150	1·2500	1·2520	1·1585	1·1605	1·0670	1·0690	·8942
$1\frac{1}{2}$	6	·16667	·10670	1·5000	1·5020	1·3933	·3953	1·2866	1·2886	1·3001

* By permission of the British Engineering Standards Association.

BRITISH STANDARD WHITWORTH SCREW THREADS—*continued.*

1	2	3	4	5	6	7	8	9	10	11
Nominal Dia. of Screw.	No. of Threads per in.	Pitch.	Depth of Thread.	Full Diameter.		Effective Diameter.		Core Diameter.		Cross Sectional Area at Bottom of Thread of Bolt.
				Bolt.	Nut.	Bolt.	Nut.	Bolt.	Nut.	
In.		In.	In.	In.	In.	In.	In.	In.	In.	Sq. In.
1¾	5	·20000	·12805	1·7500	1·7520	1·6219	1·6239	1·4939	1·4959	1·7528
2	4·5	·22222	·14230	2·0000	2·0020	1·8577	1·8597	1·7154	1·7174	2·3111
2¼	4	·25000	·16010	2·2500	2·2520	2·0899	2·0919	1·9298	1·9318	2·9249
2½	4	·25000	·16010	2·5000	2·5020	2·3399	2·3419	2·1798	2·1818	3·7318
2¾	3·5	·28571	·18295	2·7500	2·7520	2·5670	2·5690	2·3841	2·3861	4·4641
3	3·5	·28571	·18295	3·0000	3·0020	2·8170	2·8190	2·6341	2·6361	5·4496
3¼	3·25	·30769	·19700	3·2500	3·2520	3·0530	3·0550	2·8560	2·8580	6·4063
3½	3·25	·30769	·19700	3·5000	3·5020	3·3030	3·3050	3·1060	3·1080	7·5769
3¾	3	·33333	·21345	3·7500	3·7520	3·5366	3·5386	3·3231	3·3251	8·6732
4	3	·33333	·21345	4·0000	4·0020	3·7866	3·7886	3·5731	3·5751	10·0272
4½	2·875	·34783	·22270	4·5000	4·5020	4·2773	4·2793	4·0546	4·0566	12·9118
5	2·75	·36364	·23285	5·0000	5·0020	4·7672	4·7692	4·5343	4·5363	16·1477
5½	2·625	·38095	·24395	5·5000	5·5020	5·2561	5·2581	5·0121	5·0141	19·7301
6	2·5	·40000	·25615	6·0000	6·0020	5·7439	5·7459	5·4877	5·4897	23·6521

BRITISH STANDARD FINE (B.S.F.) SCREW THREADS AND THEIR TOLERANCES.*
(No. 84—1918.) (*Abstract.*)

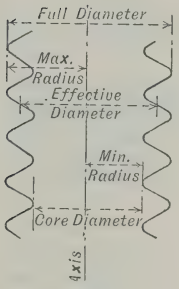


FIG. 5.

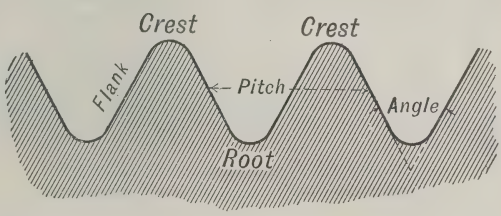


FIG. 6.

Effective Diameter of a Screw.—The effective diameter of a perfect screw having a single thread is the length of a line drawn through the axis and at right angles to it, measured between the points where the line cuts the flanks of the thread (fig. 5).

Core Diameter.—The core diameter is twice the minimum radius of a screw, measured at right angles to the axis (fig. 5).

NOTE.—From fig. 5 it will be seen that the 'core diameter' of the male thread is measured between the roots of the thread; it should be borne in mind, however, that the 'core diameter' of the female thread, being approximately the same dimension, is measured between the crests of the thread.

Full Diameter.—The full diameter is twice the maximum radius of a screw, measured at right angles to the axis (fig. 5).

Crest.—The crest is the prominent part of the thread, whether of the male screw or of the female screw (fig. 6).

Root.—The root is the bottom of the groove of the thread, whether of the male screw or of the female screw (fig. 6).

* By permission of the British Engineering Standards Association.

Flank of Thread.—The flank of the thread is the straight part of the thread which connects the crests and roots (fig. 6).

Angle of Thread.—The angle of the thread is the angle between the slopes, measured in the axial plane (fig. 6).

Pitch.—The pitch is the distance in inches or millimetres measured along a line parallel to the axis of the screw between the point where it cuts any thread of the screw and the point at which it next meets the corresponding part of the same thread (fig. 6). *The reciprocal of the pitch* measures the number of turns per inch or millimetre, as the case may be.

RECOMMENDATIONS.

Form of Thread.

1. The Association recommend that the Whitworth form of thread, as defined in clause 2, be adopted on all screws of $\frac{7}{32}$ in. diameter and above, having British Standard Fine Pitches.

Definition of Whitworth Form of Thread.

2. The following defines the Whitworth thread :—

The *Whitworth Form of Thread* is one in which the angle between the flanks, measured in the axial plane, is 55° ; one-sixth of the sharp triangle is truncated at top and bottom, the threads being rounded equally at crests and roots to a radius which is therefore 0·137329 times the pitch ; the depth of the thread is 0·640327 times the pitch.

TOLERANCES.

—	Full Diameter.	Effective Diameter.	Core Diameter.
Bolt	$3\times\cdot01\sqrt{p}$	$2\times\cdot01\sqrt{p}$	$4\times\cdot01\sqrt{p}$
Nut	$4\times\cdot01\sqrt{p}$	$2\times\cdot01\sqrt{p}$	$3\times\cdot01\sqrt{p}$

The corresponding tolerances on grade and play are :—

$$\text{Grade}=4\times\cdot01\sqrt{p}\,;\quad \text{Play}=2\times\cdot01\sqrt{p}.$$

In addition, a uniform allowance of 2 thousandths of an inch has been made throughout between the maximum bolt and the minimum nut.

For much work the above tolerances are too large. A second series of tables have therefore been formed by halving the tolerances everywhere. The Association recommend that threads made in accordance with these dimensions be known as ' British Standard Fine Threads (Close Fits).'

The basis of calculation of these tolerances is given below.

—	Full Diameter.	Effective Diameter.	Core Diameter.
Bolt	$1\cdot5\times\cdot01\sqrt{p}$	$\cdot01\sqrt{p}$	$2\times\cdot01\sqrt{p}$
Nut	$2\times\cdot01\sqrt{p}$	$\cdot01\sqrt{p}$	$1\cdot5\times\cdot01\sqrt{p}$

The corresponding tolerances and allowance are :—

$$\text{Grade tolerance}=2\times\cdot01\sqrt{p}\,;\quad \text{play tolerance}=\cdot01\sqrt{p}\,;\quad \text{allowance}=0\cdot002\text{ in.}$$

The pitches given in the following table are based on the following formulæ :—

For sizes up to and including screw threads of 1 inch diameter, $P=\frac{\sqrt[3]{d^2}}{10}$.

For sizes above 1 inch diameter, up to and including screw threads of 6 ins. diameter, $P=\frac{\sqrt[3]{d^4}}{10}$

$$\text{Pitch}=\frac{1}{\text{No. of threads per inch}}\,;\ \text{depth of thread}=0\cdot640327\times\text{pitch}\,;$$

$$\text{Rounding at crest and root}=0\cdot137329\times\text{pitch}.$$

THEORETICAL DIMENSIONS OF BRITISH STANDARD FINE SCREW THREADS.

1	2	3	4	5	6	7	8	9	10	11	12
Nominal Diameter of Screw.	No. of Threads per inch.	Pitch.	Depth of Thread.	Full Diameter.		Effective Diameter.		Core Diameter.		Cross Sectional Area at Bottom of Thread of Bolt.	Nominal Diameter of Screw.
				Bolt.	Nut.	Bolt.	Nut.	Bolt.	Nut.		
in.		in.	in.	in.	in.	in.	in.	in.	in.	sq. in.	in.
$\frac{7}{8}$	28	·03571	·02285	·2188	·2208	·1960	·1980	·1731	·1751	·0235	$\frac{7}{8}$
$\frac{15}{16}$	26	·03846	·02465	·2500	·2520	·2254	·2274	·2007	2027	·0316	$\frac{15}{16}$
$\frac{29}{32}$	26	·03846	·02465	·2813	·2833	·2566	·2586	·2320	2340	·0423	$\frac{29}{32}$
$\frac{15}{16}$	22	·04545	·02910	·3125	·3145	·2834	·2854	·2543	·2563	·0508	$\frac{15}{16}$
$\frac{15}{16}$	20	·05000	·03200	·3750	·3770	·3430	·3450	·3110	·3130	·0760	$\frac{15}{16}$
$\frac{15}{16}$	18	·05556	·03555	·4375	·4395	·4019	·4039	·3664	·3684	·1054	$\frac{15}{16}$
$\frac{15}{16}$	16	·06250	·04000	·5000	·5020	·4600	·4620	·4200	·4220	·1385	$\frac{15}{16}$
$\frac{15}{16}$	16	·06250	·04000	·5625	·5645	·5225	·5245	·4825	·4845	·1828	$\frac{15}{16}$
$\frac{15}{16}$	14	·07143	·04575	·6250	·6270	5793	·5813	·5335	·5355	·2235	$\frac{15}{16}$
$\frac{15}{16}$	14	·07143	·04575	·6875	·6895	·6418	·6438	·5960	·5980	·2790	$\frac{15}{16}$
$\frac{15}{16}$	12	·08333	·05335	·7500	·7520	·6966	·6986	·6433	·6453	·3250	$\frac{15}{16}$
$\frac{15}{16}$	12	·08333	·05335	·8125	·8145	·7591	·7611	·7058	·7078	·3913	$\frac{15}{16}$
1	11	·09091	·05820	·8750	·8770	·8168	·8188	·7586	·7606	·4520	1
$1\frac{1}{8}$	9	·11111	·07115	1·1250	1·1270	1·0539	1·0559	·9827	·9847	·7585	$1\frac{1}{8}$
$1\frac{1}{4}$	9	·11111	·07115	1·2500	1·2520	1·1789	1·1809	1·1077	1·1097	·9637	$1\frac{1}{4}$
$1\frac{3}{8}$	8	·12500	·08005	1·3750	1·3770	1·2950	1·2970	1·2149	1·2169	1·1592	$1\frac{3}{8}$
$1\frac{1}{2}$	8	·12500	·08005	1·5000	1·5020	1·4200	1·4220	1·3399	1·3419	1·4100	$1\frac{1}{2}$
$1\frac{5}{8}$	8	·12500	·08005	1·6250	1·6270	1·5450	1·5470	1·4649	1·4669	1·6854	$1\frac{5}{8}$
$1\frac{3}{4}$	7	·14286	·09150	1·7500	1·7520	1·6585	1·6605	1·5670	1·5690	1·9285	$1\frac{3}{4}$
2	7	·14286	·09150	2·0000	2·0020	1·9085	1·9105	1·8170	1·8190	2·5930	2
$2\frac{1}{4}$	6	·16667	·10670	2·2500	2·2520	2·1433	2·1453	2·0366	2·0386	3·2577	$2\frac{1}{4}$
$2\frac{1}{2}$	6	·16667	·10670	2·5000	2·5020	2·3933	2·3953	2·2866	2·2886	4·1065	$2\frac{1}{2}$
$2\frac{3}{4}$	6	·16667	·10670	2·7500	2·7520	2·6433	2·6453	2·5366	2·5386	5·0535	$2\frac{3}{4}$
3	5	·20000	·12805	3·0000	3·0020	2·8719	2·8739	2·7439	2·7459	5·9133	3

BRITISH ASSOCIATION SCREW THREADS.

No.	Approximate Dimensions in Decimals of an inch.		Threads per Inch.	Exact Dimensions in Millimetres.		No.	Approximate Dimensions in Decimals of an Inch.		Threads per Inch.	Exact Dimensions in Millimetres.	
	Diam.	Pitch.		Diam.	Pitch.		Diam.	Pitch.		Diam.	Pitch.
25	·010	·0028	353	0·25	0·072	12	·051	·0110	90·7	1·3	0·28
24	·011	·0031	317	0·29	0·080	11	·059	·0122	81·9	1·5	0·31
23	·013	·0035	285	0·33	0·089	10	·067	·0138	72·6	1·7	0·35
22	·015	·0039	259	0·37	0·098	9	·075	·0154	65·1	1·9	0·39
21	·017	·0043	231	0·42	0·11	8	·087	·0169	59·1	2·2	0·43
20	·019	·0047	212	0·48	0·12	7	·098	·0189	52·9	2·5	0·48
19	·021	·0055	181	0·54	0·14	6	·110	·0209	47·9	2·8	0·53
18	·024	·0059	169	0·62	0·15	5	·126	·0232	43·0	3·2	0·59
17	·028	·0067	149	0·70	0·17	4	·142	·0260	38·5	3·6	0·66
16	·031	·0075	134	0·79	0·19	3	·161	·0287	34·8	4·1	0·73
15	·035	·0083	121	0·90	0·21	2	·185	·0319	31·4	4·7	0·81
14	·039	·0091	110	1·0	0·23	1	·209	·0354	28·2	5·3	0·90
13	·047	·0098	101	1·2	0·25	0	·236	·0394	25·4	6·0	1·00

British Standard Brass Thread.

This is a pipe thread system adopted for thin brass tubing, gas burner fittings, and general brass work; it has a Whitworth thread of a constant pitch of $\frac{1}{36}$-in., or 26 threads per inch.

Diameter. In.	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$
Threads per inch .	26 throughout		—	—	—	—
Depth of Thread In.	0·02461 throughout		—	—	—	—
Core Diameter In.	0·0758	0·2008	0·3258	0·4508	0·5758	0·700
Tapping Size . In.	No. 47	No. 6	Q	$\frac{29}{64}$	$\frac{37}{64}$	$\frac{45}{64}$

Diameter. In.	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	—
Threads per Inch .	26 throughout		—	—	—	—
Depth of Thread In.	0·02461 throughout		—	—	—	—
Core Diameter In.	0·8258	0·9508	1·0758	1·2008	1·4508	—
Tapping Size . In.	$\frac{53}{64}$	$\frac{61}{64}$	$1\frac{5}{64}$	$1\frac{13}{64}$	$1\frac{29}{64}$	—

Whitworth's Standard Screws for Watch and Instrument Makers.

No. denoting Thousandths of an Inch in Diameter.	Threads per Inch.	No. denoting Thousandths of an Inch in Diameter.	Threads per Inch.	No. denoting Thousandths of an Inch in Diameter.	Threads per Inch.	No. denoting Thousandths of an Inch in Diameter.	Threads per Inch.
10	400	18 & 19	250	34	150	65	80
11	400	20	210	36	150	70	80
12	350	22	210	38	120	75	80
13	350	24	210	40	120	80	60
14	300	26	180	45	120	85	60
15	300	28	180	50	100	90	60
16	300	30	180	55	100	95	50
17	250	32	150	60	100	100	50

Swiss Screw Threads. (*Thury.*)

No.	Exact Dimensions in Millimetres.		Ratio of Successive Diameters	Approx. Diam. in Decimals of an Inch.	Threads per Inch.	No.	Exact Dimensions in Millimetres.		Ratio of Successive Diameters	Approx. Diam. in Decimals of an Inch.	Threads per Inch.
	Pitch.	External Diameter.					Pitch.	External Diameter.			
25	0·0718	0·254	0·879	0·0100	353·70	2	0·810	4·66	0·881	0·1835	31·35
24	0·0798	0·289	0·881	0·0114	318·30	1	0·900	5·29	0·882	0·2083	28·22
23	0·0886	0·328	0·882	0·0129	286·65	0	1	6	0·881	0·2362	25·40
22	0·0985	0·372	0·872	0·0146	255·32	— 1	1·11	6·81	0·881	0·2681	22·85
21	0·109	0·426	0·889	0·0168	233·02	— 2	1·23	7·73	0·881	0·3043	20·65
20	0·122	0·479	0·882	0·0189	208·19	— 3	1·37	8·77	0·881	0·3453	18·54
19	0·135	0·543	0·882	0·0214	188·40	— 4	1·52	9·95	0·880	0·3917	16·73
18	0·150	0·616	0·881	0·0243	169·33	— 5	1·69	11·3	0·883	0·4449	15·02
17	0·167	0·699	0·880	0·0275	152·32	— 6	1·88	12·8	0·883	0·5039	13·48
16	0·185	0·794	0·881	0·0313	137·29	— 7	2·09	14·5	0·879	0·5709	12·15
15	0·206	0·901	0·883	0·0355	123·30	— 8	2·32	16·5	0·882	0·6496	10·94
14	0·229	1·02	0·879	0·0402	110·91	— 9	2·58	18·7	0·882	0·7362	9·84
13	0·254	1·16	0·879	0·0457	100·00	—10	2·87	21·2	0·880	0·8346	8·85
12	0·282	1·32	0·886	0·0520	90·07	—11	3·19	24·1	0·880	0·9488	7·97
11	0·314	1·49	0·909	0·0587	80·89	—12	3·54	27·4	0·884	1·0787	7·17
10	0·349	1·64	0·854	0·0646	72·77	—13	3·93	31·0	0·881	1·2048	6·46
9	0·387	1·92	0·881	0·0756	65·63	—14	4·37	35·2	0·880	1·3858	5·81
8	0·430	2·18	0·879	0·0858	59·06	—15	4·86	40·0	0·881	1·5748	5·22
7	0·478	2·48	0·883	0·0976	53·13	—16	5·40	45·4	0·882	1·7874	4·84
6	0·531	2·81	0·881	0·1106	47·80	—17	6·00	51·5	0·882	2·0276	4·23
5	0·590	3·19	0·881	0·1256	43·05	—18	6·66	58·4	0·881	2·2992	3·82
4	0·650	3·62	0·881	0·1425	38·71	—19	7·40	66·3	0·882	2·6103	3·43
3	0·729	4·11	0·882	0·1618	34·84	—20	8·23	75·2	0·882	2·9607	3·08

British Association Screw Threads.

These screw threads (p. 1143) have been standardised by the British Engineering Standards Association, and the latest dimensions are given in Specification No. 93—1919.

French Standard Threads.

Diam. in Millimètres.	Pitch in Millimètres.			Diam. in Millimètres.	Pitch in Millimètres.		
	Cail or Railway.	Ducommun.	Armengaud's.		Cail or Railway.	Ducommun.	Armengaud's.
3	—	·5	—	25	3	3	3
4	—	·75	—	28	3	3	—
5	1	·75	1·4	30	3	3·5	3·4
6	1	1	—	32	3	3·5	—
7	1·25	1·2	—	35	3·5	4	3·8
7·5	—	—	1·6	38	3·5	4	—
8	1·25	1·25	—	40	4	4	4·2
9	1·5	1·5	—	42	4	4·5	—
10	1·5	1·5	1·8	45	4	4·5	4·6
12	1·5	1·75	—	47	4·5	5	—
12·5	—	—	2	50	4·5	5	5
14	2	—	—	55	—	5	5·4
15	2	2	2·2	60	—	6	5·8
17·5	—	—	2·4	65	—	6	6·2
18	2	2·5	—	70	—	7	6·6
20	2	2·5	2·6	75	—	7	7
22·5	—	—	2·8	80	—	7	7·4
23	2·5	3	—				

American Standard Threads.

(*Sellers'*.)

Dimensions in Inches.						Dimensions in Inches.					
Outside Diameter.	Diameter at Bottom of Thread.	Size for Tapping.	Pitch.	Width of Flat.	Threads per Inch.	Pitch in Millimètres.	Outside Diameter.	Diameter at Bottom of Thread.	Size for Tapping.	Pitch.	Width of Flat.	Threads per Inch.	Pitch in Millimètres.
$\frac{1}{4}$	·185	$\frac{3}{16}$	·05	·0062	20	1·27	2	1·711	$1\frac{23}{32}$	·2222	·0277	$4\frac{1}{2}$	5·644
$\frac{5}{16}$	·24	$\frac{1}{4}$	·0555	·0069	18	1·41	$2\frac{1}{4}$	1·961	$1\frac{31}{32}$	·2222	·0277	$4\frac{1}{2}$	5·644
$\frac{3}{8}$	·294	$\frac{9}{64}$	·0625	·0078	16	1·587	$2\frac{1}{2}$	2·175	$2\frac{3}{16}$	·25	·0313	4	6·35
$\frac{7}{16}$	·345	$\frac{5}{32}$	·0714	·0089	14	1·814	$2\frac{3}{4}$	2·425	$2\frac{7}{16}$	·25	·0313	4	6·35
$\frac{1}{2}$	·4	$\frac{13}{64}$	·0769	·0096	13	1·95	3	2·628	$2\frac{11}{16}$	·2857	·0357	$3\frac{1}{2}$	7·257
$\frac{9}{16}$	·454	$\frac{15}{32}$	·0833	·0104	12	2·116	$3\frac{1}{4}$	2·878	$2\frac{27}{32}$	·2857	·0357	$3\frac{1}{2}$	7·257
$\frac{5}{8}$	·506	$\frac{33}{64}$	·0909	·0114	11	2·39	$3\frac{1}{2}$	3·1	$3\frac{1}{32}$	·3077	·0384	$3\frac{1}{4}$	7·815
$\frac{11}{16}$	·62	$\frac{17}{32}$	·1	·0125	10	2·54	$3\frac{3}{4}$	3·317	$3\frac{21}{32}$	·3333	·0416	3	8·466
$\frac{3}{4}$	·731	$\frac{47}{64}$	·1111	·0139	9	2·822	4	3·567	$3\frac{27}{32}$	·3333	·0416	3	8·466
1	·837	$\frac{29}{32}$	·125	·0156	8	3·175	$4\frac{1}{4}$	3·798	$3\frac{13}{16}$	·3478	·0434	$2\frac{7}{8}$	8·834
$1\frac{1}{8}$	·939	$\frac{61}{64}$	·1428	·0178	7	3·628	$4\frac{1}{2}$	4·027	$4\frac{1}{32}$	·3636	·0454	$2\frac{3}{4}$	9·236
$1\frac{1}{4}$	1·064	$1\frac{5}{64}$	·1428	·0178	7	3·628	$4\frac{3}{4}$	4·255	$4\frac{17}{32}$	·3809	·0478	$2\frac{5}{8}$	9·676
$1\frac{3}{8}$	1·158	$1\frac{13}{64}$	·1666	·0208	6	4·233	5	4·48	$4\frac{31}{64}$	·4	·05	$2\frac{1}{2}$	10·16
$1\frac{1}{2}$	1·283	$1\frac{19}{64}$	·1666	·0208	6	4·233	$5\frac{1}{4}$	4·73	$4\frac{47}{64}$	·4	·05	$2\frac{1}{2}$	10·16
$1\frac{5}{8}$	1·389	$1\frac{25}{64}$	·1818	·0227	$5\frac{1}{2}$	4·62	$5\frac{1}{2}$	4·953	$4\frac{1}{64}$	·421	·0526	$2\frac{3}{8}$	10·68
$1\frac{3}{4}$	1·49	$1\frac{3}{4}$	·2	·025	5	5·08	$5\frac{3}{4}$	5·203	$5\frac{13}{64}$	·421	·0526	$2\frac{3}{8}$	10·68
$1\frac{7}{8}$	1·615	$1\frac{7}{8}$	·2	·025	5	5·08	6	5·423	$5\frac{7}{16}$	·444	·0555	$2\frac{1}{4}$	11·29

Comparison of U.S. Standard and Whitworth Screw Threads.

The difference in form that exists between the Whitworth and the U.S. standard screw thread is shown in fig. 7. The comparison is based upon a screw with 1 thread per inch. Clearances and interferences for the two forms of threads having any other pitch may therefore be obtained by dividing the respective values given in the diagram by the number of threads per inch under consideration.

In order to fit a U.S. standard screw into a Whitworth tapped hole or nut, round the outside corners of the screw and change the outline of the thread below the pitch line by slightly filing the flanks. These changes are indicated in the figure by the sectional areas. If it is desired to fit a Whitworth screw into a U.S. standard tapped hole or nut, drill out the hole or nut, using a drill which is larger than the tap drill required for an equal size U.S. standard tap, as shown by

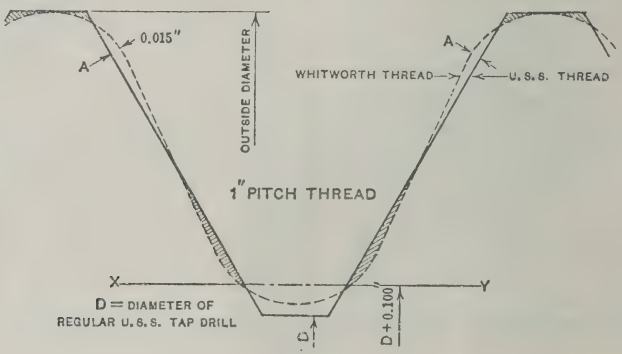


FIG. 7.

the line XY in the diagram. For 1 thread per inch, this increase in diameter of the tap drill will be 0·100 inch, and for other numbers of threads the amount of increase will be found by dividing 0·100 inch by the number of threads per inch. Thus, for 10 threads per inch, 0·100 ÷ 10 = 0·010 inch. The flanks of the Whitworth thread above the pitch line are then filed slightly as indicated at A in the illustration.

The methods mentioned are recommended only for fastening screws and ordinary threaded parts, and should not be employed where accurate threads are required.

(*G. R. R. in* '*Machinery.*')

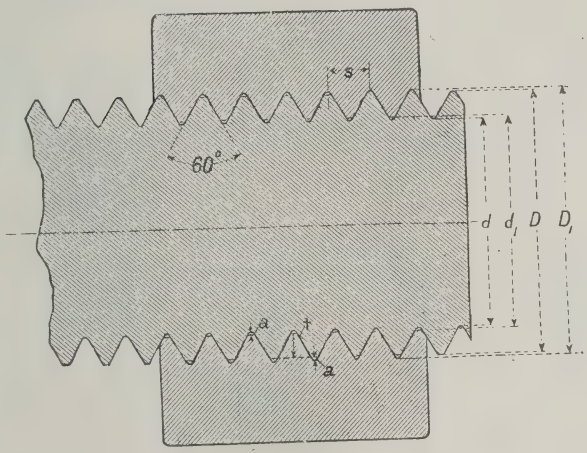
International System of Metrical Screw Threads.

(*Système International.*)

Dia. of Bolt in mm.	Pitch in mm.	Dia. of Bolt in mm.	Pitch in mm.	Dia. of Bolt in mm.	Pitch in mm.	Dia. of Bolt in mm.	Pitch in mm.	Dia. of Bolt in mm.	Pitch in mm.
3	0·55	8	1·25	18	2·5	36	4	60	5·5
3·5	0·55	9	1·25	20	2·5	39	4	64	6
4	0·7	10	1·5	22	2·5	42	4·5	68	6
4·5	0·7	11	1·5	24	3	45	4·5	72	6·5
5	0·85	12	1·75	27	3	48	5	76	6·5
6	1	14	2	30	3·5	52	5	80	7
7	1	16	2	33	3·5	56	5·5		

Fig. 8 shows the shape and proportions of the threads of the International System of metrical screw threads, also the maximum and minimum deviations allowed from the normal.

INTERNATIONAL SYSTEM OF METRICAL SCREW THREADS.



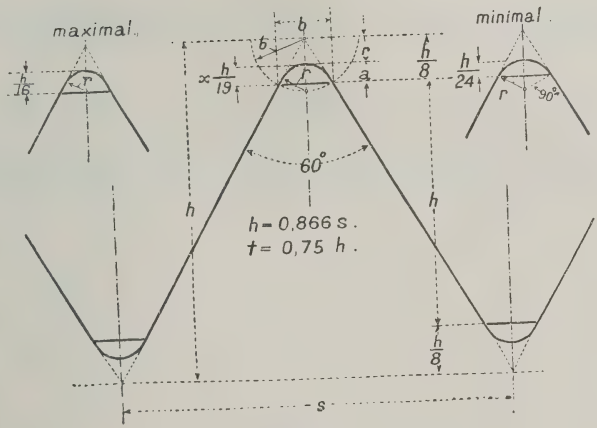


FIG. 8.

Wood Screws.

(Sometimes termed 'Screw Nails.')

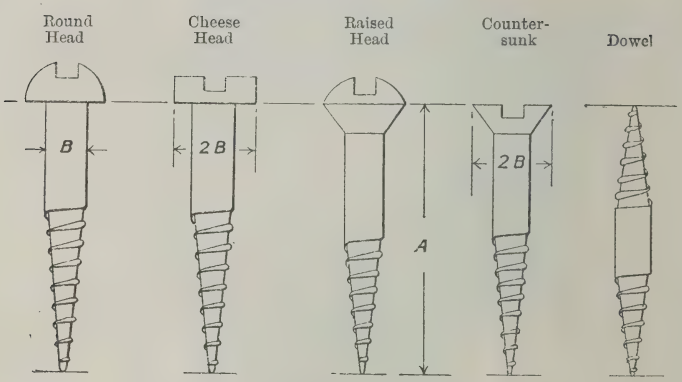


FIG. 9.

Wood screws are measured, as regards length (A), from the point to the portion normally flush with the surface of the material into which they are driven. Dowel screws are measured from point to point.

The number (size or thickness) (B) refers to the diameter of the shank or neck, and for Nettlefold's manufacture is as follows :—

Number.	Diameter.	Number.	Diameter.	Number.	Diameter.	Number.	Diameter.
	In.		In.		In.		In.
1	·066	9	·178	17	·290	25	·402
2	·080	10	·192	18	·304	26	·416
3	·094	11	·206	19	·318	27	·430
4	·108	12	·220	20	·332	28	·444
5	·122	13	·234	21	·346	29	·458
6	·136	14	·248	22	·360	30	·472
7	·150	15	·262	23	·374	31	·486
8	·164	16	·276	24	·388	32	·500

Formula for ascertaining diameter of screw :—

(·014 × No.) + ·052 in. = diameter.

Example :—(·014 × 16) + ·052 in. = ·224 + ·052 = ·276 diameter of No. 16.

The diameter of the head may be taken as twice the thickness of the neck.

Numerous lengths are obtainable in each size from ¼ in. in the smaller sizes to 9 ins. in No. 32.

Larger sizes are made to order but not gauged, and most sizes are obtainable in iron or brass.

(*H. A. Miles.*)

TABLE OF WOOD SCREWS.

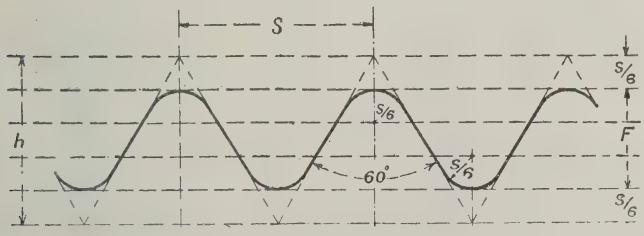
Number.	Clear Drill.	Diameter.	Diameter of Head.	Number.	Clear Drill.	Diameter.	Diameter of Head.
00	56	·046″	·120″	7	25	·149″	·300″
0	53	·059″	·140″	8	20	·161″	·330″
1	49	·073″	·160″	9	16	·171″	·365″
2	44	·086″	·175″	10	12	·189″	·390″
3	40	·098″	·200″	11	7	·201″	·420″
4	35	·110″	·225″	12	3	·213″	·450″
5	30	·128″	·250″	13	1	·228″	·475″
6	28	·140″	·275″	14	¼″	·250″	·500″

(*H. T. R. Grantham.*)

NOTE ON WOOD SCREWS.

Wood screws may be put into the hardest of wood by the following process: A screw of the same size to be used is ground down to the form of a half-round bit. A hole, equal in diameter to the size of the screw at the bottom of the thread, is drilled into the wood, and the half-round tap is screwed in. This cuts a good thread, into which the screw may easily be screwed. Before introducing the screw, give it a good greasing with tallow, so that no binding can take place.

CYCLE ENGINEERS' INSTITUTE STANDARD THREADS.



$h = 0{\cdot}866\ S.$ $F = 0{\cdot}5327\ S.$

FIG. 10.

The dimensions given below are for right-hand threads except those marked ** which are for left-hand threads only, and that marked * (i.e. $\frac{9}{16}$ in.) is for right or left hand threads.

Diameter in Inches. Decimals and Fractions.	No. of Threads per In.	Diameter in Inches. Decimals and Fractions.	No. of Threads per In.	Diameter in Inches. Decimals and Fractions.	No. of Threads per In.
·056 —	62	·154 —	40	·375 $\frac{3}{8}$	26
·064 —	62	·175 —	32	*·5625 $\frac{9}{16}$	20
·072 —	62	·1875 $\frac{3}{16}$	32	1·000 1	26
·080 —	62	·250 $\frac{1}{4}$	26	**1·290 —	24
·092 —	56	·266 —	26	1·370 —	24
·104 —	44	·281 —	26	**1·4375 $1\frac{7}{16}$	24
·125 $\frac{1}{8}$	40	·3125 $\frac{5}{16}$	26	1·5000 $1\frac{1}{2}$	24

Screw Heads.

BRITISH STANDARD HEADS FOR SMALL SCREWS.*

(No. 57—1920.) (*Abstract.*)

For all classes of head the diameter to be 1·75 times the full diameter of the thread.

The width of saw-cuts to be according to the formula,

$$S = \cdot 2\ D + \cdot 1m,$$

where

S = width of saw-cut, and D = full diameter of the thread in millimetres.

The standard depth of saw-cut to equal one-half the total depth of the head in all cases, such depth being measured at the centre of the head.

Pipe Threads.

WHITWORTH'S STANDARD THREADS FOR GAS AND WATER PIPES.

Dimensions in Inches.				Dimensions in Inches.			
Internal Diameter of Pipe.	External Diameter of Pipe.	Diameter at Bottom of Thread.	Number of Threads per Inch.	Internal Diameter of Pipe.	External Diameter of Pipe.	Diameter at Bottom of Thread.	Number of Threads per Inch.
$\frac{1}{8}$	·382	·336	28	$1\frac{7}{8}$	2·245	2·128	11
$\frac{1}{4}$	·518	·451	19	2	2·347	2·23	11
$\frac{3}{8}$	·656	·589	19	$2\frac{1}{8}$	2·467	2·351	11
$\frac{1}{2}$	·826	·734	14	$2\frac{1}{4}$	2·587	2·47	11
$\frac{5}{8}$	·902	·811	14	$2\frac{3}{8}$	2·794	2·678	11
$\frac{3}{4}$	1·04	·949	14	$2\frac{1}{2}$	3	2·882	11
$\frac{7}{8}$	1·189	1·097	14	$2\frac{5}{8}$	3·124	3·009	11
1	1·309	1·192	11	$2\frac{3}{4}$	3·247	3·13	11
$1\frac{1}{8}$	1·492	1·375	11	$2\frac{7}{8}$	3·367	3·251	11
$1\frac{1}{4}$	1·65	1·533	11	3	3·485	3·368	11
$1\frac{3}{8}$	1·745	1·628	11	$3\frac{1}{4}$	3·698	3·581	11
$1\frac{1}{2}$	1·882	1·765	11	$3\frac{1}{2}$	3·912	3·795	11
$1\frac{5}{8}$	2·022	1·965	11	$3\frac{3}{4}$	4·125	4·008	11
$1\frac{3}{4}$	2·16	2·042	11	4	4·340	4·223	11

* By permission of the British Engineering Standards Association.

BRITISH STANDARD PIPE THREADS.*
(No. 21—1909.) (*Abstract.*)

1	2	3	4	5	6	7	8	9	10	11
Nominal Bore of Tube.	Approximate Outside Diameter of Black Tube.	Gauge Diameter.	Depth of Thread.	Core Diameter, Measured at same Distance from End of Pipe as the Gauge Diameter.	Number of Threads per Inch.	Length of Thread.		Distance of Gauge Diameter from Pipe-end. (Class I. Taper Screw.)		
						L on Pipe-end. Min.	In Coupler. Min.	Standard.	Max.	Min.
Ins.	Ins.	Ins.	In.	Ins.		Ins.	Ins.	Ins.	Ins.	Ins.
$\frac{1}{8}$	$\frac{13}{32}$	·383	·0230	·337	28	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{5}{32}$ (·1563)	·18	·13
$\frac{1}{4}$	$\frac{17}{32}$	·518	·0335	·451	19	$\frac{7}{16}$	$\frac{7}{8}$	$\frac{3}{16}$ (·1875)	·22	·16
$\frac{3}{8}$	$\frac{11}{16}$	·656	·0335	·589	19	$\frac{1}{2}$	1	$\frac{1}{4}$ (·2500)	·29	·21
$\frac{1}{2}$	$\frac{27}{32}$	·825	·0455	·734	14	$\frac{5}{8}$	$1\frac{1}{4}$	$\frac{1}{4}$ (·2500)	29	·21
$\frac{5}{8}$	$\frac{15}{16}$	·902	·0455	·811	14	$\frac{5}{8}$	$1\frac{1}{4}$	$\frac{1}{4}$ (·2500)	·29	·21
$\frac{3}{4}$	$1\frac{1}{16}$	1·041	·0455	·950	14	$\frac{3}{4}$	$1\frac{1}{2}$	$\frac{3}{8}$ (·3750)	·44	·31
$\frac{7}{8}$	$1\frac{7}{32}$	1·189	·0455	1·098	14	$\frac{3}{4}$	$1\frac{1}{2}$	$\frac{3}{8}$ (·3750)	·44	·31
1	$1\frac{11}{32}$	1·309	·0580	1·193	11	$\frac{7}{8}$	$1\frac{5}{8}$	$\frac{3}{8}$ (·3750)	·44	·31
$1\frac{1}{4}$	$1\frac{11}{16}$	1·650	·0580	1·534	11	1	2	$\frac{1}{2}$ (·5000)	·58	·42
$1\frac{1}{2}$	$1\frac{23}{32}$	1·882	·0580	1·766	11	1	2	$\frac{1}{4}$ (·5000)	·58	·42
$1\frac{3}{4}$	$2\frac{3}{32}$	2·116	·0580	2·000	11	$1\frac{1}{8}$	$2\frac{1}{4}$	$\frac{5}{8}$ (·6250)	·73	·52
2	$2\frac{3}{8}$	2·347	·0580	2·231	11	$1\frac{1}{8}$	$2\frac{1}{4}$	$\frac{5}{8}$ (·6250)	·73	·52
$2\frac{1}{4}$	$2\frac{5}{8}$	2·587	·0580	2·471	11	$1\frac{1}{4}$	$2\frac{1}{2}$	$\frac{11}{16}$ (·6875)	·80	·57
$2\frac{1}{2}$	3	2·960	·0580	2·844	11	$1\frac{1}{4}$	$2\frac{1}{2}$	$\frac{11}{16}$ (·6875)	·80	·57
$2\frac{3}{4}$	$3\frac{1}{4}$	3·210	·0580	3·094	11	$1\frac{3}{8}$	$2\frac{3}{4}$	$\frac{13}{16}$ (·8125)	·95	·68
3	$3\frac{1}{2}$	3·460	·0580	3·344	11	$1\frac{3}{8}$	$2\frac{3}{4}$	$\frac{13}{16}$ (·8125)	·95	·68
$3\frac{1}{4}$	$3\frac{3}{4}$	3·700	·0580	3·584	11	$1\frac{3}{8}$	3	$\frac{7}{8}$ (·8750)	1·02	·73
$3\frac{1}{2}$	4	3·950	·0580	3·834	11	$1\frac{3}{8}$	3	$\frac{7}{8}$ (·8750)	1·02	·73
$3\frac{3}{4}$	$4\frac{1}{4}$	4·200	·0580	4·084	11	$1\frac{3}{8}$	3	$\frac{7}{8}$ (·8750)	1·02	·73
4	$4\frac{1}{2}$	4·450	·0580	4·334	11	$1\frac{3}{8}$	$3\frac{1}{4}$	1 (1·0000)	1·17	·83
$4\frac{1}{2}$	5	4·950	·0580	4·834	11	$1\frac{5}{8}$	$3\frac{1}{2}$	1 (1·0000)	1·17	·83
5	$5\frac{1}{2}$	5·450	·0580	5·334	11	$1\frac{5}{8}$	$3\frac{3}{4}$	$1\frac{1}{8}$ (1·1250)	1·31	·94
$5\frac{1}{2}$	6	5·950	·0580	5·834	11	$1\frac{5}{8}$	$3\frac{3}{4}$	$1\frac{1}{2}$ (1·2500)	1·46	1·04
6	$6\frac{1}{2}$	6·450	·0580	6·334	11	2	4	$1\frac{1}{8}$ (1·3750)	1·60	1·15
7	$7\frac{1}{2}$	7·450	·0640	7·322	10	$2\frac{1}{8}$	$4\frac{1}{4}$	$1\frac{1}{8}$ (1·3750)	1·60	1·15
8	$8\frac{1}{2}$	8·450	·0640	8·322	10	$2\frac{1}{4}$	$4\frac{1}{2}$	$1\frac{1}{2}$ (1·5000)	1·75	1·25
9	$9\frac{1}{2}$	9·450	·0640	9·322	10	$2\frac{1}{4}$	$4\frac{1}{2}$	$1\frac{1}{2}$ (1·5000)	1·75	1·25
10	$10\frac{1}{2}$	10·450	·0640	10·322	10	$2\frac{3}{8}$	$4\frac{3}{4}$	$1\frac{5}{8}$ (1·6250)	1·90	1·35
11	$11\frac{1}{2}$	11·450	·0800	11·290	8	$2\frac{1}{2}$	5	$1\frac{5}{8}$ (1·6250)	1·90	1·35
12	$12\frac{1}{2}$	12·450	·0800	12·290	8	$2\frac{1}{2}$	5	$1\frac{5}{8}$ (1·6250)	1·90	1·35
13	$13\frac{3}{4}$	13·680	·0800	13·520	8	$2\frac{5}{8}$	$5\frac{1}{4}$	$1\frac{5}{8}$ (1·6250)	1·90	1·35
14	$14\frac{3}{4}$	14·680	·0800	14·520	8	$2\frac{3}{4}$	$5\frac{1}{2}$	$1\frac{3}{4}$ (1·7500)	2·04	1·46
15	$15\frac{3}{4}$	15·680	·0800	15·520	8	$2\frac{3}{4}$	$5\frac{1}{2}$	$1\frac{3}{4}$ (1·7500)	2·04	1·46
16	$16\frac{3}{4}$	16·680	·0800	16·520	8	$2\frac{7}{8}$	$5\frac{3}{4}$	$1\frac{7}{8}$ (1·8750)	2·19	1·56
17	$17\frac{3}{4}$	17·680	·0800	17·520	8	3	6	2 (2·0000)	2·33	1·67
18	$18\frac{3}{4}$	18·680	·0800	18·520	8	3	6	2 (2·0000)	2·33	1·67

NOTES ON TABLE.

Columns 1 and 2 of the table give the nominal bore of the tube and the full diameter of the screw in the coupler for use with that tube. The diameter of this screw has been chosen, in all cases, so that the coupler can be screwed by hand on to the tube, leaving from four to eight threads exposed.

The Association have preserved the Old Whitworth Standards as far as possible in the simplified dimensions for screwing sizes on tubes up to $2\frac{1}{4}$ ins. nominal bore. The Association have been guided in fixing the gauge diameters of the larger screwing sizes by the advantages of making the outside diameters of the tubes even quarters of inches.

* By permission of the British Engineering Standards Association.

AMERICAN GAS PIPE THREADS.

Nominal Diameter in Inches.	Outside Diameter in Inches.	No. of Threads per Inch.	Nominal Diameter in Inches.	Outside Diameter in Inches.	No. of Threads per Inch.
$\frac{1}{8}$	0·405	28	$2\frac{1}{2}$	2·875	8
$\frac{1}{4}$	0·54	18	$2\frac{3}{4}$	—	—
$\frac{3}{8}$	0·675	18	3	3·5	8
$\frac{1}{2}$	0·84	14	$3\frac{1}{2}$	4·0	8
$\frac{5}{8}$	—	—	4	4·5	8
$\frac{3}{4}$	1·05	14	$4\frac{1}{2}$	5·0	8
$\frac{7}{8}$	—	—	5	5·563	8
1	1·315	$11\frac{1}{2}$	6	6·625	8
$1\frac{1}{4}$	1·66	$11\frac{1}{2}$	7	7·625	8
$1\frac{1}{2}$	1·9	$11\frac{1}{2}$	8	8·625	8
$1\frac{3}{4}$	—	—	9	9·688	8
2	2·375	$11\frac{1}{2}$	10	10·75	8
$2\frac{1}{4}$	—	—			

Bolts and Nuts.

According to the Whitworth system of standard sizes of bolts and nuts, the thickness of the bolt-head is seven-eighths of the diameter, and the nut is equal to the diameter.

In the Sellers' (American) system the heads and nuts are hexagonal, the thickness of the heads and of the nuts being equal to 1 diameter minus one-sixteenth inch. The breadth across the flats is equal to $1\frac{1}{2}$ diameters plus one-sixteenth inch.

WHITWORTH STANDARD HEXAGON BOLTS AND NUTS.

Dimensions in Inches.								
Size of Bolt and Thickness of Nut.	Nut across Flats.	Nut across Corners.	Tapping Hole.	Thickness of Bolt-head.	Size of Bolt and Thickness of Nut.	Nut across Flats.	Nut across Corners.	Tapping Hole.	Thickness of Bolt-head.
$\frac{1}{8}$	·338	·390	·093	·1093	$2\frac{3}{4}$	4·18	4·82	2·39	2·4062
$\frac{3}{16}$	·448	·517	·134	·164	$2\frac{7}{8}$	4·34	5·02	2·5	2·5156
$\frac{1}{4}$	·525	·606	·186	·2187	3	4·53	5·23	2·63	2·625
$\frac{5}{16}$	·601	·694	·241	·2734	$3\frac{1}{8}$	4·69	5·41	2·75	2·7343
$\frac{3}{8}$	·709	·819	·295	·3281	$3\frac{1}{4}$	4·85	5·60	2·85	2·8256
$\frac{7}{16}$	·820	·947	·346	·3828	$3\frac{3}{8}$	5·01	5·78	2·98	2·9531
$\frac{1}{2}$	·920	1·06	·393	·4375	$3\frac{1}{2}$	5·17	5·98	3·1	3·0624
$\frac{9}{16}$	1·01	1·16	·455	·4921	$3\frac{5}{8}$	5·36	6·19	3·23	3·1718
$\frac{5}{8}$	1·1	1·27	·508	·5468	$3\frac{3}{4}$	5·55	6·41	3·32	3·2812
$\frac{11}{16}$	1·2	1·38	·571	·6016	$3\frac{7}{8}$	5·75	6·64	3·45	3·3906
$\frac{3}{4}$	1·3	1·5	·622	·6562	4	5·95	6·87	3·57	3·5
$\frac{13}{16}$	1·39	1·6	·684	·7064	$4\frac{1}{8}$	6·16	7·11	3·69	3·6094
$\frac{7}{8}$	1·48	1·7	·732	·7656	$4\frac{1}{4}$	6·37	7·36	3·8	3·7046
$\frac{15}{16}$	1·57	1·82	·795	·8203	$4\frac{3}{8}$	6·60	7·62	3·93	3·8271
1	1·67	1·95	·84	·875	$4\frac{1}{2}$	6·82	7·88	4·05	3·9374
$1\frac{1}{8}$	1·86	2·15	·942	·9843	$4\frac{5}{8}$	7·06	8·15	4·18	4·0469
$1\frac{1}{4}$	2·05	2·36	1·06	1·0937	$4\frac{3}{4}$	7·3	8·43	4·28	4·1562
$1\frac{3}{8}$	2·21	2·55	1·16	1·2031	$4\frac{7}{8}$	7·55	8·72	4·4	4·2656
$1\frac{1}{2}$	2·41	2·78	1·28	1·3125	5	7·8	9·01	4·53	4·375
$1\frac{5}{8}$	2·57	2·97	1·37	1·4128	$5\frac{1}{8}$	8·06	9·31	4·66	4·4844
$1\frac{3}{4}$	2·75	3·18	1·49	1·5312	$5\frac{1}{4}$	8·35	9·64	4·76	4·5936
$1\frac{7}{8}$	3·02	3·48	1·59	1·6406	$5\frac{3}{8}$	8·6	9·93	4·88	4·7031
2	3·15	3·63	1·71	1·75	$5\frac{1}{2}$	8·85	10·22	5·01	4·8124
$2\frac{1}{8}$	3·34	3·85	1·84	1·8523	$5\frac{5}{8}$	9·15	10·57	5·13	4·9218
$2\frac{1}{4}$	3·54	4·09	1·93	1·9687	$5\frac{3}{4}$	9·45	10·91	5·24	5·0312
$2\frac{3}{8}$	3·75	4·33	2·05	2·0781	$5\frac{7}{8}$	9·75	11·26	5·36	5·1406
$2\frac{1}{2}$	3·89	4·49	2·18	2·1875	6	10	11·55	5·48	5·25
$2\frac{5}{8}$	4·05	4·67	2·30	2·2968					

American Standard Hexagon Nuts and Bolts (*Sellers'*).

Dimensions in Inches.

Size of Bolt and Thickness of Nut.	Nut across Flats.	Nut across Corners.	Tapping Hole.	Thickness of Bolt-head.	Size of Bolt and Thickness of Nut.	Nut across Flats.	Nut across Corners.	Tapping Hole.	Thickness of Bolt-head.
1/4	·5	·577	·185	·197	2	3·125	3·609	1·711	1·576
5/16	·597	·685	·240	·2472	2 1/4	3·5	4·042	1·961	1·7628
3/8	·687	·794	·294	·2954	2 1/2	3·875	4·475	2·175	1·97
7/16	·781	·902	·345	·3447	2 3/4	4·25	4·908	2·425	2·1528
1/2	·875	1·0106	·4	·394	3	4·625	5·341	2·628	2·364
9/16	·968	1·1187	·454	·4432	3 1/4	5	5·775	2·878	2·561
5/8	1·062	1·227	·506	·4925	3 1/2	5·375	6·208	3·1	2·758
3/4	1·25	1·443	·62	·591	3 3/4	5·75	6·641	3·317	2·954
7/8	1·437	1·66	·731	·6895	4	6·125	7·074	3·567	3·152
1	1·625	1·876	·837	·788	4 1/4	6·5	7·507	3·798	3·349
1 1/4	1·812	2·093	·939	·8864	4 1/2	6·875	7·94	4·027	3·5256
1 1/4	2	2·31	1·064	·985	4 3/4	7·25	8·373	4·255	3·743
1 3/8	2·187	2·526	1·158	1·0764	5	7·625	8·806	4·48	3·94
1 1/2	2·375	2·743	1·283	1·182	5 1/4	8	9·24	4·73	4·137
1 5/8	2·562	2·959	1·389	1·2805	5 1/2	8·375	9·67	4·953	4·334
1 3/4	2·75	3·176	1·49	1·379	5 3/4	8·75	10·106	5·203	4·531
1 7/8	2·937	3·393	1·615	1·477	6	9·125	10·539	5·423	4·728

Whitworth Standard Bolts and Nuts. Safe Load in Lbs.

Diam. of Bolt.	Diam. at Bottom of Thread.	Area at Bottom of Thread.	Approximate Safe Load in Lbs.					
			At 4,000 lbs. per Sq. In.	At 5,000 lbs. per Sq. In.	At 6,000 lbs. per Sq. In.	At 7,000 lbs. per Sq. In.	At 8,000 lbs. per Sq. In.	At 9,000 lbs. per Sq. In.
Ins.	Ins.	Sq. Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
5/16	0·134	0·014	56	70	84	98	112	126
1/4	0·186	0·027	108	135	162	189	216	243
5/16	0·241	0·046	184	230	276	322	378	414
3/8	0·295	0·068	272	340	408	478	546	614
7/16	0·346	0·094	376	470	504	628	752	846
1/2	0·393	0·121	484	605	726	850	970	1,090
9/16	0·455	0·155	620	775	930	1,085	1,240	1,395
5/8	0·508	0·204	816	1,020	1,224	1,418	1,620	1,824
11/16	0·571	0·256	1,024	1,280	1,536	1,792	2,048	2,304
3/4	0·622	0·304	1,216	1,520	1,824	2,126	2,430	2,734
13/16	0·684	0·367	1,468	1,835	2,202	2,569	2,936	3,303
7/8	0·733	0·422	1,688	2,110	2,532	2,954	3,376	3,800
15/16	0·795	0·496	1,984	2,480	2,976	3,472	3,968	4,464
1	0·840	0·554	2,216	2,770	3,324	3,880	4,430	4,990
1 1/16	0·942	0·697	2,788	3,485	4,182	4,880	5,575	6,270
1 1/8	1·067	0·894	3,576	4,470	5,364	6,250	7,145	8,035
1 3/16	1·162	1·058	4,232	5,290	6,348	7,400	8,455	9,510
1 1/4	1·287	1·299	5,196	6,495	7,794	9,030	10,320	11,610
1 5/16	1·369	1·472	5,888	7,360	8,832	10,290	11,760	13,230
1 3/8	1·494	1·753	7,012	8,765	10,518	12,210	13,950	15,700
1 7/16	1·590	1·986	7,944	9,930	11,916	13,900	15,890	17,875
1 1/2	1·720	2·311	9,244	11,555	13,866	16,170	18,489	20,790
2	1·930	2·926	11,704	14,630	17,556	20,475	23,400	26,325
2 1/4	2·180	3·733	14,932	18,665	22,398	26,125	29,855	33,590

WHITWORTH STANDARD BOLTS AND NUTS. SAFE LOAD IN LBS.—*cont.*

Diam. of Bolt.	Diam. at Bottom of Thread.	Area at bottom of Thread.	APPROXIMATE SAFE LOAD IN LBS.					
			At 4,000 lbs. per Sq. In.	At 5,000 lbs. per Sq. in.	At 6,000 lbs. per Sq. In.	At 7,000 lbs. per Sq. In.	At 8,000 lbs. per Sq. In.	At 9,000 lbs. per Sq. In.
Ins.	Ins.	Sq. Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
2¾	2·384	4·464	17,856	22,320	26,784	31,220	35,680	40,140
3	2·634	5·450	21,800	27,250	32,700	38,080	43,520	48,960
3¼	2·856	6·402	25,608	32,010	38,412	44,814	51,216	57,618
3½	3·105	7·563	30,252	37,815	45,378	57,940	60,504	68,065
3¾	3·320	8·673	34,692	43,365	52,038	60,710	69,384	78,057
4	3·573	10·027	40,108	50,135	60,162	70,190	80,216	90,240
4¼	3·804	11·365	45,480	56,825	68,190	79,575	90,960	102,305
4½	4·054	12·908	51,632	64,540	77,448	90,356	103,264	116,172
4¾	4·284	14·404	57,616	72,020	86,424	100,830	115,232	129,636
5	4·534	16·146	64,584	80,730	96,876	113,020	129,168	145,314
5¼	5·012	19·720	78,880	98,600	118,320	138,040	157,760	177,480
6	5·487	23·640	94,560	118,200	141,840	165,200	189,120	212,760

BRITISH STANDARD HEXAGONAL BRIGHT AND BLACK NUTS AND BOLT-HEADS.*
(No. 28—1908, and 190—1924.) (*Abstract.*)

Diameter of Bolt.	Nuts.					Lock Nuts.		Bolt-Heads.	
	Width across Flats.		Width across Corners.	Thickness.		Thickness.		Thickness.	
	Max.	Min.	Approximate Max.†	Max.	Min.	Max.	Min.	Max.	Min.
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
¼ (·25)	·525	·520	·61	·26 *·27*	·25	·18 *·19*	·17	·23 *·24*	·22
⅜ (·375)	·710	·705	·82	·39 *·40*	·38	·26 *·27*	·25	·34 *·35*	·33
½ (·5)	·920	·915	1·06	·51 *·52*	·50	·34 *·35*	·33	·45 *·46*	·44
⅝ (·625)	1·100	1·092	1·27	·64 *·65*	·63	·43 *·44*	·42	·56 *·57*	·55
¾ (·75)	1·300	1·292	1·50	·76 *·77*	·75	·51 *·52*	·50	·67 *·68*	·66
⅞ (·875)	1·480	1·468	1·71	·89 *·90*	·88	·59 *·60*	·58	·78 *·79*	·77
1	1·670	1·658	1·93	1·01 *1·02*	1·00	·68 *·69*	·67	·89 *·90*	·88
1¼ (1·25)	2·050	2·035	2·37	1·27 *1·28*	1·25	·85 *·86*	·83	1·11 *1·12*	1·09
1½ (1·5)	2·410	2·390	2·78	1·52 *1·53*	1·50	1·02 *1·03*	1·00	1·33 *1·34*	1·31
1¾ (1·75)	2·760	2·735	3·19	1·77 *1·78*	1·75	1·19 *1·20*	1·17	1·55 *1·56*	1·53
2	3·150	3·125	3·64	2·02 *2·03*	2·00	1·35 *1·36*	1·33	1·77 *1·78*	1·75
2¼ (2·25)	3·550	3·535	4·10	2·27 *2·28*	2·25	1·52 *1·53*	1·50	1·99 *2·00*	1·97
2½ (2·5)	3·890	3·875	4·49	2·52 *2·53*	2·50	1·69 *1·70*	1·67	2·21 *2·22*	2·19
2¾ (2·75)	4·180	4·165	4·83	2·77 *2·78*	2·75	1·85 *1·86*	1·83	2·43 *2·44*	2·41
3	4·530	4·515	5·23	3·02 *3·03*	3·00	2·02 *2·03*	2·00	2·65 *2·66*	2·63
3¼ (3·25)	4·850	4·830	5·60	3·27 *3·28*	3·25	2·19 *2·20*	2·17	2·86 *2·86*	2·84
3½ (3·5)	5·180	5·160	5·98	3·52 *3·53*	3·50	2·35 *2·36*	2·33	3·08 *3·09*	3·06
3¾ (3·75)	5·550	5·530	6·41	3·77 *3·78*	3·75	2·52 *2·53*	2·50	3·30 *3·31*	3·28
4	5·950	5·930	6·87	4·02 *4·03*	4·00	2·69 *2·70*	2·67	3·52 *3·53*	3·50
4½ (4·5)	6·820	6·795	7·88	4·53 *4·54*	4·50	3·03 *3·03*	3·00	3·97 *3·98*	3·94
5	7·800	7·775	9·01	5·03 *5·04*	5·00	3·36 *3·37*	3·33	4·41 *4·42*	4·38
5½ (5·5)	8·850	8·820	10·22	5·53 *5·54*	5·50	3·70 *3·71*	3·67	4·84 *4·84*	4·81
6	10·000	9·970	11·55	6·03 *6·04*	6·00	4·03 *4·04*	4·00	5·28 *5·29*	5·25

* By permission of the British Engineering Standards Association.

† The figures in this column represent the maximum width across the corners correct to two decimal places. The figures in italics are those for *black* nuts.

A chamfer of 30° from the face should be adopted on all bright nuts and bolt-heads. The interior diameter of the chamfer circle should not be appreciably less than the width across the flats.

British Standard Specification for Black, Iron and Steel, Cup and Countersunk Bolt-Heads, Nuts and Washers.

(See No. 325, 1928.)

British Standard Whitworth Screw Threads. Tolerances for Nuts.

1	2	3	6	7	8	9	10	11	12	13	14
Nominal Dia. of Screw.	No. of Threads per inch.	Pitch.	Full Diameter.			Effective Diameter.			Core Diameter.		
			Min.	Tol.	Max.	Min.	Tol.	Max.	Min.	Tol.	Max.
In.		In.	In.	In.	In.	In.	In.	In.	In.	In.	In.
¼	20	·05000	·2520	·0090	·2610	·2200	·0045	·2245	·1880	·0067	·1947
5/16	16	·06250	·3770	·0100	·3870	·3370	·0050	·3420	·2970	·0075	·3045
3/8	12	·08333	·5020	·0115	·5135	·4486	·0058	·4544	·3953	·0087	·4040
7/16	11	·09091	·6270	·0121	·6391	·5688	·0060	·5748	·5106	·0090	·5196
½	10	·10000	·7520	·0126	·7646	·6880	·0063	·6943	·6239	·0095	·6334
9/16	9	·11111	·8770	·0133	·8903	·8059	·0067	·8126	·7347	·0100	·7447
⅝	8	·12500	1·0020	·0141	1·0161	·9220	·0071	·9291	·8419	·0106	·8525
¾	6	·16667	1·5020	·0163	1·5183	1·3953	·0082	1·4035	1·2886	·0122	1·3008
⅞	4·5	·22222	2·0020	·0189	2·0209	1·8597	·0094	1·8691	1·7174	·0141	1·7315
1	4	·25000	2·5020	·0200	2·5220	2·3419	·0100	2·3519	2·1818	·0150	2·1968
1⅛	3·5	·28571	3·0020	·0214	3·0234	2·8190	·0107	2·8297	2·6361	·0160	2·6521
1¼	3·25	·30769	3·5020	·0222	3·5242	3·3050	·0111	3·3161	3·1080	·0166	3·1246
1⅜	3	·33333	4·0020	·0231	4·0251	3·7886	·0115	3·8001	3·5751	·0173	3·5924
1½	2·875	·34785	4·5020	·0236	4·5256	4·2793	·0118	4·2911	4·0566	·0177	4·0743
1⅝	2·75	·36364	5·0020	·0241	5·0261	4·7692	·0121	4·7813	4·5363	·0181	4·5544
1¾	2·5	·40000	6·0020	·0253	6·0273	5·7459	·0126	5·7585	5·4897	·0190	5·5087

British Standard Whitworth Screw Threads. Tolerances for Bolts.*

(No. 92—1919.) (*Abstract.*)

1	2	3	6	7	8	9	10	11	12	13	14
Nominal Dia. of Screw.	No. of Threads per inch.	Pitch.	Full Diameter.			Effective Diameter.			Core Diameter.		
			Max.	Tol.	Min.	Max.	Tol.	Min.	Max.	Tol.	Min.
Ins.		In.	In.	In.	In.	In.	In.	In.	In.	In.	In.
¼	20	·05000	2500	·0067	·2433	·2180	·0045	·2135	·1860	·0090	·1770
5/16	16	·06250	·3750	·0075	·3675	·3350	·0050	·3300	·2950	·0100	·2850
3/8	12	·08333	·5000	·0087	·4913	·4466	·0058	·4408	·3933	·0115	·3818
7/16	11	·09091	·6250	·0090	·6160	·5668	·0060	·5608	·5086	·0121	·4965
½	10	·10000	·7500	·0095	·7405	·6860	·0063	·6797	·6219	·0126	·6093
9/16	9	·11111	·8750	·0100	·8650	·8039	·0067	·7972	·7327	·0133	·7194
⅝	8	·12500	1·0000	·0106	·9894	·9200	·0071	·9129	·8399	·0141	·8258
¾	6	·16667	1·5000	·0122	1·4878	1·3933	·0082	1·3851	1·2866	·0163	1·2703
⅞	4·5	·22222	2·0000	·0141	1·9859	1·8577	·0094	1·8483	1·7154	·0189	1·6965
1	4	·25000	2·5000	·0150	2·4850	2·3399	·0100	2·3299	2·1798	·0200	2·1598
1⅛	3·5	·28571	3·0000	·0160	2·9840	2·8170	·0107	2·8063	2·6341	·0214	2·6127
1¼	3·25	·30769	3·5000	·0166	3·4834	3·3030	·0111	3·2919	3·1060	·0222	3·0838
1⅜	3	·33333	4·0000	·0173	3·9827	3·7866	·0115	3·7751	3·5731	0231	3·5500
1½	2·875	·34783	4·5000	·0177	4·4823	4·2773	·0118	4·2655	4·0546	·0236	4·0310
1⅝	2·75	·36364	5·0000	·0181	4·9819	4·7672	·0121	4 7551	4·5343	·0241	4·5102
1¾	2·5	·40000	6·0000	·0190	5·9810	5·7439	·0126	5·7313	5·4877	·0253	5·4624

* By permission of the British Engineering Standards Association.

British Standard Fine Screw Threads. Tolerances for Nuts.

1	2	3	6	7	8	9	10	11	12	13	14
Nominal Dia. of Screw.	No. of Threads per inch.	Pitch.	Full Diameter.			Effective Diameter.			Core Diameter.		
			Min.	Tol.	Max.	Min.	Tol.	Max.	Min.	Tol.	Max.
In.		In.	In.	In.	In.	In.	In.	In.	In.	In.	In
1/4	26	·03846	·2520	·0078	·2598	·2274	·0039	·2313	·2027	·0059	·2086
5/16	20	·05000	·3770	·0090	·3860	·3450	·0045	·3495	·3130	·0067	·3197
3/8	16	·06250	·5020	·0100	·5120	·4620	·0050	·4670	·4220	·0075	·4295
7/16	14	·07143	·6270	·0107	·6377	·5813	·0053	·5866	·5355	·0080	·5435
1/2	12	·08333	·7520	·0115	·7635	·6986	·0058	·7044	·6453	·0087	·6540
9/16	11	·09091	·8770	·0121	·8891	·8188	·0060	·8248	·7606	·0090	·7696
1	10	·10000	1·0020	·0126	1·0146	·9380	·0063	·9443	·8739	·0095	·8834
1 1/8	9	·11111	1·1270	·0133	1·1403	1·0559	·0067	1·0626	·9847	·0100	·9947
1 1/4	9	·11111	1·2520	·0133	1·2653	1·1809	·0067	1·1876	1·1097	·0100	1·1197
1 3/8	8	·12500	1·3770	·0141	1·3911	1·2970	·0071	1·3041	1·2169	·0106	1·2275
1 1/2	8	·12500	1·5020	·0141	1·5161	1·4220	·0071	1·4291	1·3419	·0106	1·3525
1 5/8	8	·12500	1·6270	·0141	1·6411	1·5470	·0071	1·5541	1·4669	·0106	1·4775
1 3/4	7	·14286	1·7520	·0151	1·7671	1·6605	·0076	1·6681	1·5690	·0113	1·5803
2	7	·14286	2·0020	·0151	2·0171	1·9105	·0076	1·9181	1·8190	·0113	1·8303
2 1/4	6	·16667	2·2520	·0163	2·2683	2·1453	·0082	2·1535	2·0386	·0122	2·0508
2 1/2	6	·16667	2·5020	·0163	2·5183	2·3953	·0082	2·4035	2·2886	·0122	2·3008
2 3/4	6	·16667	2·7520	·0163	2·7683	2·6453	·0082	2·6535	2·5386	·0122	2·5508
3	5	·20000	3·0020	·0179	3·0199	2·8739	·0089	2·8828	2·7459	·0134	2·7593

British Standard Fine Screw Threads. Tolerances for Bolts.*

(No. 84—1918.) (*Abstract.*)

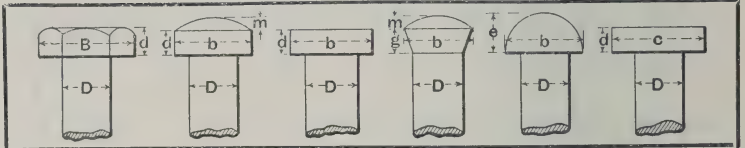
1	2	3	6	7	8	9	10	11	12	13	14
Nominal Dia. of Screw.	No. of Threads per inch.	Pitch.	Full Diameter.			Effective Diameter.			Core Diameter.		
			Max.	Tol.	Min.	Max.	Tol.	Min.	Max.	Tol.	Min.
In.		In.	In.	In.	In.	In.	In.	In.	In.	In.	In.
1/4	26	·03846	·2500	·0059	·2441	·2254	·0039	·2215	·2007	·0078	·1929
5/16	20	·05000	·3750	·0067	·3683	·3430	·0045	·3385	·3110	·0090	·3020
3/8	16	·06250	·5000	·0075	·4925	·4600	·0050	·4550	·4200	·0100	·4100
7/16	14	·07143	·6250	·0080	·6170	·5793	·0053	·5740	·5335	·0107	·5228
1/2	12	·08333	·7500	·0087	·7413	·6966	·0058	·6908	·6433	·0115	·6318
9/16	11	·09091	·8750	·0090	·8660	·8168	·0060	·8108	·7586	·0121	·7465
1	10	·10000	1·0000	·0095	·9905	·9360	·0063	·9297	·8719	·0126	·8593
1 1/8	9	·11111	1·1250	·0100	1·1150	1·0539	·0067	1·0472	·9827	·0133	·9694
1 1/4	9	·11111	1·2500	·0100	1·2400	1·1789	·0067	1·1722	1·1077	·0133	1·0944
1 3/8	8	·12500	1·3750	·0106	1·3644	1·2950	·0071	1·2879	1·2149	·0141	1·2008
1 1/2	8	·12500	1·5000	·0106	1·4894	1·4200	·0071	1·4129	1·3399	·0141	1·3258
1 5/8	8	·12500	1·6250	·0106	1·6144	1·5450	·0071	1·5379	1·4649	·0141	1·4508
1 3/4	7	·14286	1·7500	·0113	1·7387	1·6585	·0076	1·6509	1·5670	·0151	1·5519
2	7	·14286	2·0000	·0113	1·9887	1·9085	·0076	1·9009	1·8170	·0151	1·8019
2 1/4	6	·16667	2·2500	·0122	2·2378	2·1433	·0082	2·1351	2·0366	·0163	2·0203
2 1/2	6	·16667	2·5000	·0122	2·4878	2·3933	·0082	2·3851	2·2866	·0163	2·2703
2 3/4	6	·16667	2·7500	·0122	2·7378	2·6433	·0082	2·6351	2·5366	·0163	2·5203
3	5	·20000	3·0000	·0134	2·9866	2·8719	·0089	2·8630	2·7439	·0179	2·7260

* By permission of the British Engineering Standards Association.

French Standard Bolts and Nuts.

Diameter of Screw.		Diameter of Bottom of Thread.		Pitch of Thread.	Number of Threads per Inch.	Width across Flats.		Width across Corners.		Thickness of Head.	
Millimètres.	Inches.	Millimètres.	Inches.	Millimètres.		Millimètres.	Inches.	Millimètres.	Inches.	Millimètres.	Inches.
5	·196	3·30	·13	1·4	18·14	13·97	·55	16·0	·63	5	·196
7·5	·295	5·58	·22	1·6	15·87	17·27	·68	19·81	·78	7	·275
10	·393	7·87	·31	1·8	14·11	22·35	·88	25·90	1·02	9	·355
12·5	·492	9·90	·39	2	12·70	26·42	1·04	30·48	1·20	11	·433
15	·589	12·19	·48	2·2	11·55	30·48	1·20	35·05	1·38	13	·51
17·5	·688	14·73	·58	2·4	10·58	34·54	1·36	39·87	1·57	15	·59
20	·787	16·76	·66	2·6	9·77	38·60	1·52	44·45	1·75	17	·67
22·5	·885	19·20	·76	2·8	9·07	42·67	1·68	49·27	1·94	19	·75
25	·984	21·34	·84	3	8·46	46·74	1·84	53·84	2·12	21	·83
30	1·178	25·90	1·02	3·4	7·47	54·86	2·16	63·50	2·50	24	·94
35	1·376	30·48	1·20	3·8	6·68	62·99	2·48	72·64	2·86	27	1·06
40	1·574	35·56	1·40	4·2	6·05	71·12	2·80	82·04	3·23	31	1·22
45	1·770	39·62	1·56	4·6	5·52	79·25	3·12	91·44	3·60	35	1·38
50	1·968	44·19	1·74	5	5·08	87·37	3·44	100·80	3·97	38	1·50
55	2·162	48·76	1·92	5·4	4·70	95·50	3·76	110·2	4·34	41	1·65
60	2·356	52·83	2·08	5·8	4·38	103·63	4·08	119·8	4·72	44	1·73
65	2·554	57·42	2·26	6·2	4·09	111·76	4·40	129·0	5·08	48	1·89
70	2·752	61·97	2·44	6·6	3·85	118·88	4·72	138·4	5·45	52	2·05
75	2·950	66·04	2·60	7·0	3·63	128·02	5·04	147·8	5·82	56	2·20
80	3·149	70·61	2·78	7·4	3·43	136·14	5·36	157·5	6·20	60	2·36

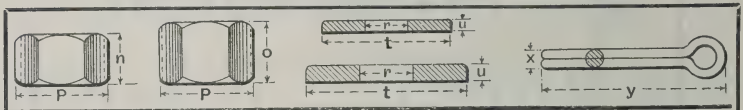
FRENCH BOLTS. (SIZES GENERALLY USED.)



Diameter of Bolt. *D*	Pitch of Thread.	Diameter of Hexagon Heads. *B*	Diameter of Heads for Iron *b*	Side of Square for Wood. *c*	Height of Spherical Heads. *e*	Height of Counter Sink. *g*	Height of Round and Square Heads. *d*	Height of Rise. *m*
mm.	mm.	mm.	mm.	mm.	mm.	mm.	mm.	mm.
8	1¼	16	14	18	6	4	5	1
9	1¼	18	15	20	9	5	6	1
10	1½	20	16	24	10	5	6	1
12	1½	24	20	28	12	6	7	1
14	2	28	22	32	13	7	8	1
15	2	30	24	35	14	7	9	2
16	2	32	26	36	15	8	10	2
18	2	36	28	42	16	9	11	2
20	2½	40	32	46	18	10	12	2
22	2½	44	36	54	20	11	14	3
25	3	50	40	58	22	12	15	3
28	3	56	44	65	24	14	17	4
30	3	60	48	70	26	15	18	4
32	3½	64	52	74	28	16	19	4
35	4	70	56	80	31	17	21	5

(The actual depth of the thread is generally equal to ·75 of the pitch.)

FRENCH NUTS, WASHERS, AND BOLT-PINS. (SIZES GENERALLY USED.)



Nuts.				Washers.				Bolt-pins.	
Diameter of bolt. *D*	Height of Nut. *n*	Diameter of Hexagon. *P*	Extra Stout. *o*	Diameter of Hole. *r*	Outside Diameter for Iron. *s*	Outside Diameter for Wood. *t*	Thickness. *u*	Length. *y*	Diameter. *x*
mm.	mm.	mm.	mm.	mm.	mm.	mm.	mm.	mm.	mm.
8	8	16	11	8½	19	22	2	16	2
9	9	18	12	9½	21	24	2	18	2
10	10	20	14	10½	25	28	3	20	3
12	12	24	16	12½	29	33	3	24	3
14	14	28	18	14½	34	38	3	28	3
15	15	30	20	15½	37	41	4	30	4
16	16	32	21	16½	40	45	4	32	4
18	18	36	24	18½	45	50	5	36	5
20	20	40	27	20½	50	55	5	40	5
22	22	44	29	22½	55	61	6	44	6
25	25	50	33	23½	62	68	6	50	6
28	28	56	37	29	69	75	7	56	7
30	30	60	40	31	74	80	7	60	7
32	32	64	42	33	78	85	7	64	8
35	35	70	46	36	86	94	8	70	9

ROUND BOLT HEADS.

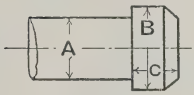


FIG. 11.

PROPORTIONS OF ROUND-HEADED BOLTS.

A	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$
B	$\frac{3}{8}$	$\frac{9}{16}$	$\frac{3}{4}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{9}{16}$	$1\frac{3}{4}$	$1\frac{7}{8}$	$2\frac{1}{16}$	$2\frac{1}{4}$	$2\frac{3}{8}$
C	$\frac{3}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{15}{16}$	$1\frac{1}{16}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{5}{16}$

CAPPED NUTS.

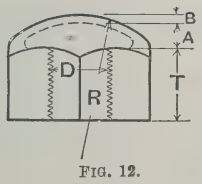


FIG. 12.

D = dia. of bolt; R = D

T = D; A = $\frac{D}{2}$; B = $\frac{1}{8}$ in.

Nut Locking.

In the common locking arrangement, in which a thick and thin nut are employed, the thin nut should have a thickness approximately equal to two-thirds that of the standard nut.

LIST OF SPLIT PINS AND BOLT-END LENGTHS FOR NUT LOCKING.

Diameter of Bolt or Pin	Diameter of Split Pin.	Length of Split Pin through Shank.	Length of Split Pin through Nut.	Length over Grip for Drilled Bolt.	Length over Grip for Undrilled Bolt.
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
$\frac{3}{8}$	$\frac{3}{32}$	1	$1\frac{1}{4}$	$\frac{5}{8}$	$\frac{1}{2}$
$\frac{7}{16}$	$\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$\frac{7}{8}$	$\frac{5}{8}$
$\frac{1}{2}$	$\frac{3}{16}$	$1\frac{1}{2}$	2	$1\frac{1}{16}$	$\frac{3}{4}$
$\frac{9}{16}$	$\frac{5}{32}$	$1\frac{3}{4}$	$2\frac{1}{4}$	$1\frac{3}{16}$	$\frac{7}{8}$
$\frac{5}{8}$	$\frac{1}{4}$	2	$2\frac{1}{2}$	$1\frac{5}{8}$	1
$\frac{11}{16}$	$\frac{1}{4}$	$2\frac{1}{4}$	$2\frac{3}{4}$	$1\frac{1}{16}$	$1\frac{1}{8}$
1	$\frac{1}{4}$	$2\frac{1}{4}$	3	$1\frac{5}{8}$	$1\frac{1}{4}$
$1\frac{1}{8}$	$\frac{1}{4}$	$2\frac{1}{2}$	3	$1\frac{3}{4}$	$1\frac{3}{8}$
$1\frac{1}{4}$	$\frac{1}{4}$	$2\frac{1}{2}$	$3\frac{1}{4}$	$1\frac{7}{8}$	$1\frac{1}{2}$
$1\frac{3}{8}$	$\frac{1}{4}$	$2\frac{1}{2}$	4	$2\frac{1}{8}$	$1\frac{5}{8}$
$1\frac{1}{2}$	$\frac{5}{16}$	3	$4\frac{1}{4}$	$2\frac{3}{8}$	$1\frac{3}{4}$
$1\frac{5}{8}$	$\frac{5}{16}$	$3\frac{1}{4}$	$4\frac{1}{4}$	$2\frac{1}{2}$	$1\frac{7}{8}$
$1\frac{3}{4}$	$\frac{5}{16}$	$3\frac{1}{2}$	$4\frac{1}{4}$	$2\frac{5}{8}$	2
$1\frac{7}{8}$	$\frac{5}{16}$	$3\frac{3}{4}$	$4\frac{1}{2}$	$2\frac{3}{4}$	$2\frac{1}{8}$
2	$\frac{5}{16}$	$3\frac{3}{4}$	$4\frac{1}{2}$		

NOTE.—Length of a split pin is always measured overall.

('*Machinery*.')

British Standard Whitworth Castle Nuts.*

(190—1924.) (*Abstract*).

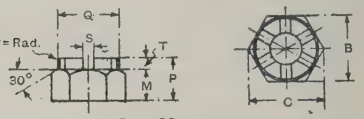


FIG. 13.

Thickness of hexagon and from face of nut to bottom of slot.	Min.	M	0·240	0·365	0·490	0·615	0·740	0·990	1·230	1·480	1·730
	Max.		0·250	0·375	0·500	0·625	0·750	1·000	1·250	1·500	1·750
Thickness of Castle nut.	Min.	P	0·334	0·505	0·677	0·802	0·974	1·270	1·558	1·855	2·198
	Max.		0·344	0·515	0·687	0·812	0·984	1·280	1·578	1·875	2·218
Diameter of castellated portions.	Min.	Q	0·500	0·685	0·895	1·065	1·265	1·635	2·015	2·370	2·715
	Max.		0·510	0·695	0·905	1·085	1·285	1·655	2·035	2·390	2·735
Approximate width of slot .		S	0·084	0·125	0·165	0·165	0·208	0·250	0·290	0·333	0·416
Approximate depth of slot .		T	0·094	0·140	0·187	0·187	0·234	0·280	0·328	0·375	0·468
Radius on castellated portion		R	0·05	0·06	0·06	0·07	0·08	0·09	0·11	0·13	0·14

To Lock a Nut.

(1).

In the absence of any of the regular nut-locking devices the work can still be done with an ordinary nut and washer. The stud or bolt used is first filed flat on one side at A, fig. 14, to

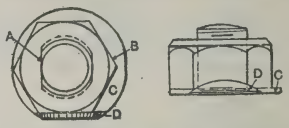


FIG. 14.

the depth of the threads. Next, the washer is given a few taps on one edge to elongate the hole, so that it will not turn when on the bolt. The nut B and washer C are then assembled in the usual manner, and when tightened up a chisel is inserted beneath the washer and the edge turned up at D, and tightened close to the side of the nut by hammering

This is an improvised method, but it acts just as well as the regular locking devices.

(*H. Moore*, '*The Mechanical World*,' *June* 22, 1923.)

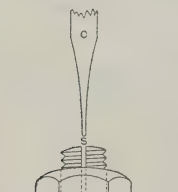


FIG. 15.

(2).

A slot S (fig. 15) is cut in the bolt with a hack-saw, and when the nut is tightly screwed home a cold chisel wedge C, whose width is less than the diameter of the bolt, is driven into the slot S (and then withdrawn) so as to spread out the end of the bolt, thus preventing the nut from unscrewing.

A serrated steel wedge can be driven into the slot if the nut is not required to be unscrewed again. (*H. R. Kempe.*)

* By permission of the British Engineering Standards Association.

LOCK NUTS FOR FEEDING SCREWS OF MACHINE TOOLS.

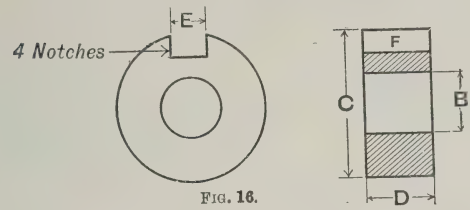


FIG. 16.

Dimensions in Inches.					Number of Threads per Inch.	Dimensions in Inches.					Number of Threads per Inch.
B.	C.	D.	E.	F.		B.	C.	D.	E.	F.	
$\frac{1}{2}$	1	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{3}{32}$	14	$1\frac{5}{8}$	$2\frac{7}{8}$	$1\frac{3}{4}$	$\frac{3}{8}$	$\frac{3}{16}$	11
$\frac{5}{8}$	$1\frac{1}{4}$	$\frac{5}{16}$	"	"	14	$1\frac{3}{4}$	$3\frac{1}{16}$	$\frac{7}{8}$	"	"	11
$\frac{3}{4}$	$1\frac{1}{2}$	$\frac{3}{8}$	"	$\frac{1}{8}$	14	$1\frac{7}{8}$	$3\frac{1}{4}$	$1\frac{5}{16}$	"	"	11
$\frac{7}{8}$	$1\frac{7}{8}$	$\frac{7}{16}$	$\frac{1}{4}$	"	11	2	$3\frac{1}{2}$	1	"	"	11
1	$1\frac{3}{4}$	$\frac{1}{2}$	"	"	11	$2\frac{1}{4}$	4	1	$\frac{7}{16}$	$\frac{7}{32}$	8
$1\frac{1}{8}$	2	$\frac{9}{16}$	$\frac{5}{16}$	$\frac{5}{32}$	11	$2\frac{1}{2}$	$4\frac{3}{8}$	$1\frac{1}{8}$	"	"	8
$1\frac{1}{4}$	$2\frac{1}{4}$	$\frac{5}{8}$	"	"	11	$2\frac{3}{4}$	$4\frac{1}{2}$	$1\frac{1}{4}$	$\frac{1}{2}$	$\frac{1}{4}$	8
$1\frac{3}{8}$	$2\frac{3}{8}$	$\frac{11}{16}$	"	"	11	3	$5\frac{1}{4}$	$1\frac{3}{8}$	"	"	8
$1\frac{1}{2}$	$2\frac{5}{8}$	$\frac{3}{4}$	"	"	11						

A pair of these nuts is used for locking purposes.

WING NUTS.

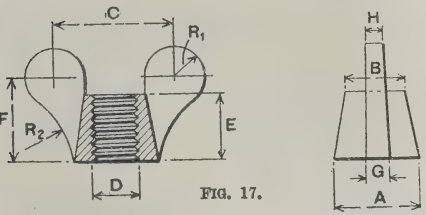


FIG. 17.

WING NUT PROPORTIONS.

The following table gives various dimensions for wing nuts threaded for bolts from $\frac{1}{4}$-inch to 1-inch Whitworth. The dimensions chosen are representative of usual practice.

D.	A.	B.	C.	E.	F.	G.	H.	R_1.	R_2.
	in.	in.	in.	in.	in.	in.	in.	in.	in.
$\frac{1}{4}$" Whit.	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{2}$	$\frac{5}{16}$	$\frac{3}{32}$	$\frac{3}{32}$	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{3}{16}$
$\frac{5}{16}$ "	$\frac{7}{16}$	$\frac{5}{16}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{1}{8}$	$\frac{3}{32}$	$\frac{1}{16}$	$\frac{5}{32}$	$\frac{1}{4}$
$\frac{3}{8}$ "	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{3}{32}$	$\frac{3}{16}$	$\frac{5}{16}$
$\frac{7}{16}$ "	$\frac{9}{16}$	$\frac{7}{16}$	$\frac{7}{8}$	$\frac{9}{16}$	$\frac{5}{32}$	$\frac{5}{32}$	$\frac{1}{16}$	$\frac{7}{32}$	$\frac{3}{8}$
$\frac{1}{2}$ "	$\frac{5}{8}$	$\frac{1}{2}$	1	$\frac{5}{8}$	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{3}{32}$	$\frac{1}{4}$	$\frac{7}{16}$
$\frac{9}{16}$ "	$\frac{11}{16}$	$\frac{9}{16}$	$1\frac{1}{16}$	$\frac{11}{16}$	$\frac{3}{16}$	$\frac{7}{32}$	$\frac{3}{32}$	$\frac{9}{32}$	$\frac{1}{2}$
$\frac{5}{8}$ "	$\frac{3}{4}$	$\frac{5}{8}$	$1\frac{1}{8}$	$\frac{3}{4}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{1}{16}$	$\frac{5}{16}$	$\frac{9}{16}$
$\frac{11}{16}$ "	$\frac{13}{16}$	$\frac{11}{16}$	$1\frac{1}{4}$	$\frac{13}{16}$	$\frac{7}{32}$	$\frac{9}{32}$	$\frac{3}{32}$	$\frac{11}{32}$	$\frac{5}{8}$
$\frac{3}{4}$ "	$\frac{7}{8}$	$\frac{3}{4}$	$1\frac{3}{8}$	$\frac{7}{8}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{1}{16}$	$\frac{3}{8}$	$\frac{11}{16}$
$\frac{13}{16}$ "	$\frac{15}{16}$	$\frac{13}{16}$	$1\frac{7}{16}$	$\frac{15}{16}$	$\frac{9}{32}$	$\frac{11}{32}$	$\frac{3}{32}$	$\frac{13}{32}$	$\frac{3}{4}$
$\frac{7}{8}$ "	1	$\frac{7}{8}$	$1\frac{1}{2}$	1	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{1}{16}$	$\frac{7}{16}$	$\frac{13}{16}$
$\frac{15}{16}$ "	$1\frac{1}{16}$	$\frac{15}{16}$	$1\frac{9}{16}$	$1\frac{1}{16}$	$\frac{11}{32}$	$\frac{13}{32}$	$\frac{3}{32}$	$\frac{15}{32}$	$\frac{7}{8}$
1 "	$1\frac{1}{8}$	1	$1\frac{5}{8}$	$1\frac{1}{8}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{16}$	$\frac{1}{2}$	$\frac{15}{16}$
$1\frac{1}{16}$ "	$1\frac{3}{16}$	$1\frac{1}{16}$	$1\frac{11}{16}$	$1\frac{3}{16}$	$\frac{13}{32}$	$\frac{15}{32}$	$\frac{3}{32}$	$\frac{17}{32}$	1
$1\frac{1}{8}$ "	$1\frac{5}{16}$	$1\frac{1}{8}$	$1\frac{13}{16}$	$1\frac{5}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{16}$	$\frac{9}{16}$	$1\frac{1}{16}$
$1\frac{3}{16}$ "	$1\frac{7}{16}$	$1\frac{3}{16}$	$1\frac{15}{16}$	$1\frac{7}{16}$	$\frac{15}{32}$	$\frac{17}{32}$	$\frac{3}{32}$	$\frac{19}{32}$	$1\frac{1}{8}$
$1\frac{1}{4}$ "	$1\frac{1}{2}$	$1\frac{1}{4}$	$2\frac{1}{16}$	$1\frac{1}{2}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{1}{16}$	$\frac{5}{8}$	$1\frac{3}{16}$
$1\frac{5}{16}$ "	$1\frac{9}{16}$	$1\frac{5}{16}$	$2\frac{3}{16}$	$1\frac{9}{16}$	$\frac{17}{32}$	$\frac{19}{32}$	$\frac{3}{32}$	$\frac{21}{32}$	$1\frac{1}{4}$
$1\frac{3}{8}$ "	$1\frac{5}{8}$	$1\frac{3}{8}$	$2\frac{5}{16}$	$1\frac{5}{8}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{1}{16}$	$\frac{11}{16}$	$1\frac{5}{16}$
$1\frac{7}{16}$ "	$1\frac{11}{16}$	$1\frac{7}{16}$	$2\frac{7}{16}$	$1\frac{11}{16}$	$\frac{19}{32}$	$\frac{21}{32}$	$\frac{3}{32}$	$\frac{23}{32}$	$1\frac{3}{8}$
$1\frac{1}{2}$ "	$1\frac{3}{4}$	$1\frac{1}{2}$	$2\frac{9}{16}$	$1\frac{3}{4}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{1}{16}$	$\frac{3}{4}$	$1\frac{7}{16}$
$1\frac{9}{16}$ "	$1\frac{13}{16}$	$1\frac{9}{16}$	$2\frac{11}{16}$	$1\frac{13}{16}$	$\frac{21}{32}$	$\frac{23}{32}$	$\frac{3}{32}$	$\frac{25}{32}$	$1\frac{1}{2}$
$1\frac{5}{8}$ "	$1\frac{7}{8}$	$1\frac{5}{8}$	$2\frac{13}{16}$	$1\frac{7}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{1}{16}$	$\frac{13}{16}$	$1\frac{9}{16}$
$1\frac{11}{16}$ "	$1\frac{15}{16}$	$1\frac{11}{16}$	$2\frac{15}{16}$	$1\frac{15}{16}$	$\frac{23}{32}$	$\frac{25}{32}$	$\frac{3}{32}$	$\frac{27}{32}$	$1\frac{5}{8}$
$1\frac{3}{4}$ "	2	$1\frac{3}{4}$	3	2	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{1}{16}$	$\frac{7}{8}$	$1\frac{11}{16}$
$1\frac{13}{16}$ "	$2\frac{1}{16}$	$1\frac{13}{16}$	$3\frac{1}{16}$	$2\frac{1}{16}$	$\frac{25}{32}$	$\frac{27}{32}$	$\frac{3}{32}$	$\frac{29}{32}$	$1\frac{3}{4}$
$1\frac{7}{8}$ "	$2\frac{1}{8}$	$1\frac{7}{8}$	$3\frac{1}{8}$	$2\frac{1}{8}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{1}{16}$	$\frac{15}{16}$	$1\frac{13}{16}$
$1\frac{15}{16}$ "	$2\frac{3}{16}$	$1\frac{15}{16}$	$3\frac{3}{16}$	$2\frac{3}{16}$	$\frac{27}{32}$	$\frac{29}{32}$	$\frac{3}{32}$	$\frac{31}{32}$	$1\frac{7}{8}$
1 "	$1\frac{1}{2}$	1	$1\frac{3}{16}$	$\frac{3}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{16}$	$\frac{1}{2}$	$\frac{3}{4}$
$1\frac{1}{16}$ "	$1\frac{9}{16}$	$1\frac{1}{16}$	$1\frac{5}{16}$	$\frac{13}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{1}{16}$	$\frac{9}{16}$	$\frac{13}{16}$
$1\frac{1}{8}$ "	$1\frac{5}{8}$	$1\frac{1}{8}$	$1\frac{7}{16}$	$\frac{7}{8}$	$\frac{11}{32}$	$\frac{3}{8}$	$\frac{1}{16}$	$\frac{5}{8}$	$\frac{7}{8}$
$1\frac{3}{16}$ "	$1\frac{11}{16}$	$1\frac{3}{16}$	$1\frac{9}{16}$	$\frac{15}{16}$	$\frac{23}{32}$	$\frac{25}{32}$	$\frac{3}{32}$	$\frac{27}{32}$	$1\frac{1}{16}$
$1\frac{1}{4}$ "	$1\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{11}{16}$	$\frac{15}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{16}$	$\frac{1}{2}$	$\frac{15}{16}$
$1\frac{5}{16}$ "	$1\frac{13}{16}$	$1\frac{5}{16}$	$1\frac{13}{16}$	$1\frac{1}{16}$	$\frac{25}{32}$	$\frac{27}{32}$	$\frac{3}{32}$	$\frac{29}{32}$	1
$1\frac{3}{8}$ "	$1\frac{7}{8}$	$1\frac{3}{8}$	$1\frac{15}{16}$	$1\frac{1}{8}$	$\frac{13}{32}$	$\frac{15}{32}$	$\frac{3}{32}$	$\frac{17}{32}$	$1\frac{1}{8}$
$1\frac{7}{16}$ "	$1\frac{15}{16}$	$1\frac{7}{16}$	$2\frac{1}{16}$	$1\frac{3}{16}$	$\frac{27}{32}$	$\frac{29}{32}$	$\frac{3}{32}$	$\frac{31}{32}$	$1\frac{3}{16}$
$1\frac{1}{2}$ "	2	$1\frac{1}{2}$	$2\frac{3}{16}$	$1\frac{1}{4}$	$\frac{7}{8}$	$\frac{15}{16}$	$\frac{1}{16}$	1	$1\frac{1}{4}$
$1\frac{9}{16}$ "	$2\frac{1}{16}$	$1\frac{9}{16}$	$2\frac{5}{16}$	$1\frac{5}{16}$	$\frac{29}{32}$	1	$\frac{3}{32}$	$1\frac{1}{32}$	$1\frac{5}{16}$
$1\frac{5}{8}$ "	$2\frac{1}{8}$	$1\frac{5}{8}$	$2\frac{7}{16}$	$1\frac{3}{8}$	$\frac{15}{16}$	$1\frac{1}{16}$	$\frac{1}{16}$	$1\frac{1}{16}$	$1\frac{3}{8}$
$1\frac{11}{16}$ "	$2\frac{3}{16}$	$1\frac{11}{16}$	$2\frac{9}{16}$	$1\frac{7}{16}$	$\frac{31}{32}$	$1\frac{1}{32}$	$\frac{3}{32}$	$1\frac{3}{32}$	$1\frac{7}{16}$
$1\frac{3}{4}$ "	$2\frac{5}{16}$	$1\frac{3}{4}$	$2\frac{11}{16}$	$1\frac{1}{2}$	1	$1\frac{1}{16}$	$\frac{1}{16}$	$1\frac{1}{8}$	$1\frac{1}{2}$
$1\frac{13}{16}$ "	$2\frac{7}{16}$	$1\frac{13}{16}$	$2\frac{13}{16}$	$1\frac{9}{16}$	$1\frac{1}{32}$	$1\frac{3}{32}$	$\frac{3}{32}$	$1\frac{5}{32}$	$1\frac{9}{16}$
$1\frac{7}{8}$ "	$2\frac{1}{2}$	$1\frac{7}{8}$	$2\frac{15}{16}$	$1\frac{5}{8}$	$1\frac{1}{16}$	$1\frac{1}{8}$	$\frac{1}{16}$	$1\frac{3}{16}$	$1\frac{5}{8}$
$1\frac{15}{16}$ "	$2\frac{9}{16}$	$1\frac{15}{16}$	3	$1\frac{11}{16}$	$1\frac{3}{32}$	$1\frac{5}{32}$	$\frac{3}{32}$	$1\frac{7}{32}$	$1\frac{11}{16}$
2 "	$2\frac{5}{8}$	2	$3\frac{1}{16}$	$1\frac{3}{4}$	$1\frac{1}{16}$	$1\frac{3}{16}$	$\frac{1}{16}$	$1\frac{1}{4}$	$1\frac{3}{4}$
$2\frac{1}{16}$ "	$2\frac{11}{16}$	$2\frac{1}{16}$	$3\frac{3}{16}$	$1\frac{13}{16}$	$1\frac{5}{32}$	$1\frac{7}{32}$	$\frac{3}{32}$	$1\frac{9}{32}$	$1\frac{13}{16}$
$2\frac{1}{8}$ "	$2\frac{3}{4}$	$2\frac{1}{8}$	$3\frac{5}{16}$	$1\frac{7}{8}$	$1\frac{3}{32}$	$1\frac{1}{4}$	$\frac{1}{16}$	$1\frac{5}{16}$	$1\frac{7}{8}$
$2\frac{3}{16}$ "	$2\frac{13}{16}$	$2\frac{3}{16}$	$3\frac{7}{16}$	$1\frac{15}{16}$	$1\frac{7}{32}$	$1\frac{9}{32}$	$\frac{3}{32}$	$1\frac{11}{32}$	$1\frac{15}{16}$
$2\frac{1}{4}$ "	$2\frac{7}{8}$	$2\frac{1}{4}$	$3\frac{9}{16}$	2	$1\frac{1}{16}$	$1\frac{5}{16}$	$\frac{1}{16}$	$1\frac{3}{8}$	2

Keys and Keyways.

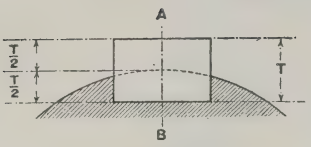


FIG. 18.

BRITISH STANDARD KEYS, KEY BARS, AND KEYWAYS.*

(No. 46—1909.) (*Abstract.*)

The keyway shall be cut in the shaft to a depth equal to one-half the thickness of the standard key it is intended to receive, and this depth shall be measured at the centre line (A–B) of the keyway.

Designation.	Shafts. Over.	Shafts. To	Key. Nominal Width.	Key. Nominal Thickness.	Key Bar. Min. Width.	Key Bar. Min. Thickness.	Key Bar. Max. Width.	Key Bar. Max. Thickness.	Keyway. Min. Width.	Keyway. Tolerance.	Keyway. Max. Width.
	Ins. Diam.	Ins. Diam.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	In.	Ins.
B.S.K. 1	1·25	1·50	0·375	0·2500	0·377	0·2520	0·379	0·2540	0·3740	−0·0010	0·375
B.S.K. 4	2·50	3·00	0·750	0·5000	0·752	0·5020	0·754	0·5040	0·7480	−0·0020	0·750
B.S.K. 7	4·00	5·00	1·250	0·8125	1·252	0·8145	1·255	0·8175	1·2480	−0·0020	1·250
B.S.K.10	7·00	8·00	2·000	1·3750	2·002	1·3770	2·006	1·3810	1·9970	−0·0030	2·000

Handles.

PROPORTIONS OF COCK HANDLES.

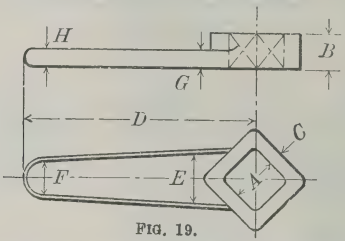


FIG. 19.

A.	B.	C.	D.	E.	F.	G.	H.
$\frac{1}{2}$	$\frac{1}{2}$	$\frac{3}{16}$	5	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{1}{4}$	$\frac{3}{16}$
$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{16}$	6	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{1}{4}$	$\frac{3}{16}$
$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{16}$	7	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{1}{4}$	$\frac{3}{16}$
$\frac{7}{8}$	$\frac{7}{8}$	$\frac{1}{4}$	8	1	$\frac{7}{8}$	$\frac{5}{16}$	$\frac{1}{4}$
1	1	$\frac{1}{4}$	9	$1\frac{1}{8}$	1	$\frac{5}{16}$	$\frac{1}{4}$
$1\frac{1}{8}$	$1\frac{1}{8}$	$\frac{1}{4}$	10	$1\frac{1}{4}$	1	$\frac{5}{16}$	$\frac{1}{4}$
$1\frac{1}{4}$	$1\frac{1}{4}$	$\frac{5}{16}$	11	$1\frac{1}{2}$	$1\frac{1}{8}$	$\frac{3}{8}$	$\frac{5}{16}$
$1\frac{3}{8}$	$1\frac{3}{8}$	$\frac{5}{16}$	12	$1\frac{5}{8}$	$1\frac{1}{4}$	$\frac{3}{8}$	$\frac{5}{16}$
$1\frac{1}{2}$	$1\frac{1}{2}$	$\frac{3}{8}$	13	$1\frac{3}{4}$	$1\frac{1}{2}$	$\frac{7}{16}$	$\frac{3}{8}$
$1\frac{5}{8}$	$1\frac{1}{2}$	$\frac{3}{8}$	13	$1\frac{3}{4}$	$1\frac{3}{8}$	$\frac{7}{16}$	$\frac{3}{8}$
$1\frac{11}{16}$	$1\frac{1}{2}$	$\frac{7}{16}$	13	$1\frac{3}{4}$	$1\frac{3}{8}$	$\frac{7}{16}$	$\frac{3}{8}$
$1\frac{3}{4}$	$1\frac{1}{2}$	$\frac{7}{16}$	$13\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{3}{8}$	$\frac{7}{16}$	$\frac{7}{16}$
$1\frac{5}{16}$	$1\frac{1}{2}$	$\frac{7}{16}$	$13\frac{1}{2}$	2	$1\frac{3}{8}$	$\frac{1}{2}$	$\frac{7}{16}$

(*'Machinery,' June* 3, 1921.)

* By permission of the British Engineering Standards Association.

PROPORTIONS OF MACHINE HANDLES.

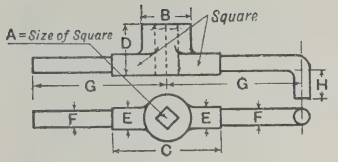


FIG. 20.

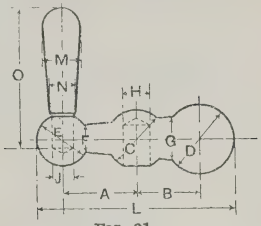


FIG. 21.

TABLE—FIG. 20.

Size of Square	A.	B.	C.	D.	E.	F.	G.	H.
$\frac{1}{2}$	$\frac{9}{16}$	1	2	$1\frac{1}{2}$	$\frac{9}{16}$	$\frac{1}{2}$	3	$2\frac{1}{2}$
$\frac{5}{8}$	$\frac{11}{16}$	$1\frac{1}{4}$	$2\frac{1}{2}$	2	$\frac{11}{16}$	$\frac{5}{8}$	4	$2\frac{1}{2}$
$\frac{3}{4}$	$\frac{13}{16}$	$1\frac{3}{4}$	3	$2\frac{1}{4}$	$\frac{13}{16}$	$\frac{3}{4}$	5	3
$\frac{7}{8}$	$\frac{15}{16}$	$1\frac{3}{4}$	$3\frac{1}{2}$	$2\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$	6	$3\frac{1}{2}$
1	$1\frac{1}{16}$	$1\frac{3}{4}$	4	3	1	1	$7\frac{1}{2}$	4
$1\frac{1}{8}$	$1\frac{3}{16}$	$2\frac{1}{4}$	$4\frac{1}{4}$	$3\frac{1}{2}$	$1\frac{1}{8}$	$1\frac{1}{8}$	$8\frac{1}{4}$	4
$1\frac{1}{4}$	$1\frac{5}{16}$	$2\frac{3}{8}$	$4\frac{3}{4}$	$3\frac{3}{4}$	$1\frac{1}{8}$	1	9	4

TABLE—FIG. 21.

A.	B.	C.	D.	E.	F.	G.	H.	J.	L.	M.	N.	O.
$1\frac{1}{2}$	$1\frac{1}{4}$	1	$1\frac{1}{4}$	$\frac{7}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{5}{16}$	$3\frac{3}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$2\frac{1}{2}$
$2\frac{1}{4}$	2	$1\frac{1}{8}$	$1\frac{3}{8}$	1	$\frac{9}{16}$	$\frac{11}{16}$	$\frac{1}{2}$	$\frac{5}{16}$	$5\frac{7}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	3
3	$2\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{1}{8}$	$\frac{9}{16}$	$\frac{13}{16}$	$\frac{9}{16}$	$\frac{5}{16}$	$7\frac{1}{16}$	$\frac{7}{8}$	$\frac{7}{8}$	$3\frac{1}{2}$
$3\frac{3}{4}$	$3\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{5}{8}$	$1\frac{1}{4}$	$\frac{11}{16}$	$\frac{15}{16}$	$\frac{5}{8}$	$\frac{5}{16}$	9	$1\frac{1}{16}$	$1\frac{1}{16}$	4
$4\frac{1}{2}$	$4\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{3}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{11}{16}$	$\frac{5}{16}$	$10\frac{5}{16}$	1	$1\frac{7}{32}$	$4\frac{1}{2}$

PROPORTIONS OF MACHINE HAND WHEELS.

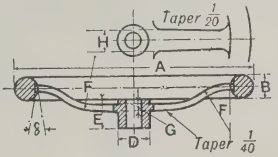


FIG. 22.

A.	B.	C.	D.	E.	F.	G.	H.
4	$\frac{3}{8}$	$\frac{3}{64}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{16}$	$\frac{1}{16}$	$\frac{3}{8}$
5	$\frac{7}{16}$	$\frac{1}{16}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{7}{8}$	$\frac{5}{32}$	$\frac{7}{16}$
6	$\frac{1}{2}$	$\frac{1}{64}$	$\frac{5}{8}$	$\frac{5}{8}$	$1\frac{1}{16}$	$\frac{3}{16}$	$\frac{1}{2}$
7	$\frac{3}{8}$	$\frac{3}{32}$	1"	1"	$1\frac{1}{4}$	$\frac{7}{32}$	$\frac{5}{8}$
8	$\frac{7}{16}$	$\frac{7}{64}$	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{7}{16}$	$\frac{1}{4}$	$\frac{3}{4}$
9	$\frac{1}{2}$	$\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$\frac{9}{32}$	$\frac{13}{16}$
10	1	$\frac{5}{64}$	$1\frac{3}{8}$	$1\frac{3}{8}$	$1\frac{11}{16}$	$\frac{5}{16}$	$\frac{15}{16}$
12	$1\frac{1}{4}$	$\frac{3}{32}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$2\frac{1}{4}$	$\frac{3}{8}$	$1\frac{1}{4}$

Spanners.

British Standard Spanners.*

(No. 192—1924.) (*Abstract.*)

The angle between the head and the shank of spanners should be 15°, 30°, or 45°.

Dimensions of British Standard Spanners.

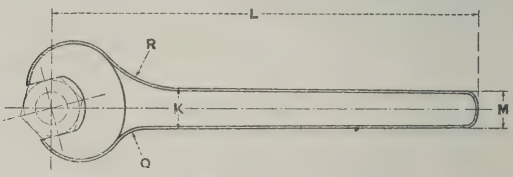


Fig. 23.—Single-ended B.S. Spanner.

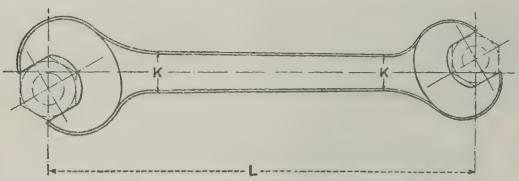


Fig. 24.—Double-ended B.S. Spanners.

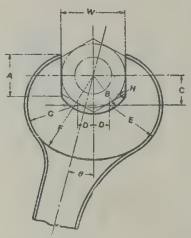


Fig. 25.—Key Diagram of Head of B.S. Spanner.

Fig. 26.—Front View of B.S. Spanners.

Box Spanners or Keys.

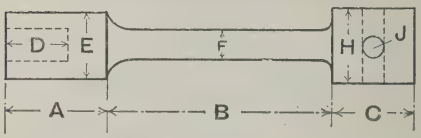


Fig. 27.

Dimensions in Inches.								
Nut.	A.	B.	C.	D.	E.	F.	H.	J.
$\frac{5}{8}$	$1\frac{1}{4}$	$3\frac{1}{2}$	1	$\frac{7}{8}$	$1\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{8}$	$\frac{7}{16}$
$\frac{3}{4}$	$1\frac{3}{8}$	$3\frac{1}{2}$	$1\frac{1}{4}$	1	$1\frac{5}{8}$	$\frac{7}{8}$	$1\frac{1}{4}$	$\frac{9}{16}$
$\frac{7}{8}$	$1\frac{3}{4}$	4	$1\frac{3}{4}$	$1\frac{1}{4}$	$2\frac{1}{4}$	1	$1\frac{1}{2}$	$\frac{11}{16}$
1	2	$4\frac{1}{2}$	2	$1\frac{1}{2}$	$2\frac{1}{4}$	$1\frac{1}{4}$	2	
$1\frac{1}{8}$	$2\frac{3}{8}$	5	$2\frac{1}{4}$	$1\frac{5}{8}$	$2\frac{3}{4}$	$1\frac{5}{8}$	$2\frac{1}{4}$	$\frac{15}{16}$
$1\frac{1}{4}$	$2\frac{5}{8}$	$5\frac{1}{2}$	$2\frac{1}{2}$	$1\frac{3}{4}$	3	$1\frac{1}{2}$	$2\frac{1}{2}$	$1\frac{1}{16}$
$1\frac{1}{2}$	3	$5\frac{1}{2}$	3	2	$3\frac{1}{4}$	$1\frac{5}{8}$	$2\frac{3}{4}$	$1\frac{1}{16}$

* By permission of the British Engineering Standards Association.

DIMENSIONS OF BRITISH STANDARD SPANNERS.

All dimensions are in inches.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Dia. of Bolt.		Dimensions of Spanner.																	
B.S.W.	B.S.F.	Max.	Min.	A	B	C	D	E	F	G	H	J	K	L	M	N	P	Q	R
—	$\frac{7}{32}$	0·418	0·415	0·234	0·239	0·178	0·104	0·313	0·501	0·295	0·021	0·167	0·312	3·25	0·275	0·112	0·099	0·261	0·748
—	$\frac{1}{4}$	0·450	0·447	0·273	0·263	0·200	0·112	0·355	0·565	0·336	0·022	0·180	0·336	3·50	0·296	0·121	0·107	0·281	0·806
$\frac{1}{4}$	*$\frac{9}{32}$ $\frac{5}{16}$	0·530	0·527	0·313	0·307	0·232	0·132	0·410	0·653	0·387	0·050	0·212	0·395	5	0·348	0·142	0·126	0·331	0·949
$\frac{5}{16}$	$\frac{3}{8}$	0·605	0·602	0·391	0·359	0·277	0·151	0·496	0·786	0·470	0·063	0·242	0·451	5	0·398	0·162	0·144	0·378	1·083
$\frac{3}{8}$	$\frac{7}{16}$	0·715	0·712	0·469	0·427	0·331	0·178	0·593	0·939	0·563	0·075	0·286	0·533	6	0·470	0·192	0·170	0·447	0·280
$\frac{7}{16}$	$\frac{1}{2}$	0·825	0·822	0·547	0·494	0·384	0·206	0·689	1·090	0·654	0·088	0·330	0·615	6	0·542	0·221	0·196	0·516	1·477
$\frac{1}{2}$	$\frac{9}{16}$	0·925	0·922	0·583	0·546	0·419	0·231	0·747	1·186	0·708	0·100	0·370	0·690	8	0·608	0·248	0·220	0·578	1·656
$\frac{9}{16}$	$\frac{5}{8}$	1·015	1·012	0·656	0·604	0·466	0·253	0·833	1·321	0·791	0·113	0·406	0·757	8	0·667	0·272	0·242	0·634	1·817
$\frac{5}{8}$	*$\frac{11}{16}$	1·105	1·102	0·729	0·661	0·513	0·276	0·920	1·455	0·873	0·125	0·442	0·824	9	0·726	0·296	0·263	0·691	1·978
*$\frac{11}{16}$	$\frac{3}{4}$ *$\frac{13}{16}$	1·205	1·202	0·802	0·723	0·562	0·301	1·009	1·596	0·958	0·138	0·482	0·899	9	0·792	0·323	0·287	0·753	2·157
$\frac{3}{4}$	$\frac{7}{8}$	1·305	1·302	0·875	0·785	0·611	0·326	1·098	1·736	1·043	0·150	0·522	0·974	10	0·857	0·350	0·311	0·816	2·336
*$\frac{13}{16}$	*$\frac{15}{16}$	1·395	1·392	0·880	0·824	0·632	0·348	1·127	1·790	1·068	0·163	0·558	1·041	10	0·917	0·374	0·332	0·872	2·497
$\frac{7}{8}$	1	1·485	1·482	0·948	0·880	0·677	0·371	1·209	1·918	1·146	0·175	0·594	1·108	12	0·976	0·398	0·353	0·928	2·658
*$\frac{15}{16}$	—	1·585	1·582	1·016	0·941	0·724	0·396	1·294	2·052	1·227	0·188	0·634	1·182	12	1·041	0·425	0·377	0·991	2·837
1	$1\frac{1}{8}$	1·675	1·672	1·083	0·997	0·769	0·418	1·376	2·181	1·305	0·200	0·670	1·250	15	1·100	0·449	0·399	1·047	2·998
$1\frac{1}{8}$	$1\frac{1}{4}$	1·870	1·864	1·125	1·090	0·826	0·467	1·465	2·335	1·386	0·225	0·748	1·395	15	1·229	0·501	0·445	1·169	3·348
$1\frac{1}{4}$	*$1\frac{3}{8}$	2·060	2·054	1·250	1·203	1·914	0·514	1·622	2·583	1·534	0·250	0·824	1·537	18	1·353	0·552	0·490	1·288	3·688
*$1\frac{3}{8}$	$1\frac{1}{2}$	2·230	2·224	1·375	1·309	0·998	0·557	1·775	2·823	1·680	0·275	0·892	1·664	18	1·465	0·598	0·531	1·394	3·992
$1\frac{1}{2}$	*$1\frac{5}{8}$	2·420	2·414	1·500	1·422	1·086	0·604	1·933	3·073	1·830	0·300	0·968	1·805	21	1·590	0·649	0·576	1·313	4·332
*$1\frac{5}{8}$	$1\frac{3}{4}$	2·590	2·584	1·625	1·528	1·170	0·647	2·086	3·313	1·976	0·325	1·036	1·932	21	1·702	0·694	0·616	1·619	4·637
$1\frac{3}{4}$	2	2·770	2·764	1·750	1·637	1·256	0·692	2·240	3·556	2·122	0·350	1·108	2·066	23	1·820	0·742	0·659	1·731	4·958
*$1\frac{7}{8}$	—	3·030	3·024	1·875	1·780	1·359	0·757	2·418	3·844	2·289	0·375	1·212	2·260	23	1·991	0·812	0·721	1·894	5·424
2	$2\frac{1}{4}$	3·160	3·154	2·000	1·869	1·435	0·789	2·559	4·062	2·425	0·400	1·264	2·357	26	2·076	0·847	0·752	1·975	5·657

* The Association recommends that for general use the nuts and bolts corresponding with these spanner sizes be dispensed with.

Pipe Flanges.

Flanges and Bolts of Cast-iron Pipes.

Diameter of Pipe.	Diameter of Flange.	Flange Thickness.	Size of Bolts.	Number of Bolts.	Bolt Circle.	Weight per Foot.
Ins.	Ins.	Ins.	Ins.		Ins.	Lbs.
1½	6	¾	½	3	4½	—
2	6½	⅞	½	4	4	8·74
2½	7½	1	⅝	4	5½	10·58
3	8	1	⅝	4	6½	14·8
3½	8½	1	¾	4	6½	22·1
4	9¼	1 1/16	¾	6	7½	30·8
4½	10	1 1/16	¾	6	7¾	36·3
5	11	1⅛	¾	6	8¼	52
6	12½	1⅛	¾	6	—	58·7
7	14	1 3/16	⅞	8	11½	71·8
8	15	1 3/16	⅞	8	12½	79·1
9	16½	1¼	¾	8	14	—
10	17½	1¼	¾	10	14¾	—
11	19	1 5/16	⅞	10	16	—
12	20	1 5/16	⅞	10	17	—

For steam pipes allow 2½ inches per 100 feet for expansion.

British Standard Pipe Flanges.

(No. 10—Part 1—1928.)

Relates to Pipe Flanges (for Land Use) for Gas Pipes up to 30 lbs. pressure per square inch, and Water Pipes up to 175 lbs. pressure per square inch.

(No. 10—Part 2—1926.) (*Abstract.*)*

Table D.—For Working Steam Pressures up to 50 lbs. per sq. in.

Nominal Pipe Size.	Diameter of Flange.	Diameter of Bolt Circle.	Number of Bolts.	Diameter of Bolts.	Thickness of Flanges.		
					Cast Iron	Cast Steel and Bronze.	Stamped or Forged Iron or Steel.
Ins.	Ins.	Ins.		Ins.	Ins.	Ins.	Ins.
⅛	3¾	2⅝	4	½	½	⅜	3/16
¼	4	2¾	4	½	½	⅜	3/16
¾	4½	3¼	4	½	½	⅜	3/16
1	4¾	3 7/16	4	½	⅝	½	¼
1¼	5¼	3¾	4	½	⅝	½	¼
1½	6	4½	4	⅝	¾	9/16	5/16
2	6½	5	4	⅝	¾	9/16	5/16
2½	7¼	5⅝	4	⅝	¾	9/16	⅜
3	8	6½	4	¾	⅞	11/16	⅜
3½	8½	7	4	⅞	⅞	11/16	½
4	9	7½	8	⅝	⅞	11/16	½
5	10	8½	8	⅝	1	¾	½
6	11	9¼	8	⅝	1	¾	½
7	12	10¼	8	¾	1⅛	⅞	⅝
8	13½	11½	8	¾	1⅛	⅞	⅝
9	14½	12¾	8	¾	1⅛	⅞	⅝
10	16	14	8	¾	1¼	1	¾
12	18	16	12	¾	1⅜	1⅛	¾
14	20¾	18½	12	⅞	1⅜	1⅛	⅞
15	21¾	19½	12	⅞	1½	1¼	⅞
16	22¾	20½	12	⅞	1½	1¼	⅞
18	25¼	23	12	⅞	1⅝	1⅜	1
20	27½	25¼	16	⅞	1¾	1½	1
21	29	26¼	16	⅞	1¾	1½	1
24	32½	29¾	16	1	1⅞	1⅝	1⅛

Bolt holes.—For ½-in. and ⅝-in. bolts the diameters of the holes to be 1/16-in. larger than the diameter of the bolts, and for larger sizes of bolts ½-in. Bolt holes to be drilled off centre lines.

* By permission of the British Engineering Standards Association.

Tables E. F, H, and J relate to pressures up to 100 lbs., 150 lbs., 250 lbs., and 350 lbs., respectively.

TABLE K.—For Working Steam Pressures up to 450 lbs. per sq. in.

Nominal Pipe Size.	Actual Outside Diameter of Wrought Pipe.	Diameter of Flange.	Diameter of Bolt Circle.	Number of Bolts.	Diameter of Bolts.	Thickness of Flange. Cast Steel and Bronze; Stamped or Forged Steel.
ins.	ins.	ins.	ins.		ins.	ins.
$\frac{1}{2}$	$\frac{27}{32}$	$4\frac{1}{2}$	$3\frac{1}{4}$	4	$\frac{5}{8}$	$\frac{3}{4}$
$\frac{3}{4}$	$1\frac{1}{16}$	$4\frac{1}{2}$	$3\frac{1}{4}$	4	$\frac{5}{8}$	$\frac{3}{4}$
1	$1\frac{5}{16}$	5	$3\frac{3}{4}$	4	$\frac{5}{8}$	$\frac{7}{8}$
$1\frac{1}{4}$	$1\frac{11}{16}$	$5\frac{1}{2}$	$3\frac{7}{8}$	4	$\frac{5}{8}$	$\frac{7}{8}$
$1\frac{1}{2}$	$1\frac{29}{32}$	6	$4\frac{1}{2}$	4	$\frac{3}{4}$	1
2	$2\frac{3}{8}$	$6\frac{1}{2}$	5	8	$\frac{5}{8}$	1
$2\frac{1}{2}$	3	$7\frac{1}{4}$	$5\frac{3}{4}$	8	$\frac{3}{4}$	$1\frac{1}{8}$
3	$3\frac{1}{2}$	8	$6\frac{1}{2}$	8	$\frac{3}{4}$	$1\frac{1}{4}$
$3\frac{1}{2}$	4	9	$7\frac{1}{4}$	8	$\frac{7}{8}$	$1\frac{1}{4}$
4	$4\frac{1}{2}$	$9\frac{1}{2}$	$7\frac{7}{8}$	8	$\frac{7}{8}$	$1\frac{3}{8}$
5	$5\frac{1}{2}$	11	$9\frac{1}{2}$	12	$\frac{7}{8}$	$1\frac{5}{8}$
6	$6\frac{1}{2}$	12	$10\frac{1}{4}$	12	$\frac{7}{8}$	$1\frac{5}{8}$
7	$7\frac{1}{2}$	$13\frac{1}{2}$	$11\frac{1}{2}$	12	1	$1\frac{3}{4}$
8	$8\frac{1}{2}$	$14\frac{1}{2}$	$12\frac{1}{2}$	12	1	$1\frac{7}{8}$
9	$9\frac{1}{2}$	16	14	16	1	2
10	$10\frac{1}{2}$	17	15	16	1	2
12	$12\frac{1}{2}$	$19\frac{1}{4}$	17	16	$1\frac{1}{8}$	$2\frac{1}{4}$
14	15	$22\frac{1}{2}$	20	16	$1\frac{1}{4}$	$2\frac{3}{8}$
15	16	$23\frac{3}{4}$	$21\frac{1}{2}$	20	$1\frac{1}{4}$	$2\frac{1}{2}$
16	17	$24\frac{3}{4}$	$22\frac{1}{2}$	20	$1\frac{1}{4}$	$2\frac{5}{8}$

Bolt holes.—For $\frac{1}{2}$-in. and $\frac{5}{8}$-in. bolts the diameters of the holes to be $\frac{1}{16}$-in. larger than the diameter of the bolts, and for larger sizes of bolts $\frac{1}{8}$-in. Bolt holes to be drilled off centre lines.

TABLE M.—Welded-on Flanges for Pipe Lines for Working Steam Pressures up to 250 lbs.

Nominal Pipe Size.	Actual Outside Diameter of Wrought Pipe.	Diameter of Flange.	Diameter of Bolt Circle.	Number of Bolts.	Diameter of Bolts.	Thickness of Flange. Steel (stamped or forged).
ins.	ins.	ins.	ins.		ins.	ins.
2	$2\frac{3}{8}$	6	$4\frac{1}{2}$	4	$\frac{5}{8}$	$\frac{11}{16}$
$2\frac{1}{2}$	3	$6\frac{1}{2}$	5	4	$\frac{5}{8}$	$\frac{3}{4}$
3	$3\frac{1}{2}$	$7\frac{1}{4}$	$5\frac{3}{4}$	8	$\frac{5}{8}$	$\frac{3}{4}$
$3\frac{1}{2}$	4	8	$6\frac{1}{2}$	8	$\frac{5}{8}$	$\frac{7}{8}$
4	$4\frac{1}{2}$	$8\frac{1}{2}$	7	8	$\frac{5}{8}$	$\frac{7}{8}$
5	$5\frac{1}{2}$	10	$8\frac{1}{2}$	8	$\frac{3}{4}$	1
6	$6\frac{1}{2}$	11	$9\frac{1}{4}$	8	$\frac{3}{4}$	$1\frac{1}{8}$
7	$7\frac{1}{2}$	12	$10\frac{1}{4}$	12	$\frac{3}{4}$	$1\frac{1}{8}$
8	$8\frac{1}{2}$	$13\frac{1}{2}$	$11\frac{1}{2}$	12	$\frac{3}{4}$	$1\frac{1}{4}$
9	$9\frac{1}{2}$	$14\frac{1}{2}$	$12\frac{3}{4}$	12	$\frac{3}{4}$	$1\frac{1}{4}$
10	$10\frac{1}{2}$	16	14	12	$\frac{7}{8}$	$1\frac{3}{8}$
12	$12\frac{1}{2}$	18	16	16	$\frac{7}{8}$	$1\frac{1}{2}$
14	15	$20\frac{3}{4}$	$18\frac{1}{2}$	16	1	$1\frac{5}{8}$
15	16	$21\frac{1}{4}$	$19\frac{1}{2}$	16	1	$1\frac{5}{8}$
16	17	$22\frac{3}{4}$	$20\frac{1}{2}$	16	1	$1\frac{5}{8}$
18	19	$25\frac{1}{4}$	23	20	1	$1\frac{5}{8}$
20	21	$27\frac{3}{4}$	$25\frac{1}{2}$	20	$1\frac{1}{8}$	2
21	22	29	$26\frac{1}{2}$	24	$1\frac{1}{8}$	2
24	25	$32\frac{1}{2}$	$29\frac{3}{4}$	24	$1\frac{1}{4}$	$2\frac{1}{4}$

Bolt holes.—For $\frac{1}{2}$-in. and $\frac{5}{8}$-in. bolts the diameters of the holes to be $\frac{1}{16}$-in. larger than the diameter of the bolts, and for larger sizes of bolts $\frac{1}{8}$-in. Bolt holes to be drilled off centre lines.

Tables L and P relate to pressures up to 150 lbs. and 350 lbs. per sq. in., respectively.

AMERICAN STANDARD PIPE FLANGES.

Pipe Size. Inches.	Pipe Thickness, Inches.	Thickness nearest Fraction, Inches.	Stress on Pipe per Square Inch at 200 lbs.	Radius of Fillet, Inches.	Flange Diameter, Inches.	Flange Thickness at Hub for Iron Pipe, Inches.	Flange Thickness at Edge, Inches.	Width Flange Face, Inches.	Bolt Circle Diameter, Inches.	Number of Bolts.	Bolt Size Diameters, Inches.	Bolt Length, Inches.	Stress on each Bolt per Square Inch at Bottom of Thread at 200 lbs.
2	·409	$\frac{7}{16}$	460	$\frac{1}{8}$	6	1	$\frac{5}{8}$	2	$4\frac{3}{4}$	4	$\frac{1}{2}$ $\frac{5}{8}$	2	825
$2\frac{1}{2}$	·429	$\frac{7}{16}$	550	$\frac{1}{8}$	7	$1\frac{1}{8}$	$\frac{11}{16}$	$2\frac{1}{4}$	$5\frac{1}{2}$	4	$\frac{5}{8}$ $\frac{5}{8}$	$2\frac{1}{4}$	1,050
3	·448	$\frac{7}{16}$	690	$\frac{1}{8}$	$7\frac{1}{2}$	$1\frac{1}{4}$	$\frac{3}{4}$	$2\frac{1}{4}$	6	4	$\frac{5}{8}$ $\frac{5}{8}$	$2\frac{1}{2}$	1,330
$3\frac{1}{2}$	·466	$\frac{1}{2}$	700	$\frac{1}{8}$	$8\frac{1}{2}$	$1\frac{1}{4}$	$\frac{13}{16}$	$2\frac{1}{2}$	7	4	$\frac{5}{8}$ $\frac{5}{8}$	$2\frac{1}{2}$	2,530
4	·486	$\frac{1}{2}$	800	$\frac{1}{8}$	9	$1\frac{3}{8}$	$\frac{15}{16}$	$2\frac{1}{2}$	$7\frac{1}{2}$	4	$\frac{5}{8}$ $\frac{5}{8}$	$2\frac{3}{4}$	2,100
$4\frac{1}{2}$	·498	$\frac{1}{2}$	900	$\frac{1}{8}$	$9\frac{1}{4}$	$1\frac{3}{8}$	$\frac{15}{16}$	$2\frac{3}{4}$	$7\frac{3}{4}$	8	$\frac{5}{8}$ $\frac{5}{8}$	3	1,430
5	·525	$\frac{1}{2}$	1,000	$\frac{1}{8}$	10	$1\frac{1}{2}$	$\frac{15}{16}$	$2\frac{1}{4}$	$8\frac{1}{2}$	8	$\frac{5}{8}$ $\frac{5}{8}$	3	1,630
6	·563	$\frac{9}{16}$	1,060	$\frac{1}{8}$	11	$1\frac{1}{2}$	1	$2\frac{1}{4}$	$9\frac{1}{2}$	8	$\frac{5}{8}$ $\frac{5}{8}$	3	2,360
7	·60	$\frac{5}{8}$	1,120	$\frac{1}{8}$	$12\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{16}$	$2\frac{1}{4}$	$10\frac{3}{4}$	8	$\frac{5}{8}$ $\frac{5}{8}$	$3\frac{1}{4}$	3,200
8	·639	$\frac{5}{8}$	1,280	$\frac{1}{8}$	$13\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{1}{8}$	$2\frac{1}{4}$	$11\frac{3}{4}$	8	$\frac{5}{8}$ $\frac{5}{8}$	$3\frac{1}{4}$	4,190
9	·678	$\frac{11}{16}$	1,310	$\frac{3}{16}$	15	$1\frac{3}{4}$	$1\frac{1}{8}$	3	$13\frac{1}{4}$	12	$\frac{5}{8}$ $\frac{5}{8}$	$3\frac{1}{4}$	3,610
10	·713	$\frac{3}{4}$	1,330	$\frac{3}{16}$	16	2	$1\frac{3}{16}$	3	$14\frac{1}{4}$	12	$\frac{5}{8}$ $\frac{5}{8}$	$3\frac{1}{2}$	2,970
12	·79	$\frac{13}{16}$	1,470	$\frac{3}{16}$	19	2	$1\frac{1}{4}$	$3\frac{1}{4}$	17	12	$\frac{5}{8}$ $\frac{5}{8}$	$3\frac{3}{4}$	4,280
14	·864	$\frac{7}{8}$	1,600	$\frac{3}{16}$	21	2	$1\frac{3}{8}$	$3\frac{1}{2}$	$18\frac{3}{4}$	12 $\frac{1}{8}$	$\frac{7}{8}$ $\frac{7}{8}$	$4\frac{1}{4}$	4,280
15	·904	$\frac{15}{16}$	1,600	$\frac{3}{16}$	$22\frac{1}{4}$	2	$1\frac{3}{8}$	$3\frac{3}{4}$	20	16 $\frac{1}{8}$	$\frac{7}{8}$ $\frac{7}{8}$	$4\frac{1}{4}$	3,660
16	·946	1	1,600	$\frac{3}{16}$	$23\frac{1}{2}$	$2\frac{1}{4}$	$1\frac{7}{16}$	$3\frac{3}{4}$	$21\frac{1}{4}$	16 $\frac{1}{8}$	$\frac{7}{8}$ $\frac{7}{8}$	$4\frac{1}{4}$	4,210
18	1·02	$1\frac{1}{16}$	1,690	$\frac{3}{16}$	25	...	$1\frac{9}{16}$	$3\frac{3}{4}$	$22\frac{3}{4}$	16 $\frac{1}{8}$	1 1	$4\frac{3}{4}$	4,540
20	1·09	$1\frac{1}{8}$	1,780	$\frac{3}{16}$	$27\frac{1}{2}$	...	$1\frac{11}{16}$	$3\frac{3}{4}$	25	20 $\frac{1}{8}$	1 $1\frac{1}{8}$	5	4,490
22	1·18	$1\frac{3}{16}$	1,850	$\frac{1}{4}$	$29\frac{1}{2}$	...	$1\frac{13}{16}$	$3\frac{3}{4}$	$27\frac{1}{4}$	20 $\frac{1}{8}$	1 $1\frac{1}{4}$	$5\frac{1}{4}$	4,320
24	1·25	$1\frac{1}{4}$	1,920	$\frac{1}{4}$	$31\frac{1}{2}$ 32	...	$1\frac{1}{8}$	4	$29\frac{1}{4}$ 29	20 $\frac{1}{8}$	1 $1\frac{1}{4}$	$5\frac{1}{2}$	5,130
26	1·30	$1\frac{5}{16}$	1,980	$\frac{1}{4}$	$33\frac{3}{4}$ $34\frac{1}{4}$	...	$1\frac{3}{8}$	$4\frac{1}{4}$	$31\frac{1}{4}$ $31\frac{3}{4}$	24 $\frac{1}{8}$	1 $1\frac{1}{4}$	$5\frac{3}{4}$	5,080
28	1·38	$1\frac{3}{8}$	2,040	$\frac{1}{4}$	36 $36\frac{1}{2}$	...	$1\frac{7}{16}$ $2\frac{1}{16}$	$4\frac{1}{4}$	$33\frac{1}{2}$ 34	28 $\frac{1}{8}$	1 $1\frac{1}{4}$	6	5,000
30	1·48	$1\frac{1}{2}$	2,000	$\frac{1}{4}$	38 $38\frac{1}{4}$	...	$1\frac{1}{2}$ $2\frac{1}{8}$	4 $4\frac{1}{2}$	$35\frac{1}{2}$ 36	28 $\frac{1}{8}$	$1\frac{1}{8}$ $1\frac{3}{8}$	$6\frac{1}{4}$	4,590
36	1·71	$1\frac{3}{4}$	1,920	$\frac{1}{4}$	$44\frac{1}{2}$ $45\frac{1}{4}$	...	$1\frac{3}{4}$ $2\frac{3}{8}$	$4\frac{1}{2}$ $4\frac{5}{8}$	42 $42\frac{3}{4}$	32 $\frac{1}{8}$	$1\frac{1}{8}$ $1\frac{3}{8}$	$6\frac{1}{2}$	5,790
42	1·87	2	2,100	$\frac{1}{4}$	51 $52\frac{3}{4}$	...	$1\frac{7}{8}$ $2\frac{5}{8}$	$4\frac{1}{2}$ $5\frac{3}{8}$	$48\frac{1}{2}$ $49\frac{1}{2}$	36 $\frac{1}{8}$	$1\frac{1}{4}$ $1\frac{1}{2}$	$7\frac{1}{4}$	5,700
48	2·17	$2\frac{1}{4}$	2,130	$\frac{1}{4}$	$57\frac{1}{2}$ $59\frac{1}{2}$	...	2 2	$4\frac{3}{4}$ $5\frac{3}{4}$	$54\frac{1}{4}$ 56	44 $\frac{1}{8}$	$1\frac{3}{8}$ $1\frac{1}{2}$	$7\frac{3}{4}$	6,090

The double sets of figures for flanges of pipe sizes above 24 in. are intended one for pressures of 200 lbs. and the other for less; this also applies to the two sizes of bolts.

TABLE OF GENERAL DIMENSIONS, FLEXIBILITY, ETC., OF JOINTS FOR THE VICTAULIC SYSTEM OF FLEXIBLE PIPE LINES.

Victaulic Joints for Mild Steel Expanded-End Pipes.						Victaulic Joints for Thickened-End Pipes (Cast-Iron).					
Bore.	A.	B.	C.	Angularity to adjacent pipe.	Gap between pipe ends	Bore.	A.	B.	C.	Angularity to adjacent pipe.	Gap between pipe ends
1¼	3 3/16	4⅞	1¾	1 in 16	⅛	3	5 9/16	7¼	2	1 in 32	⅛
2	3 11/16	6	1¾	1 in 21	⅛	4	6¾	8½	2¼	1 in 40	⅛
2½	4 7/16	6⅝	1¾	1 in 26	⅛	5	8¼	10½	2¼	1 in 50	⅛
3	5	7⅛	1¾	1 in 30·5	⅛	6	9½	11½	2¼	1 in 58	⅛
3½	5⅝	8	2	1 in 34·5	⅛	7	10⅝	13¾	2⅝	1 in 68	⅛
4	6¼	8⅝	2	1 in 38·5	⅛	8	11⅝	14¼	2⅝	1 in 76	⅛
5	7¼	9⅝	2	1 in 46·5	⅛	9	12¼	15⅝	2¾	1 in 85	⅛
6	8 7/16	11	2½	1 in 55	⅛	10	14 3/16	17	2¾	1 in 94	⅛
7	9⅞	12½	2½	1 in 63	⅛	12	16½	19¼	2¾	1 in 112	⅛
8	11	13¾	2½	1 in 72	⅛						
9	12⅛	14⅝	2⅝	1 in 80	⅛						
10	13⅛	16¼	2½	1 in 88	⅛						
12	15⅛	18¼	2½	1 in 104	⅛						

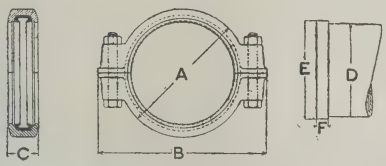


Fig. 28.

DIMENSIONS OF PIPE-ENDS FOR VICTAULIC SYSTEM OF FLEXIBLE JOINTS, AS STANDARDISED BY TUBE MAKERS.

Expanded Ends.				Thickened Ends.			
Nom. Bore.	O.D. Pipe. (D.)	O.D. Expanded End. (E.)	Length of Expanded End. (F.)	Nom. Bore.	O.D. Pipe. (D.)	O.D. Thickened Ends. (E.)	Length of Thickened End. (F)
1¼	1⅜	2	5/8	3	3·76	4	5/8
2	2⅜	2⅝	5/8	4	4·8	5	11/16
2½	2⅞	3¼	5/8	5	5·9	6¼	11/16
3	3½	3 13/16	5/8	6	6·98	7¼	11/16
3½	4	4 5/16	11/16	7	8·06	8¼	13/16
4	4½	4 13/16	11/16	8	9·14	9¼	13/16
5	5½	5 13/16	11/16	9	10·2	10⅝	13/16
6	6½	6¾	11/16	10	11·26	11½	13/16
7	7½	7⅞	11/16	12	13·6	14	13/16
8	8½	9	13/16				
9	9½	10	13/16				
10	10½	11	13/16				
12	12½	13	13/16				

Screw Taps and Tapping.

Whitworth's Standard Taps.

Dimensions in Inches.						Threads per Inch.
Outside Diameter.	Full Length.	Length of Screw Part.	Length of Square.	Size of Square.	Diameter at Bottom of Thread.	
1/16	1 1/2	5/8	3/16	—	·0413	60
3/32	1 5/8	3/4	3/16	—	·0671	48
1/8	1 3/4	7/8	1/4	—	·093	40
5/32	1 7/8	7/8	1/4	—	·112	32
3/16	2	1	5/16	—	·134	24
7/32	2 1/8	1 1/16	5/16	—	·165	24
1/4	2 1/4	1 1/8	3/8	—	·186	20
5/16	2 1/2	1 3/8	7/16	5/32	·241	18
3/8	2 3/4	1 1/2	1/2	7/32	·295	16
7/16	3 1/4	1 3/4	1/2	9/32	·346	14
1/2	3 1/2	2	9/16	5/16	·393	12
9/16	3 3/4	2	9/16	5/16	·455	12
5/8	4	2 1/4	5/8	3/8	·508	11
11/16	4 1/4	2 3/8	11/16	3/8	·571	11
3/4	4 1/2	2 1/2	3/4	7/16	·622	10
13/16	4 3/4	2 5/8	3/4	7/16	·684	10
7/8	5	2 7/8	13/16	1/2	·732	9
15/16	5 1/2	3	13/16	1/2	·795	9
1	5 1/2	3 1/4	7/8	5/8	·84	8
1 1/8	6	3 1/2	1	11/16	·942	7
1 1/4	6 1/2	3 3/4	1 1/8	13/16	1·067	7
1 3/8	7 1/2	4 1/4	1 1/8	13/16	1·161	6
1 1/2	8	4 3/4	1 1/4	1	1·286	6
1 5/8	8 1/2	5 1/4	1 1/4	1	1·368	5
1 3/4	9	5 1/2	1 3/8	1 1/8	1·494	5
1 7/8	9 1/2	5 3/4	1 3/8	1 1/8	1·59	4 1/2
2	10	6	1 1/2	1 5/16	1·715	4 1/2
2 1/8	10 1/2	6 1/2	1 1/2	1 3/8	1·84	4 1/2
2 1/4	11	6 3/4	1 5/8	1 1/2	1·93	4
2 3/8	11 1/2	7 1/4	1 5/8	1 5/8	2·054	4
2 1/2	12	7 3/4	1 3/4	1 5/8	2·18	4
2 5/8	12 1/2	8 1/4	1 3/4	1 3/4	2·304	4

Table of B.A. Drill Sizes for Tapping.

Number.	Standard.		Working.		
	Clear.	Tap.	Clear.	Hard Brass Tapping.	Soft Iron Tapping.
0	15/64	12	15/64	10	9
1	4	19	3	15	15
2	13	26	12	24	23
3	20	30	19	29	28
4	28	34	27	32	31
5	30	40	30	37	36
6	35	44	34	43	42
7	40	48	39	46	45
8	44	51	43	49	48
9	48	53	48	52	51
10	51	55	50	54	53
11	53	57	53	55	55
12	55	62	55	57	57
13	56	65			
14	61	70			

(H. T. R. Grantham.)

TABLE OF DRILL AND TAPPING SIZES.

Whitworth.		Metric.		Gas.		British Association.				
Size	Drill.	Size.	Drill.	Size.	Drill.	Size.	Drill.	Clearance.	Diam. of Head.	
1/8	No. 38			1/8	11/32	0	9–11	15/64	27/64	Drill Gauge.
3/16	„ 25			1/4	29/64	1	15–17	3–4	23/64	
1/4	„ 9			3/8	19/32	2	24–25	12–13	21/64	
5/16	1/4			1/2	47/64	3	28–29	19–20	9/32	
3/8	19/64			3/4	61/64	4	31–33	27–28	1/4	
7/16	13/32	5	3·9 or 24	1	1 3/64	5	36–39	30	15/64	
1/2	27/64	5½	4·2 – 19	1¼	1 17/32	6	42–43	34–35	3/16 Full	9
9/16	29/64	6	4·5 – 16	1½	1 49/64	7	45–48	39–40	11/64	16
5/8	33/64	7	5·4 – 3	2	2 3/32	8	48–51	42–44	5/32	24
11/16	37/64	8	6·25 – D	2½	2 25/32	9	51–53	48	9/64	30
3/4	41/64	9	7 – J	3	3 9/32	10	53–55	50–51	1/8	32
13/16	11/16	10	8 – 5/16	3½	3 3/4	11	55 56	53		
7/8	47/64	12	9·5 – V	4	4 1/4	12	57–61	55		
15/16	51/64	14	11·5 – 29/64			13	63	56		
1	27/32	16	13 0 – 33/64			14	69	61		
1 1/8	1 3/16	18	14·5 – 37/64			15	71	·035 or 65		
1¼	1 1/16	20	16·5 – 11/64			16	74	·031 – 68		
1 3/8	1 5/32	22	18 – 45/64			17	76	·027 – 70		
1½	1 9/32	24	20 – 53/64			18	78	·024 – 72		
1 5/8	1 3/8	26	21 – 53/64			19	79	·021 – 74		
1 3/4	1 19/32	28	23 – 29/32			20		·019 – 75		
1 7/8	1 27/32	30	24·5 – 31/32			21		·017 – 77		
2	1 63/64	32	1 3/64			22		·015 – 78		
2¼	2 1/32	36	30			23		·013 – 80		
2½	2 7/32	40	1 5/16			24		·011		
						25		·010		

(*H. T. R. Grantham.*)

TAPPING.

It is a common occurrence for a tap to break when tapping hard metal. If the tap is ground away on the point, so that the first five or six threads are gradually bevelled, the risk of breakage may be largely avoided. The effect of grinding the end of the tap is to break up the chips into smaller pieces, and thus prevent clogging the threads of the tap. This will be found to be especially advantageous in tapping very hard rolled steel.

An easy rule for approximating tap-drill sizes is to deduct 1½ times the pitch from the outside diameter of the tap. In the case of a ⅜-in. tap, 16-pitch thread, for instance, 1½ times 1/16 = 3/32, and 3/8 − 3/32 = 9/32 = the diameter of the drill required for a ⅜-in. tap.

TAPPING LUBRICANTS.

Copper is tapped best with two-flute taps, using milk as a lubricant. Paraffin is well known to be the most suitable oil for use on aluminium, but is sometimes ruled out on account of its expense. Both the above pay when quantities of work merit the trouble occasioned by the adoption of a new lubricant. Paraffin must never be employed with ferrous metals—it tears badly.

CORRECT TAP GRINDING.

Grinding a tap consists usually of grinding the taper at the end of the tap, and the clearance back of the cutting edge thus formed. This taper may be long, as in nut taps; short, as in plug taps; or almost none, as in bottoming taps. The principle remains the same in each case. What is required is that each flute shall have exactly the same taper, just enough clearance so that it will cut freely, and not enough to needlessly weaken the cutting edge.

Prior to the advent of tap-grinding machinery nearly all taps were ground by hand, with a resultant lack of uniformity in taper and clearance. Fig. 29 shows a section of a four-flute tap with each flute ground differently in the matter of clearance or backing off. The flute A, at the left, is ground with a straight-line clearance just sufficient to have the heel of the flute actually clear and not drag. This calls for an angle of 15°, as indicated, and results in a weak cutting edge, but the strongest possible with a straight-line clearance at the heel.

The flute B, at the top, shows a 15° convex clearance. A glance shows this to be excessive. Flute C, at the right, shows a straight-line clearance of 5°, and nearly half of the flute at the back is not cleared at all, leaving the tap to ride on that portion and keep the front of the flute from cutting. Flute D, at the bottom, shows a perfect grind for ordinary work. It is a 5° convex

clearance, and shows that with this small but sufficient clearance angle at the cutting edge, the back is perfectly cleared.

In addition to the correct clearance of taps, a uniform angle of taper for all flutes is necessary. The results of different tapers obtained by hand-grinding are indicated in fig. 29. Even a slight variation from the uniform taper angle means power wasted and taps broken. As the blunt angle flute strikes the side of a drilled hole, it throws the cutting end of the tap off centre and imparts to it a wabbling motion. This means a tapped hole that is oversized, allowing the screw to fit loosely. Tap A is correctly ground and will tap a hole true to size, properly located and

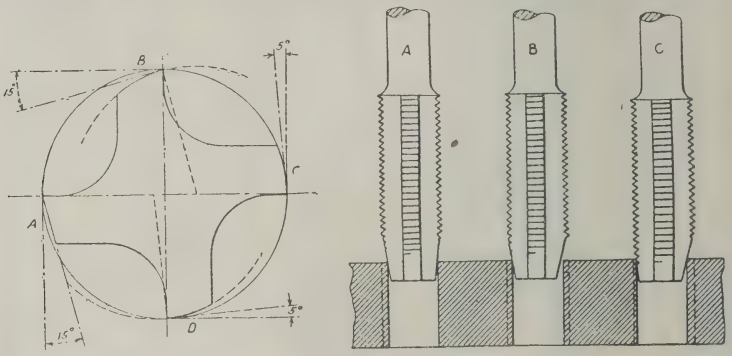


FIG. 29.

with smooth, accurately shaped threads. Tap B is ground with one flute at a more blunt angle than the other. This blunt flute is shown just as it strikes the drilled hole. Tap C shows a similar tap after it has been crowded over. The condition developed is readily seen.

(*' Railway Mechanical Engineer.'*)

THE "GUN" TAP.

A method of grinding a tap to give a cutting action similar to that of a turning tool is shown in fig. 30, which shows a ' Gun ' tap made by the Greenfield Tap & Die Corporation of America. The makers claim that the slope B of the face of the leading cutting edges (which are slightly hook-shaped, the edge being ground to give a negative rake) at A curls and projects the chips forward in an axial direction in front of the tap as tapping proceeds, the form of the chips, in steel, being spiral and more or less continuous. In this way the vicinity of the cutting operation at each flute is

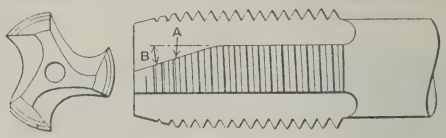


FIG. 30.

kept clear of swarf and the tap cuts freely, even through a long hole. The tap does not have to be backed out of long holes to clear a collection of chips from the flutes.

It is stated that the cutting pressure necessary with the ' Gun ' tap is 50 to 60 per cent. less than with an ordinary tap for the same size of thread. In deep holes the ' Gun ' tap has cut a full thread over four diameters long at a cutting pressure of 300 lbs., whereas an ordinary tap pulled up after tapping a full thread for a length of two diameters, the load being then 1,100 lbs., and stalling the machine.

Pipe Thread Tap Heads.

The Sheffield Small Tool Manufacturers have adopted the following list of sizes for the squares of pipe thread taps:—

No. of □.	Size of □ Inches. Fraction.	Size of □ Inches. Decimal.	Whitworth Thread Taps. Inches.	International Thread Taps. Millimetres.	Gas Tap. Inches.
0	$\frac{7}{64}$	·109	$\frac{1}{8}$	3 to $3\frac{1}{2}$	—
1	$\frac{9}{64}$	·140	$\frac{5}{32}$	4 to $4\frac{1}{2}$	—
2	$\frac{5}{32}$	·156	$\frac{3}{16}$	5	—
3	$\frac{13}{64}$	·203	$\frac{1}{4}$	6 to 7	—
4	$\frac{15}{64}$	·234	$\frac{5}{16}$ and $\frac{3}{8}$	8 to 10	$\frac{1}{8}$
5	$\frac{9}{32}$	·281	$\frac{7}{16}$	11 to 12	—
6	$\frac{5}{16}$	·312	$\frac{1}{2}$ and $\frac{9}{16}$	13 to 14	$\frac{1}{4}$
7	$\frac{25}{64}$	·390	$\frac{5}{8}$ and $\frac{11}{16}$	15 to 17	$\frac{3}{8}$
8	$\frac{29}{64}$	·453	$\frac{3}{4}$ and $\frac{13}{16}$	18 to 21	$\frac{1}{2}$
9	$\frac{35}{64}$	·546	$\frac{7}{8}$ and $\frac{15}{16}$	22 to 23	—
10	$\frac{41}{64}$	·640	1	24 to 26	$\frac{5}{8}$ and $\frac{3}{4}$
11	$\frac{23}{32}$	·718	$1\frac{1}{8}$	27 to 30	—
12	$\frac{13}{16}$	·812	$1\frac{1}{4}$	31 to 33	$\frac{7}{8}$ and 1
13	$\frac{27}{32}$	·843	$1\frac{3}{8}$	34 to 36	—
14	$\frac{29}{32}$	·906	$1\frac{1}{2}$	37 to 41	$1\frac{1}{4}$
15	$1\frac{5}{64}$	1·078	$1\frac{5}{8}$ and $1\frac{3}{4}$	42 to 46	$1\frac{1}{2}$ and $1\frac{3}{4}$
16	$1\frac{11}{64}$	1·203	$1\frac{7}{8}$ and 2	47 to 52	—
17	$1\frac{3}{8}$	1·375	$2\frac{1}{8}$ and $2\frac{1}{4}$	54 to 58	2 and $2\frac{1}{4}$
18	$1\frac{1}{2}$	1·500	$2\frac{3}{8}$ and $2\frac{1}{2}$	60 to 64	—
19	$1\frac{5}{8}$	1·625	$2\frac{5}{8}$ and $2\frac{3}{4}$	66 to 70	—
20	$1\frac{3}{4}$	1·750	$2\frac{7}{8}$ and 3	72 to 76	$2\frac{1}{2}$ to 3

Intermediate diameters have same squares as sizes below.

The heads of fluted hand reamers have same size squares as Whitworth taps and International thread taps, respectively.

The sizes of squares are stated in fractions and in decimals of inches, to facilitate measurement with either the mechanics' rule or the micrometer calliper.

No. of □ may be quoted when ordering tap wrenches.

Work-tables of Machine Tools.

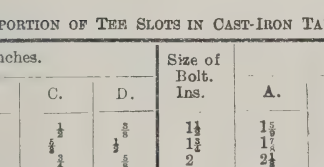


Fig. 31.

Proportion of Tee Slots in Cast-Iron Tables.

Size of Bolt. Ins.	Inches. A.	B.	C.	D.	Size of Bolt. Ins.	Inches. A.	B.	C.	D.
$\frac{1}{2}$	$\frac{9}{16}$	1	$\frac{1}{2}$	$\frac{3}{8}$	$1\frac{1}{4}$	$1\frac{5}{8}$	$2\frac{5}{8}$	$1\frac{1}{2}$	$1\frac{1}{16}$
$\frac{5}{8}$	$\frac{11}{16}$	$1\frac{3}{16}$	$\frac{5}{8}$	$\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{7}{8}$	3	$1\frac{3}{4}$	$1\frac{1}{2}$
$\frac{3}{4}$	$\frac{13}{16}$	$1\frac{7}{16}$	$\frac{3}{4}$	$\frac{5}{8}$	2	$2\frac{1}{8}$	$3\frac{1}{4}$	2	$1\frac{5}{8}$
$\frac{7}{8}$	$\frac{15}{16}$	$1\frac{5}{8}$	$\frac{7}{8}$	$\frac{13}{16}$	$2\frac{1}{4}$	$2\frac{3}{8}$	$3\frac{7}{8}$	$2\frac{1}{4}$	$1\frac{1}{2}$
1	$1\frac{1}{16}$	$1\frac{13}{16}$	1	$\frac{7}{8}$	$2\frac{1}{2}$	$2\frac{5}{8}$	$4\frac{1}{2}$	$2\frac{1}{4}$	$1\frac{7}{8}$
$1\frac{1}{8}$	$1\frac{3}{16}$	$1\frac{15}{16}$	$1\frac{1}{8}$	$\frac{15}{16}$	$2\frac{3}{4}$	$2\frac{7}{8}$	$4\frac{3}{4}$	$2\frac{3}{4}$	$1\frac{7}{8}$
$1\frac{1}{4}$	$1\frac{5}{16}$	$2\frac{1}{16}$	$1\frac{1}{4}$	$1\frac{1}{16}$	3	$3\frac{1}{8}$	5	3	$1\frac{7}{8}$

Keeping Bolts in Place in Open-slotted Tables.

When bolts have to be used for repeat work in drilling machine or other open-slot types of tables, the flopping about and dropping of these bolts becomes a time-wasting nuisance. Fig. 32 shows two methods of holding them firmly in position, so that work can be removed

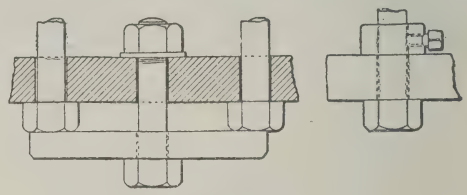


Fig. 32.

and a fresh piece substituted with the minimum of inconvenience. In the one case a clamp is suspended from a bolt and drawn up against the heads; in the other a loose collar is slipped down the bolt and tightened with a set-screw, an easier way than troubling to thread the bolt right down and putting a nut on. ('*Machinery.*')

Pulley Arms and Bosses.

Diam. of Pulley. Ins.	Section of Arms.		Depth of Rib round the Boss. Ins.	Thickness of Metal round Shaft for Shaft Diameters in Ins.				
	at Boss. Ins.	at Rim. Ins.		1½ to 2.	2½ to 3.	3½ to 4.	4½ to 5.	5½ to 6.
12	1¾ × ½	1¼ × ½	⅞	1	1⅛	1¼	1⅜	1½
15	1⅞ × ⅝	1⅜ × ⅝	⅞	1	1⅛	1¼	1⅜	1½
18	2 × ¾	1½ × ⅝	1	1	1⅛	1¼	1⅜	1½
21	2¼ × ⅞	1⅝ × ¾	1⅛	1⅛	1¼	1⅜	1½	1⅝
24	2½ × 1 5/16	1⅞ × ¾	1⅛	1⅛	1¼	1⅜	1½	1⅝
27	2⅝ × 1	2 × ⅞	1¼	1⅛	1¼	1⅜	1½	1⅝
30	2⅞ × 1	2¼ × ⅞	1¼	1⅛	1¼	1⅜	1½	1¾
33	2⅞ × 1 1/16	2¼ × ⅞	1⅜	1½	1⅜	1½	1⅝	1¾
36	3 × 1 1/16	2⅝ × ⅞	1½	1½	1⅜	1½	1⅝	1¾
39	3⅛ × 1⅛	2½ × 1	1½	1½	1⅜	1½	1⅝	1⅞
42	3⅛ × 1⅛	2⅝ × 1	1⅝	1⅝	1⅜	1½	1¾	1⅞
45	3⅜ × 1⅛	2⅝ × 1	1⅝	1⅝	1½	1⅝	1¾	2
48	3½ × 1¼	2⅞ × 1 1/16	1¾	1⅜	1⅛	1⅝	1¾	2
54	3⅝ × 1½	3 × 1 3/16	1¾	—	1⅝	1⅞	1⅞	2⅛
60	3¾ × 1½	3½ × 1⅛	2	—	1⅝	1⅞	1⅞	2⅛
66	4 × 1⅜	3½ × 1⅛	2	—	1⅞	1⅞	2	2¼
72	4½ × 1⅜	3½ × 1 3/16	2⅛	—	1⅞	2	2¼	2⅜
84	4½ × 1½	3⅝ × 1¼	2¼	—	2	2¼	2⅜	2½

Pulleys up to 17 ins. diam., 4 arms; from 18 ins. to 6 ft. diam., 6 arms; 6 ft. diam. upwards, 8 arms.

Bolt Spacing.

Fig. 33 shows minimum spacings and pitches for Whitworth studs, bolts, and nuts, which will give sufficient space to enable a spanner to be used. Being for use with hexagonal nuts, a movement of 60° full is allowed for the spanner.

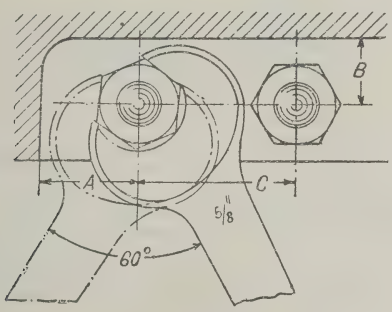


FIG. 33.

BOLT-SPACING TABLE.

Bolt Diameter.		Space A		Space B.		Pitch C.	
In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.
$\frac{1}{4}$	6·4	$\frac{5}{8}$	15·9	$\frac{7}{16}$	11·1	1	25·4
$\frac{5}{16}$	7·9	$\frac{11}{16}$	17·5	$\frac{1}{2}$	12·7	$1\frac{1}{8}$	28·6
$\frac{3}{8}$	9·5	$\frac{15}{16}$	23·8	$\frac{5}{8}$	15·9	$1\frac{7}{16}$	36·5
$\frac{7}{16}$	11·1	1	25·4	$\frac{13}{16}$	20·6	$1\frac{1}{2}$	38·1
$\frac{1}{2}$	12·7	$1\frac{1}{16}$	27·0	$\frac{7}{8}$	22·2	$1\frac{3}{4}$	44·4
$\frac{9}{16}$	14·3	$1\frac{1}{8}$	28·6	$\frac{15}{16}$	23·8	$1\frac{13}{16}$	46·0
$\frac{5}{8}$	15·9	$1\frac{5}{16}$	33·3	1	25·4	2	57·1
$\frac{11}{16}$	17·5	$1\frac{3}{8}$	34·9	$1\frac{1}{16}$	27·0	$2\frac{5}{16}$	58·7
$\frac{3}{4}$	19·1	$1\frac{7}{16}$	36·5	$1\frac{1}{8}$	28·6	$2\frac{1}{2}$	63·5
$\frac{13}{16}$	20·6	$1\frac{1}{2}$	38·1	$1\frac{1}{4}$	31·7	$2\frac{9}{16}$	65·0
$\frac{7}{8}$	22·2	$1\frac{5}{8}$	41·3	$1\frac{5}{16}$	33·3	$2\frac{3}{4}$	69·8
1	25·4	$1\frac{7}{8}$	47·6	$1\frac{1}{2}$	38·1	3	76·2
$1\frac{1}{8}$	28·6	$2\frac{1}{8}$	54·0	$1\frac{7}{8}$	42·9	$3\frac{1}{2}$	88·9
$1\frac{1}{4}$	31·7	$2\frac{3}{8}$	60·3	$1\frac{7}{8}$	47·6	4	101·6
$1\frac{3}{8}$	34·9	$2\frac{1}{8}$	63·5	$2\frac{1}{8}$	54·0	$4\frac{3}{8}$	111·1
$1\frac{1}{2}$	38·1	$2\frac{5}{8}$	66·7	$2\frac{1}{4}$	57·1	$4\frac{5}{8}$	117·5
$1\frac{5}{8}$	41·3	$2\frac{3}{4}$	69·8	$2\frac{3}{8}$	60·3	$4\frac{7}{8}$	120·7
$1\frac{3}{4}$	44·4	$2\frac{7}{8}$	73·0	$2\frac{1}{2}$	63·5	5	127·0
2	50·8	$3\frac{3}{8}$	85·7	$2\frac{5}{8}$	66·7	$5\frac{5}{8}$	142·9
$2\frac{1}{4}$	57·1	$3\frac{7}{8}$	98·4	3	76·2	$6\frac{1}{2}$	158·8
$2\frac{1}{2}$	63·5	$4\frac{1}{4}$	108·0	$3\frac{1}{8}$	79·4	$6\frac{7}{8}$	174·6
$2\frac{3}{4}$	69·8	$4\frac{3}{8}$	111·1	$3\frac{3}{8}$	85·7	$7\frac{3}{8}$	187·3
3	76·2	$4\frac{7}{8}$	123·8	$3\frac{3}{4}$	95·2	$7\frac{3}{4}$	196·9

(*W. O. Hart*, '*Mechanical World*,' *June* 22, 1923.)

Cast-Iron Muff Couplings.

Dimensions in Inches.					
Dia. of Shaft.	A. Dia. of Muff.	B. Thickness of Lug.	Length of Coupling	D. Dia. of Bolt.	No. of Bolts.
1¼	3½	13/16	5¼	⅝	2
1½	4¼	13/16	6½	⅝	2
2	5½	15/16	8½	¾	2
2½	6¼	15/16	9¾	¾	3
2¾	7¼	1	11½	¾	3
3	7½	1	13	⅞	3
3½	8½	1 1/16	15	⅞	3
4	9½	1⅛	16½	1	3
4½	10½	1 3/16	18	1	3
5	11½	1¼	20	1	3

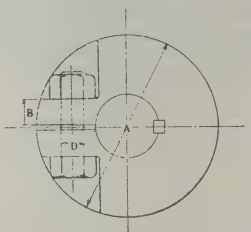


FIG. 34.

Washer Removably Attached to Screw.

There are cases where it is advisable to have a washer that cannot drop from its screw, and yet where it is impossible or impracticable to press or otherwise rigidly fasten it in place under the screw head, because it would be in the way of removal. The advantage is often considerable, for it is out of the question to lose the washer. Anyone who as ever dropped a washer into the working parts of some complicated machine, or lost it when it was needed in a hurry, will know how much the dropping of a washer may mean.

Fig. 35 shows a washer that cannot possibly drop off its screw, yet can be easily removed from the screw whenever desired. The arrangement can be applied to any screw and washer so long as they are still soft. The shank of the screw under the head, and the inner face of the washer, are each provided with a V-groove. Into the groove in the washer is sprung an oval-shaped piano-wire ring, so made that while it firmly seats in the washer, it will pinch the sides of the groove in the screw shank. The dimensions of the groove in the washer and of the wire must be such as will allow the washer with the ring in place to be slid over the shank without difficulty. The groove in the screw should be a slight distance away from the head, as shown, so as to make sure that the ring will always seat on the screw shank.

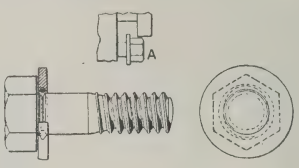


FIG. 35.

Since the depth of the groove in the screw is about the same as that of the thread, the strength of the screw is not diminished, so that no disadvantage of any kind is incurred to offset any part of the advantage gained.

At A is shown one out of many cases where an overhanging part would not permit the use of a pressed-on washer, but where the removable washer, while always in place, is never in the way. As soon as the screw has been loosened a few turns the washer is slid on the shank. From there on, as the screw is turned, the washer will either jump over successive threads or the threads will turn through it and the washer will be left on the threaded end when the screw is out.

(*'American Machinist.'*)

Rivet for Aircraft Construction and Other Purposes.

This rivet, the invention of Lieutenant J. W. Iseman, was designed for use in Duralumin construction work on airships in positions where it is difficult to hammer at one end and support the other, but it is also adaptable for other purposes, and is a very ingenious arrangement.

The rivet is shown in fig. 36 in position both before and after riveting. A hole ending in a conical point is drilled through the head of the rivet in the direction of the axis, as shown at A.

The base of the cone starts at the end of the nominal grip length of the rivet, this being equal to the thickness of the riveted plates. The end of the rivet is slotted up to the grip with two slots at right angles.

A pin of the same diameter as the hole and ending in a similar cone is tapped into the rivet until the end is flush with the head, as shown at B. This pin causes the slotted end to spread out as the straight side of the pin travels along the side of the cone. When the pin is home, the sides of the cone are parallel to the axis of the rivet as at B.

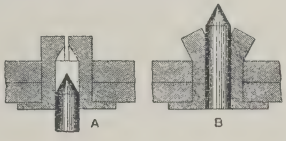


FIG. 36.

It is found that friction and binding between the rivet and the pin are quite sufficient to hold the latter in place.

Tests of these rivets have shown that with an outside diameter of 0·127 in. and a pin 0·053 in. in diameter, a shear strength of 450 lb., or 35,500 lb. per square inch, can be reached.

SECTION XXIII

THE ATMOSPHERE — WIND — AIR RESISTANCE — WINDMILLS—TRANSMISSION OF POWER BY COMPRESSED AIR—AIR COMPRESSORS—PNEUMATIC DESPATCH TUBES

(pp. 1181–1206)

(With Contributions by Edward Butler, M.I.Mech.E., and R. A. Sturgeon, Assoc.M.Inst.C.E.)

SECTION XXIII

THE ATMOSPHERE—WIND—AIR-RESISTANCE—WINDMILLS—AIR COMPRESSION—AIR COMPRESSORS—PNEUMATIC DESPATCH TUBES.

The Atmosphere.

Air consists, by volume, of oxygen 20·84 parts, of nitrogen, 79·16 parts; and by weight, of oxygen 23·15 parts, and of nitrogen 76·85 parts. Also 0·03 to 0·04 per cent., by volume, of carbonic acid.*

One cubic foot of ordinary dry air at normal pressure, temp. 62° F., weighs 532·5 grains (saturated 529 grains); or 1 lb. of air contains 13·146 cubic feet.

Dry air is 819 times lighter than water. Aqueous vapour, 1·4 per cent. by volume.

At 60° a cubic foot of saturated air contains 0·00083 lb. of moisture. This increases to 0·00285 lb. at 100° and 0·030 lb. at 200°. These are exactly the same as the weight of a cubic foot of saturated steam at the given temperatures.

The viscosity of air in C.G.S. units is 1·48 at 15° C. (59° F.) and 1·49 at 16·5° C. (61·7° F.).

The mean weight of a column of air 1 inch square and 45 miles high is 15 lbs.

Density of air $= \left(\frac{1}{4}\right)^{\frac{h}{6\cdot8}} = (0\cdot8156)^{h}$; Pressure of air, $p = 14\cdot7\,(0\cdot8156)^{h}$ lbs. per sq. in.

$$h = \frac{\log.\frac{p}{14\cdot7}}{-0\cdot0885228}$$

where h = height in miles above surface of earth.

At 6·8 miles above the surface of the earth the air is 4 times lighter than at the earth's surface; at 13·6 miles, 16 times; at 20·4 miles, 64 times. Total weight of atmosphere, 5,300 billion tons.

According to a writer in *Nature*, the result of experiments made with forty balloons (sent up in Great Britain) has been that the isothermal layer was reached more than thirty times. Its average height was 35,000 ft. (10·7 kiloms.); it is rather higher with a high barometer. The mean temperature found was — 53·6° F. (— 77° C.), considerably higher than the Continental mean. The extremes observed in England were — 22° F., near Ditcham Park, Hants, on July 24, and — 73° F., near Oxford, on September 12, 1907. On the same day, November 11, 1907, the temperature was — 42° F. at Ditcham at 36,000 ft. altitude, — 58° F. in Oxfordshire at 38,500 ft., and — 74° F. near Manchester at 37,000 ft. The existence of an isothermal layer in equilibrium in which the fall of temperature suddenly stops, and yet in which such great differences of temperature occur at spots not very far from one another, is difficult to understand, especially considering the low atmospheric pressure at these heights.

To consume 1 lb. of coal requires 11½ lbs., or 150 cubic feet, of air to combine with it.

The work stored up in 1 cu. ft. of air at atmospheric pressure, i.e. the work it would do if allowed to expand into an absolute vacuum = 5,188 ft. lbs.

The expansion of air, like all other elastic fluids, is uniform at all temperatures.

* Air contains 15 parts of 'neon' and 5 parts of 'helium,' in one million volumes (*Dewar*)

If h = height of barometrical column of mercury ; K = number of square inches in section of tube ; 13,568 ozs., or 848 lbs. = weight of cubic foot of mercury at mean temperature : p = number of pounds pressure of air on square inch of surface ; then

Pressure of atmosphere on K sq. ins. $= p \times K$. Weight of column of mercury $= \frac{Kh}{1728} \times 848$.

But column of mercury balances the pressure of the atmosphere ; therefore

$$p \times K = \frac{Kh}{1728} \times 848, \qquad \text{or,} \qquad p = \frac{848}{1728} \times h = \frac{53h}{108} = \frac{h}{2} \text{ very nearly.}$$

Hence it appears that *the pressure of the atmosphere in pounds per square inch is very nearly equal to half the column of mercury in the barometer-tube in inches.*

The column of mercury in the barometer in this country varies from 28 to 31 ins. ; therefore the mean column is 29·5 ins. ; and by the foregoing result the mean pressure of the atmosphere will be about 14·7 lbs. per square inch.

Rankine assumes a standard English atmosphere to be 14·704 lbs. per square inch in the latitude of London, or equal to a mercury column 29·922 ins. high at 32° F., which = 30 ins. at 57·8° F.

The metrical atmosphere, 760 mm. (29·922 ins.), at 0° C. (32° F.) = 1·0333 kilograms per square cm., or 14·6967 lbs. per square inch, at Paris = 14·686 lbs. at New York.

To find the barometric height, H, at any latitude, l,

$$H = 760 \times \frac{(1 + 0{\cdot}00531 \sin^2 48° 50')}{(1 + 0{\cdot}00531 \sin^2 l)} \text{ mm.,}$$

where 48° 50′ is the latitude of Paris.

The altitude or head of the atmosphere at uniform density will be that of a column of water 33·95 ft. high divided by the specific gravity of air (·0012046), or 28,183 ft. The velocity due at the foot of this head will be : $V = 8{\cdot}02 \sqrt{28183} = 1346{\cdot}4$ ft. per second, the velocity at which the air will pass into a vacuum. The motions and effects of gases by the force of gravity are precisely the same as those of liquids. Under like conditions a conduit will discharge 30·55 times more air than water.

Liquid Air.

Liquid air is produced by passing highly compressed air through a counter flow heat exchanger followed by an expansion device. On expansion the air is cooled and the cold air thus produced passes back through the heat exchanger, so cooling the incoming compressed air. The cooling is thus self-intensive, and finally results in liquefaction of part of the expanded air. The unliquefied part escapes through the heat exchanger and so keeps up the cooling effect. The expansion may take place through a nozzle, or in an expansion engine with the output of external work. The initial pressure used varies in different systems from about 45 atms. to 200 atms. About 8 to 12 per cent. of the air supply is liquefied, and the power required is from 1 to 2 h.p. hrs. per lb. of liquid produced, according to size and type of plant employed.

The critical temperature of air, above which it cannot be liquefied, is −140° C., and the corresponding critical pressure is 39 atms. The boiling point at atmospheric pressure is about −190° C., and the latent heat of evaporation about 50 calories. The specific gravity of the liquid is about 0·88.

If liquid air (21 per cent. O_2) is allowed to evaporate quietly it gives off a vapour containing about 7 per cent. O_2 owing to the fact that nitrogen boils at a temperature 12° C. lower than that of oxygen. The remaining liquid consequently becomes richer in oxygen as evaporation proceeds, and the boiling point rises until finally a temperature of −184° C. is reached (the boiling point of liquid oxygen).

The chief use of liquid air is in the process of obtaining oxygen and nitrogen from the atmosphere by means of liquefaction and distillation. Argon and neon are also obtained in the same way by additions to the ordinary oxygen or nitrogen apparatus.

Liquid air as such is used industrially for obtaining very low temperatures. One application is the freezing of mercury vapour during the evacuation of radio valves where the evacuation is performed by mercury vapour pumps. The mercury vapour is prevented from reaching the valves by the insertion of a liquid air trap in the pump circuit. A somewhat kindred use is the charcoal pump, in which specially prepared charcoal is cooled in liquid air. When so cooled the charcoal absorbs a large volume of air, and so can produce a higher vacuum than is obtainable by any other known method.

Liquid oxygen is used in breathing apparatus either for mine rescue work or for high altitude flying. It is also used in conjunction with special cartridges as an explosive for use in quarries

and mines. The chief advantages of this explosive lie in the facts (1) that it can generally be manufactured on the spot, as suitable material for the cartridges can be obtained almost anywhere, and (2) that neither the cartridge nor the liquid oxygen is in itself explosive. The actual soaking of the cartridge which renders it explosive is only performed immediately before insertion into the bore hole.

Velocity of Air.

When air passes into an air of less density, the velocity of its passage is measured by the difference of the densities of the two.

H and h = densities of the air in inches of mercury ; t = temperature (F.°) at the time of passage ; and V = velocity of air in feet per second.

$$V = 1347{\cdot}4\sqrt{\frac{H-h}{h}\left(1+0{\cdot}002088t\right)}.$$

Pressure against a Plane Surface moved through the Air.

P = pressure against plane. a = area of plane, in square feet. v = velocity, in feet per second. v_1 = velocity in miles per hour.

$$P = {\cdot}00142av^2 \text{ lbs.}^* \qquad P = {\cdot}0029av_1^2 \text{ lbs.}^*$$

$P = ({\cdot}003 + {\cdot}00004q)\,av_1^2$ lbs.,† where q = ratio of sides of rectangular plate.

The quantity ·003 decreases as v_1 increases, being ·0029 for 90 miles, and ·0032 for 0 to 25 miles.

Horse-power required to move a Plane Surface against the Air.

H.P. = horse-power. a = area of plane, in square feet. v = velocity, in feet per minute.

$$\text{H.P.} = {\cdot}000{,}000{,}000{,}010{,}6av^3.$$

TABLE SHOWING HORSE-POWER CONSUMED IN MOVING A PLANE SURFACE AGAINST THE AIR.

Speed in		H.P. per Sq.Ft. of Effective Surface.	Speed in		H.P. per Sq. Ft. of Effective Surface.
Miles per Hour.	Feet per Min.		Miles per Hour.	Feet per Min.	
10	880	0·00722	50	4,400	0·903
15	1,320	0·0244	75	6,600	3·05
20	1,760	0·0514	100	8,800	7·23
25	2,200	0·110	150	13,200	24·4
30	2,640	0·195	200	17,600	57·8

As a result of tests, M. Eiffel found that, for velocities of from 60 ft. to 130 ft. per second, the resistance offered by air to a moving surface is practically proportional to the square of the velocity, although the exponent increases slightly and gradually when velocities exceeding 100 ft. per second are employed. The resistance or pressure in pounds, P, is represented by the product KSV^2, where K is a constant depending on the size and shape of the surface tested ; S is the area of the surface in square feet, and V is the velocity in miles per hour. At ordinary temperatures and a barometric pressure of 760 mm., M. Eiffel found that the value of the coefficient K ranged from 0·00286 to 0·00327, the latter value being apparently a maximum attained only with large surfaces. Thus, for a circular surface of 1 sq. ft., the value of K was 0·00286, for a square surface of the same area 0·00295, and for an area of 10 sq. ft. 0·00323. He also found that the resistance of an inclined surface is practically the same as that of a normal one, when the perpendicular to the surface makes an angle less than 30° with the direction of motion. For greater angles the resistance proportionately decreases (zero at 90°). On pressure boards ranging from 25 sq. ft. to 100 sq. ft. in area, which were mounted at the top of a 50 ft. tower, Dr. T. E. Stanton found, from a large number of experiments, that the mean value of this constant was 0·0032.

Dr. Hutton deduced that the resistance of air varied as the square of the velocity nearly, and to an inclined surface as the 1·84 power of the sine × cosine.

The figure of a plane makes no appreciable difference in the resistance, but the convex surface of a hemisphere, with a surface double the base, has only half the resistance.

At high velocities the resistance becomes nearly a constant quantity.

The resistance of the air to a train of cars in a dead calm was found to be $\frac{1}{200}$ of their weight.

The velocity of a train of cars which gave a resistance of $\frac{1}{200}$ of the load with a fair wind was 34½ miles per hour, and only 27½ with an adverse wind.

* T. E. Stanton, National Physical Laboratory. † Hagen.

Flow of Air.

Weight of Air Passing Through a Nozzle.

The weight of air at atmospheric pressure passing through a properly-designed nozzle into any vacuum of not less than 8 lbs. may be calculated as follows (Leblanc) :—

W = lb. weight per hour ; A = area of nozzle neck in square inches ; T = atmospheric temperature in deg. Cent.

$$W = \frac{20330A}{\sqrt{T + 273}}$$

Air Friction.

Recent experiments on the flow of air through smooth sheet steel ducts varying in size from 12 ins. to 48 ins. in diameter, at velocities ranging from 1,000 ft. to 2,400 ft. per min., appear to show that the average value of the coefficient of friction is 0·0035.

Impulse of Wind.

V = velocity of wind. D = direct impulse in pounds on one square foot.

$D = V^2 \times \cdot00271$, where V is in miles per hour
$D = V^2 \times \cdot00312$, where V is in knots per hour } *

Velocity in Statute Miles per Hour.	Impulse in Pounds per Square Foot.	Velocity in Knots per Hour.	Impulse in Pounds per Square Foot.	Kind of Wind.	Velocity in Statute Miles per Hour.	Impulse in Pounds per Square Foot.	Velocity in Knots per Hour.	Impulse in Pounds per Square Foot.	Kind of Wind.
1	·00271	1	·00312	Light air.	22	1·31	22	1·51	Moderate gale.
2	·0108	2	·0125		24	1·56	24	1·80	
3	·0244	3	·0281		26	1·83	26	2·11	Fresh gale.
4	·0434	4	·0499	Light wind.	28	2·12	28	2·45	
5	·0678	5	·0780		30	2·44	30	2·81	
6	·0976	6	·112	Light breeze.	32	2·78	32	3·19	Strong gale.
7	·133	7	·153		34	3·13	34	3·61	
8	·173	8	·200	Moderate breeze.	36	3·51	36	4·04	
9	·220	9	·253		38	3·99	38	4·51	Heavy gale.
10	·271	10	·312	Fresh breeze.	40	4·34	40	4·99	
11	·328	11	·378		45	5·22	45	6·32	Storm.
12	·390	12	·449		50	6·78	50	7·80	
13	·458	13	·527		60	9·76	60	11·2	Hurricane.
14	·531	14	612		70	13·3	70	15·3	
15	·610	15	·702	Strong breeze.	80	17·3	80	20·0	
16	·694	16	·799		90	22·0	90	25·3	
17	·783	17	·902		100	27·1	100	31·2	
18	·878	18	1·01						
19	·978	19	1·13						
20	1·08	20	1·25						

Model experiments carried out in a wind tunnel at Washington show that a wind blowing at 76 miles per hour produces a pressure of 22 lb. per sq. ft. on a tall building.

Wind Velocity by Robinson Anemometer.

If

v = velocity of anemometer cups in miles per hour ; V = velocity of wind in miles per hour ; then,

$$V = 2v + 4.$$

Instrument fails to record velocities below 4 miles per hour.

(*W. H. Dines*, 'Report of Astronomer Royal, 1921.')

* Recent researches have shown that the constants ·00271 and ·00312 are apparently too small by about 6 per cent.

WINDMILLS.

(Contributed by Edward Butler, M.I.Mech.E.)

The driving shaft of a windmill situated on level ground should, with a tower of moderate height, be tilted *upwards* at an angle of 6° to 8°. If on a low hill with the fall against the prevailing wind the shaft is better horizontal, but when situated on a steep hill the best result will be obtained by tilting the shaft *downwards* a few degrees.

The shaft should be mounted so that the driving-wheel faces the wind in either direction and at a height ranging from 20-25 ft. on high ground, and 50-70 ft. on low ground, and so that it is clear of buildings or trees.

The tower should, in ordinary circumstances, be of sufficient strength to hold up against a 40-mile storm with a wind pressure of 8 lbs. per sq. ft. On exposed elevations subject to wind velocities exceeding this, a windmill unless specially constructed is not suitable.

WIND ACTION.

Velocity per hour in miles.	Pressure per sq. ft. in lbs.	Description of Wind.	Action on Windmill.
5	0·125	Faint breeze . .	Will not drive.
8	0·33	Breeze . . .	Will start against light load.
10	0·5	Fresh breeze . .	Will run slowly.
15	1·125	Good wind . .	Will run steadily.
20	2·	Strong wind . .	Gives best service.
25	3·125	Very strong wind .	Maximum useful effect.
30	4·5	Gale . . .	Should be partly furled or held out of the wind.
40	8·	Storm . . .	Must be totally put out of action.
60	18·	Violent storm .	Not safe.
80	30·	Hurricane . .	„ „
100	50·	Tornado . . .	„ „

Power of Windwheels.

The total momentum of wind is,

$$\frac{0{\cdot}08}{32{\cdot}2} \times \text{area} \times \text{velocity}^2 = 0{\cdot}0025\ AV^2, \text{ approximately.}$$

The turning moment at starting, for each vane, equals

$$0{\cdot}002\ AV^2 \cos\theta r,$$

and when in motion is

$$0{\cdot}0015\ AV^2 \cos\theta r \text{ for moderate speed,}$$

the constant ·0015 decreasing to 0·0012 with increase of speed;

where A = area of vane; V = velocity of wind; θ = angle to the axis of rotation; r = mean radius; 0·08 is the mean weight of one cubic foot of air. Velocity in ft. per sec. is to miles per hour as 22 is to 15.

HORSE-POWER OF WINDWHEELS.

Shaft or b.h.p. of a 4-armed wheel with cloth sails

$$= \frac{v^3 r^2}{512477}\ (\textit{Rankine}), \quad \frac{4\ Av^3}{1145833}\ (\textit{Weisbach}).$$

The brake h.p. of a multi-vaned wheel

$$= \frac{0{\cdot}6 \times 0{\cdot}78\ (D^2 - d^2) \times V^3}{33000};$$

where,

A = area of one vane; D = total diameter of wheel; d = clearance diameter at centre; r = radius of sail tips; V = velocity in feet per second.

A windwheel (Dutch pattern) with four cloth sails, each 24 ft. in length and 6 ft. in breadth, will develop 4-5 b.h.p. in a 20-mile wind.

Windwheels with multiple vanes have a considerable advantage in starting with low velocity winds and in slow running.

The power of a windwheel is not so great as is popularly supposed. A 16 ft. wheel, for instance, with multiple cambered steel vanes, only develops about 1 b.h.p. in a 15-mile wind, and this is reduced to less than one-half in a 10-mile wind, although increased to nearly double in a 20-mile wind.

Construction.

According to windmill practice (old type) the wheel may have 4, 5, or 6 cloth sails of rectangular form, the length being equal to five times the breadth; but the total surface should not exceed one-third of the whole surface described by the radial arms carrying the sails. The angles of the cross bars for supporting the sails, which commence at one-seventh from the wheel centre, are usually 24°, 21°, 18°, 9°, 3°, from the plane of motion.

While more power can be gained by increasing the sail surface to more than one-half the wheel area, the difficulty is such with the old practice of using cloth sails that the sail area seldom exceeds one-third and is more often reduced to one-fourth.

The angles of the sails to the plane of motion, according to Smeaton, should be 20°, 19°, 18°, 16°, 12½°, 7°; and again, to Forrester, as 24°, 21°, 18°, 14°, 9°, 3°.

In multi-vane windwheels, the vanes—either oil-soaked wood slats, or cambered steel plates of light construction and galvanised—are carried by a series of 6-8 steel or wooden arms, fixed to a steel boss. The arms are braced to an extension of the wheel boss by tension rods. The vanes are fastened to two sets of cross-bars, connecting the arms together, in sections of 8 to 15, according to wheel diameter and angle to the plane of motion; this varies from 20° to 40°, but is generally 25° in flat or cambered vanes of uniform angle. The steeper the angle the narrower the vanes, to reduce whirl resistance. For direct connected pumps the angle is steeper than for geared down pumps. Steel vanes are cambered more for rigidity than for any increased driving efficiency. There is some advantage in forming the angle to suit the radius, but is not of such consequence as with wide cloth sails. What is of more importance is for the shaft to run in well lubricated roller bearings and for the turntable to move freely so that the wheel shall be held square to a light wind.

Power Control.

In windwheels with wide cloth sails, these are furled to suit either the load or the wind. In wheels with multiple vanes the usual practice is to fit a governor-vane on one side, at a right-angle to the wheel shaft, and for the tail-vane to be swivelled to the turntable, and to be held end-on either by a spring or a weight, or, better still, by a series of weights chained together. These weights should be proportioned for the governor-vane so as to pull the turntable, and the wheel with it, partly out of the wind as soon as the velocity increases 20-25 m.p.h., and to bring round the wheel edge-on to the wind in a storm.

Wind Pumps.

The popularity of wind-driven pumps is due to the introduction of the multiple-vaned wheel and to automatic governing, in combination with light easily-erected steel towers, the average power developed ranging from 1 to 6 shaft h.p., with a wind velocity of 15 m.p.h. (22 ft. per sec.), a 20-mile wind nearly doubling this power, and a 10-mile wind generating about half the power obtained by a 15-mile wind.

In ordinary practice the pumping capacity of a wind engine, as expressed in pounds of water raised per minute by pressure-head in feet, rarely exceeds from 25 to 35 per cent. of the wheel h.p., but is influenced by wheel diameter, the angle of the vanes, and the method of transmitting the power to the plunger.

In geared-down transmissions, the wheel-shaft pinion should be situated at the side of the pump wheel, and not above or below it. This disposition reduces friction and maintains correct alignment and mesh of the cogs.

Owing to the very inadequate fly-wheel effect of a windwheel, it works at considerable disadvantage in driving single-acting pumps direct, and only less so when geared down; double-acting, or better, 3-throw single-acting pumps, oppose a more equable torque, but are impracticable for small installations. For this reason a rack or mangle transmission is peculiarly applicable to a wind-driven pump, as shown in the higher duty and efficiency in the table, p. 1187.

The location of a wind pump is generally influenced more by the available water supply than from any consideration of available wind-power, the tower usually being constructed high enough—50 to 70 ft.—to clear all obstacles around for a distance of 300 to 400 ft. Under normal conditions eight hours' pumping only can generally be relied upon as a daily average; so that to make an installation successful a storage-tank or reservoir should be provided, the minimum capacity of which may with advantage equal from four to seven days' pumping, although a larger reserve is often allowed for.

TABLE SHOWING THE WORKING OF SIX SELECTED WIND ENGINES DURING TRIALS OF 150 HOURS, BY THE ROYAL AGRICULTURAL SOCIETY.

Name of Engine.	Diam. of Wheel in Feet.	Total No. of Gallons Pumped.	Total Pressure Head in Feet.	Total Revolutions of Wheel.	Total No. of Double Strokes of Pump.	Transmission.	Ratio of Wheel to Pump.	Particulars of Pumps. Diam. in Ins.	Particulars of Pumps. Stroke in Ins.	Single or Double Acting.	Average Water H.P. per Minute.
Canadian-Imperial	16	79,000	200	308,000	42,000	rack gear	5	4	22	double	·53
Henry Sykes, Ltd.	16	49,000	200	210,000	210,000	direct crank	1	2·5	8	,,	·33
J. W. Titt . .	16	46,000	200	285,000	285,000	,, ,,	1	3·25	6·12	,,	·30
Thomas & Son .	16	41,000	200	307,000	122,000	geared ,,	2·5	4	8	single	·275
R. Warner & Co. .	16	40,000	200	330,000	160,000	,, ,,	2	3·5	5	double	·268
J. W. Titt . .	16	36,000	200	230,000	88,000	,, ,,	2·5	4·5	8	,,	·242

TABLE SHOWING PERFORMANCES OF WIND-PUMPING ENGINES, SHOWING QUANTITY IN IMPERIAL GALLONS PUMPED AGAINST A TOTAL HEAD OF 200 FEET, AS REGISTERED BY METER, PER DAY OF 10 HOURS.

Average Wind Velocity in miles per hour for each day.	Results of Two Engines with 8 ft. diameter wheels.		Results of Six Engines with 12 ft. diameter wheels.		Results of Seven Engines with 16 ft. diameter wheels.		Results of Three Engines with 20 ft. diameter wheels.		Result of Engine with 30 ft. diameter wheel.	
	Highest.	Lowest.	Highest.	Lowest.	Highest.	Lowest.	Highest.	Lowest.	Highest.	Lowest.
5	—	—	650	320	1,200	350	—	—	940	—
8	—	—	1,500	900	3,000	600	1,600	800	4,000	—
10	—	—	2,000	950	4,000	1,600	3,500	1,200	7,000	—
12	—	—	2,400	1,000	4,200	2,000	4,100	3,800	8,000	—
15	400	—	4,000	1,250	5,600	3,750	6,000	4,000	13,000	—
18	700	—	3,500	1,700	5,200	4,500	—	—	19,000	—
20	900	700	4,200	1,400	6,930	4,200	—	—	21,000	—
22	800	—	4,500	1,500	7,600	3,300	—	—	23,000	—
24	—	—	4,950	1,600	10,400	3,600	—	—	24,000	—
	CORRECTED AVERAGE CAPACITY IN GALLONS PER HOUR RAISED 50 FEET.									
15	250		1,000		1,500		2,200		5,000	

AIR COMPRESSION.

RELATION BETWEEN PRESSURE AND VOLUME.

If

p = actual pressure of air in a container;
P = gauge ,, ,, ,,
v = volume of air at pressure p;

p_1 = actual pressure of air when compressed;
P_1 = gauge ,, ,, ,,
v_1 = volume of air when compressed;

then,

$$\frac{p}{p_1} = \frac{P + 14{\cdot}7}{P_1 + 14{\cdot}7} = \frac{v_1}{v},$$

the compressed air being *at the same temperature* as the air before compression—that is to say, if the heat due to the compression is allowed to become dissipated.

If the air is compressed from atmospheric pressure (i.e. if gauge pressure P = 0), then

$$\frac{14\cdot7}{P_1 + 14\cdot7} = \frac{v_1}{v},$$

or,

$$P_1 = 14\cdot7\frac{v - v_1}{v_1}, \quad \text{and} \quad v_1 = v\frac{14\cdot7}{P_1 + 14\cdot7}.$$

Example 1.—What gauge pressure (P_1) would be required to reduce 5 cub. ft. (v) of air at atmospheric pressure down to 3 cub. ft. (v_1), at the same temperature?

$$P_1 = 14\cdot7\,\frac{5-3}{3} = 9\cdot80 \text{ lbs.}$$

Example 2.—What volume (v_1) will 1 cub. ft. of air occupy when compressed to 30 lbs. gauge pressure (P_1), at the same temperature?

$$v_1 = 1\,\frac{14\cdot7}{30 + 14\cdot7} = 329 \text{ cub. ft.}$$

If the heat is not allowed to disappear when the air is compressed, then

$$\frac{14\cdot7}{P_1 + 14\cdot7} = \left(\frac{v_1}{v}\right)^{1\cdot408}$$

or,

$$P_1 = 14\cdot7\frac{v^{1\cdot408} - v_1{}^{1\cdot408}}{v_1{}^{1\cdot408}}, \quad \text{and} \quad v_1 = v\left(\frac{14\cdot7}{P_1 + 14\cdot7}\right)^{\cdot7102}$$

If $P_1 = 30$, as in the case of *Example* 2, then the value of v_1 instead of being ·329, will be

$$(\cdot329)^{\cdot7102} = \cdot454 \text{ cub. ft.}$$

Work and Horse-power in Compressed Air.

(1)

Work.

The effective work, W, stored up in 1 cu. ft. of compressed air (or the work required to compress the air) at a gauge pressure (pressure above atmosphere) of P_1 lbs. per sq. in. is

$$W = 352\cdot94\,(P_1 + 14\cdot7)\left(1 - \left(\frac{14\cdot7}{P_1 + 14\cdot7}\right)^{0\cdot29}\right) \text{ ft. lbs.}$$

Example.—1 cu. ft. of compressed air at a gauge pressure of 10 lbs. (the resulting heat not being allowed to disappear when the compression takes place) will contain:

$$352\cdot94 \times 24\cdot7\left(1 - \left(\frac{14\cdot7}{24\cdot7}\right)^{0\cdot29}\right) = 352\cdot94 \times 24\cdot7(1 - 0\cdot86028) = 1{,}218 \text{ ft. lbs. of}$$

effective work, *i.e.* the work it would do in expanding down to atmospheric pressure.

Horse-Power.

The H.P. required to produce 1 cu. ft. per min. of compressed air at P_1 lbs. pressure, is

$$W \div 33{,}000, \quad \text{or,} \quad H.P. = 0\cdot0107\,(P_1 + 14\cdot7)\left(1 - \left(\frac{14\cdot7}{P_1 + 14\cdot7}\right)^{0\cdot29}\right).$$

Example.—To produce 1,000 cu. ft. per min. of compressed air at 10 lbs. pressure would require

$$H.P. = 1{,}218 \div 33{,}000 \times 1{,}000 = 36\cdot9.$$

(2)

Work.

The effective work, w, stored up in 1 cu. ft. of air at atmospheric pressure when compressed to a gauge pressure of P_2 lbs. per sq. in., is

$$w = 352{\cdot}94 \times 14{\cdot}7 \left(\left(\frac{P_1 + 14{\cdot}7}{14{\cdot}7} \right)^{0{\cdot}29} - 1 \right) = 5188{\cdot}2 \left(\left(\frac{P_1 + 14{\cdot}7}{14{\cdot}7} \right)^{0{\cdot}29} - 1 \right) \text{ ft. lbs.}$$

Example.—1 cu. ft. of atmospheric air, when compressed to a gauge pressure of 10 lbs. (the resulting heat not being allowed to disappear when the compression takes place), will contain

$$5188{\cdot}2 \left(\left(\frac{24{\cdot}7}{14{\cdot}7} \right)^{0{\cdot}29} - 1 \right) = 5188{\cdot}2 \quad 1{\cdot}1624 - 1) = 843 \text{ ft. lbs. of effective work.}$$

Horse-Power.

The H.P. required to compress 1 cu. ft. per min. of air at atmospheric pressure to P_1 lbs. gauge pressure, is

$$W \div 33{,}000, \quad \text{or,} \quad \text{H.P.} = 0{\cdot}157 \left(\left(\frac{P_1 + 14{\cdot}7}{14{\cdot}7} \right)^{0{\cdot}29} - 1 \right).$$

Example.—To compress 1,000 cu. ft. per min. of atmospheric air to 10 lbs. gauge pressure would require

$$\text{H.P.} = 843 \div 33{,}000 \times 1{,}000 = 25{\cdot}6.$$

(3)

$$\frac{W}{w} = \left(\frac{P_1 + 14{\cdot}7}{14{\cdot}7} \right)^{0{\cdot}7102}.$$

Example.—If $P_1 = 10$ lbs.,

$$\frac{W}{w} = \left(\frac{24{\cdot}7}{14{\cdot}7} \right)^{0{\cdot}7102} = 1{\cdot}446.$$

INCREASE OF TEMPERATURE OF AIR BY COMPRESSION.

If air at atmospheric pressure and at a temperature $t_1°$ F. be compressed suddenly to a gauge pressure P_1, then the temperature will be raised to $t_2°$ F., where

$$t_2° = (459 + t_1°) \left(\frac{P_1 + 14{\cdot}7}{14{\cdot}7} \right)^{0{\cdot}29} - 459.$$

Example.—If air at 32° F. temperature ($t_1°$) be compressed to 10 lbs. gauge pressure (P_1), then temperature $t_2°$ of air will be raised to

$$t_2° = 491 \quad \frac{24{\cdot}7}{14{\cdot}7} \Big)^{0{\cdot}29} - 459 = 491 \times 1{\cdot}162 - 459 = 112° \text{ F}$$

PRESSURE OF AIR PRODUCED BY INCREASE OF TEMPERATURE.

If air at atmospheric pressure in a closed vessel and at $t_1°$ F. be raised to $t_2°$ F., then the gauge pressure, P_1, of the air will be

$$P_1 = 14{\cdot}7 \left(\frac{459 + t_2}{459 + t_1} \right)^{3{\cdot}45} - 14{\cdot}7.$$

Example.—If $t_1° = 32$, $t_2° = 112$, then $P_1 = 14{\cdot}7 \left(\frac{571}{491} \right)^{3{\cdot}45} - 14{\cdot}7 = 24{\cdot}7 - 14{\cdot}7 = 10$ lbs.

INCREASE OF VOLUME OF AIR BY INCREASE OF TEMPERATURE.

If a volume, V_1, of air at atmospheric pressure and at $t_1°$ F. be raised to $t_2°$ F., the air being allowed to expand freely, then the volume V_2 at $t_2°$ F. will be

$$V_2 = V_1 \frac{461 + t_2°}{461 + t_1°}.$$

Example.—If volume V_1 of air at 32° ($t_1°$) be 1,000, then volume V_2 at 525° ($t_2°$) will be:

$$V_2 = 1{,}000 \frac{461 + 525}{461 + 32} = 1{,}000 \times \frac{986}{493} = 2{,}000.$$

The following table gives the increase in volume from 32° F :—

INCREASE OF VOLUME OF AIR BY INCREASE OF TEMPERATURE FROM 32° F.

F.°	V.	F.°	V.	F.°	V.	F.°	V.	F.°	V.
32	1,000	49	1,034	66	1,069	83	1,103	100	1,138
33	1,002	50	1,037	67	1,071	84	1,105	110	1,158
34	1,004	51	1,039	68	1,073	85	1,107	120	1,178
35	1,006	52	1,041	69	1,075	86	1,110	130	1,199
36	1,008	53	1,043	70	1,077	87	1,112	140	1,219
37	1,010	54	1,045	71	1,079	88	1,114	150	1,239
38	1,012	55	1,047	72	1,081	89	1,116	160	1,260
39	1,014	56	1,049	73	1,083	90	1,118	170	1,280
40	1,016	57	1,051	74	1,085	91	1,120	180	1,300
41	1,018	58	1,052	75	1,087	92	1,122	190	1,320
42	1,020	59	1,055	76	1,089	93	1,124	200	1,341
43	1,022	60	1,057	77	1,091	94	1,126	210	1,361
44	1,024	61	1,059	78	1,093	95	1,128	212	1,365
45	1,026	62	1,061	79	1,095	96	1,130	300	1,544
46	1,028	63	1,063	80	1,097	97	1,132	400	1,746
47	1,030	64	1,065	81	1,099	98	1,134	500	1,949
48	1,032	65	1,067	82	1,101	99	1,136	525	2,000

AIR COMPRESSORS.

The mechanical efficiency of air compressors is about 60 per cent. at 5 pounds per square inch, increasing to 80 per cent. at 40 pounds per square inch and above. It is quite usual for the best makes to attain an efficiency exceeding 90 per cent., but for general purposes it is wiser to assume the lower figure. In the case of machines of small output and low pressure, the efficiency is frequently below 50 per cent.

In providing a driving unit for an air compressor, the type of machine should be considered before fixing the size of the driver. Thus, the turning effort required by a single-acting, single-cylinder air compressor may vary considerably above and below the mean, and allowance must be made for this, particularly in the case of direct-coupled drives.

DESIGN OF AIR COMPRESSORS.

The best practice adopted in the design of air compressors is as follows :—The maximum terminal pressures are, for one stage, 80 lbs. to 100 lbs. per sq. in. ; two stages, 600 lbs. per sq. in. ; three stages, 1,500 lbs. per sq. in. ; four stages, 2,500 lbs. to 3,000 lbs. per sq. in. ; five stages, 4,500 lbs. per sq. in. ; and above 4,500 lbs. per sq. in., six or seven stages to 6,000 lbs. per sq. in., which is about the highest pressure for which an air compressor has been constructed. In the case of very small machines these pressures, for a given number of stages, are often exceeded in order to reduce complication. For example, machines have been constructed and work quite satisfactorily when compressing to 2,500 lbs. per sq. in. in two stages. The temperatures attained, however, make it necessary to carry out the cylinder lubrication by soapy water.

(*J. M. Ford, in a paper read before the Greenock Association of Shipbuilders and Engineers.*)

'BROTHERHOOD' AIR COMPRESSORS FOR MODERATE PRESSURES.

(*Peter Brotherhood, Ltd., Peterborough.*)

These compressors are made in a wide range of sizes, for small capacities of the single-acting trunk-piston type and for larger capacities of the double-acting type. The former are lubricated automatically, and the latter by forced lubrication.

BROTHERHOOD AIR COMPRESSOR

FIG. 1.—Details of Valves.

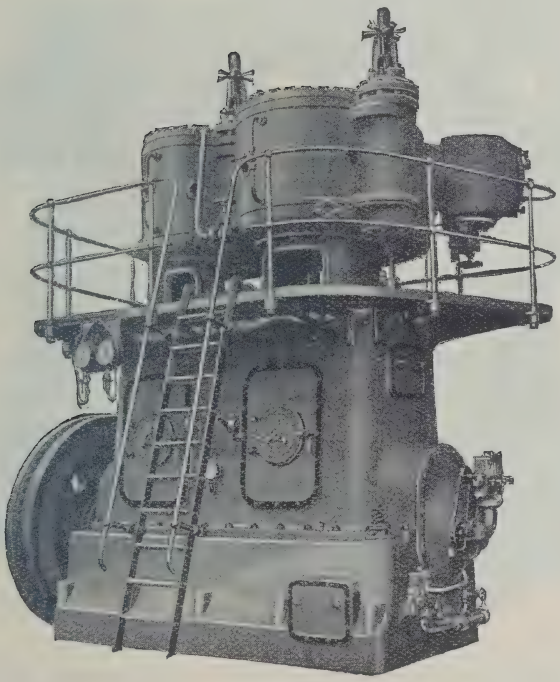


FIG. 2.

BROTHERHOOD AIR COMPRESSOR.

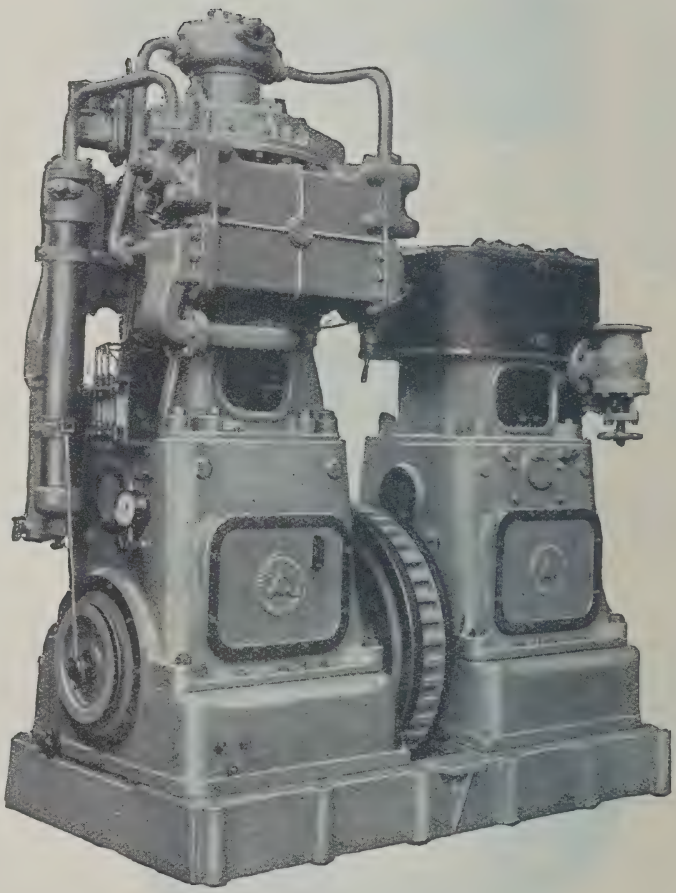


FIG. 3.

The single-acting types are designed for one or two stage, working up to a free air capacity of about 200 cu. ft. per minute, and are suitable for driving by belt, motor, or engine.

The double-acting types include single-line simple and tandem two-stage machines up to about 800 cu. ft. of free air per minute, and two-line simple and two-stage up to about 6,000 and 5,000 cu. ft., respectively.

The valves, which are illustrated on fig. 1, p. 1191, combine a large opening with very small moving mass, and will operate continuously at very high speeds.

The general construction of the double-acting types is shown on fig. 2, which illustrates a two-stage machine of 4,000 cu. ft. free air capacity.

The governing adopted varies according to the form of drive employed; for belt or motor drive the compressor is unloaded and reloaded as the air in the receiver rises or falls above and below the normal. For steam drive the speed is regulated and the air discharge controlled by a combined centrifugal and pressure governor.

'Brotherhood' High-Pressure Compressors for Air or Gas.

(*Peter Brotherhood, Ltd., Peterborough.*)

Brotherhood multi-stage compressors embody an exceptionally wide range of types resulting from development and research extending over sixty years. Standard types deal with pressures ranging from 600 to 6,000 lbs. per sq. in., in three to six stages of compression.

With the exception of compressors for oxygen, all piston and gland packings are of the metallic ring type, and except for special purposes, all classes have the working parts enclosed and lubricated under pressure.

Fig. 3 shows a three-stage air compressor for auxiliary air service for fuel injection and manœuvring in a motor ship.

The capacity of the compressor is 300 cu. ft. of free air per minute compressed to 1,000 lbs. per sq. in. at 350 r.p.m. The compressor is driven by a Brotherhood Force Lubricated Steam Engine developing 140 b.h.p.

Westinghouse Air Compressors.

(*Westinghouse Brake and Saxby Signal Co., Ltd.*, 82 *York Rd., King's Cross, London, N.* 1.)

The valve motion of the Westinghouse type of steam-driven air compressor is simple and compact, being entirely in the tophead, and all its parts are so arranged as to readily permit of examination or replacement. Should repairs to the valve mechanism be necessary, the tophead can be quickly removed and replaced by another, so that no unnecessary delay need occur through the compressor being put of service, pending repair. The air valves are contained in separate cases, so placed that they can readily be taken out.

The compressors are arranged vertically, and consist of a steam cylinder connected by a centre-piece to an air cylinder, or air cylinders, for either simple, compound, or three-stage compression. The three-stage compressor, figs. 4 and 5, is so constructed that air for compression is inhaled only on the upstroke of the piston, whilst on the downward stroke air at atmospheric pressure is drawn into and circulates around the inner walls of the cylinder and piston, to be again expelled on the succeeding upstroke; thus a supply of cool air is drawn into the interior of the compressor at every alternate stroke, and the compressor is air-cooled internally as well as externally; the result being that air compressed up to 300 lbs. per square inch is delivered at a temperature not exceeding 266° F., rendering the employment of water-cooled jackets unnecessary with this type of compressor.

For service on electric railways, the Westinghouse Brake & Saxby Signal Co., Ltd., manufacture air compressors driven direct by special slow-speed electric motors and also geared compressors driven by high-speed motors. Both of these types are equally suitable for industrial service.

Belliss High-speed Forced-lubricated Compressor.

(*Belliss & Morcom Ltd., Birmingham.*)

This type of compressor, fig. 6, p. 1195, embodies the best features of the Belliss engine, including the system of forced lubrication originated by the firm in 1890 (see Section XXX., Part III.). The working parts are generally the same as those for the 'O' type engine described in the section referred to. The air cylinders are water-jacketed and two-stage, as desirable for pressures in excess of 60 lbs. per sq. in., except in the case of quite small compressors.

WESTINGHOUSE THREE-STAGE AIR COMPRESSOR.

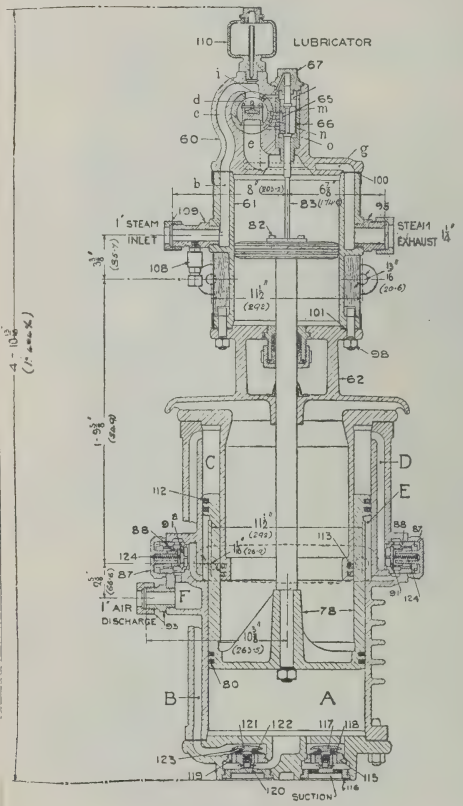


FIG. 4.

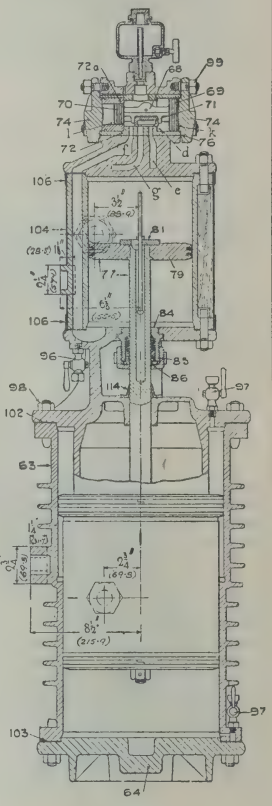


FIG. 5.

The compressor shown is adapted for being directly connected to an electric motor or for rope or belt driving. When operated at constant speed the air supply is controlled by an automatic valve on the inlet as shown, the valve being of the equilibrium double-beat type and con-

BELLISS AIR COMPRESSOR.

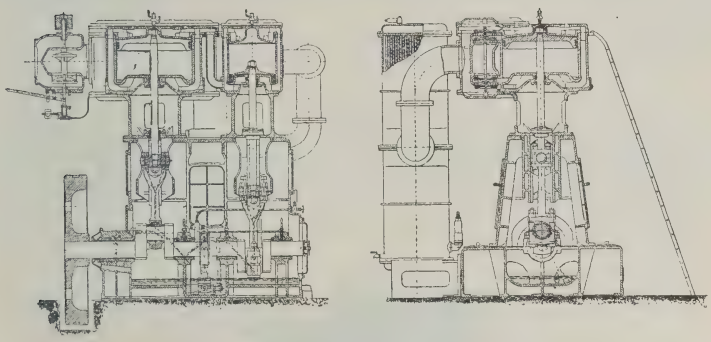


FIG. 6.—The Belliss Two-Stage Air Compressor.

trolled by an air relay, so designed that on the air pressure exceeding the intended pressure by 3 to 5 per cent. the air supply is cut off entirely, the pistons running *in vacuo* until the demand again causes the pressure to fall. As the air supply is not throttled the system is a highly efficient one. Special tests have demonstrated that the power required to run the compressor light

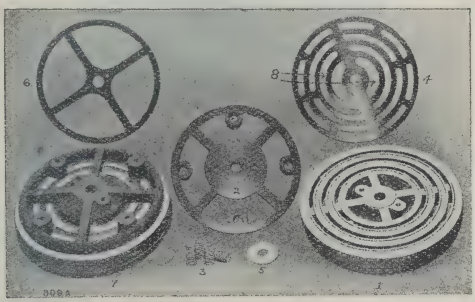


FIG. 7.—Details of Valves used in Belliss Compressors.

may be only 6 per cent. of the full power, and under 9 per cent. for the compressor and motor combined.

One of the chief points in the design of these compressors is the air valves. These are of the Rogier-Hoerbiger patent thin plate type, fig. 7, devised as the result of experience gained during the construction of the Simplon Tunnel, and which have come to be recognised as one of the best types of valves for air compressors. The valves consist of light discs of tempered

steel, perforated to give a multiple opening and falling on a cast-iron grid with suitable cushion and guard plates above. These valves are very reliable, noiseless in action, and a minimum amount of power is lost in operating them.

The Belliss compressors are also made with self-contained steam cylinders, controlled by the makers' patent relay type air governor, designed to vary the speed of the compressor in accordance with the demand for air, there being also a centrifugal governor directly connected to the crankshaft to prevent the speed under any conditions exceeding a predetermined maximum.

The Holman Single-Stage Compressor.

The Holman 'S' type compressor is a single-acting, duplex vertical machine. It is capable of working continuously at pressures up to 100 lbs. or intermittently to 110 lbs. per sq. in. It is made in four sizes, the largest having a displacement of 250 cub. ft. per min., and may be arranged for belt drive or for direct coupling to an electric motor or other prime mover.

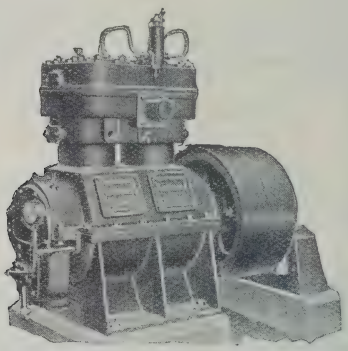


Fig. 8.—Holman 'S.' Type Air Compressor.

The cylinders and top portion of the casing are cast in one piece and the cylinder cover is removable. The suction and delivery flanges being on the cylinder, the cover may be removed without disturbing either suction or delivery air pipes.

The bearings are of die cast white metal and are adjustable, with the exception of the connecting rod small end which is solid bronze bushing. All bearings are supplied with oil under pressure from a valveless, gear-type pump driven from the crank shaft.

The valves are of the Rogler Hoerbiger type; all parts, with the exception of the seats, are interchangeable and the plates are of stainless steel. Each valve is situated in its own chamber, and water passages between suction and delivery chambers prevent the pre-heating of the incoming air.

Unloading is effected by the holding open of the inlet valve. The mechanism for this is controlled by a pilot valve which, connected to the air receiver, may be set to operate when the pressure in the latter reaches a predetermined maximum. The difference in pressure between unloading and resuming load may also be adjusted.

The three smaller sizes of this type are also incorporated in portable sets which may be either petrol engine or electric motor driven.

THE HOLMAN ROTARY PISTON AIR COMPRESSOR.

(Hele-Shaw Beacham Type.)

This type of compressor which was developed from the Hele-Shaw Rotary Oil Pump, has recently been simplified and is now becoming popular, especially where a light-weight, high-speed unit is required.

It is a single-stage machine with three cylinders, formed in one casting, which rotate on ball and roller bearings about a fixed water-cooled valve in which the air inlet and discharge passages are formed.

When the cylinder body is rotated a reciprocating motion is imparted to the pistons, each making an inward and outward stroke per revolution. The floating rings are mounted on ball bearings and are free to rotate. For this reason the gudgeon-pin bearings do not rotate, their motion being simply oscillatory, similar to the gudgeon-pin bearings of a single-acting engine.

During the outward stroke air is drawn into a cylinder through an opening at its inner end, a suitably positioned port in the central valve communicating with the inlet passage. At the commencement of the inward stroke the inlet port is covered, and the air is compressed until the delivery port is uncovered, when the air passes along the delivery passage through a non-return valve and thence to the air receiver.

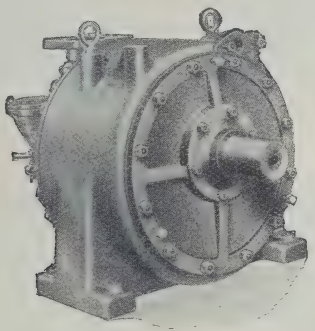


FIG. 9.—Holman 'H.S.B.' Type Compressor Unit.

Thus the same cycle of operations takes place as in a reciprocating type compressor with mechanically operated valves. One of the advantages of this design is that the machine may be run, with high overall efficiency, at higher speeds than is possible with an ordinary reciprocating machine, thus permitting direct coupling to an electric motor or other prime mover. Other advantages include: positive rotary valve action without operating mechanism, oil sealing for pistons and valve, reduced piston side thrust, dynamic running balance, uniform torque, compact form and light weight.

Two sizes of this type of compressor are made, the smaller for capacities up to 80, and the larger to 160 cub. ft. per min. displacement. Both are supplied as either stationary or portable sets; their light weight, small size and vibrationless running make them extremely suitable for the latter.

Besides the usual purposes for which portable compressors are employed, such as road construction, etc., these sets, because of their extreme portability, are very suitable for use in the more inaccessible districts which would be most difficult, if not impossible, to reach with compressors of the ordinary type.

The volumetric efficiency of the Holman Rotary Compressor is exceptionally high, and the isothermal efficiency is practically constant over the whole range of delivery pressures.

In the smaller sizes the compressor is readily portable, and is very compact.

THE HOLLAND 'SLM' ROTARY AIR COMPRESSOR.

(Bernard Holland & Co., 17 Victoria St., London, S.W. 1.)

(1) This Rotary Air Compressor, figs. 10, 11, consists essentially of a fixed casing or stator in which is a rotor fitted with blades free to move radially. As the rotor is less in diameter than the bore of the stator or casing and is placed eccentrically within it, there is a crescent-shaped space between

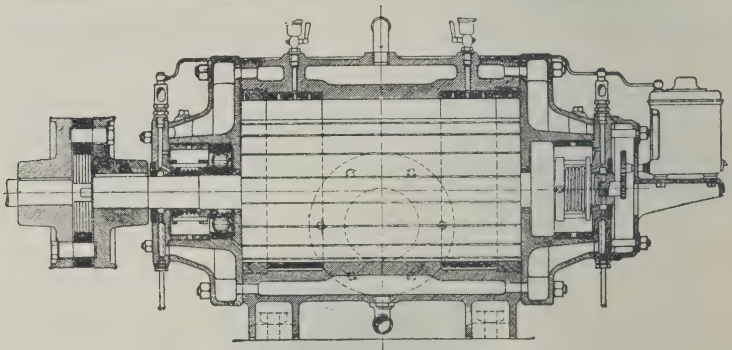


FIG. 10.—The Holland 'SLM' Rotary Air Compressor.

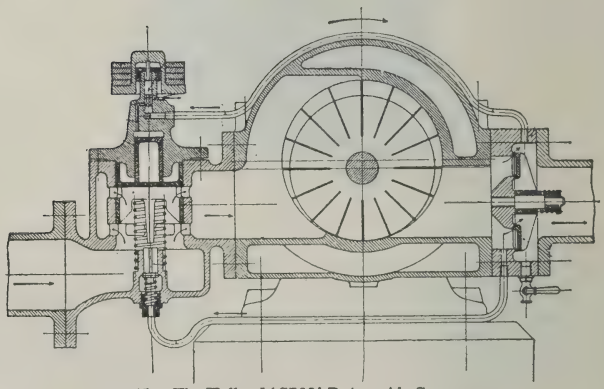


FIG. 11.—The Holland 'SLM' Rotary Air Compressor.

them. When the rotor is revolved, the blades, maintained by centrifugal force in contact with the casing, divide the crescent-shaped space into cells, which, each in turn, expand from a minimum to a maximum cubic capacity (suction period) and then decrease to a minimum cubic capacity (compression and delivery period).

Floating rings, one near each end of the rotor and revolving in unison with it, take the greater part of the thrust of the blades, so restricting and controlling their pressure on the wall of the casing; the consequent low friction and the absence of reciprocating parts and valves allow of this compressor being run at high speeds of revolution without vibration or perceptible wear after long use.

Metallic glands of special design dispense with the use of stuffing-boxes and packing. The lubrication is by pump, and is automatic.

Air-cooled single-stage compressors are made for pressures up to 10 lbs. per sq. in., and water-cooled single-stage machines for pressures up to 60 lbs. per sq. in. Water-cooled two-stage machines are supplied for pressures up to 150 lbs. per sq. in. and, dependent on size, up to 200 lbs. per sq. in. They are made in a number of standardsizes ranging from 6 cu. ft. to 4,000 cu. ft. delivery of free air per minute. For certain duties much larger units can be built.

The pumps are particularly suitable for continuous duty, coupled direct to electric motors and other fast-running prime movers; they are also arranged for belt and chain drive. They occupy little space, run without vibration, and need but inexpensive foundations.

(2) These machines can be so arranged as to perform the dual service of compressing and exhausting, one and the same machine being operated, at will, either as a compressor or as a vacuum pump. (See p. 1201.)

Air-Compressor Packing.

'BRITIMP' HOLLOW RING PACKING.

(*British Metallic Packings Co.* (1929), *Ltd.*, 7 *Princes Street, Westminster, S.W.* 1.)

This packing is largely used for high-pressure air and gas compressors, also for water-pumps, hydraulic accumulators, stop valves, emergency valves, expansion joints, gun recoil chambers,

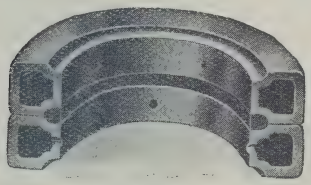


FIG. 12.

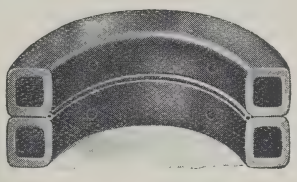


FIG. 13.

etc. The packing consists of hollow circular rings filled with graphite. The inner walls of the rings contain small holes through which the graphite is squeezed, thus providing the necessary lubrication. The rings are made in various forms to suit individual conditions. Two examples are shown in figs. 12 and 13.

See also pp. 1751, 1792, and 1903.

Air-Compressor Lubrication.

An oil suitable for air-compressors should always be highly filtered, because such oil will have a minimum tendency to form carbon deposits. Oils compounded with vegetable or animal oils are quite unsuitable for air cylinders.

Use only a straight-run, highly filtered, pure mineral oil of the lowest natural flash test consistent with the viscosity necessary to lubricate the cylinders and provide a proper piston seal.

The correct viscosities (worked out by the members of the Compressed Air Society, U.S.A., all of whom are builders of air-compressors) are given in the following tables, showing the proper specifications for both paraffin and asphaltic oils of various viscosities.

PHYSICAL TESTS OF PARAFFIN-BASE OILS. (*W. F. Osborne.*)

	Minimum.	Average.	Maximum.
Gravity, Baumé	28–32 deg.	25–30 deg.	25–27 deg.
Flash point, open cup	375–400 deg. F.	400–425 deg. F.	425–500 deg. F.
Fire	425–450 deg. F.	450–475 deg. F.	475–575 deg. F.
Viscosity (Saybolt) at 100 deg. F.	120–180 sec.	230–315 sec.	Up to 1,500 secs.
Colour	Yellowish	Reddish	Dark red to green
Congealing point (pour test)	20–25 deg. F.	30 deg. F.	35–45 deg. F.

PHYSICAL TESTS OF ASPHALTIC BASE OILS.

	Minimum.	Average.	Maximum.
Gravity, Baumé	20–22 deg.	19·8–21 deg.	19·5–20·5 deg.
Flash point, open cup	305–325 deg. F.	315–335 deg. F.	330–375 deg. F.
Fire	360–380 deg. F.	370–400 deg. F.	385–400 deg. F.
Viscosity (Saybolt) at 100 deg. F.	175–225 sec.	275–325 sec.	475–750 sec.
Colour	Pale yellow	Pale yellow	Pale yellow
Congealing point (pour test)	0 deg. F.	0 deg. F.	0 deg. F.

Cooling Water for Air Compressors.

Cooling water for cylinder jackets, cylinder heads, and intercoolers should always be turned on before the compressor is started. In case of stage compression, the greatest efficiency is obtained with the outlet water from the intercooler at a low temperature. The water supply to air cylinders and intercooler should be independent of each other, and the flow controlled by a valve on the inlet. The water should be shut off when the compressor is stopped, and in freezing weather all jacket drains should be opened. Outlet water should flow into open funnels so that frequent temperature readings can be made, a rise in temperature indicating deficient cooling, carbonised discharge valves, or leaking piston rings. The use of the thermo-siphon cooling system is not recommended.

NOTE ON AIR-COMPRESSOR OPERATION.

A bad practice in air-compressor operation is to put the air-discharge pipes through the intake duct. This serves directly to defeat the gain of the cool duct, because the hot discharge pipe heats the incoming air. On one such installation the incoming air was heated 6° F. by the presence of the discharge pipe in the intake duct. In this case there was a direct throwing away of about 1½ per cent. of power, which amounted to throwing away 11½ lbs. of coal every time 1,000 lbs. was shovelled into the furnace, day in and day out. In estimating such a heating loss a rough figure is 1 per cent. loss for each 5° F. intake-temperature rise.

VACUUM PUMPS.

THE BAILEY PATENT SLIDE VALVE 'DRY' VACUUM PUMP.

(Sir W. H. Bailey & Co., Ltd., Manchester.)

In the 'K' Vacuum Pump the slide-valve is without a stuffing-box, and the spring delivery valve is not carried by the slide-valve, but works vertically on a stationary seat, easily accessible even when the machine is running. By this arrangement the slide-valve does not run in the interior of the valve chest, but on the outside of it, and is, consequently, easy to fit and adjust; it has no stuffing-box to its spindle.

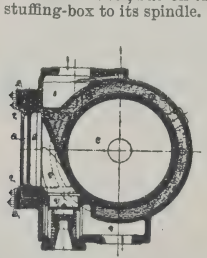


FIG. 14.

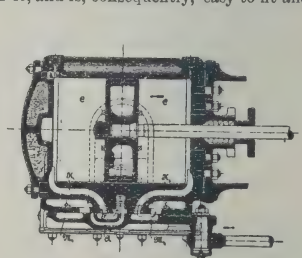


FIG. 15.

Fig. 14 shows a transverse section, and fig. 15 a longitudinal section of the standard air cylinder of the 'K' pump. The cylinder and back covers are water-jacketed, the front cover is cast in one with the cylinder, and a joint between the cylinder and the frame is avoided. In the larger sizes the front cover is water-jacketed.

The action of the valve is as follows :—

The air or gases enter by the suction pipe S and pass to the spaces S1 and S2. In fig. 14 the piston is passing towards the left-hand side and draws air into the cylinder from S2 through the space M2 and the port K2. At the same time it discharges the air drawn in during the preceding stroke by K1, P, and D, the slide-valve being set in relation to the crank so that the spaces S1 or S2 and D are put into communication at the right moment with the corresponding ends of the cylinder. Besides the ports M1 and M2, through which the gases pass into the cylinder, and the port P1 through which they are discharged, the slide-valve has a compensating or equalising port A which communicates between the discharge side of the piston and the suction side, thus equalising the pressure. When the piston has reached the dead point this port opens, for an instant of time, the communication between the two sides of the piston, the cylinder being cut off from the inlet and outlet by the slide-valve. The tension or pressure in the clearance space is thus reduced almost to the suction tension or pressure, and the piston can now draw air in much sooner, the effective suction stroke or volumetric efficiency being thereby increased. In addition to the slide-valve there is a spring valve V. The slide-valve governs the beginning and the end of both suction and discharge strokes. The spring valve has no effect on the volumetric efficiency of the pump, as without it the same volume would be drawn in, but it is necessary in order to reduce the power required to drive to the lowest possible limit. The slide-valve opens the suction and discharge at the beginning of each stroke and closes at the dead point so that the pump can be used for any degree of vacuum. The opening of the discharge port at a time when the air or gases are not yet compressed to atmospheric pressure would permit the air in the discharge pipe to flow back into the cylinder, causing back pressure and loss of power. The spring valve prevents this, as it separates the external air from the cylinder and prevents it from entering the cylinder when the slide-valve opens the discharge. When the air or gases in the cylinder are compressed to atmospheric pressure the spring valve opens automatically and the discharge commences. The influence of the spring valve therefore only affects the power for driving as previously stated ; it works vertically, is always accessible even when the machine is running, and is noiseless.

The single type pumps will maintain a vacuum of over 29·8 ins. of mercury when exhausting from a closed vessel. When a higher vacuum is necessary, the Bailey Double Vacuum Pump is recommended. This pump will maintain a vacuum of over 29·95 ins. of mercury at normal temperature and with a barometric pressure of 30 ins.

These pumps are particularly suitable as air pumps for condensers for steam engines, central condensation plants, cooling towers, condensers for evaporating plants, vacuum drying plants, distillers, filtering plants, etc.

The Holland 'SLM' Rotary Vacuum Pump.

(*Bernard Holland & Co.*, 17 *Victoria St.*, *London*, *S.W.* 1.)

This machine is rotary and is similar in general design to the Holland 'SLM' Compressor; it runs without vibration at the same high speeds of revolution and possesses all the characteristics of that machine, including the ability to perform, at will, the dual service of exhausting and compressing when arranged with the appropriate inlet and outlet piping. The single-stage air-cooled pumps will maintain a vacuum of 60 per cent. or 18 ins. of mercury with a barometric pressure of 30 ins., and under the same conditions the single-stage water-cooled pumps will produce a vacuum of 96 per cent. or 28·8 ins. of mercury, and the two-stage water-cooled machines, when exhausting from a closed circuit, will maintain a vacuum of over 99·9 per cent. or 29·97 ins. of mercury.

These machines, like the compressors, will run continuously with little attention, and maintain their efficiency after long use.

See also p. 1198.

Snifting Valves.

These valves open for the purpose of allowing the passage of air, but close if liquids attempt to pass ; they are used in hydraulic systems, and also in motor-cars. In a simple form the arrangement is shown by fig. 16.

B is a light float with a leather washer at the top ; O is an outlet pipe ; X is the pressure system. Air is free to pass in and out, but if liquid is passing out, B is carried up and seals the outlet pipe.

In connection with air vessels for pumps, a snifting valve is utilised to recharge such with air ; two cocks may be used, one at the top and one at the bottom of the vessel, so

that water may be drawn off below and air re-admitted above. If it be desired to admit it automatically while at work, a 'snifting valve,' a separate air-pump or an air-injector may be adopted. The valve, fig. 17, is placed between the suction and delivery valves, and

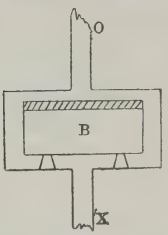


FIG. 16.

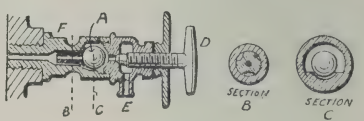


FIG. 17.

consists of a gun-metal chamber, F, containing a ball, A, which rolls backward and forward on account of the pump stroke. When rolled leftward, air is drawn inward at E, while the escape of water on return stroke is prevented by a rightward roll, and the whole may be put out of action or adjusted by a screwed plug, D.

PNEUMATIC DESPATCH TUBES.

Tubes.

The tubes used in the Pneumatic Despatch Service (telegrams only) of the British Post Office are 3, 2½, and 1½ inches in diameter, the 2½-inch being the size in most general use. The tubes are manufactured in lengths of about 28 feet. The tubes are of lead, and are enclosed in cast-iron pipes for mechanical protection. Each length of lead pipe is laid in a wooden trough as soon as manufactured, so that it may be handled without fear of bending. A tightly fitting polished steel 'mandril,' 1 foot long, and of the exact gauge of the tube, attached to a strong chain, and lubricated with soft soap, is then drawn through the entire length of the tube. This operation ensures the tube being smooth, cylindrical, and uniform throughout. The jointing of the lead tubes is effected as follows: Previous to the placing of the tubes in the trench, a strong chain is passed through it, and a polished steel mandril, 6 inches long and slightly less in diameter than the diameter of the tube, is heated and attached to the chain, and pushed half its length into the end of the tube already laid; the new length of tube is then forced over the projecting end of the mandril until the tube ends (which have been previously cut flat) butt perfectly together; a plumber's wipe joint is then made. By this means the tube is made perfectly air-tight, and the mandril keeps the surface of the tube under the joint as smooth as at any other part of its length. After the joint has been completed, the mandril is drawn out by the chain attached to it, the next length is drawn on and the process repeated.

Carriers.

The carriers in which the messages are placed consist of a cylindrical tube of gutta-percha covered with felt: a 'skirt' projects beyond the open end of the carrier. The pressure behind causes the skirt to expand until it exactly fits the tube and thus prevents the air leaking past. The front of the carrier is provided with a buffer or piston, formed of several discs of felt, which exactly fits the tube. To prevent the messages getting out of the carrier, the end is closed by an elastic band, which can be stretched sufficiently to allow the messages to be put in.

The length, over all, of a carrier for a 2½-inch tube is 6⅔ inches; the inside measurements of the gutta-percha tube or message holder being—length 4⅔ inches, diameter 1½ inches. The weight of such a carrier when empty is a little over 3 ounces.

The 3- and 2½-inch carriers will, on the average, take 65 and 25 messages respectively.

The speed of the carrier may be roughly stated to range from 20 to 30 miles an hour for tubes not exceeding a mile in length.

Working Pressure and Vacuum.

The standard pressure is 10 lbs. per sq. in., and the standard vacuum 6½ lbs. per sq. in.

For a given effective pressure, carriers travel most rapidly in pressure tubes when the air supply is *intermittent*, *i.e.* when the message tube is only connected with the compressed air mains during such times as carriers are actually traversing the tubes. The reason of this is, that

the compressed air with which the portion of the tube in front of the carrier is full, and which has to be displaced by the advancing carrier, weighs more than the uncompressed air, and consequently requires a greater amount of power to move it. The difference in the two cases is, however, not great, but the higher the pressure the greater is the difference in speed due to this cause. With 6 lbs. pressure, the carrier takes only 3 per cent. longer time with continuous than with intermittent working, whilst with 14 lbs. pressure it takes nearly 6 per cent. longer time.

In the case of carriers propelled by the exhaustion of the air, if the exhaustion is not commenced until the carrier is inserted at the further end, then the carrier commences to move as soon as the exhaustion at that end reaches a few ounces only, and the full exhaustion effect is not felt until the carrier has been moving for some time. When, however, the exhaustion is continuous the carrier on its insertion practically starts off under the full pressure: hence its speed is greater than in the first case. The continuous system of vacuum working, where one tube is used for the 'up' traffic (*i.e.* the traffic from the distant to the central or 'power' station) only, is easily effected, whether the traffic is heavy or light; but the intermittent system for the 'down,' or pressure working, which, as pointed out, gives the greatest speed, cannot always be followed out, as during the busy hours of the day the carriers have to be despatched in such quick succession that the current of air in the tube is practically continuous.

Power Required to Work Tubes.

The actual power required for working *continuously* the tubes at a pressure of 10 lbs. per square inch (the standard pressure) is, theoretically, 3·35 horse-power for a 2¼-inch tube 1 mile long; but in practice 50 per cent. requires to be added to allow for losses from various causes, *i.e.* the actual horse-power should be 5. With 10 lbs. pressure the time occupied in the transit of a carrier through a tube 1 mile long is 2½ minutes.

In order to neutralise the intermittent action of the pumps, and ensure a uniform flow of air, reservoirs or 'containers' are provided. These are arranged to have sufficient capacity to give out a reserve supply when the load on the tubes becomes abnormal, as occasionally happens.

For a given transit time the actual horse-power required is much less in the case of vacuum than in the case of pressure working. Thus, for example, the transit time for 10 lbs. pressure is the same as for 6½ lbs. vacuum, but the horse-power required in the two cases would be as 1·83 to 1. Inasmuch, however, as the tubes cannot practically be worked only by vacuum, the calculations for transit time and horse-power must be based on pressure working. The actual power necessary must be largely governed by local conditions, and no definite rule can be laid down. The power required to work any tube or number of tubes is usually computed, however, on the estimated traffic in carriers per hour which each tube is required to carry. This figure multiplied into the volume of the tube and divided by 60 gives the volume of air swept out per minute (if there is a number of tubes the figures are of course aggregated), and this affords the capacity of the pump at the required pressure.

Pumping Machinery.

The air compressors used for this system are driven by electric motors, oil, steam, or gas engines, and are provided according to the present standard practice with one or more cylinders arranged to draw from the vacuum container and compress the air thus drawn into the pressure container. Automatically controlled auxiliary valves are interposed in the inlet and delivery mains of the pump. They are controlled pneumatically by means of a by-pass connection with the pressure container in such manner that if (*a*) the pressure in the pressure container is low and the vacuum in the vacuum container high, so that the pump is deprived of air to compress, the auxiliary inlet valve on the vacuum main opens to atmosphere and admits free air for compression until the pressure in the pressure container is restored, when the valve shuts. On the other hand, (*b*) if the vacuum is low and the pressure is high, so that the pump would have to discharge against considerable back pressure the air it exhausts from the vacuum container, the auxiliary valve in the pressure main opens to atmosphere until the vacuum is restored. A non-return valve on the container side of each auxiliary valve prevents the loss of pressure or vacuum whilst either auxiliary is open to atmosphere. This system, besides economising in plant and space, absorbs and utilises certain losses which are inevitable with independent pumping for both services.

Transit Time.

Table I. enables the transit time for any diameter and length of tube at any pressure to be calculated, and from Table II. can be ascertained the pressure required to drive a carrier through any tube in any time. The chart, fig. 18 (p. 1206), based on Table I., shows the Transit Time variation for different lengths of tubes of three sizes, 1¼ in., 2¼ ins., and 3 ins., each worked at 10 lbs. pressure.

I.—Tables for calculating Time of Transit of a Carrier through a Tube.

Formula: $t = \cdot00872\sqrt{l^3} \times \frac{1}{\sqrt{P}} \times \frac{1}{\sqrt{d}} = A \times B \times C$ secs.

$A = \cdot00872\sqrt{l^3}$.

l	A	l	A	l	A	l	A	l	A
Yds.		Yds.		Yds.		Yds.		Yds.	
50	3·08	1,050	297	2,050	809	3,050	1,470	4,050	2,250
100	8·72	1,100	318	2,100	839	3,100	1,510	4,100	2,290
150	16·0	1,150	340	2,150	869	3,150	1,540	4,150	2,330
200	24·7	1,200	362	2,200	900	3,200	1,580	4,200	2,370
250	34·5	1,250	385	2,250	931	3,250	1,620	4,250	2,420
300	45·3	1,300	409	2,300	962	3,300	1,650	4,300	2,460
350	57·1	1,350	433	2,350	993	3,350	1,690	4,350	2,500
400	69·8	1,400	457	2,400	1,030	3,400	1,730	4,400	2,550
450	83·2	1,450	482	2,450	1,060	3,450	1,770	4,450	2,590
500	97·5	1,500	507	2,500	1,090	3,500	1,810	4,500	2,630
550	112	1,550	532	2,550	1,120	3,550	1,840	4,550	2,680
600	128	1,600	558	2,600	1,160	3,600	1,880	4,600	2,720
650	144	1,650	584	2,650	1,190	3,650	1,920	4,650	2,770
700	161	1,700	611	2,700	1,220	3,700	1,960	4,700	2,810
750	179	1,750	638	2,750	1,260	3,750	2,000	4,750	2,850
800	197	1,800	666	2,800	1,290	3,800	2,040	4,800	2,900
850	216	1,850	694	2,850	1,330	3,850	2,080	4,850	2,950
900	235	1,900	722	2,900	1,360	3,900	2,120	4,900	2,990
950	255	1,950	751	2,950	1,400	3,950	2,170	4,950	3,040
1,000	276	2,000	780	3,000	1,430	4,000	2,210	5,000	3,080

$B = \frac{1}{\sqrt{P}}$.

P	B	P	B	P	B	P	B	P	B
Lbs.		Lbs.		Lbs.		Lbs.		Lbs.	
·25	2·000	4·25	·485	8·25	·348	12·25	·286	16·25	·248
·50	1·410	4·50	·471	8·50	·343	12·50	·283	16·50	·246
·75	1·150	4·75	·459	8·75	·338	12·75	·280	16·75	·244
1·00	1·000	5·00	·447	9·00	·333	13·00	·277	17·00	·243
1·25	·894	5·25	·436	9·25	·329	13·25	·275	17·25	·241
1·50	·816	5·50	·426	9·50	·324	13·50	·272	17·50	·239
1·75	·756	5·75	·417	9·75	·320	13·75	·270	17·75	·237
2·00	·707	6·00	·408	10·00	·316	14·00	·267	18·00	·236
2·25	·667	6·25	·400	10·25	·312	14·25	·265	18·25	·234
2·50	·633	6·50	·392	10·50	·309	14·50	·263	18·50	·232
2·75	·603	6·75	·385	10·75	·305	14·75	·260	18·75	·231
3·00	·577	7·00	·378	11·00	·301	15·00	·258	19·00	·229
3·25	·555	7·25	·371	11·25	·298	15·25	·256	19·25	·228
3·50	·534	7·50	·365	11·50	·295	15·50	·254	19·50	·226
3·75	·517	7·75	·359	11·75	·292	15·75	·252	19·75	·225
4·00	·500	8·00	·354	12·00	289	16·00	·250	20·00	·224

$C = \frac{1}{\sqrt{d}}$.

d	C	d	C	d	C	d	C	d	C
Ins.		Ins.		Ins.		Ins.		Ins.	
1·00	1·000	3·25	·555	5·50	·426	7·75	·359	10·00	·316
1·25	·894	3·50	·534	5·75	·417	8·00	·354	10·25	·312
1·50	·816	3·75	·517	6·00	·408	8·25	·348	10·50	·309
1·75	·756	4·00	·500	6·25	·400	8·50	·343	10·75	·305
2·00	·707	4·25	·485	6·50	·392	8·75	·338	11·00	·301
2·25	·667	4·50	·471	6·75	·385	9·00	·333	11·25	·298
2·50	·633	4·75	·459	7·00	·378	9·25	·329	11·50	·295
2·75	·603	5·00	·447	7·25	·371	9·50	·324	11·75	·292
3·00	·577	5·25	·436	7·50	·365	9·75	·320	12·00	·289

Example: Required, the transit time, t, of a carrier through a tube 4 ins. in diameter (C) and 2,000 yds. long (A), worked at 7 lbs. pressure (B). $t = 780 \times 8 \times \cdot500 = 147$ secs.

II.—Tables for calculating Pressure required to drive a Carrier through a Tube in a Given Time.

Formula : $P = \cdot 00007604\, l^3 \times \frac{1}{t^2} \times \frac{1}{d} = A_1 \times B_1 \times C_1$.

$A_1 = \cdot 00007604\, l^3$.

l (Yds.)	A_1	l (Yds.)	A_1	l (Yds.)	A_1	l (Yds.)	A_1	l (Yds.)	A_1
50	9·51	1,050	88,000	2,050	655,000	3,050	2,160,000	4,050	5,050,000
100	76·0	1,100	101,000	2,100	704,000	3,100	2,270,000	4,100	5,240,000
150	257	1,150	116,000	2,150	756,000	3,150	2,380,000	4,150	5,430,000
200	608	1,200	131,000	2,200	810,000	3,200	2,490,000	4,200	5,630,000
250	1,190	1,250	149,000	2,250	866,000	3,250	2,610,000	4,250	5,840,000
300	2,050	1,300	167,000	2,300	925,000	3,300	2,730,000	4,300	6,050,000
350	3,260	1,350	187,000	2,350	987,000	3,350	2,860,000	4,350	6,260,000
400	4,870	1,400	209,000	2,400	1,050,000	3,400	2,990,000	4,400	6,480,000
450	6,930	1,450	232,000	2,450	1,120,000	3,450	3,120,000	4,450	6,700,000
500	9,500	1,500	257,000	2,500	1,190,000	3,500	3,260,000	4,500	6,930,000
550	12,700	1,550	283,000	2,550	1,260,000	3,550	3,400,000	4,550	7,240,000
600	16,400	1,600	312,000	2,600	1,340,000	3,600	3,550,000	4,600	7,400,000
650	20,900	1,650	342,000	2,650	1,420,000	3,650	3,700,000	4,650	7,650,000
700	26,100	1,700	374,000	2,700	1,500,000	3,700	3,850,000	4,700	7,890,000
750	32,100	1,750	408,000	2,750	1,580,000	3,750	4,010,000	4,750	8,150,000
800	38,900	1,800	443,000	2,800	1,670,000	3,800	4,170,000	4,800	8,410,000
850	46,700	1,850	482,000	2,850	1,760,000	3,850	4,340,000	4,850	8,670,000
900	55,400	1,900	522,000	2,900	1,850,000	3,900	4,510,000	4,900	8,950,000
950	65,200	1,950	564,000	2,950	1,950,000	3,950	4,690,000	4,950	9,220,000
1,000	76,000	2,000	608,000	3,000	2,050,000	4,000	4,870,000	5,000	9,510,000

$B_1 = \frac{1}{t^2}$.

t (Secs.)	B_1	t (Secs.)	B_1	t (Secs.)	B_1	t (Secs.)	B_1	t (Secs.)	B_1
5	·0400	105	·0000907	205	·0000238	305	·0000107	405	·00000610
10	·0100	110	·0000826	210	·0000227	310	·0000104	410	·00000595
15	·00444	115	·0000756	215	·0000216	315	·0000101	415	·00000581
20	·00250	120	·0000694	220	·0000207	320	·00000977	420	·00000567
25	·00160	125	·0000640	225	·0000198	325	·00000947	425	·00000554
30	·00111	130	·0000592	230	·0000189	330	·00000918	430	·00000541
35	·000816	135	·0000520	235	·0000181	335	·00000891	435	·00000529
40	·000625	140	·0000510	240	·0000174	340	·00000865	440	·00000517
45	·000494	145	·0000476	245	·0000167	345	·00000840	445	·00000505
50	·000400	150	·0000444	250	·0000160	350	·00000816	450	·00000494
55	·000331	155	·0000416	255	·0000154	355	·00000794	455	·00000483
60	·000278	160	·0000391	260	·0000148	360	·00000772	460	·00000473
65	·000237	165	·0000367	265	·0000142	365	·00000751	465	·00000463
70	·000204	170	·0000346	270	·0000137	370	·00000731	470	·00000453
75	·000178	175	·0000326	275	·0000132	375	·00000711	475	·00000443
80	·000156	180	·0000309	280	·0000128	380	·00000693	480	·00000434
85	·000138	185	·0000292	285	·0000123	385	·00000675	485	·00000425
90	·000123	190	·0000277	290	·0000119	390	·00000658	490	·00000417
95	·000111	195	·0000263	295	·0000115	395	·00000641	495	·00000408
100	·000100	200	·0000250	300	·0000111	400	·00000625	500	·00000400

$C_1 = \frac{1}{d}$.

d (Ins.)	C_1	d (Ins.)	C_1	d (Ins.)	C_1	d (Ins.)	C_1	d (Ins.)	C_1
1·00	1·000	3·25	·308	5·50	·182	7·75	·129	10·00	·100
1·25	·800	3·50	·286	5·75	·174	8·00	·125	10·25	·0976
1·50	·667	3·75	·267	6·00	·167	8·25	·121	10·50	·0952
1·75	·571	4·00	·250	6·25	·160	8·50	·118	10·75	·0930
2·00	·500	4·25	·235	6·50	·154	8·75	·114	11·00	·0909
2·25	·444	4·50	·222	6·75	·148	9·00	·111	11·25	·0889
2·50	·400	4·75	·211	7·00	·143	9·25	·108	11·50	·0870
2·75	·364	5·00	·200	7·25	·138	9·50	·105	11·75	·0851
3·00	·333	5·25	·190	7·50	·133	9·75	103	12·00	·0833

Example : Required, the pressure, P, to drive a carrier through a tube 5 ins. in diameter (C_1) and 2.000 yds. long (A_1) in 200 secs. (B_1). P = 608,000 × ·0000250 × ·200 = 3 lbs.

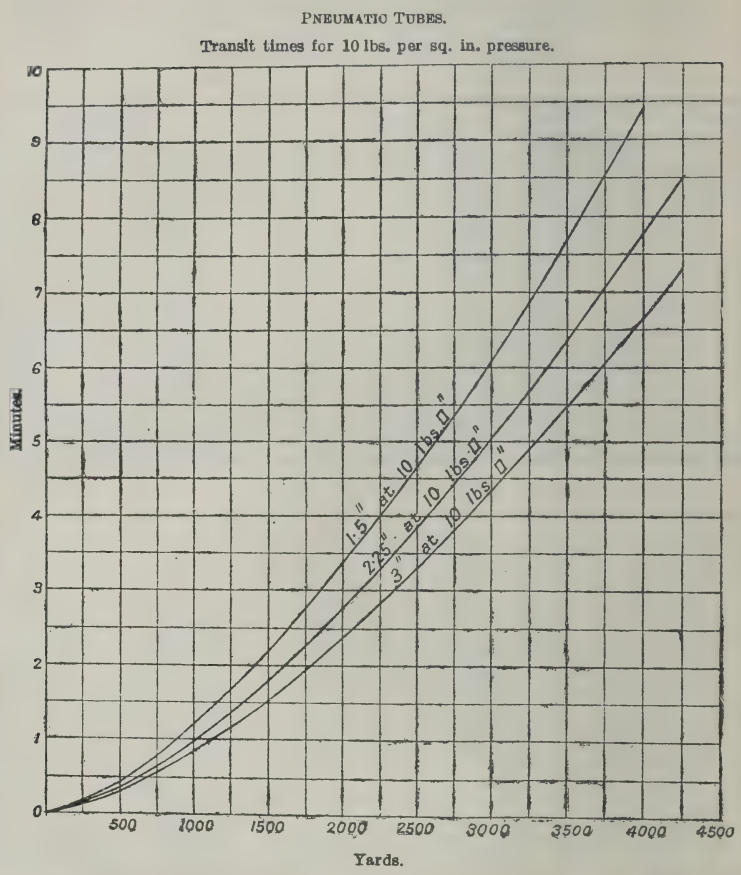


FIG. 18.

PNEUMATIC HOUSE TUBES.

These are, in the British Post Office Service, usually of brass or copper, and in some cases are operated on lines similar to the foregoing system, but considerable modifications have been introduced.

A very general system for warehouses is a circular one with loops into the various apartments which it is desired to connect.

The air is continually circulated by means of a blower of the Roots type which is joined at one end of the tube, whilst the other is open to atmosphere.

Carriers are despatched from a station by being inserted in the 'return' pipe of the loop and are received on the 'flow' pipe, suitable receiving apparatus being provided for the interception of a withdrawal of the carrier without interrupting the flow of air, which is made to travel through a bye-pass.

SECTION XXIV

REFRIGERATION—REFRIGERATING MACHINERY—ICE MAKING—COLD STORAGE

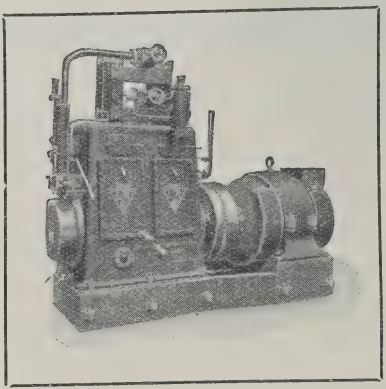
(pp. 1209-1235.)

(Contributed by Hal Williams, M.I.Mech.E., M.I.E.E., Consulting Engineer, and A. G. Clausen, A.M.I.Mech.E.)

POLAR

REFRIGERATING PLANTS ICE-MAKING MACHINERY

CO_2 AND NH_3 SYSTEMS



Vertical quick revolution, twin cylinder, single acting CO_2 compressor, enclosed crankcase type, fitted with forced lubrication and direct coupled to variable-speed electric motor. Made at our Works, Warrington.

THE

LIVERPOOL REFRIGERATION CO., LTD.

COLONIAL HOUSE LIVERPOOL ENGLAND

F. 1209

SECTION XXIV

REFRIGERATION—REFRIGERATING MACHINERY—ICE MAKING—COLD STORAGE.*

(Contributed by Hal Williams, M.I.Mech.E., M.I.E.E., Consulting Engineer, and A. G. Clausen, A.M.I.Mech.E.)

Refrigeration, or the mechanical production of low temperatures for commercial use, is a process of removing heat at a low level and discharging it at a high level on the thermometer scale.

In a steam-power plant heat is applied to evaporate water in a boiler; a small proportion of the heat supplied is converted into useful work in the engine, and the balance which cannot be utilised is rejected to the condensing water.

A refrigerating machine is simply a reversed heat engine; the cold air machines are now obsolete, and in all modern plant heat is taken up by the evaporation of some suitable working fluid termed the 'refrigerant,' and more heat is applied, generally in the form of power, to compress the resulting vapour to such a pressure that it can be condensed.

The sum of the heat taken up by evaporation and that added as power is then discharged in a condenser, to water of the prevailing normal temperature, the refrigerant is condensed, and as a liquid is in a state to complete the cycle of operations.

The performance of a refrigerating machine is the ratio of the number of heat units abstracted to those applied as power, and for a reversible cycle with a perfect fluid is $\frac{T_1}{T - T_1}$

where

T = absolute temperature of condensation;
T_1 = „ „ „ evaporation.

A refrigerating machine working on the compression system—which is practically universal—is shown diagrammatically in fig. 1 (see p. 1210) and consists of four essential parts:

(1) The evaporator or refrigerator, in which the refrigerant boils or evaporates at a low pressure and temperature.
(2) A gas pump or compressor to withdraw the vapour formed in the evaporator and compress it to such a pressure that it can be condensed by the water supply available.
(3) The condenser in which the compressed gas is condensed at a high pressure.
(4) The regulating or expansion valve, provided with fine adjustment to control the rate of flow of the condensed liquid from the condenser into the evaporator.

Starting with the liquid refrigerant in the condenser, the thermal cycle is as follows:

The liquid at condenser pressure and temperature is permitted to flow through the regulating valve into the evaporator.

On passing the valve a proportion immediately evaporates, taking up its latent heat from the remainder, which in this way is cooled to the evaporator temperature.

The remainder evaporates at constant pressure and temperature, taking up its latent heat from the air, brine, water, or other substance that is in contact with the exterior surface of the evaporator, and which is correspondingly reduced in temperature.

The proportion of the liquid evaporated to cool the remainder to the temperature of evaporation is not available for refrigeration, and represents a loss as compared to the ideal or reversible cycle.

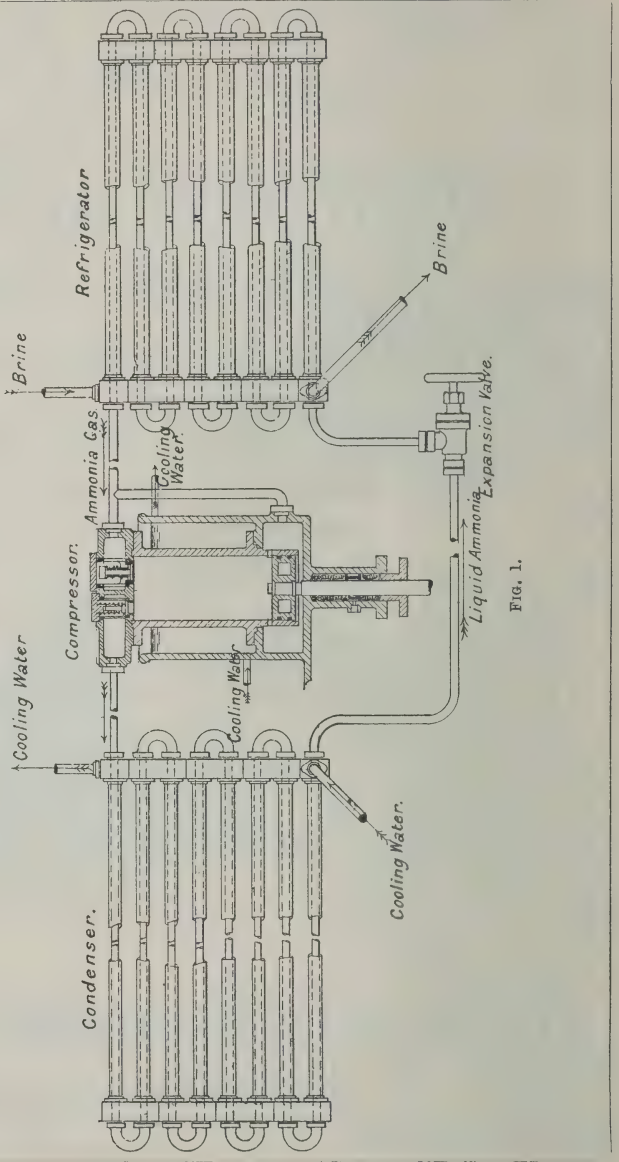
The liquid should therefore be cooled to as low a temperature as is practicable by ordinary means before passing the regulating valve, and its specific heat, considered as a fraction of its latent heat, should be low.

The vapour leaving the evaporator may be either:

(*a*) In a dry saturated state;
(*b*) Slightly superheated, or having its temperature a few degrees above its saturation temperature.
(*c*) In a wet state, or carrying with it some particles of unevaporated liquid.

With (*a*) and (*b*) the condition of working known as 'dry compression' is obtained and the work of compression, converted into heat, raises the temperature of the refrigerant so that it is discharged from the compressor in a superheated condition.

* For further information the reader is referred to 'Mechanical Refrigeration,' by Hal Williams (Sir Isaac Pitman & Sons, Ltd.), from which publication certain of the tables and data included in this section have been condensed.



THE COMPRESSION SYSTEM.

FIG. 1.

A small proportion of the heat of compression can be absorbed by water jackets on the discharge side of the compressor. Owing to the low initial temperatures, complete water jacketing cannot be used as with air compressors, as it would cause losses through initial superheating.

With (*c*) the heat of compression is either partially or wholly taken up by the evaporation of the entrained liquid, and the state of working known as ' wet compression ' is obtained when the incoming vapour carries just sufficient liquid to maintain the refrigerant during compression at the temperature corresponding to its saturation pressure.

In practice the refrigerant is as a general rule at some point between the states giving either completely dry or wet compression.

In the condenser the gas, if superheated, must first be cooled down to its saturation temperature. when the process of condensation proceeds.

In order to permit the transfer of heat, the temperature of evaporation must be below that to which the air or brine is to be cooled, and similarly the temperature of condensation must be above that of the cooling water used.

The greatest efficiency is obtained when the temperature of evaporation is as high and the temperature of condensation is as low, as possible, so that the surfaces for heat transfer should be large to keep the necessary differences small.

The refrigeration effected per pound of refrigerant can be obtained directly from the tables of properties of the various fluids, by deducting the total heat of the liquid at the condenser temperature from the total heat of the saturated vapour at the evaporator temperature.

The volume of gas that must be pumped by the compressor can be obtained directly in the same way.

The amount by which the actual swept volume of the compressor must be greater than the net volume of the gas, will depend upon the characteristics of any particular compressor and the conditions of working, and averages about 25 per cent.

The power required to drive the compressor can be obtained directly from the Mollier or from the entropy diagrams, or from indicator diagrams for the compressor.

The heat that must be transferred to the condensing water is the sum of the heat abstracted or effective refrigeration, and the heat equivalent of the work done in compression.

The quantity of water required will vary directly with the permitted rise in temperature.

The selection of a working fluid is governed by its limits of pressure and volume over the normal range of temperatures, its critical pressure, and the relative values of its specific heat as a liquid and its latent heat.

Other factors are its action, if any, on the materials used for the construction of machinery or on lubricating oils, its poisonous properties, and its inflammability.

There are many fluids that can be used, but by far the most important are Ammonia and CO_2. Sulphuric ether was often used in the early days of refrigeration, and SO_2 has been employed to a considerable extent on the Continent of Europe.

SO_2 methyl-chloride, occasionally ethyl-chloride, and other fluids, are also used for the very small machines of the household type that are now made in large numbers.

Ammonia Compression Refrigerating Machines.

TABLE I.—PROPERTIES OF ANHYDROUS AMMONIA, NH_3.

Temperature. ° F.	Absolute Pressure. Lbs. per sq. in.	Volume Vapour. Cu. ft. per lb.	Total Heat.		Latent Heat. B.Th.U.'s per lb.
			Liquid. B.Th.U.'s per lb.	Vapour. B.Th.U.'s per lb.	
−22	17·34	15·43	19·2	604·3	585·1
−13	21·99	12·36	28·9	607·5	578·6
− 4	27·59	9·99	38·6	610·5	571·9
5	34·27	8·15	48·3	613·3	565·0
14	42·18	6·70	58·2	616·1	557·9
23	51·47	5·56	68·0	618·6	550·6
32	62·29	4·64	77·9	621·0	543·1
41	74·80	3·90	87·9	623·2	535·3
50	89·19	3·30	97·9	625·2	527·3
59	105·60	2·80	108·1	627·1	519·0
68	124·30	2·39	118·3	628·8	510·5
77	145·40	2·05	128·5	630·2	501·7
86	169·20	1·77	138·9	631·5	492·6
95	195·80	1·53	149·4	632·6	483·2
104	225·40	1·33	159·9	633·4	473·5

TABLE II.—REFRIGERATING EFFECT OBTAINED PER CUBIC FOOT SWEPT VOLUME OF COMPRESSOR IN B.Th.U.'s WITH DIFFERENT CONDENSER AND EVAPORATOR TEMPERATURES.

Temperature Evaporation. ° F.	Suction Gauge Pressure.	Condensing Temperatures.		
		68° F.	86° F.	104° F.
		Corres ponding gauge pre ssures.		
		109·6	154·5	210·7
−22	2·6	19·8	16·9	13·9
−13	7·3	27·8	24·5	21·1
− 4	12·9	36·7	32·5	28·7
5	19·6	46·5	42·2	38·3
14	27·5	58·0	54·5	50·5
23	36·8	71·0	67·0	62·5
32	47·6	87·0	82·5	78·5

TABLE III.—COMPRESSOR I.H.P. PER CUBIC FOOT SWEPT VOLUME PER MINUTE WITH DIFFERENT CONDENSER AND EVAPORATOR TEMPERATURES.

Temperature of Evaporation. ° F.	Suction Gauge Pressure.	Condensing Temperatures.		
		68° F.	86° F.	104° F.
		Corres ponding gauge pre ssures.		
		109·6	154·5	210·7
−22	2·6	0·167	0·183	0·205
−13	7·3	0·190	0·220	0·250
− 4	12·9	0·205	0·245	0·282
5	19·6	0·220	0·270	0·316
14	27·5	0·226	0·282	0·347
23	36·8	0·220	0·290	0·363
32	47·6	0·200	0·288	0·377

Ammonia is generally considered to be the most satisfactory and economical refrigerant. It has a high latent heat, and over the normal temperature range the pressures and sizes of parts are comparable to ordinary steam and air compressor practice.

The gas has a highly pungent odour and is irrespirable, so that great care must be exercised in working or in overhauls, to prevent dangerous leakages.

It is exceedingly soluble in water, and one volume of water at 60° F. will absorb 600 volumes of gas.

Ammonia attacks copper and its alloys, so that none of these must be used for parts of the machine with which it comes in contact.

The figures given in Tables II. and III. are for compressors working with dry or practically dry compression. They represent the average results that are obtained with machines of moderate size, but the variations in performance and power naturally vary to some extent with the characteristics of different makes.

With wet compression, both the number of B.Th.U.'s abstracted and compressor I.H.P. are reduced per unit compressor volume, and theoretically the performance, or ratio of $\frac{\text{B.Th.U.'s}}{\text{I.H.P.}}$, is slightly superior, though in practice dry compression gives the best results.

Carbonic Acid Refrigerating Machines.

CO_2 occupies a unique position in the group of fluids used for refrigeration in that its critical temperature (88·4° F.) often comes within the working range; also its specific heat is high, and its latent heat relatively low.

Its efficiency with condensing water at about 50° F. is but little inferior to that of the ammonia machine, but falls away more rapidly with an increasing condenser temperature. There is, however, no break in the curves of capacity at the critical temperature.

CO_2 is inodorous, and it is stated that air containing up to 8 per cent. can be inhaled without danger, though at higher percentages it causes stupefaction and suffocation.

The quantities used in machines are, however, such that even if the whole charge were to escape, in any ordinary situation, it is hardly likely to produce a dangerous concentration, and on this account the Board of Trade regulations permit CO_2 machinery to be installed in the engine-room or similar space on board ship, whereas ammonia machines must be housed in a special deck-house.

CO_2 does not attack any metals, so that condensers can be constructed of copper piping, and these advantages have given it a predominant position for marine refrigeration.

Although the pressures are high, the diameters of all parts are small, so that there are no constructional difficulties.

TABLE IV.—PROPERTIES OF CARBONIC ACID.

Temperature. ° F.	Absolute Pressure in Atmospheres.	Volume Vapour. Cu. ft. per lb.	Total Heat.		Latent Heat. B.Th.U.'s per lb.
			Liquid. B.Th.U.'s per lb.	Vapour. B.Th.U.'s per lb.	
−22	14·48	0·432	−24·78	102·0	126·7
−13	16·92	0·367	−21·03	102·2	123·3
− 4	19·67	0·313	−17·17	102·3	119·5
5	22·75	0·267	−13·16	102·1	115·3
14	26·19	0·228	− 8·99	101·7	110·7
23	30·03	0·195	− 4·62	100·9	105·6
32	34·29	0·167	0·00	99·83	99·83
41	39·01	0·142	4·93	98·3	93·37
50	44·22	0·120	10·28	96·24	85·95
59	49·96	0·101	16·22	93·44	77·22
68	56·26	0·084	23·08	89·56	66·49
77	63·16	0·067	31·62	83·8	52·17
86	70·70	0·047	45·45	72·46	27·00
88·4	72·86	0·035	59·23	59·23	0·00

TABLE V.—REFRIGERATING EFFECT OBTAINED PER CUBIC FOOT SWEPT VOLUME OF COMPRESSOR IN B.TH.U.'S WITH DIFFERENT CONDENSER AND EVAPORATOR TEMPERATURES.

Temperatures of Evaporation. ° F.	Suction Gauge Pressure in Atmospheres.	Condensing Temperatures.		
		68° F.	86° F.	104° F.
		Corresponding Pressure in Atmospheres.		
−22	13·48	132	104	82
−13	15·92	160	134	106
− 4	18·67	196	162	132
5	21·75	236	200	166
14	25·19	282	242	200
23	29·03	334	288	242
32	33·29	396	348	294

TABLE VI.—COMPRESSOR I.H.P. PER CUBIC FOOT SWEPT VOLUME PER MINUTE WITH DIFFERENT CONDENSER AND EVAPORATOR TEMPERATURES.

Temperatures of Evaporation. ° F.	Suction Gauge Pressure in Atmospheres.	Condensing Temperatures.		
		68° F.	86° F.	104° F.
		Corresponding Pressure in Atmospheres.		
−22	13·48	1·33	1·45	1·58
−13	15·92	1·42	1·57	1·73
− 4	18·67	1·46	1·65	1·85
5	21·75	1·47	1·69	1·95
14	25·19	1·43	1·69	1·99
23	29·03	1·34	1·62	1·97
32	33·29	1·22	1·50	1·90

The figures given in Tables V. and VI. are for ordinary CO_2 compressors working with dry compression. They represent the average results that are obtained with compressors of moderate size, but the variations in performance and power naturally vary to some extent with the characteristics of different makes.

To obtain the best results in working, a CO_2 machine should be run with the suction gas in a dry saturated state, or even slightly superheated. When leather piston and gland packings were used this was impracticable, as the leathers were burnt by the temperatures reached with dry compression. All modern CO_2 machines have piston rings and metallic gland packings, so that there is now no difficulty in running a CO_2 machine to give the best results that can be obtained.

With high condensing temperatures the performance of a CO_2 machine can be appreciably improved by permitting the cooling of the liquid by its own partial evaporation to take place in two stages.

The compression also takes place in two stages: in the first the vapour from the liquid that has effected refrigeration, plus that from the second stage of the partial evaporation for liquid cooling, is compressed to the pressure at which the first stage of partial evaporation ends, when the whole of the vapour is compressed to the condensing pressure.

Other Refrigerants.

Both the theoretical and practical performance of other fluids used approximate closely to the results obtained with Ammonia, but the B.Th.U.'s extracted and compressor I.H.P. as shown in the tables should be divided by the following factors :

For Sulphur Dioxide	2·89
For Methyl Chloride	1·94
For Ethyl Chloride	6·57

The principal characteristics of these refrigerants, taken at 32° F. are :

TABLE VII.—REFRIGERANTS.

Refrigerant.	Absolute Pressure. Lbs. per sq. in.	Volume Vapour. Cu. ft. per lb.	Specific Heat of Liquid.	Latent Heat. B.Th.U.'s per lb.
Sulphur Dioxide .	22·45	3·41	0·322	156·8
Methyl Chloride .	36·63	2·68	0·467	174·6
Ethyl Chloride .	8·90	8·88	0·376	168·9

With very small machines, which receive practically no skilled attention, low working pressures are an advantage in reducing the risks of leakage, but pressures below the atmosphere should be avoided, as inward leaks of air cause difficulties in working, and any moisture carried in may combine with SO_2, forming sulphuric acid, which corrodes the parts.

Compressors.

Both single-acting and double-acting compressors are made in a large variety of designs and sizes, and during the last few years there have been great developments in high-speed forced lubrication machines suitable for direct coupling to electric motors.

With the exception of machines made by Messrs H. J. West & Co., Ltd., who employ a sleeve valve, all valves are of the automatic type, the high-speed compressors using light ring and plate valves of similar character to those developed for air compressors.

Most single-acting ammonia compressors are made of the enclosed type and work with ammonia at the suction pressure in the crankcase. The gland is of the rotary type on the crankshaft, and it is sealed with oil from the pressure lubricating pump. These machines have trunk pistons, the lower portion of which forms the crosshead, and is fitted with oil scraper rings. The upper forms the piston proper, and the suction ports in the cylinder walls lie between the limits of the travel of the upper and lower parts.

The suction valves are placed at the top of the piston, and the delivery valves in a spring-loaded false or safety head.

The arrangement enables the maximum valve port area to be provided with a minimum clearance space; the gas passes through the compressor in one direction, so that the upper portion can be effectively water-jacketed, and it is claimed that the 'Uniflow' principle increases the compressor efficiency.

The safety head is a valuable feature, as it will lift and save any risk of accident in the event of liquid ammonia or oil passing into the compressor.

These machines are always of the vertical type, and are made with from one to four cylinders.

Double-acting compressors are made both in the horizontal and vertical types, and the valves are arranged either in hemispherical end covers or round the circumference at each end of the cylinder.

Double-acting compressors usually have the gland constructed in two portions, with a central space or lantern between them. The lantern space is connected to the suction side of the compressor, so that the outer portion of the gland is subjected to the low and steady suction pressure only, and any gas leaking past the inner portion is retained in the system.

CO_2 compressors are now almost invariably machined from high carbon steel forgings to ensure strength and the necessary gas-tightness, and for the same reasons all stop valves and fittings are of mild steel and machined from the solid.

Double-acting compressors are general except in the smallest sizes; the gland is provided with a lantern which is sealed with oil at a rather higher pressure than the compressor is delivering against.

This higher pressure is obtained by using a differential piston in a cylinder, connecting the greater area to the gas delivery and the lesser area to the oil space in the gland, a small hydraulic pump being used to make up any oil leakage.

In refrigerating machinery, absolute gas tightness is essential, and it is usual to test compressors and all other parts subject to gas pressure by hydraulic test to about three times the normal maximum pressure; also to test them when immersed in water with compressed air to about one and a half times the normal maximum pressure to show up any minute leaks or porosity in materials.

Condensers.

Condensers are made in three general types:

Submerged Condensers consist of a coil, or series of coils, of piping connected to headers, contained in a tank or casing, generally of circular form, through which the water is passed.

Atmospheric Condensers consist of coils in vertical stacks, over which the water flows, erected in the open air and standing over a collecting tank. These are often made of large size, and the water continuously recirculated so that they function also as recooling towers and only require a small make up supply.

Double Pipe Condensers consist of a number of concentric pipes, arranged in series as shown in Fig. 1 (p. 1210); the water passes through the inner pipe and the gas condenses in the annulus between the two pipes. The gas and water traverse the pipes in opposite directions, so that the condenser is counter-current, and on account of the high water velocities the rate of heat transfer is high.

Evaporators.

A refrigerating machine may be used either to cool brine or water that is in turn circulated through the cooling appliances used, or it may be arranged so that the evaporator piping itself forms the cooling appliances, and this method is referred to as direct expansion.

Brine Cooling Evaporators may be either of submerged coil or double pipe construction, or for ammonia or other low pressure refrigerants they may be of shell and tube form, the ammonia evaporating in the shell around the tubes through which the brine is circulated in either one or more passes, depending upon the quantity and velocity required.

Evaporators for Cooling Water are frequently made of the open type where the water trickles over the outside of the coils to avoid the danger of freezing up in any confined spaces.

The Direct Expansion System may be applied for several purposes, but is generally only used for cooling cold storage rooms.

The piping may be fixed directly in the rooms, or nested together to form an air-cooling battery, with a fan to circulate air over it and through the rooms by a system of ducts.

This type of air cooler must have a circulation of brine maintained over the piping to keep the surfaces free from frost where temperatures below freezing-point are required, or otherwise the accumulation of snow and frost would rapidly choke it up.

Heat Transference.

The proportions of the condenser and evaporator are determined by the rates of heat transfer that can be obtained.

The rate of heat transfer is directly proportional to the temperature difference. It varies as a power of the velocity V, and may be generally taken as $V^{0.75}$.

The rate of heat transfer varies with the temperature, viscosity, density, and state of the fluids in contact with the surfaces, and to secure satisfactory operation over a long life it is necessary to allow for deposits on and deterioration of pipe surfaces.

The rate of heat transfer through iron pipe and under the conditions usually prevailing in refrigerating machinery may be taken as being approximately as follows:

	B.Th.U.'s per sq. ft. surface per hour and per 1° F. mean temp. difference.
In submerged condensers	40 to 45
Atmospheric condensers	60 to 70
Double pipe condensers with a water velocity of 3 feet per second	100 to 120
Submerged brine coolers	35 to 40
Double pipe or shell and tube brine coolers with a brine velocity of 3 feet per second	80 to 90
Direct expansion or brine piping in a room with natural air circulation only	2·0 to 2·5
Air coolers with fans and forced air circulation and an air velocity of 10 feet per second	6 to 7

To conform with general practice, pipe surfaces should be so proportioned that the mean temperature difference between the refrigerant and the brine or water is 10° to 15° F., and in the piping for rooms a difference of 15° to 20° between the air and the refrigerant or brine.

Indicator Diagrams.

The indicator diagram gives an exact measure of the indicated horse-power developed in a compressor, and if it is compared with an indicator diagram taken when the machine is new and in good condition, it will disclose any defects in working such as leaky valves or piston rings.

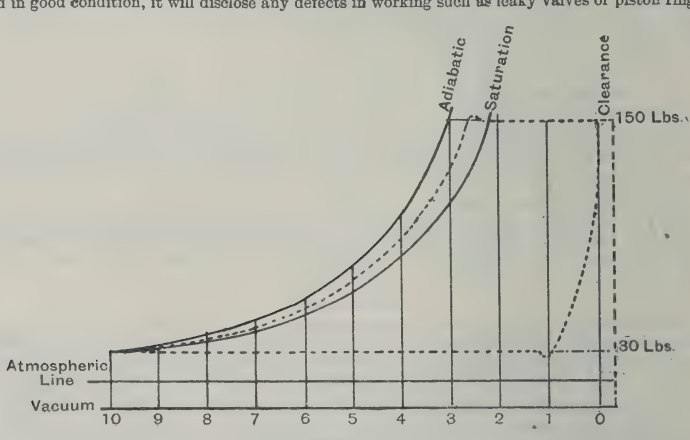


FIG. 2.

In order completely to analyse the indicator diagram, it is necessary to draw on it the curves for adiabatic compression, and for compression at saturation temperatures, as is shown in fig. 2.

With ammonia, the exponent for adiabatic compression is 1·29 and the compression curve follows the formula $P \cdot V^{1\cdot29} = C$.

The saturation curve can easily be plotted from the volumes given in the table of properties.

The actual compression curve will lie between these two, and in the case of dry compression it will be close to the adiabatic line.

The horizontal suction and delivery lines should practically coincide with the gauge pressures. If the gauges read correctly, and the difference is more than one or two pounds, it shows that there is a restriction through the valves which may be due to either too heavy springs, the valves not working freely, or insufficient areas in the ports.

At the points where the valves open, there is always a peak on the pressure line due to the difference in pressure required to lift them from their seats, but these peaks should only be small.

The re-expansion of the gas remaining in the clearance space at the end of the stroke follows approximately the same curve as that formed on the compression stroke.

The effective swept volume of a compressor, which is indicated by the length of the suction line, is thus reduced with the proportion of the clearance volume, but as the power required is correspondingly reduced, a moderate amount of clearance does not seriously affect the performance, though it may make a considerable difference in the output of compressors with otherwise similar dimensions.

If the compression curve approaches or overlaps the adiabatic line, it indicates a leakage from the condenser side, back into the compressor through the delivery valves or the joint rings of their seats.

If the compression curve approaches the saturation line, it indicates a leakage either through the suction valves or past the piston rings.

If the diagrams from each end of a double-acting compressor are compared, and a falling away of the compression line occurs at one end only, it indicates that the leakage is at the suction valve there. If both ends are similar, it may be either the valves, the piston rings, or both that require attention.

Ammonia Absorption Machines.

Ammonia absorption machines are only used in special cases where there is sufficient live or preferably waste steam which can be utilised.

The arrangement is shown diagrammatically in fig. 3, p. 1218.

In place of the compressor the ammonia is absorbed by cool weak liquor, which, being thus enriched, is pumped into a generator or boiler, where it is heated, and the ammonia driven off into the condenser.

An interchanger is provided to heat up the cool rich liquor passing to the generator, and at the same time cool down the hot weak liquor returning to the absorber.

The small 'Electrolux' refrigerator working on the absorption system makes most ingenious use of Dalton's law of partial pressures by introducing an inert gas into what is normally the low pressure side of the system. The total pressure throughout is maintained constant, so that circulation of the liquid can be effected on the thermo-syphon principle without any moving parts. At the same time the partial pressure of the ammonia in the evaporator is low enough to effect refrigeration.

Small and Automatic Refrigerating Machines.

The use of very small refrigerating machines, both for commercial and for domestic purposes, is now extending very rapidly, and their employment has been considerably stimulated by the recent 'Pure Food Act.'

With the exception of the Electrolux machine referred to, practically all these machines work on the compression system. They are so equipped that they are completely automatic in their operation, and standard models are driven by electric motors ranging from $\frac{1}{4}$th b.h.p. to 2 b.h.p.

A low-pressure refrigerant, generally SO_2 or methyl chloride, is used to simplify the construction and to minimise the risk of leakage at the gland, compressors are of the enclosed type, and the crankshaft gland is usually provided with a collar on the shaft, maintained by means of a spring in contact with a floating ring attached to the crank case by a syphon.

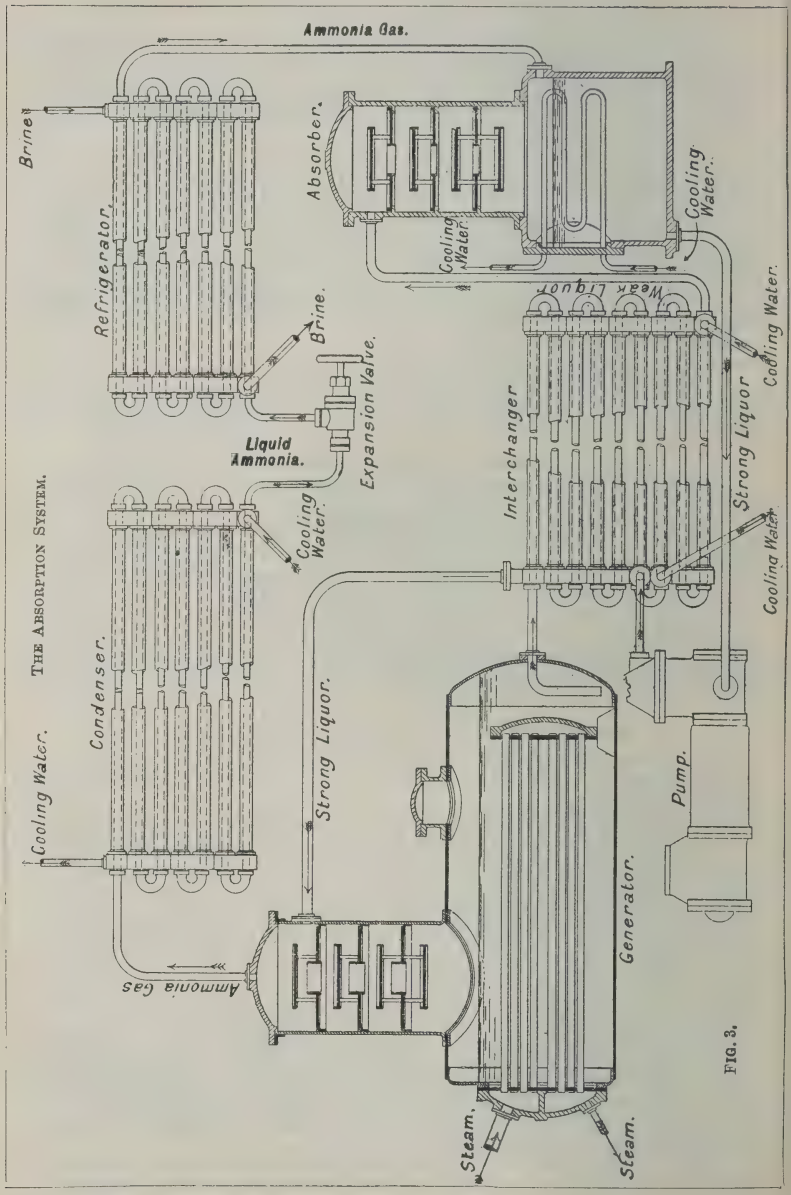
The compressors are started and stopped at predetermined temperatures by means of a thermostat operated mercoid switch, and a similar switch is fitted as a safety device to stop the motor if the condensing pressure rises to some point in excess of the normal working pressure.

In the smallest machines the condensers are air cooled, a fan either separate or framed in the compression fly-wheel being used to circulate air over it. In larger sizes the condenser is water cooled, and the flow of water is governed by a diaphragm controlled valve operated by the condenser pressure so that water only flows when the machine is in operation.

Automatic regulating valves are either of the float type operating by the level of the liquid refrigerant, or of the diaphragm type with a needle valve operated by the difference in pressure.

By far the largest numbers of small machines use SO_2 as a refrigerant, and a float type of regulating valve works in a receiver on the evaporator side, this receiver is also provided with an overflow pipe to return direct into the crank case any lubricating oil that has passed over.

There is a strong tendency for automatic features to be applied to rather larger units than the small self-contained plants, and many of these are working satisfactorily, but difficulties arise where more than one separate duty has to be performed.



THE ABSORPTION SYSTEM.

FIG. 3.

Ice Making.

The manufacture of ice is one of the principal uses of refrigeration, and the method now almost universally employed is the 'can' system, where the ice is made by placing cans or moulds containing water in a tank containing brine cooled to a temperature of 10° F. to 15° F.

When distilled and de-aerated water is used, the ice made is clear and transparent, but if ordinary water is frozen without agitation, the ice is soft and opaque, due to the minute bubbles of air dissolved in the water.

By agitating the water during the time of freezing, these air bubbles are swept off the ice surface, and the block made is clear, with the exception of a central core formed of the dissolved salts, the extent of the core depending on the hardness of the water.

Agitation can be effected by stirrers, but the best system in all except the smallest plants is that of blowing air through the water.

The time required for freezing can ice with brine at 15° F. is approximately :

Thickness of ice in inches	3	4	6	8	10	12
Time of freezing in hours	6	10	18	30	46	64

Ice moulds are made in a great variety of sizes to produce blocks of definite weights up to 4 cwts. each, which is a usual size for plants of large capacity, these blocks measuring about 22 in. wide, 11 in. thick, and 60 in. long.

The moulds must be made with a slight taper to enable the ice to slide out freely, and usually a number are secured together in a row, and handled as one unit by a crane.

The ice tank is arranged to contain the evaporator, which may either be of coil piping, or of the shell and tube type, and a vigorous circulation of the brine in the tank is maintained by means of a propeller or centrifugal pump.

A row of ice moulds when lifted from a tank is dipped into a thawing tank containing warm water to release the blocks of ice from the cans. These are then lifted, placed in a swinging tipping frame, and turned over so that the blocks slide out.

The empty row of moulds is then refilled with the correct quantity of water on one operation by a filling tank. One cubic foot of ice at 32° F. weighs 57·5 lbs., or 30 cubic inches weigh one pound.

In a temperate climate, and with a medium or large plant, about 230 B.Th.U.'s are required to produce one pound of ice from water at 65° F., and about 60 kw. hours are required with a motor-driven plant per ton of ice made. In warmer and tropical climates these figures are considerably increased.

Refrigerating Power.

The output of a refrigerating machine is often stated in Tons Refrigeration, and the standard American rating is then generally referred to.

One ton refrigeration is then equal to 288,000 B.Th.U.'s per day, or 12,000 B.Th.U.'s per hour when working with an evaporator temperature of 5° F. (— 15° C.) and a condensing temperature of 86° F. (30° C.).

Cold Storage.

The average temperatures maintained in cold storage rooms are as follows :

Butter—frozen .	10° F. to 12° F.	Hops	28° F.
Butter—short storage	32° F. „ 35° F.	Meat—frozen . .	15° F. to 16° F.
Cheese . . .	35° F. „ 40° F.	chilled . .	29° F. „ 30° F.
Eggs . . .	29° F. „ 31° F.	fresh . .	36° F. „ 38° F.
Furs and Fabrics .	30° F.	Fish—frozen . .	10° F. „ 12° F.
Fruit—Apples . .	32° F. „ 35° F.	fresh . .	30° F. „ 32° F.
Pears . .	35° F. „ 36° F.	Dried Fish—frozen .	15° F. „ 16° F.
Peaches .	32° F. „ 38° F.	for short storage	32° F. „ 35° F.
Oranges .	45° F. „ 50° F.		

Cold storage temperatures should, as far as possible, be maintained constant without variation. Both this and the maintenance of a definite temperature are particularly important where the temperature required is in the neighbourhood of the freezing-point. For produce that is frozen hard, an exact storage temperature is not quite so important, but variations should also be as far as possible eliminated.

The refrigerating power required for cold storage rooms varies very greatly with their size, shape, insulation, quantities of warm goods brought in and methods of working, but for ordinary storage rooms is about :

Capacity of Cold Rooms in Cu. Ft.	Temperature. 30° to 35° F. Tons Refrigeration.	10° to 15° F. Tons Refrigeration.
1,000	0·5	0·8
10,000	2·5	4·0
100,000	12·0	20·0

These figures assume continuous operation, and for reduced hours must be proportionately increased. The surface and extent of the cooling appliances will vary with the refrigerating power required and conditions of working.

The quantity of piping fixed in a room is usually determined by empirical rules based on practical results, which vary somewhat with different makers' and users' experience and give the following averages:

Capacity of Cold Rooms in Cu. Ft.	Temperature. Length of 1¼ in. Piping to be fixed in Stores.	
	30° to 35° F.	10° to 15° F.
1,000	225	450
10,000	1,125	2,250
100,000	6,000	12,000

Where other sizes of piping are used, the length is adjusted to provide a similar surface.

Calcium Brine.

Calcium Chloride Brine of a density ranging from 1·18 to 1·25 is usually employed. In estimating the weight of calcium chloride required to produce brine of a given density regard must be had to the fact that the commercial calcium chloride contains a large amount of water of crystallisation. An approximate rule is to multiply the density by 20 and subtract 20·1, the remainder gives the number of pounds of commercial calcium chloride required for each gallon of brine of the required density; for example, take the required density at 1·20, then $1{\cdot}20 \times 20 - 20{\cdot}1 = 24 - 20{\cdot}1 = 3{\cdot}9$ lbs. per gallon of brine.

Corrosion by Brines.

An important problem of refrigeration engineers is the corrosion by brines, either sodium chloride or calcium chloride, of copper, steel, or grey cast iron. An investigation by a French authority, M. Pierre, shows that the active element of this corrosion is due to the formation of an electrolytic couple, due to the association of air, brine, and the metal, and thus the dissolution of air should be avoided by submerging the return brine pipe so that it delivers below the level of the fluid in the main container. Strong brines cause more rapid corrosion, and the presence of magnesium chloride increases the activity. Lime or soda should be used with them to neutralise the formation of hydrochloric acid.

Insulation.

The thermal conductivity of the materials used for insulation may vary to some extent with individual samples, and with all loose fillings with the density to which they are packed.

The following figures are taken from the tests made at the National Physical Laboratory for the Food Investigation Board:

Material.	Density in lbs. per cu. ft.	Conductivity in B.Th.U.'s per hour per sq. ft. per 1 in. thickness and 1° F. difference in temperature.
Cork slab	11·3	0·306
" "	12·5	0·331
Granulated cork, fine	—	0·298
" " coarse	5·3	0·326
" "	7·3	0·345
Silicate cotton	13·0	0·302
" "	21·0	0·290
Dry charcoal	11·5	0·370
Kieselguhr	30·0	0·560
Wood—Pitch pine	42·0	1·036
Concrete	—	8·200

These figures are taken at normal cold storage temperatures. The conductivity increases slightly with an increase in temperature.

To obtain the conductivity of a compound wall, the reciprocals of the conductivities of its components must be added together, and this will give the reciprocal of the whole.

Thus, if a 6-in. concrete wall is insulated with 4-in. compressed cork slabs, the reciprocal of its conductivity

$$= \frac{1}{8 \cdot 2} \times 6 + \frac{1}{0 \cdot 33} \times 4 = 12 \cdot 85;$$

and its conductivity

$$= \frac{1}{12 \cdot 85} = 0 \cdot 078.$$

If, therefore, there is a difference in temperature of 40° F., the heat leakage would be 3·12 B.Th.U.'s per sq. ft. per hour, or about 75 B.Th.U.'s per sq. ft. per day.

It is not advisable to attempt to make any allowance for the rate of heat transfer between the air and the two wall surfaces, as while this may vary very greatly, it is in any case so relatively small an item that it does not appreciably affect the result.

In an insulated wall, the jointing material for cork slabs, the timber studding for loose fillers, and unavoidable inperfections of workmanship will make the conductivity of the material as erected rather greater than the test figures given, and the results should be increased by 10 per cent. to 15 per cent. to allow for this.

The leakage of heat through the insulated walls, floor, and ceiling of a cold store practically always forms the greater proportion of the total refrigerating power required to cool it, so it is necessary that the insulation provided should reduce this leakage to a minimum.

The first essential is that the material used should have a low co-efficient for the transfer of heat; it should have no attraction for moisture, be uninflammable, inodorous, and have a high specific heat.

It should pack well without tendency to settle and leave void spaces, and it should have no tendency to spontaneous combustion under any conditions.

Compressed cork slabs erected in cement or bitumen, and having their surfaces finished in hard-setting cement, provide the most satisfactory and permanent construction, but granulated cork and silicate cotton, or slag wool, are also largely used, and other materials that are employed are pumice stone, charcoal, kieselguhr, sawdust, and shavings.

Loose filling materials are packed between wooden studs and joists, erected against walls, floors, and ceilings, and covered by T. & G. boarding with or without waterproof paper.

An air space is often provided against external walls, the construction then being illustrated in figs. 4, 5, and 6.

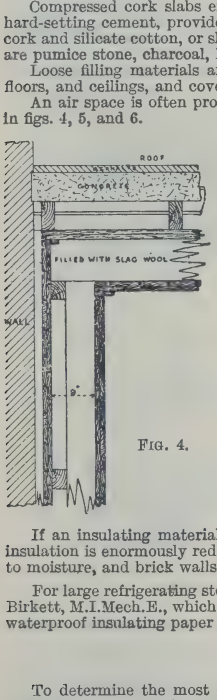


FIG. 4.

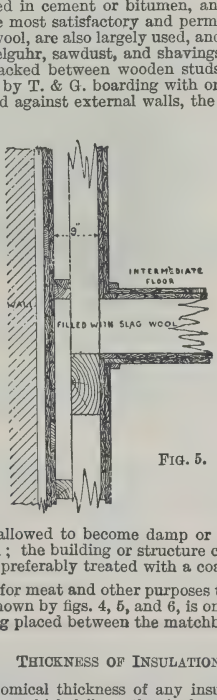


FIG. 5.

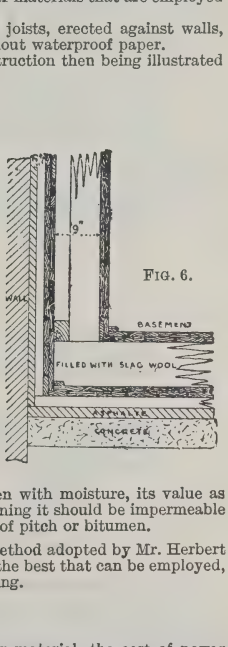


FIG. 6.

If an insulating material is allowed to become damp or sodden with moisture, its value as insulation is enormously reduced; the building or structure containing it should be impermeable to moisture, and brick walls are preferably treated with a coating of pitch or bitumen.

For large refrigerating stores for meat and other purposes the method adopted by Mr. Herbert Birkett, M.I.Mech.E., which is shown by figs. 4, 5, and 6, is one of the best that can be employed, waterproof insulating paper being placed between the matchboarding.

THICKNESS OF INSULATION.

To determine the most economical thickness of any insulating material, the cost of power to drive the refrigerating machine, which falls as the thickness of insulation increases, should be balanced against the interest on the increasing cost, and the decrease in the net or useful storage capacity.

The usual practice is to use a thickness of 4 ins. of compressed cork slabs for chill, and 6 ins. for freezing temperatures, or, alternatively, 6 and 9 ins. of granulated cork or silicate cotton. These figures are largely worked to, not entirely as giving equal insulating value, but as economical standards for the commercial thickness of cork slabs and sizes of timber.

Paper Insulation.

(*The Willesden Paper and Canvas Works, Ltd., Willesden Junction, London, N.W. 10.*)

A special description of paper known as 'Willesden' paper has for many years been largely used in the insulation of cold storage chambers. The usual method of constructing these chambers is to erect vertical studding at certain distances apart, and to cover this on both sides with grooved and tongued matched boarding, then to cover that in its turn with the Willesden paper, and over the paper to again fix another surface of matched boarding. There is then left a space between the two erections of matched boarding and paper equal to the width of the studding used, and this space is filled with some insulating material, such as charcoal, slag wool or rice husks, etc., and the work is done. The main use of the paper is to exclude air, and the Willesden material is preferred because of its durability and strength, and because it is not affected by damp nor liable to harbour fungoid growth.

'Neponset,' 'Ko-sat,' and 'Coted' are special papers of the highest grade, which are impervious to air and moisture under the severest conditions, and are largely used for cold storage insulation. They are made in rolls 36 ins. wide containing 1,000 sq. ft.

Cork Insulation.

The application of compressed cork slabs for insulation purposes is becoming more general, both in the U.S. and in England. Being in the form of slabs which are manufactured in various shapes and dimensions, they are easily fitted to the walls of the chamber. By being built together in various layers, breaking joint and cemented with a waterproof cement, they form an impermeable insulating lining to the chamber.

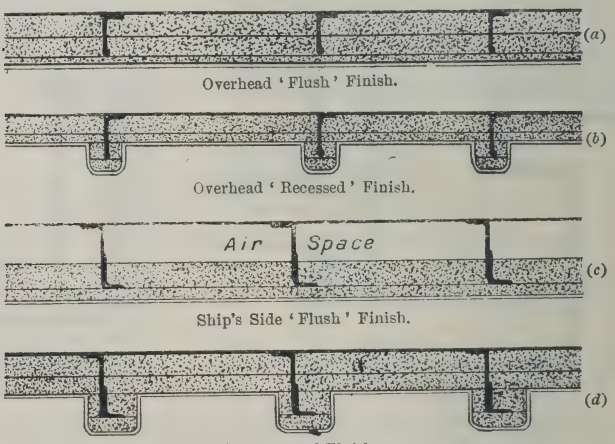


Fig. 7.—'Anderson' Jointless Insulation on Cork Slabs.

The interior facing to the chamber is usually of a special cement, preferably without joint.

With this system of insulation (fig. 7) a less thickness is required as compared with the insulating materials of a loose granular nature, and it has the further advantage of giving an increased capacity to the chamber. *Messrs. Thomas Anderson Ltd., Liverpool*, have had considerable success in its application. The efficacy of their preparation is illustrated by a recent case in which the insulation of a steamship, which had been submerged for weeks, on being raised was found intact, not requiring renewal or repair.

The material '*Fosalsil*' (p. 435) is very suitable for insulation purposes.

Insulation on Board Ship.

The insulation of cold chambers on board ship follows the same general principles and method that apply to insulation work carried out on land.

Owing to the way in which the ship's side and deck beams break up the surfaces, and to the absence of pure rectangular spaces, cork slabs are not so simple to use as loose filling

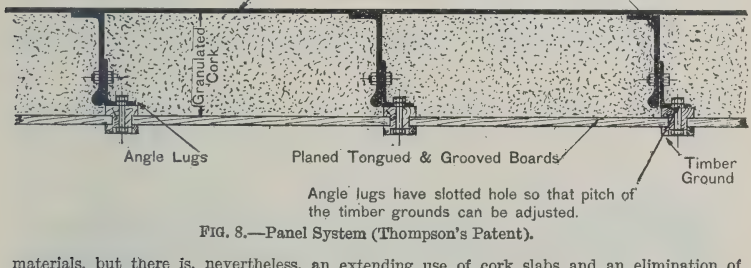


FIG. 8.—Panel System (Thompson's Patent).

materials, but there is, nevertheless, an extending use of cork slabs and an elimination of woodwork, inner linings, and air spaces.

One of the latest methods is the panel system of insulation (Thompson's patents), which, while it is particularly adapted to ships, is also applicable to work on land.

The construction is shown by figs. 8 and 9, and the advantages claimed are:

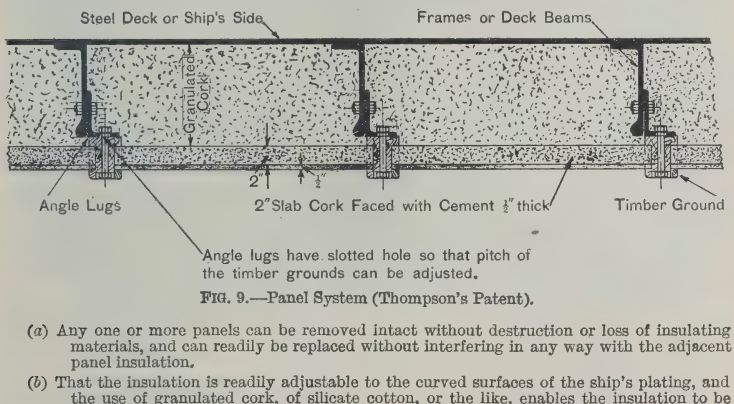


FIG. 9.—Panel System (Thompson's Patent).

(*a*) Any one or more panels can be removed intact without destruction or loss of insulating materials, and can readily be replaced without interfering in any way with the adjacent panel insulation.

(*b*) That the insulation is readily adjustable to the curved surfaces of the ship's plating, and the use of granulated cork, of silicate cotton, or the like, enables the insulation to be thoroughly well packed, no matter how curved or cut up the surface.

(*c*) When cement-faced cork slabs are used, the size of each slab being comparatively small, the liability to surface crack is slight.

(*d*) The wood grounds are exposed on the front faces, their condition can always be examined, and as they are ventilated, the danger of dry rot is greatly reduced.

(*e*) Repairs are easily effected, each panel being separate from all the others.

Operating Refrigerating Machinery.

As refrigerating machinery is solely concerned with temperatures, the gauges connected to the compressor suction and delivery are marked both with a pressure scale and a scale of the equivalent temperatures. The temperature scale should be used for regulating the working of the machine and in making any logs or records, as the differences are then obtained direct without reference to tables of properties.

The gauge on the suction side of the compressor indicates the pressure and temperature of evaporation, and that on the delivery side the pressure and temperature of condensation.

To operate a plant successfully, or analyse its working, it is also necessary to have thermometers to show the temperatures of:

1. The suction gas returning to the compressor from each separate evaporator circuit.
2. The discharge gas leaving the compressor.
3. The liquid refrigerant leaving the condenser.
4. The liquid refrigerant immediately before the regulating valves.
5. The brine leaving the evaporator.
6. The brine returning to the evaporator from each separate brine circuit.
7. The water entering the condenser.
8. The water leaving the condenser.
9. The temperature of the cold storage rooms, or, where a machine is used for special purposes, the temperature of the materials that are cooled.

The first four of these points are frequently omitted in commercial plants, but the second four are essential.

The suction gas thermometer readings should coincide with the evaporator gauge readings unless the suction gas is superheated, and thermometers in this position are particularly useful where there are several separate expansion circuits, as they provide the best indication for the quantity of liquid to be admitted to each, and for the manipulation of the regulating valves.

The discharge gas thermometer readings will be above the condenser gauge readings unless the compressor is run wet compression, and the elevation of the thermometer above the gauge temperature is an index of the dryness of the gas returning to the compressor.

The temperature of the liquid refrigerant will coincide with the condenser gauge reading unless the liquid is sub-cooled either by the use of a counter-current condenser or separate liquid cooler.

Sub-cooling of the liquid, particularly with CO_2, is always an advantage where it is practicable.

The regulation of the quantity of brine flowing through the separate circuits, and consequently of the refrigerating effect, is controlled by the difference between the brine flow and return temperatures.

The quantity of water flowing through the condenser is regulated by the difference in temperature between the inlet and outlet.

It is most important that the machine should be maintained fully charged with its refrigerant, and that supplies should be added from time to time to make good the inevitable small losses that occur with even the most carefully operated plant.

With a proper charge and well-proportioned installation, the condensing temperature should be from 10° to 15° F. above the mean condensing water temperature, and the evaporating temperature from 10° to 15° F. below the mean temperature of the brine, or about 15° to 20° F. below the air temperature when cold storage rooms are cooled by the direct expansion system.

At the same time the suction valve ports on the compressor should be frosted over, and, in the absence of thermometers, the delivery should be as hot as the hand can comfortably bear.

Closing down the regulating valve reduces the evaporating temperature and increases the gas discharge temperature, while opening up the regulating valve increases the evaporating temperature and decreases the discharge gas temperature.

The dryness of the suction gas is indicated by the discharge gas temperature, and while the aim should be to work as nearly as possible dry compression, the conditions of working that best suit individual makes of compressors and proportions of plant can only be determined by experience with them.

An undercharge of refrigerant is indicated by a high delivery temperature, a low condensing gauge temperature, and a low evaporator temperature, which does not rise as it should when the regulating valve is opened.

An overcharge is indicated by a cold delivery temperature, a high condenser gauge temperature, and an evaporator temperature which does not fall at the normal rate when the regulating valve is closed.

Of the two, an undercharge is the more serious, as it results in a reduced refrigerating effect, while an overcharge practically only affects the power required.

The refrigerant should be kept in pure condition by occasionally purging out any air or foreign gases, and the oil separator should be regularly drained to prevent oil passing over into the condenser and evaporator.

Refrigeration on Board Ship.

The use of refrigerating machinery on board ship may be roughly subdivided into three groups:

I. Small plants to cool provision storage rooms and to make small quantities of ice.

II. Large passenger vessels that are arranged to carry refrigerated cargo.

III. Large vessels used primarily for refrigerated cargo.

CO_2 refrigerating machines are almost invariably used for all classes and sizes of machines, and evaporative liquid cooling of the CO_2 is frequently employed.

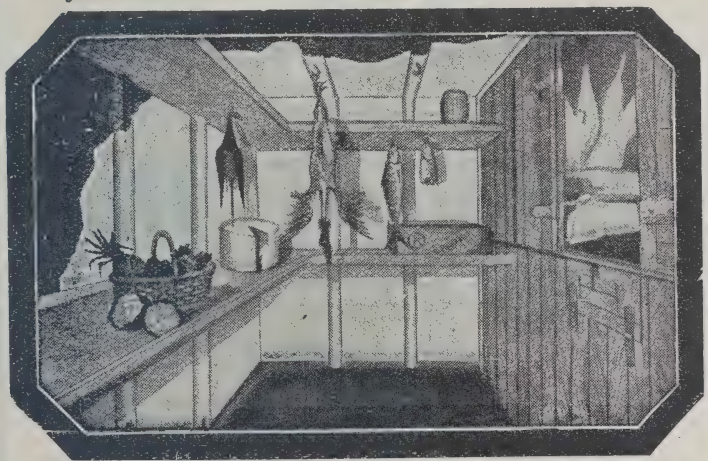
On steamships the larger machines are practically always steam-driven and incorporate their own steam condensers, but the smaller machines are now generally driven by electric motors, and on motor ships an electric drive is always used.

Compressors running at low and medium speeds are generally preferred, and while direct-coupled sets are sometimes installed, the drive is generally through double helical or other speed-reduction gears.

ON ADMIRALTY LIST.

'ANDERSON' INSULATION

JOINTLESS AND UNIT SYSTEMS.



For COLD *Storage and Provision Chambers, Ice Boxes and all Refrigerated Spaces,* HEATED *Surfaces, Bulkheads and Engine Room Casings.*

CORK SECTIONAL COVERING FOR BRINE AND COLD WATER PIPES.

Magnesia Sectional and Plastic Coverings for Steam Pipes, Boilers, Turbines, etc.

THOMAS ANDERSON LTD. (*Insulating Engineers*).
54-62, DERBY ROAD, LIVERPOOL.
Wires: 'Centrifugal, Liverpool.' Phone: 1070 Bootle.
BRANCHES: LONDON, GLASGOW, BELFAST, SOUTHAMPTON, BARROW-IN-FURNESS.

F. 1224

ON ADMIRALTY LIST.

'ANDERSON'

TRADE

MARK.

INSULATION.

JOINTLESS AND UNIT SYSTEMS.

For **COLD** Storage and Provision Chambers, Ice Boxes :: :: and all Refrigerated Spaces. :: ::

HEATED Surfaces, Bulkheads and Engine Room Casings.

CORK SECTIONAL COVERING FOR BRINE AND COLD WATER PIPES.

Magnesia Sectional and Plastic Coverings for Steam Pipes, :: :: Boilers, Turbines, etc., etc. :: ::

THOMAS ANDERSON,

Insulating Engineers, LIMITED

54 to 62, DERBY ROAD, LIVERPOOL.

'Phone : 1070 Bootle. **Wires : 'Centrifugal, Liverpool'**

BRANCHES: London, Glasgow, Belfast, Southampton and Barrow-in-Furness.

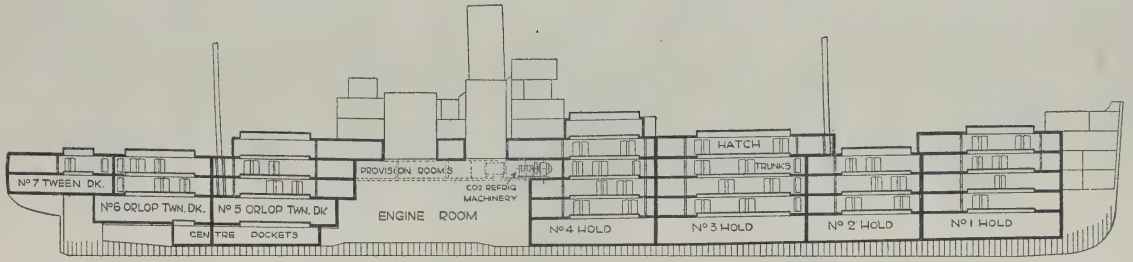


FIG. 10.

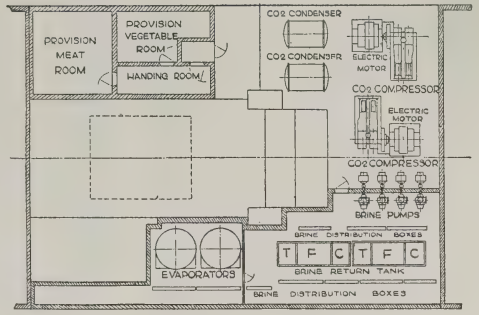


FIG. 11.

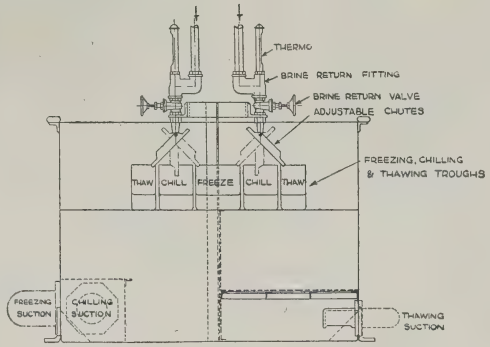


FIG. 12.

The third group includes the increasing numbers of vessels used solely for the carriage of tropical fruits, but by far the most valuable and important marine installations are those intended mainly for the carriage of chilled beef.

Some of these have an insulated capacity of upwards of 600,000 cu. ft., and a typical section of such a vessel, together with the arrangement of the refrigerating machinery supplied by *J. & E. Hall, Ltd.*, is shown in figs. 10, 11, and 12.

In these vessels both chilled and frozen meat can be carried, and the insulated spaces are cooled by piping in them through which cold brine is circulated.

The brine pipes are arranged in a number of circuits of limited length so that the return temperatures shall only show a small rise, and each of these circuits is separately controlled.

In modern practice a number of the spaces are, in many cases, also provided with air-cooling batteries and fans, so that cold air circulation can be used either to supplement the brine piping or as a complete alternative.

The method of cooling by cold air circulation is often preferred for fruit and for other cargoes that are carried at a temperature above the freezing-point.

The carriage of chilled beef calls for very precise regulation of the temperatures to be maintained in the various spaces.

This is effected by what is known as the three-temperature brine system, with mains providing for the delivery of brine at suitable temperatures for freezing, chilling, and thawing service, to enable any temperature from freezing to chilling to be maintained in the various spaces to suit the class of produce carried, and individual requirements of shippers.

The great majority of the large meat-carrying installations are arranged on the open brine circulating system, but the closed system is preferred by several important shipowners.

Both systems have certain advantages; both produce excellent results in practice, and the balance between them may be said to be a matter of individual preference.

The general principle common to both systems is that the brine returning from the piping in the chill spaces is pumped directly back to the chilling delivery header, but the heat absorbed by the brine in its passage through the chilling circuits is eliminated by means of an injection of the freezing brine into the chilling system, there being a corresponding overflow to the freezing system.

When carrying mixed cargoes, it is often an advantage to run one of the evaporators at the chilling and the other at the freezing temperature, instead of having both on the low temperature, and to use the system of brine injection, or attemperating, as it is often termed.

In the open system the brine returns into an open return tank or tanks placed in an insulated space adjoining the machine room, which also contains the headers, valves, cocks, and thermometers required to control the flow of brine.

The arrangement is shown in figs. 11 and 12, and the detailed arrangement of the piping for the same vessel in fig. 13.

An example of the closed brine system is shown diagrammatically in fig. 14, p. 1228, for a vessel fitted up by the *Liverpool Refrigeration Co., Ltd.*

Figs. 15 and 16 (p. 1229) show the form of main attemperating valve and attemperating cock of the Liverpool Refrigeration Co., Ltd.; these, as will be seen, adjust the proportion of chilled or cold brine in such a manner that the one supply is cut off in proportion to the amount the other is opened up.

The flow of brine through the circuits is sometimes controlled by flowmeters placed in them. In other cases no special arrangements are made for measuring quantities, and the control is made entirely on the thermometer readings, but each return has a bye-pass connection, so that the brine may be diverted into an open tank for testing purposes if required.

A large modern installation may involve up to about 200 separate brine circuits, and the control room for a closed brine system fitted up by the *Haslam Foundry & Engineering Company* is shown in fig. 17 (p. 1230).

The refrigerating machinery for these installations always consists of at least two units, each complete with its condenser and evaporator, and capable of independent working, each being of sufficient capacity to maintain the required temperatures in the cargo spaces at sea if the other is out of action.

In some important cases there are three machines, two of which have ample power to maintain the cargo with one independent unit in reserve.

The machines are often of the Duplex type, with a cross compound engine driving the CO_2 compressors from the tail rods, and arranged so that either side can be used independently. Large single machines are made with tandem compound steam cylinders driving one compressor direct.

There are also a few examples of triple compressor machines with three-cylinder engines, as shown in fig. 18 (p. 1231), illustrating an installation by the *Liverpool Refrigeration Co., Ltd.*

The CO_2 condensers are constructed of solid drawn copper piping contained in cast-iron casings. The evaporators are generally of solid drawn steel piping contained in built steel casings, and in modern practice both brine and water circulating pumps are of the centrifugal type direct coupled to electric motors.

The pump connections to the suction and delivery mains are made to valve boxes, permitting an interchange of duties on the respective chilling and freezing circuits, and for one pump being always available as a standby for either duty.

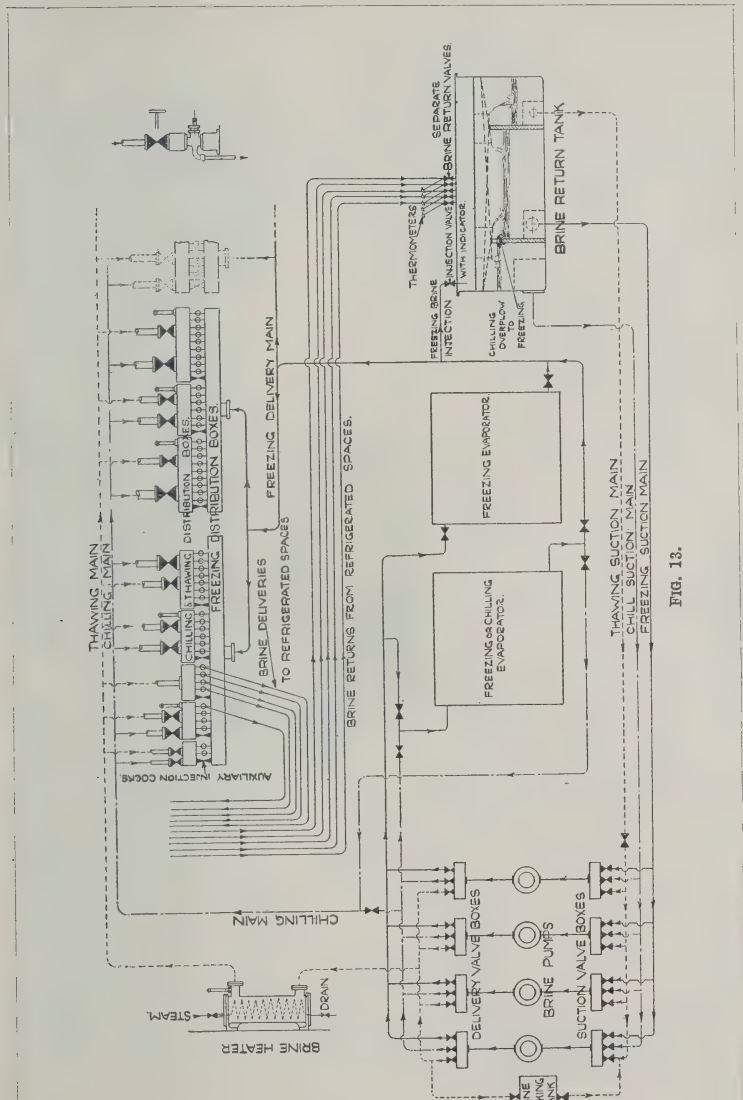

THAWING MAIN
CHILLING MAIN
DISTRIBUTION BOXES
CHILLING & THAWING
FREEZING DISTRIBUTION BOXES
AUXILIARY INJECTION COCKS
BRINE DELIVERIES TO REFRIGERATED SPACES
FREEZING DELIVERY MAIN
BRINE RETURNS FROM REFRIGERATED SPACES.
THERMOMETERS
SEPARATE BRINE RETURN VALVES
FREEZING BRINE INJECTION
INJECTION VALVE WITH INDICATOR
CHILLING OVERFLOW TO FREEZING
BRINE RETURN TANK
FREEZING EVAPORATOR
FREEZING & CHILLING EVAPORATOR.
THAWING SUCTION MAIN
CHILL. SUCTION MAIN
FREEZING SUCTION MAIN
CHILLING MAIN
STEAM
DRAIN
BRINE HEATER
DELIVERY VALVE BOXES
BRINE PUMPS
SUCTION VALVE BOXES
BRINE MAKING TANK

FIG. 13.

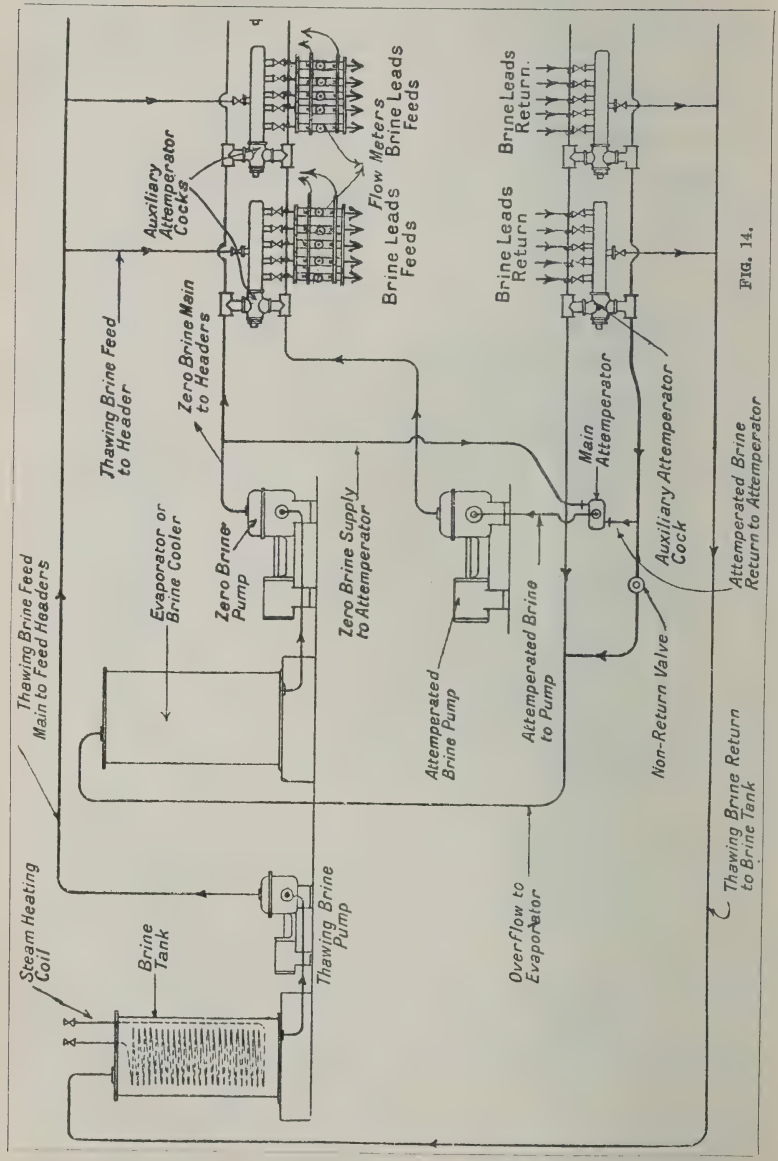

Steam Heating Coil
Brine Tank
Thawing Brine Pump
Thawing Brine Feed Main to Feed Headers
Thawing Brine Feed to Header
Evaporator or Brine Cooler
Zero Brine Pump
Zero Brine Main to Headers
Zero Brine Supply to Attemperator
Auxiliary Attemperator Cocks
Flow Meters
Brine Leads Feeds
Brine Leads Feeds
Brine Leads Return
Brine Leads Return
Attemperated Brine Pump
Attemperated Brine to Pump
Main Attemperator
Auxiliary Attemperator Cock
Non-Return Valve
Attemperated Brine Return to Attemperator
Overflow to Evaporator
Thawing Brine Return to Brine Tank

FIG. 14.

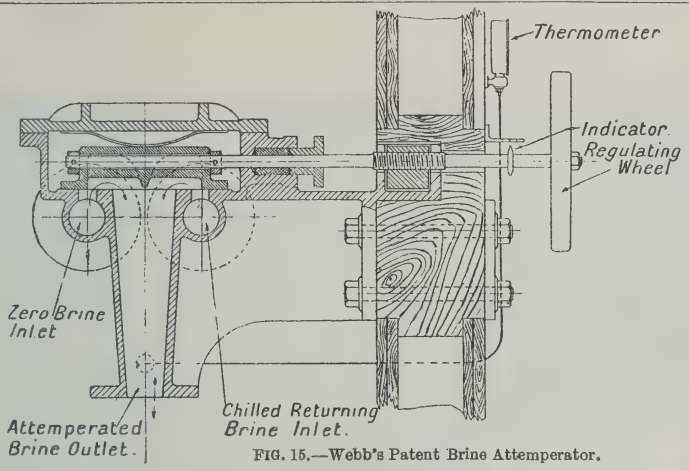


FIG. 15.—Webb's Patent Brine Attemperator.

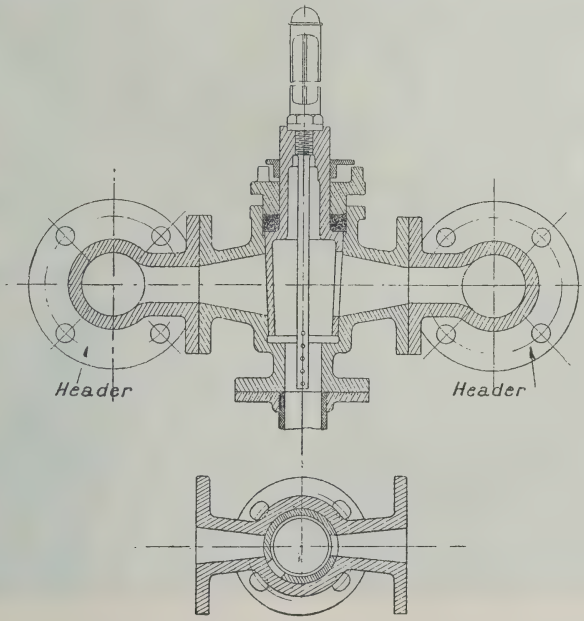


FIG. 16.—Webb's Patent Attemperator Cock (Section).

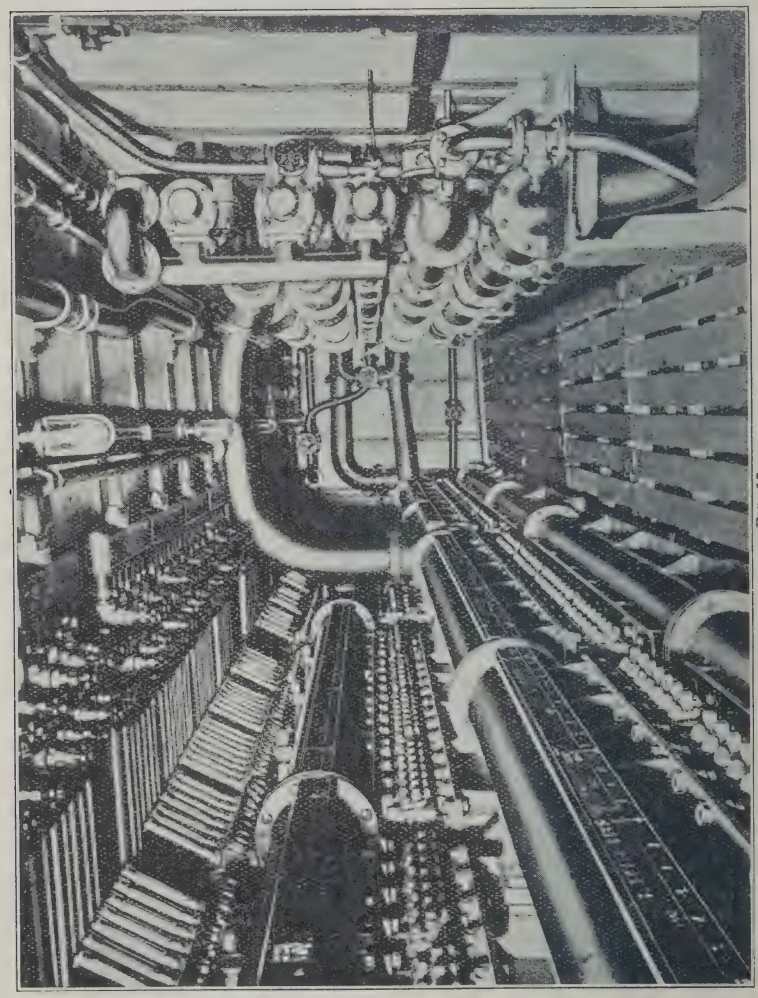


FIG. 17.

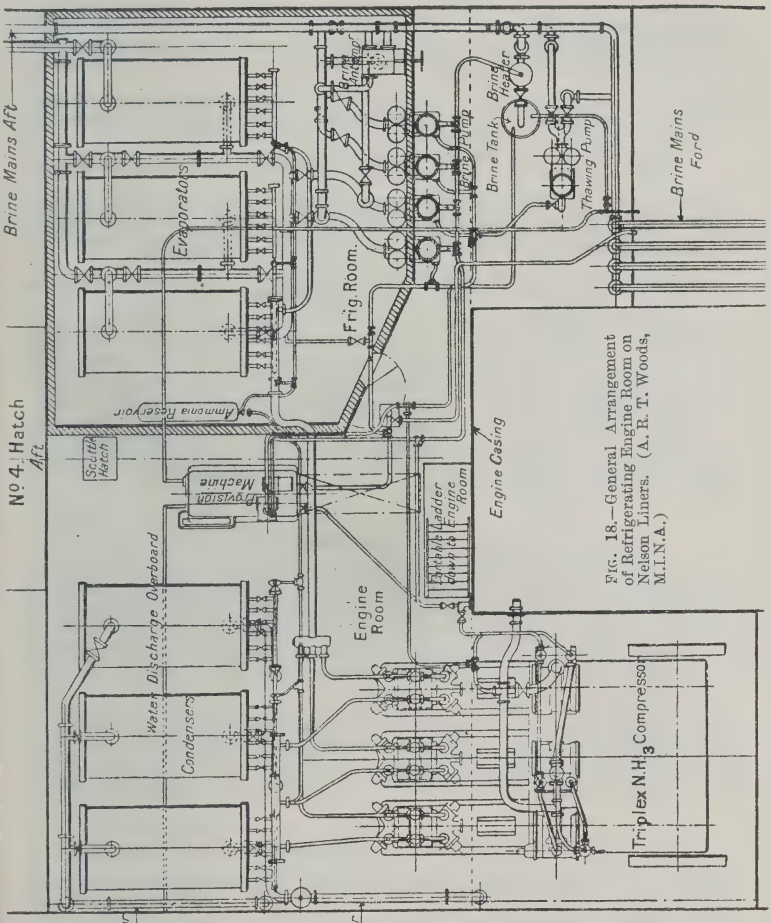


FIG. 18.—General Arrangement of Refrigerating Engine Room on Nelson Liners. (A. R. T. Woods, M.I.N.A.)

Water Coolers.

The 'Heenan' Patent Water Cooler.

The latest type of cooler recently introduced by Messrs. Heenan & Froude, Ltd., Worcester, claims a minimum consumption of water per brake horse-power-hour, and has the advantage of being compact and occupying comparatively small space. The apparatus is also self-contained. It is of the stationary type, and has no moving parts beyond the fan which circulates the air, and the pump which circulates the water.

The apparatus (as shown by fig. 19) consists of an outer casing, which is rectangular in shape, built up of mild steel plates. The base of this casing forms a tank into which the water drains after cooling, and previous to re-circulation.

The cooling effect is obtained by evenly distributing the heated water over a number of screens set up one above the other in the body of the cooler.

The water to be cooled, after passing through a strainer, enters at the top of the cooler and falls on to the screens through nozzles which feed perforated troughs. In its fall through the screens it meets a current of cold air which is forced upward through the casing by a centrifugal fan, the discharge from which is introduced to the underside of the lowest screen.

The screens are formed of phosphor-bronze metallic wool of standard size and thickness, held together between wire netting which is attached to angle-steel frames.

The cooling effect is positive and continuous, the cooling surface being always efficient, and the volume of air being a predetermined quantity which is not dependent on climatic conditions. By suitably proportioning the size of the machine for the duty which it has to undertake, water can be cooled down to a temperature closely approaching that of the prevailing wet bulb thermometer. Thus in hot weather—in temperate as well as in tropical climates—re-cooled water can be provided at a temperature considerably below the dry bulb temperature of the surrounding atmosphere. This feature renders the machine especially suitable for dealing with the circulating water of refrigerating plants, for which purpose large numbers of 'Heenan' coolers have been supplied. The practical value of the cooler is due not only to the saving in water but to the improved efficiency of the refrigerating plant, resulting in lower compression pressures and reduced power consumption.

'Pulsometer' Refrigerating Plants.

The Pulsometer Engineering Co., Ltd., are manufacturers of apparatus for working on both the NH_3 (Ammonia), and CO_2 (Carbon Dioxide) compression systems, and their machines are designed and manufactured on the principle that high efficiency as well as reliable and satisfactory service with a minimum amount of attention is of first importance, and apart from the large number of plants of various capacities they supply for land service both in this country and most countries abroad, they are large suppliers of marine type plants for ordinary ships' stores cooling and icemaking, and also for magazine cooling, refrigerating, and icemaking on board war vessels.

A considerable number of cargo and passenger boats have been equipped with this type, and the manufacturers have recently developed an improved design in the form of an enclosed type quick running splash lubrication machine which is particularly suited for both land and marine use. Though marked by many up-to-date improvements, it retains most of the features for which the early designs were so well known, i.e. simplicity and ease of operation with the minimum amount of attention, freedom from breakdown, and low cost of maintenance.

Negretti & Zambra's Low Temperature Electrical Distance Thermometer.

This Thermometer (which depends upon the variation of an electrical resistance by change of temperature) is specially designed for cold-storage work suitable for dairies, butter factories, and meat storage.

The advantage of its installation is that it is possible to read in a central position the temperatures of a number of different points situated some distance away, i.e. the temperatures of cold chambers can be read in the engine room or the manager's office, which is a great advantage and assists the economical working of the plant.

The elements or thermometer bulbs are placed in the cold chambers, brine pipes, etc., and connected up to the indicator by means of lead-covered wire, the electric current being provided by an accumulator. These elements vary according to the purpose for which they are required, either for air temperatures, or for brine or water temperatures.

THE 'HEENAN' PATENT COOLER. TYPE 'V.'

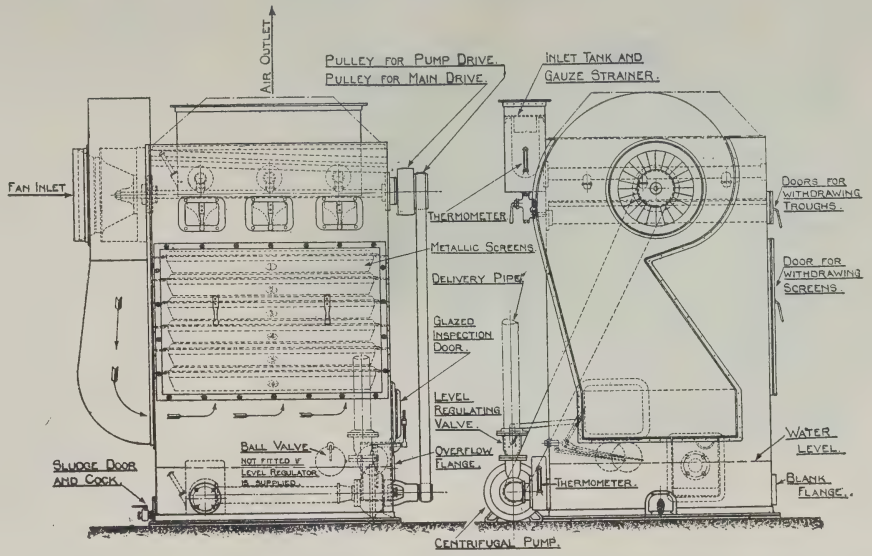


FIG. 19.

The indicator (fig. 20) is mounted in a cast-iron moisture-proof case sprayed and stoved white, the scale being graduated to cover the temperatures required, such as cold stores – 20 to + 100° F., etc. There is a multiple point switchboard incorporated in the case, by means of which the temperature of any point can be read on the indicator as desired.

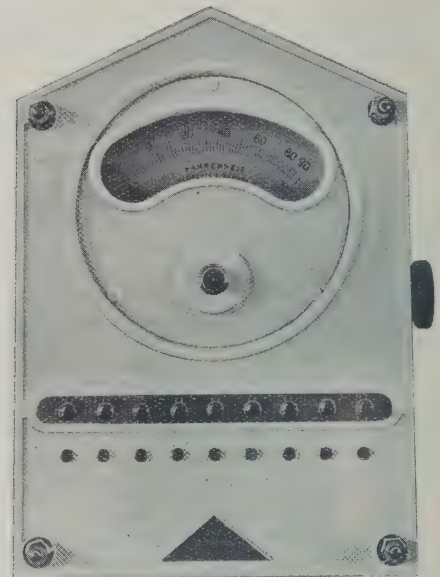


FIG. 20.

All the connections are enclosed in the case, so that they are adequately protected from moisture, dust, etc.

The connecting wire used is a 3-core cable, lead-covered for protection.

LLOYD'S RULES FOR THE SURVEY OF REFRIGERATING MACHINERY AND APPLIANCES OF FROZEN MEAT STORES.

(Abstract.)

Power of Machinery.—The refrigerating machinery is to be of approved construction and of sufficient power to maintain a temperature of 15° F. simultaneously in all insulated chambers when running 18 hours per day.

Motive Power.—Each compressor is to be driven by independent means, or, alternatively, the motive power is to be in duplicate.

Where the compressors are electrically driven, a spare electric generator is to be fitted, unless the power is supplied by a public generating station.

Where electric transformers are installed on the premises, a spare transformer is to be fitted.

Hydraulic and other Tests.—In the case of new refrigerating machinery and appliances, the tests recommended for the various parts are as follows, viz.:

(*a*) CO_2 compressors, separators, condenser and evaporator coils, headers and connections to be tested by hydraulic pressure to 3,000 lbs. per sq. in., and afterwards by air pressure to 1,500 lbs. per sq. in., whilst submerged in water.

(*b*) NH_3 compressors and cast iron or steel connections to be tested by hydraulic pressure to 600 lbs. per sq. in. Where compound compressors are fitted, the low pressure compressor and cast-iron or steel connections to be tested by hydraulic pressure to 200 lbs. per sq. in.

(*c*) NH_3 condenser, evaporator and air cooler coils to be tested by hydraulic pressure to 1,500 lbs. per sq. in., and afterwards by air pressure to 500 lbs. per sq. in., preferably whilst submerged in water.

(*d*) NH_3 plant, after erection in place, to be tested by air pressure; the gas delivery side (*i.e.* from the compressor to the liquid expansion valves) to 200 lbs., and the return side, including the direct expansion piping for the chambers (*i.e.* from the liquid expansion valves to the compressor suction valves) to 150 lbs. per sq. in.

(*e*) Brine piping, after erection in place, to be tested by air or water pressure to 90 lbs. per sq. in.

(*f*) Gas evaporator casings of cast iron and gas condenser casings, whether of cast iron or steel, where of enclosed type, to be tested by hydraulic pressure to 15 lbs. per sq. in., or to double the working pressure, whichever is the greater.

(*g*) Gas evaporator wrought-iron and steel casings, where of enclosed type, to be tested by hydraulic pressure to 30 lbs., if gravitation type, or to 50 lbs. per sq. in. if pressure type, or to double the working pressure, whichever is the greater.

Cooling Tests.—(*a*) Upon completion of the installation under special survey, the refrigerating machinery is to be tested under working conditions, and the insulated chambers are to be cooled down simultaneously to about 15° F.

(*b*) The whole of the refrigerating power may be used in order to reduce the temperature, but the specified temperature is to be maintained in the insulated chambers for a period of 12 hours with the spare compressor unit out of action.

Brine Cooling System.—(*a*) Where the brine system of refrigeration is employed the cooling pipes in each refrigerated chamber of over 5,000 cu. ft. capacity are to be arranged in not less than two sections, and the cooling surface is to be sufficient to maintain the working temperature of the chamber when one of the sections is out of use. Isolating valves or cocks are to be fitted to each section of cooling pipes.

(*b*) The brine circulating pipes and brine tanks should not be galvanised on the inside.

(*c*) In cases where internally galvanised tanks or cooling pipes are fitted, the brine cooling and return tanks, if of closed type, are each to be provided with a ventilating pipe or pipes led to the open air in a situation where no danger will be incurred from the issuing gas, and each ventilating pipe is to be fitted with a wire gauze diaphragm which can easily be removed for renewal.

(*d*) Where the tanks are not closed the compartments in which they are situated are to be efficiently ventilated.

Direct Expansion System.—Where the direct expansion system of refrigeration is employed, the cooling pipes in each refrigerated chamber of over 5,000 cu. ft. capacity are to be arranged in not less than two sections, and the cooling surface is to be sufficient to maintain the working temperature of the chamber when one of the sections is out of use. An expansion valve and a return gas valve or cock are to be fitted to each section of cooling pipes.

SECTION XXV

PART I

HYDRAULICS — GENERAL DATA OF WATER — WATER PRESSURE — HORSE POWER REQUIRED TO PUMP WATER — PRESSURE IN HYDRAULIC MAINS — DIMENSIONS AND STRENGTH OF HYDRAULIC PIPES AND CYLINDERS — PIPE UNIONS AND JOINTS (pp. 1239-1248)

PART II

PUMPS - STEAM PUMPS — PUMP PACKING — CENTRIFUGAL PUMPS—TURBINE PUMPS—AIR LIFT PUMPS (pp. 1249-1281)

PART III

THE DEVELOPMENT OF WATER-POWER

Partly Contributed by Arthur Surveyer, D.Eng., M.E.I.C., M.Inst.C.E.)
(pp. 1283-1307)

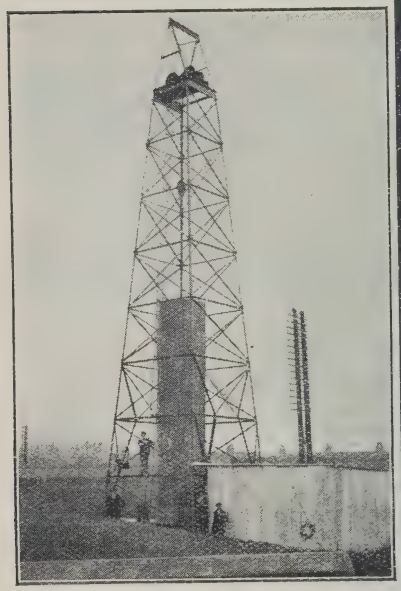
PART IV

WATER PRESSURE ENGINES—PELTON WHEELS—TURBINES
(pp. 1309-1348)

PART V

HYDROSTATIC PRESS—HYDRAULIC RAMS—HYDRAULIC LIFTS —HYDRAULIC BOLT FORCERS—HYDRAULIC RIVETERS
(pp. 1349-1356)

• JUST A REMINDER THAT



IN EVERYTHING CONNECTED WITH UNDERGROUND WATER SUPPLIES FREE ADVICE AND ESTIMATES ARE OBTAINABLE FROM

LE GRAND SUTCLIFF & GELL, LTD.

SOUTHALL LONDON, W.

'Grams: Legrand, Southall 'Phones: Southall 2211, 4 lines GTII

F. 1239

SECTION XXV

PART I

HYDRAULICS—GENERAL DATA OF WATER—WATER PRESSURE—HORSE POWER REQUIRED TO PUMP WATER—PRESSURE IN HYDRAULIC MAINS—DIMENSIONS AND STRENGTH OF HYDRAULIC PIPES AND CYLINDERS—PIPE UNIONS AND JOINTS.

General Data Regarding Water.

An imperial gallon of water at 62° F. weighs 10 lbs. avoirdupois. (See also page 124.)

Gallons × 277·420 = cu. ins. (*International Bureau of Standards, Paris*, 1921.)

Gallons × 0·16055 = cu. ft.
Cu. ft. × 6·2288 = gallons.
Cu. in. × 0·0036047 = gallons.
Cu. ft. × 62·288 = lbs. weight.
Lbs. × 0·016053 = cu. ft.

Gallons × 0·004464 = tons.
Tons × 224 = gallons.
Cu. ft. × 0·027807 = tons.
Tons × 35·962 = cu. ft.

A pipe one yard long holds approximately (actually, 1·52 per cent. less) as many lbs. of water as the square of its diameter in inches; thus, a 6-inch pipe holds approximately 36 lbs. of water in each yard length.

VISCOSITY (ν)* OF WATER IN C.G.S. UNITS AT VARIOUS TEMPERATURES.

(Calculated from data in *The Philosophical Transactions of the Royal Society*, A., Vol. 214.—*H. R. K.*)

C.°	8·6	10·2	10·5	10·7	11·1	11·3	11·5	14·3	15·0	16·0	20·5	23·0	25·0
F.°	47·5	50·4	50·9	51·3	51·9	52·3	52·7	57·7	59·0	60·8	68·9	73·4	77·0
ν	0·0136	0·0131	0·0130	0·0129	0·0128	0·0127	0·0126	0·0115	0·0114	0·0110	0·0100	0·0093	0·0090

(Approximately the viscosity at 69° F. is 0·01. The viscosity varies approximately 2·06 per cent. for each C.°, or 1·2 per cent. for each F.°

Water-Pressure.

If

H = head of water in feet; P = lbs. pressure per square inch,

$$H = P \times 2{\cdot}311. \qquad P = H \times {\cdot}4326.$$

10 feet head of water gives a pressure of 4½ lbs. per square inch, approximately,

27·71 inches of water gives a pressure of 1 lb. per square inch.

* See note p. 994.

TABLE SHOWING WATER PRESSURE CORRESPONDING TO VARIOUS FEET HEAD.

Feet Head.	Pressure in Lbs. per Square Inch.	Feet Head.	Pressure in Lbs. per Square Inch.	Feet Head.	Pressure in Lbs. per Square Inch.	Feet Head.	Pressure in Lbs. per Square Inch.	Feet Head.	Pressure in Lbs. per Square Inch.	Feet Head.	Pressure in Lbs. per Square Inch.
1	·43	21	9·08	41	17·8	61	26·4	81	35·0	110	47·6
2	·87	22	9·52	42	18·2	62	26·8	82	35·5	120	51·9
3	1·30	23	9·95	43	18·6	63	27·3	83	35·9	130	56·2
4	1·73	24	10·4	44	19·0	64	27·7	84	36·3	140	60·6
5	2·16	25	10·8	45	19·5	65	28·1	85	36·8	150	64·9
6	2·60	26	11·3	46	19·9	66	28·6	86	37·2	160	69·2
7	3·03	27	11·7	47	20·4	67	29·0	87	37·6	170	73·5
8	3·46	28	12·1	48	20·8	68	29·4	88	38·1	180	77·9
9	3·89	29	12·5	49	21·2	69	29·8	89	38·5	190	82·2
10	4·33	30	13·0	50	21·6	70	30·3	90	38·9	200	86·5
11	4·76	31	13·4	51	22·1	71	30·7	91	39·4	210	90·8
12	5·19	32	13·8	52	22·5	72	31·1	92	39·8	220	95·2
13	5·62	33	14·3	53	22·9	73	31·6	93	40·2	230	99·5
14	6·06	34	**14·7** *	54	23·4	74	32·0	94	40·7	240	103·8
15	6·49	35	15·2	55	23·8	75	32·4	95	41·1	250	108·1
16	6·92	36	15·6	56	24·2	76	32·9	96	41·5	260	112·2
17	7·35	37	16·0	57	24·7	77	33·3	97	42·0	270	116·8
18	7·79	38	16·4	58	25·1	78	33·7	98	42 4	280	121·1
19	8·22	39	16·9	59	25·5	79	34·2	99	42·8	290	125·5
20	8·65	40	17·3	60	26·0	80	34·6	100	43·3	300	129·8

* Atmospheric pressure.

TABLE SHOWING THEORETICAL HORSE-POWER REQUIRED TO RAISE 1,000 GALLONS (160·6 cubic feet) OF WATER PER HOUR TO VARIOUS HEIGHTS.

Height in Feet.	Horse-power.	Height in Feet.	Horse-power.	Height in Feet.	Horse-power.	Height in Feet.	Horse-power.	Height in Feet.	Horse-power.	Height in Feet.	Horse-power.
1	·005	21	·106	41	·207	61	·308	81	·409	110	·556
2	·010	22	·111	42	·212	62	·313	82	·414	120	·606
3	·015	23	·116	43	·217	63	·318	83	·419	130	·657
4	·020	24	·121	44	·222	64	·323	84	·424	140	·707
5	·025	25	·126	45	·227	65	·328	85	·429	150	·758
6	·030	26	·131	46	·232	66	·333	86	·434	160	·808
7	·035	27	·136	47	·237	67	·338	87	·439	170	·859
8	·040	28	·141	48	·242	68	·343	88	·444	180	·910
9	·045	29	·146	49	·247	69	·348	89	·449	190	·960
10	·051	30	·152	50	·253	70	·354	90	·455	200	1·010
11	·056	31	·157	51	·258	71	·359	91	·460	210	1·061
12	·061	32	·162	52	·263	72	·364	92	·464	220	1·111
13	·066	33	·167	53	·268	73	·369	93	·470	230	1·162
14	·071	34	·172	54	·273	74	·374	94	·475	240	1·212
15	·076	35	·177	55	·278	75	·379	95	·480	250	1·263
16	·081	36	·182	56	·283	76	·384	96	·485	260	1·313
17	·086	37	·187	57	·288	77	·389	97	·490	270	1·364
18	·091	38	·192	58	·293	78	·394	98	·495	280	1·414
19	·096	49	·197	59	·298	79	·399	99	·500	290	1·465
20	·101	40	·202	60	·303	80	·404	100	·505	300	1·515

In practice about 33 per cent. more power than the theoretical may be required.

TABLE SHOWING HORSE-POWER REQUIRED TO PUMP WATER TO GIVEN VERTICAL HEIGHTS.
(Calculated at 33 per cent. above theoretical value.)

Gallons per Hour.	Height in Feet.											
	20	30	40	50	60	70	80	90	100	200	300	400
	Horse-power.											
300	·0403	·0605	·0806	·1008	·1209	·1411	·1612	·1814	·2015	·4031	·6046	·8062
350	·0470	·0705	·0941	·1176	·1411	·1646	·1881	·2116	·2351	·4703	·7054	·9405
400	·0537	·0806	·1075	·1344	·1612	·1881	·2150	·2418	·2687	·5374	·8062	1·075
450	·0605	·0907	·1209	·1512	·1814	·2116	2418	·2721	·3023	·6046	·9069	1·209
500	·0672	·1008	·1344	·1680	·2015	·2351	·2687	·3023	·3359	·6718	1·008	1·344
550	·0739	·1108	·1478	·1847	·2217	·2586	·2956	·3325	·3695	·7390	1·108	1·478
600	·0806	·1209	·1612	·2150	·2418	·2822	·3225	·3628	·4031	·8062	1·209	1·612
650	·0873	·1310	·1747	·2183	·2620	3057	·3493	·3930	·4367	·8733	1·310	1·747
700	·0940	·1411	·1881	·2351	·2822	·3292	·3762	·4232	·4703	·9405	1·411	1·881
750	·1008	·1512	·2015	·2520	·3023	·3527	·4031	·4535	5039	1·008	1·512	2·015
800	·1075	·1612	·2150	·2687	·3225	·3762	·4300	·4837	·5374	1·075	1·612	2·150
850	·1142	·1713	·2284	·2855	·3426	·3997	·4568	·5139	·5710	1·142	1·713	2·284
900	·1209	·1814	·2418	·3023	·3628	·4232	·4837	·5442	·6046	1·209	1·814	2·418
950	·1276	·1915	·2503	·3191	·3829	·4467	·5106	·5744	·6382	1·276	1·915	2·503
1,000	·1344	·2015	·2687	·3359	·4031	·4703	·5374	·6046	·6718	1·344	2·015	2·687
2,000	·2687	·4031	·5374	·6718	·8062	·9405	1·075	1·209	1·344	2·687	4·031	5·374
3,000	·4031	·6046	·8062	1·008	1·209	1·411	1·612	1·814	2·015	4·031	6·046	8·062
4,000	·5374	·8062	1·075	1·344	1·612	1·881	2·150	2·418	2·687	5·374	8·062	10·75
5,000	·6718	1·008	1·344	1·680	2·015	2·351	2·687	3·023	3·360	6·718	10·08	13·44
6,000	·8062	1·209	1·612	2·015	2·418	2·822	3·225	3·628	4·031	8·062	12·09	16·12
7,000	·9405	1·411	1·881	2·351	2·822	3·292	3·762	4·232	4·703	9·405	14·11	18·81
8,000	1·075	1·612	2·150	2·687	3·225	3·762	4·300	4·837	5·374	10·75	16·12	21·50
9,000	1·209	1·814	2·418	3·023	3·628	4·232	4·837	5·442	6·046	12·09	18·14	24 18
10,000	1·344	2·015	2·687	3·359	4·031	4·703	5·374	6 046	6·718	13·44	20·15	26·87
20,000	2·687	4·031	5·374	6·718	8·062	9·405	10·75	12·05	13·44	26·87	40·31	53·74
30,000	4·031	6·046	8·062	10·08	12·09	14·11	16·12	18·14	20·15	40·31	60·46	80·62
40,000	5·374	7·062	10·75	13·44	14·12	18·81	21·50	24·18	26·87	53·74	70·62	107·5
50,000	6·718	10·08	13·44	16·80	20·15	23·51	26·87	30·23	33·59	67·18	100·8	134·4
60,000	8·062	12·09	16·12	20·15	24·18	28·22	32·25	36·28	40·31	80·62	120·9	161·2
70,000	9·405	14·11	18·81	23·5	28·22	32·92	37·62	42·32	47·03	94·05	141·1	188 1

(H. R. Kempe.)

Horse-power required to Pump Water.

1 H.P. indicated will raise about 3,000 gallons of water per hour 50 feet high.

To pump 1 gallon of water per minute against a pressure of p lbs. per square inch requires $p \times \cdot0007$ H.P.

Allowing 40 per cent. for friction, the following shows the H.P. required to pump 1 gallon of water per minute at various pressures :—

Pressure per square inch, in lbs.	500	700	1000	1200	1500	2000	2250	2500	5000
Horse-power	·49	·69	·98	1·18	1·47	1·96	2·21	2·45	4·90

The *cost of lifting* 1 ton 50 feet is, in the case of the London Hydraulic Power Company, sometimes as low as $\frac{1}{4}d$.

Speed of Water should not, as a rule, exceed 100 feet per minute with 700 lbs. pressure. In the mains of the London Hydraulic Power Company the delivery is 1,040 gallons per minute, at a velocity averaging 2·83 feet per second, or 170 feet per minute. The loss of head due to this velocity is 22·896 feet per 1,000 yards, by the formula :—

$$\frac{(\text{Gallons per minute})^2 \times (\text{length of pipe in yards})}{(3 \times \text{diameter of pipe in inches})^5}.$$

The most distant point of the main is 5,320 yards, or just over 3 miles, from the accumulators. In a circuit of 5 miles the normal difference of pressure, or loss of head, was 20 feet head.

Pressure in Hydraulic Mains.

The average working pressure is 700 lbs. per square inch with accumulators, but varies from 300 to 1,500 lbs. per square inch. The pressure in the mains of the London Hydraulic Power Company averages 700 lbs., the actual pressure at the accumulators being 750 lbs.

700 lbs. per square inch = 549·78 lbs. per circular inch, equivalent to a head of 1,618 feet.

EFFECTIVE PRESSURE WITH 700 LBS. AT ACCUMULATOR.

Ratio of multiplying power . .	2 to 1	4 to 1	6 to 1	8 to 1	10 to 1	12 to 1	14 to 1	16 to 1
Multiply by . .	·8	·6	·72	·67	·63	·59	·54	·5
Effective pressure in lbs. . . .	560	532	504	469	441	413	378	350

Dimensions of Hydraulic Pipes and Cylinders.

TABLE OF THICKNESS OF HYDRAULIC PIPES.

Description.	Wrought Iron, to stand 4,000 Lbs. per Square Inch.									Cast Iron, to stand 700 Lbs. per Square Inch.					
Internal diameter, in inches .	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{7}{8}$	2	$2\frac{1}{4}$	$2\frac{1}{2}$	3	3
External diameter in inches .	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{7}{8}$	2	$2\frac{1}{8}$	$2\frac{1}{4}$	$2\frac{3}{8}$	$3\frac{1}{4}$	$3\frac{9}{16}$	$3\frac{11}{16}$	$4\frac{1}{4}$	$5\frac{1}{4}$
Number of threads per inch . .	14	14	14	11	11	11	11	11	11	11	11							

(1)

The external diameter, D, may be calculated from the formula :—

$$D = d\left(\frac{p}{f} + 1\right),$$

where,

d = diameter of bore of pipe, in inches ; p = pressure, in lbs., per square inch ; f = 4,000 for wrought iron, and 1,200 for cast iron.

(2)

$$T = \frac{(P + 100)\,D}{0{\cdot}4\,S} + 0{\cdot}333\left(1 - \frac{D}{100}\right),$$

where,

T = thickness of pipe in ins.; D = interior diameter of pipe in ins.; P = pressure of water in lbs. per sq. in.; S = ultimate tensile strength of cast-iron in lbs. per sq. in. (taken as 18,000 lbs.).

This formula gives the weight and thickness for good, tough, and elastic cast-iron pipes covering the ordinary range of static pressures of public water supplies.

TABLE CALCULATED FROM FOREGOING FORMULA.

Head of water in ft. . .	115	150	200	250	300
Pressure in lbs. . . .	50	65	87	109	130
Diameter in Ins.	Thickness in Ins.				
4	—	—	—	0·40	0·45
6	—	—	—	0·46	0·50
8	—	—	—	0·52	0·55
10	—	—	—	0·60	0·63
12	—	0·57	0·61	0·65	0·69
14	—	0·61	0·65	0·70	0·75
16	—	0·65	0·70	0·75	0·81
20	—	0·73	0·79	0·85	0·92
24	—	0·80	0·87	0·95	1·03
30	—	0·92	1·00	1·10	1·20
36	0·93	1·03	1·13	1·25	1·36
42	1·01	1·14	1·27	1·40	—
48	1·15	1·25	1·40	1·55	—
54	1·23	1·35	1·53	—	—
60	1·35	1·50	1·70	—	—

(*R. D. Wood & Co., America.*)

(3)

$$T = \frac{(P + L)\,R}{3{,}300} + 0{\cdot}25.$$

where

T = thickness of pipe in ins.; P = static pressure in lbs. per sq. in.; R = inside radius of pipe in ins.; L = pressure in lbs. allowed for water hammer; 0·25 = a constant allowance for inequality in manufacture, deterioration, and rough handling; 3,300 = one-fifth of the tensile strength of cast-iron, which is taken to be 16,500.

The values of L—the allowance for water hammer—for different sizes of pipe are as follows, the allowance decreasing as the diameter and thickness of pipe increase :—

Diameter of Pipe, Ins.	Value of 'L,' Lbs.	Corresponding Head, Ft.
3, 4, 6, 8, or 10	120	277
12 or 14	110	254
16 or 18	100	231
20	90	208
24	85	196
30	80	185
36	75	173
42, 48, 54 or 60	70	162

For a static pressure of 100 lbs. per sq. in. a 6 in. pipe need be only 0·10 in. thick. Such thin pipe, however, could not well be made or handled, and allowance must be made (as above) for shock due to water hammer, for roughness in handling, for inequality in manufacture, and for deterioration. (*D. Fitzgerald, America.*)

Thickness of Hydraulic Cylinders.

If T = thickness of metal in inches, P = bursting pressure in lbs. per square inch, R = radius of cylinder in inches, and C = constant (16,000 for cast iron, 34,000 for brass, 50,000 for wrought iron, and 70,000 for steel),

$$T = \frac{PR}{C - P}.$$

If D = external diameter of cylinder in inches, and d = internal diameter in inches, P and C being as above, then

$$D = d\sqrt{\frac{C + P}{C - P}}.$$

Strength of Hydraulic Cylinders.

If f = allowable stress in lbs. per sq. in.; P = working pressure in lbs. per sq. in.; d = inside diameter in ins.; D = outside diameter in ins.; then

$$D = d \times \text{anti-log}\left(\frac{P}{p} \cdot 4343\right). \qquad \textit{(Clark.)}$$

EXAMPLE.—If P = 3,000 lbs. per sq. in.; f = 8,000 lbs. per sq. in.; d = 30 ins.; then

$$\frac{P}{p} \cdot 4343 = \frac{3,000}{8,000} \cdot 4343 = \cdot 16286$$

anti-log.* of ·16286 = 1·455 (page 99);

therefore $D = 30 \times 1 \cdot 455 = 43 \cdot 65$ ins.

Size of Hydraulic Mains.

The largest mains used for hydraulic purposes are 7½ inches in diameter. The working stress on the metal due to working pressure is 2,800 lbs. per square inch. The mains are usually tested to a water-pressure of 2,500 lbs. per square inch before laying, and to a pressure of 800 to 1,600 lbs. per square inch after laying.

Pipe Unions and Joints.

For wrought-iron pipes up to 2 inches in diameter, such as are used for the connecting-pipes to hydraulic machines, the unions are generally of brass, and of the form shown by fig. 1. For

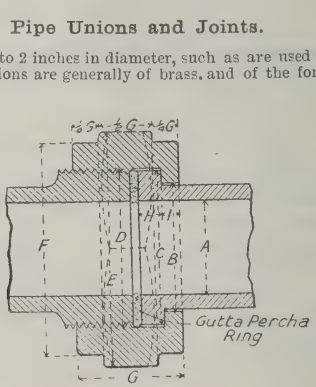


FIG. 1.

main cast-iron pipes the unions are made by means of elliptical flanges (fig. 1) formed on the ends of the pipes, the inner faces of the flanges being turned so that one pipe can enter a little distance into the other; a recess is left between them, to receive a gutta-percha ring, whose outside diameter is about 4 times the inner.

* Number corresponding to logarithm.

DIMENSIONS OF PRESSURE UNIONS.

700 Lbs. per Square Inch.									1,500 Lbs. per Square Inch.								
Dimensions of Parts.									Dimensions of Parts.								
A	B	C	D	E	F	G	H	I	A	B	C	D	E	F	G	H	I
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	In.	In.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	In.	In.
$\frac{1}{2}$	$\frac{5}{8}$	$1\frac{1}{8}$	$1\frac{5}{16}b$	$1\frac{43}{64}$	$1\frac{5}{8}$	1	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$1\frac{1}{8}$	$1\frac{5}{16}b$	$1\frac{55}{64}$	$1\frac{3}{4}$	$1\frac{1}{8}$	$\frac{1}{4}$	$\frac{1}{4}$
$\frac{3}{4}$	$1\frac{1}{8}$	$1\frac{7}{16}$	$1\frac{7}{8}f$	$2\frac{3}{64}$	2	$1\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{3}{4}$	$1\frac{1}{8}$	$1\frac{7}{16}$	$1\frac{7}{8}f$	$2\frac{3}{64}$	$2\frac{1}{8}$	$1\frac{1}{8}$	$\frac{1}{4}$	$\frac{1}{4}$
1	$1\frac{1}{8}$	$1\frac{11}{16}$	$1\frac{7}{8}f$	$2\frac{13}{32}$	$2\frac{5}{16}$	$1\frac{1}{8}$	$\frac{1}{4}$	$\frac{1}{4}$	1	$1\frac{1}{8}$	$1\frac{11}{16}$	$1\frac{7}{8}f$	$2\frac{3}{64}$	$2\frac{7}{16}$	$1\frac{1}{8}$	$\frac{5}{16}$	$\frac{5}{16}$
$1\frac{1}{4}$	$1\frac{11}{16}$	$1\frac{15}{16}$	$2\frac{1}{16}b$	$2\frac{5}{64}$	$2\frac{5}{16}$	$1\frac{1}{8}$	$\frac{1}{4}$	$\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{11}{16}$	$2\frac{1}{16}$	$2\frac{1}{16}b$	$3\frac{1}{16}$	$2\frac{15}{16}$	$1\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{16}$
$1\frac{1}{2}$	$1\frac{15}{16}$	$2\frac{3}{16}$	$2\frac{5}{8}b$	$3\frac{3}{16}$	$2\frac{7}{8}$	$1\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{16}$	$1\frac{1}{2}$	$1\frac{15}{16}$	$2\frac{3}{16}$	$2\frac{3}{8}$	$3\frac{11}{32}$	$3\frac{1}{8}$	2	$\frac{3}{8}$	$\frac{3}{8}$
$1\frac{3}{4}$	$2\frac{3}{16}$	$2\frac{7}{16}$	$2\frac{9}{16}f$	$3\frac{11}{32}$	$3\frac{1}{8}$	2	$\frac{5}{16}$	$\frac{5}{16}$	$1\frac{3}{4}$	$2\frac{3}{16}$	$2\frac{7}{16}$	$2\frac{9}{16}f$	$3\frac{25}{64}$	$3\frac{3}{8}$	$2\frac{1}{4}$	$\frac{3}{8}$	$\frac{3}{8}$
2	$2\frac{7}{16}$	$2\frac{13}{16}$	$3f$	$3\frac{67}{64}$	$3\frac{5}{8}$	$2\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{16}$	2	$2\frac{7}{16}$	$2\frac{13}{16}$	$3f$	$4\frac{3}{64}$	$3\frac{7}{8}$	$2\frac{1}{2}$	$\frac{7}{16}$	$\frac{7}{16}$

b = bare. f = full.

DIMENSIONS OF CAST-IRON PRESSURE FLANGES, 700 LBS.

Dimensions of Parts.								Dimensions of Parts.							
A	B	C	D	E	F	G	H	A	B	C	D	E	F	G	H
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
$2\frac{1}{2}$	$3\frac{5}{8}$	$5\frac{1}{2}$	$8\frac{1}{4}$	$4\frac{1}{2}$	$1\frac{3}{8}$	1	$\frac{5}{8}$	4	$5\frac{1}{2}$	$7\frac{5}{8}$	$10\frac{3}{4}$	7	$1\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{5}{8}$
3	$4\frac{1}{8}$	$6\frac{1}{4}$	$9\frac{1}{4}$	$5\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{1}{8}$	1	$4\frac{1}{2}$	$6\frac{3}{8}$	$8\frac{5}{8}$	12	$7\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{5}{8}$	$1\frac{1}{4}$
$3\frac{1}{2}$	$5\frac{1}{8}$	$7\frac{1}{4}$	$10\frac{1}{4}$	$6\frac{3}{8}$	$1\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{1}{8}$	5	7	$9\frac{1}{2}$	13	$8\frac{1}{4}$	2	$1\frac{5}{8}$	$1\frac{1}{4}$

Mains fitted with flanged joints of the description shown by fig. 2 can be made absolutely tight and free from leakage. The flanges are placed in a horizontal position in laying.

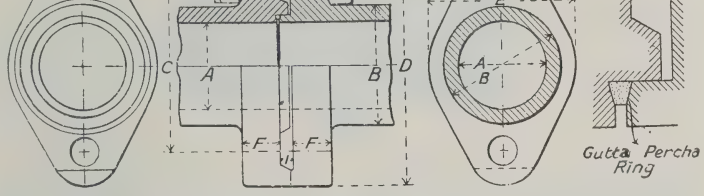


FIG. 2.

Mr. Ellington (engineer of the London Hydraulic Company) found that fractures in the pipes occurred by the breaking off of one of the bolt lugs. By placing the lugs a little farther back on the pipe the strength was found to be greater. It is probable that the slight flexibility of the pipe line which this kind of joint allows is an important element in its success.

On low-pressure hydraulic systems ordinary socket joints can be used.

In the *Philosophical Magazine* the Earl of Berkeley and Dr. Burton have described a simple form of joint packing, which is, they state, tight to start with and maintains this tightness up to the highest pressures they have employed, which have attained 1,500 atmospheres, or

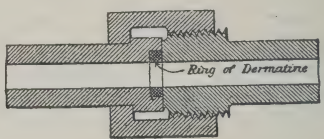


FIG. 3.

say, 10 tons per square inch. The pipe ends (fig. 3) are faced, and a recess is cut in one to take a ring of dermatine; this ring must be slightly thicker than the depth of the recess so that it is slightly compressed when the joint is tightened up. Either red or black dermatine may be used, but the red, being harder, is more easily machined. The joint is said to be equally good for water or oil, and is as suitable for a flanged joint as for a screwed coupling. It is also said to be as satisfactory with large pipes as with small.

THE 'VICTAULIC' SYSTEM OF FLEXIBLE PIPE LINES.

This system (fig. 4) employs pipes with shouldered ends, which are engaged with a certain amount of clearance by the split housing of the joint, which, while coupling the pipes together, permits disalignment or movement due to expansion of the pipes.

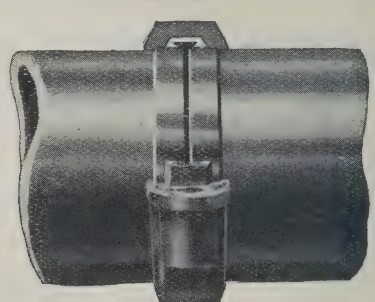


FIG. 4.

Within the split housing is a special design of double 'U' washer, which exercises an initial grip upon the pipe ends and is self-sealing as the pressure rises. By a special proportioning of the buttress-shaped walls of this leak-proof ring, the joint is also self-sealing against a vacuum.

A feature of the joint is its extreme ease in fitting, since there are only two bolts and the joint is not dependent for its tightness upon heavy mechanical pressure. The makers claim that the average time taken in fitting their 6-in. joints by unskilled labour is only two minutes.

The leak-proof ring, being of a fine quality rubber, is of a very durable nature; for oil or gases, acids, etc., a special mixture is used.

The joint is not supplied for use with steam.

Fittings, valves, etc., are all supplied with the ends shouldered or expanded in the same way as the pipe ends, and complete installations can be rapidly built up.

The dimensions of pipe ends are standardised by all the leading tube makers (see table on page 1169), so that all joints and tubes of the same nominal bore will match up together.

Cement Joints for Cast-Iron Water Mains

(1.)

The pipe is placed and spaced in the usual manner. A thin backing of the best dry jute is used instead of oakum, as the jute is free from oil and grease, which should be avoided. Portland cement is placed dry on a piece of canvas (usually a cement sack ripped open) and moistened so that when thoroughly mixed by hand it will be of such a consistency that when gripped tight it will take the form of the hand and when dropped 12 ins. will crumble. The canvas containing the cement is placed under the bell, and the cement tamped into place by hand with a caulking iron until the bell is about half full. It is then caulked with heavy blows until the cement is thoroughly packed in the back of the socket. The process is continued until the bell is packed solid out to the face. A small bead of neat cement is then put on (this is essential), the caulking iron being used as a trowel, and, as soon as the initial set of the cement in the bead has taken place, the joint is covered with earth to protect it from the air and sun. In back-filling the excavated material is always settled with water, which helps to cure the exposed portion of the joint. The joint is allowed to stand forty-eight hours before the pressure is turned on and the main put into service. At Long Beach, California, there are now sixty miles of cast-iron water mains, ranging from 4 ins. to 24 ins. in diameter, laid with joints of this type. All these pipes are under pressures ranging from 40 lbs. to 80 lbs. per sq. in., and are giving satisfaction.

(*Clark H. Shaw, the American Society of Civil Engineers.*)

(2.)

A method of cement jointing cast-iron water pipes is described by Mr. G. W. Pracy in the *Journal of the American Waterworks Association*. The spigot end of the pipe projects into the bell just the same as for a lead joint, and two rings of jute or yarn are placed round the pipe and driven into the bell in the regular manner. The jute used must be free from oil or tar. Neat cement of standard Portland cement brand is used. The mix is very dry, the proportions being about 13 lbs. to 14 lbs. of cement to 1 lb. of water. The operator takes a handful of cement, and with an ordinary caulking iron forces the cement into the joint until full. He then takes a caulking hammer and caulks the joint until the cement is rammed in as hard as possible. The joint is then again filled and the operation repeated until the joint is completely filled with the rammed cement. At first a bead was put around outside the bell, but this was discontinued as being unnecessary. After the joint is finished it is covered with earth while the cement is hardening. In one case over a mile of piping jointed in this manner has been in use under 175 lbs. per square inch pressure for more than two years without requiring any repairs. It is claimed the method is cheaper than lead joints, and one indirect advantage is that the cement joint is an insulating joint, which tends to reduce damage from electrolytic action.

Pipe Joints for Petrol.

Pipe joints for petrol at ordinary atmospheric temperatures can be made up tight by first filling between the screw threads of the pipe with a thin paste made of gum shellac dissolved in a 10 per cent. solution of ammonia. The screw threads of both pipe and fittings should be thoroughly cleaned of grease before application of the paste.

Concrete-Lined Metal Pipes.

It has been American practice for some years past to line metal pipes with a thin coating of cement, which usually does not exceed three-eighths of an inch in thickness, applied by brush, cement gun, or by hand tamping. The experience gained with such pipes, some of which have now been in continuous service over periods of eight to ten years, showed that no appreciable incrustation had taken place even with bad waters. The iron or steel pipe lined with cement was found to have the further advantage that the interior surface could be rendered perfectly smooth—making its coefficient of friction not appreciably different from that of a new cast-iron pipe.

Stanton Centrifugally Concrete-lined Pipes.

A rotary motion is imparted to the pipe by four friction wheels driven by chains and operated by a hand-controlled variable-speed motor. The pipe having been trued up, internal rings are placed in position in order to prevent the concrete coming out at the ends of the pipe during the spinning process. During the first spin the pipe is rotated slowly over a period of about ten minutes, during which time the sand and cement concrete is shovelled into each end of the pipe by hand. Owing to the centrifugal force the cement does not fall down, but adheres to the surface of the pipe, and further consolidation is effected by passing a 3-in. diameter rod with a bronze sleeve through the pipe and pressing this by hand against the revolving internal surface. After the correct thickness has been deposited the pipe is spun at a higher speed, which drives the coarser particles of the concrete towards the iron surface of the pipe and gradually expels the water and slurry from the internal surface. During the spin the water and slurry form a thin revolving film, which, as soon as the pipe is brought to rest, falls to the bottom of the pipe. The internal end rings are then removed and the water brushed out. The concrete lining is now quite hard, and after three or four weeks in the open the pipe is ready for service. In the larger pipes the lining is about $\frac{1}{2}$ in. in thickness, but it may be increased up to 1 in. or more if so desired.

Pipes from 4 ins. to 72 ins. diameter may be lined.

The concrete lining is so finished at the socket end that when the joint is made the water in the pipe is only in contact with concrete and does not touch the iron in any part.

A pipe may be drilled and tapped without the concrete unduly breaking away at the edges of the hole, while the operation of running and caulking the joint can be carried out without detrimental effect on the lining. The effect of exposure to the weather has been investigated by leaving a 48-in. diameter pipe in the open for two years, and although subjected to effect of sun and frost no alteration has been detected either in the metal or in the concrete. A modified form of construction is represented by a series of 60-in. diameter pipes made for the Birmingham Corporation. The shells of these pipes are made from $\frac{3}{8}$-in. steel plates bent and welded into a 5-ft. diameter circle. The steel tube is so light that when the finished shell is placed on the ground it takes up a more or less oval form, being distorted in diameter by nearly $\frac{3}{4}$ in. These tubes are filled with a concrete lining $1\frac{1}{2}$ in. in thickness, which, when formed by spinning, is sufficiently strong to prevent all distortion and give to the pipe a perfectly round shape.

The process of centrifugal lining has also been most successfully applied to the new spun cast-iron pipes which are being manufactured at Stanton in sizes from 4 ins. up to 15 ins. in diameter. The spun cast-iron pipe has not only the advantage that it is nearly 25 per cent. lighter than the ordinary vertical sand-cast pipe, but the way in which it is made ensures a perfectly round and true pipe, avoiding the necessity of re-truing before the lining process.

SECTION XXV

PART II

PUMPS — STEAM PUMPS — PUMP PACKING — CENTRIFUGAL PUMPS—TURBINE PUMPS—AIR LIFT PUMPS.

PUMPS.

The surface of the water to be raised should not be more than 25 feet below the movable valve.

Especially for small pumps, the suction pipe of a pump should fall all the way from the pump to the supply well. Any upward bend in the suction pipe introduces an air trap and may seriously diminish the performance of the pump.

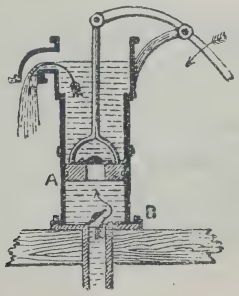


FIG. 1.

FIG. 2.

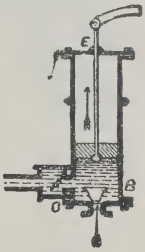


FIG. 3.

THE SUCTION PUMP.

This form of pump, which is shown by fig. 1. consists of a cylinder, called 'the barrel,' with valve, B, in which works a piston, A, called 'the box,' with valve in it, both valves opening upwards. The piston is attached to a rod and worked by a handle. When the handle is raised the piston is depressed: the valve A opens upwards, allowing the water or air below it to escape into upper part of cylinder. When the piston is raised the valve in it is closed, the water above is lifted up through the spout, the water from below lifts the valve at B, and rushes up to fill the vacuum caused by the raising of the piston.

If the valves are very dry a little oil should be poured on them. A little water poured on the piston before commencing to work it, makes it air-tight.

LIFTING PUMP.

The suction pump is converted into a lifting pump by attaching a pipe in place of a spout at C (fig. 2), with a valve in it, opening upwards to prevent the return of the water. A stuffing-box is provided for the piston-rod to work in at E. Each time the piston, A, is raised the water above it is forced up the pipe, and, as the piston descends, the valve C prevents the return of the water from above. It is clear that the man who pumps must exert at each upward stroke of the piston as much strength as is necessary not only to raise a cylinder full from below, but also to lift the column of water in the pipe above.

The valve C is not necessary to the action of the pump, because, the barrel being full, the valve B retains all the water. It is used more as an additional check. The pump would also act if only valve C were in and B removed.

Single-acting Force Pump.

This form of pump, shown by fig. 3, is used to force up water to a height. The piston is made without a valve. The delivery pipe is at bottom of piston: it has a valve opening upwards at C to prevent the return of the water. When the piston is raised the valve B opens and the water rushes into the cylinder: when the piston descends valve B closes, valve C opens, and the water is forced up the pipe. In this pump the force is required for the down stroke. The top of the cylinder E need not be closed; a little water poured in above the piston makes it air-tight.

Plunger Pump.

Sometimes a solid plunger, A (fig. 4), is substituted for the piston, to avoid the expense of turning the cylinder true, and also because it resists better the wear from grease and dirt. At each stroke it raises a volume of water equal to that of the plunger.

Double-acting Force Pump.

This pump (fig. 5) sends up water both at the up and down stroke. At the up stroke the water above the piston is forced through the valve A, while that below enters the piston at the valve D; at the down stroke the water is forced through the valve C, whilst it enters above the piston at the valve B.

Fig. 4.

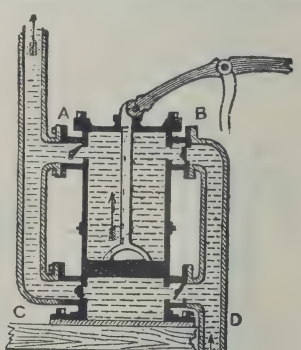


Fig. 5.

In all force pumps the packing of the piston or box is of great importance; it is commonly made of waste tow soaked in tallow and rammed tight; or for great pressures, cup leather packings are used.

Working Barrels, Buckets, and Valves.

These should be made of brass, gun-metal, or phosphor bronze, which metals, although costly in the first instance, will be found more durable, to require less repair, and to be more accurate in their operation than iron, being much less liable to injure the leather attached to the bucket and valve, the parts most frequently in want of repair.

An alloy which will withstand the action of water under pressure in a metal valve, and which is used by one of the largest firms in the country, consists of the following ingredients:—Copper, 10; tin, 1; zinc, $\frac{7}{16}$; lead, $\frac{1}{8}$.

The suction-pipe should be two-thirds the diameter of working barrel.

Pump Balancing.

To ensure equal work on the up and down strokes of a pump, it must be balanced. A complete balance provides for the whole weight of metal hung on the crank pin or beam, and one half the water work. The case of a bucket and plunger pump is perhaps the least simple, as in a bore-hole pump with a plunger at the surface delivering to a height above the surface.

Let A = area of bucket; a = area of plunger; W = weight of moving parts hung on crank pin and immersed in air; w = ditto immersed in water, e.g. the bucket and rods; H = height from water-level when pumping to plunger, in feet; h = height from plunger to point of delivery, in feet; B = balance weight to act at same lever arm as crank pin; then

$$B = W + 0{\cdot}875\, w + 0{\cdot}216 \left\{ [A \times H] + [(A - a) \times h] - [a \times h] \right\}.$$

The coefficient 0·216 represents one half the work, and the relation of pressure to head, or $0{\cdot}5 \times 0{\cdot}43$.

The correction for the different specific gravity of a gun-metal bucket may be neglected. Modern rods ultimately become as heavy as water, but at first have some negative weight. If a balance weight moves through a greater range than the cross-head, it must be correspondingly reduced.

Pumping from Wells.

For wells under 30 feet a 4-inch working barrel with a 9-inch or 10-inch stroke should be used. This size pump will throw 24 gallons per minute from a 20-feet well, and is as large as one man can conveniently work, and the best adapted for general use. For wells above 30 feet and under 70 feet a 3½-inch working barrel with a 9-inch or 10-inch stroke. For wells above 70 feet deep three 3-inch barrels, worked with a three-throw crank wheel and pinion in cast-iron case with fly-wheel, etc.

Artesian Well Pumps.

When the water level in a bore-hole is far below the surface, say 25 to 300 or 400 ft., it is necessary to place the pump barrel down the bore-hole, and it is usually desirable that it shall be lower than the rest level of the water, so as to be charged for starting work.

A bore-hole pump consists of a barrel of gun-metal with a bore at least 1 in. to 1¼ in. less than the diameter of the well lining tube. The barrel is screwed upon a rising main, which must be at least ⅛ in. to ¼ in. bigger than the barrel. The pump bucket and lower valve can thus be drawn up through the rising main for repair or inspection. A 4-in. pump will have a stroke of 1 to 3 ft. as desired, and a speed of 15 to 30 up strokes per minute. For durability it is desirable with very deep pumps to run at slow speeds. The pump rod terminates in a gun-metal rod or plunger, which passes through a stuffing box in the casting which forms the head of the pump. This head is flanged to the lining tube and screwed upon the rising main, and a branch is cast on one side for the water. If the lift above surface is considerable, it is possible partially or wholly to balance the lift of water to the surface by the use of a plunger, which may, if required, be as large in diameter as the bucket. However arranged, in order to equalise the work on the up and down strokes, it is proper to balance half the water lifting, and all the weight hung on the crank pin, so that the duty of the motive power may be evenly distributed in up and down strokes. Comparatively short pumps can be balanced by a weight in the main spur wheel on the crank shaft. When deeper, this is not sufficient, and a weighted lever may be linked to the crosshead pin and fitted with a balance weight on its outer or long end.

Some makers hang a heavy weight upon a pin in the face of the spur wheel, other makers attach wire ropes to the pump crosshead and pass them over a wheel and hang a weight, in guides, upon the ropes. Some engineers prefer to employ wooden pump rods, on account of their large displacement, so as to reduce the balance weights necessary.

A deep-well pump, unless small and slowly driven by a windmill, should be two to three diameters in length, of plain cylindrical form, with alternate grooves, with square-cut corners, and plain body. The grooves should be ¼ in. × $\frac{1}{16}$ in. and the space between, $\frac{5}{16}$ in., the two extreme spaces about ⅝ in. to ¾ in. No rings or packing leathers should be used.

The bucket valve may be mushroom, ball, or flat dermatine, the foot valve the same, in a taper casing, or foot piece which fits in the taper foot piece of the barrel. Under the bucket is a female screw which fits upon the thread projecting above the guard of the lower valve. When it is desired to remove the foot valve for inspection, it is lifted out by the bucket and pump rods, the bucket being lowered upon the valve and screwed to it; the valve then comes up with the bucket.

These pumps are usually driven from a second motion shaft by wheel and pinion, the second motion shaft carrying a fast and loose pulley driven by a belt from the engine or main shaft.

In country house work an oil-engine or an electric motor may be the driving agent, speed being reduced by more pairs of wheels or by larger driven wheels and pulleys. Or an electric motor may be worm-geared to the crank-shaft direct, in which case it is sometimes desirable to interpolate a friction clutch, so that the motor can be started off the load. Larger plants, such as the pumps for small town supplies, are perhaps best driven by gas-engines and suction gas plants.

Steel worms working in phosphor bronze wheels are said to give the best results. The wheel should be hollow faced so as partially to embrace the worm, and the angle of the teeth across the face of the wheel should be about 45°, so that the worm may require to be double or treble threaded to secure this. A thrust collar, or rather several collars, of propeller thrust type, are required on the worm shaft; but it is probably better to employ two worms, right and left, and two wheels connected to the same work, so that the thrusts balance, and no thrust block is required. The wheel should run in oil, which should be not too viscous. If a thrust block is employed, it should not be enclosed too narrowly, especially if near to the worm, as this has been found to prevent free access of oil.

When the water level in a bore-hole is not too deep below the surface, the air lift pump may be used with advantage, but it falls off in efficiency at great depths, and may often necessitate the boring of the well to depths very much greater than would be necessary merely to give the water yield. (*See also* Air Lift Pump, p. 1276.)

Discharge from Reciprocating Pumps.

TABLE OF CONSTANTS WHICH, MULTIPLIED BY EFFECTIVE NUMBER OF STROKES PER MINUTE, GIVE THE DISCHARGE IN GALLONS PER HOUR.

Stroke of Pump, in Inches.	Diameter of Pump, in Inches.												
	2	2½	3	3½	4	4½	5	5½	6	7	8	9	10
1	0 6795	1·061	1·528										
1½	1·019	1·592	2·292	3·120									
2	1·359	2·123	3·057	4·160									
2½	1·698	2·654	3·821	5·200	6·795								
3	2·038	3·185	4·586	6·240	8·154	10·31							
3½	2·378	3·715	5·350	7·280	9·513	12·03	14·86	17·98					
4	2·718	4·247	6·115	8·320	10·87	13·74	16·98	20·54	24·45				
4½	3·057	4·777	6·879	9·360	12·23	15·45	19·11	23·16	27·50				
5	3·397	5·308	7·644	10·40	15·59	17·19	21·23	25·68	30·06	41·60			
5½	3·737	5·839	8·408	11·44	14·95	18·91	23·35	28·25	33·63	45·76			
6	4 077	6·370	9·173	12·48	16·31	20·63	25·48	30·81	36·68	49·92	65·20	82·52	
6½	4·416	6·901	9·937	13·52	17·67	22·35	27·60	33·39	39·73	54 08	70·63	89·40	
7	4·756	7·432	10·70	14·56	19·02	24·07	29·73	35·95	42·79	58·24	76·06	96·27	
7½	5·096	7·962	11·46	15·60	20·38	25·79	31·85	38·52	45·84	62·40	81·49	103·1	
8	5·436	8·494	12·23	16·64	21·74	27·50	33·97	41·09	48·91	66·56	86·93	110·0	135·8
8½	5·778	9·024	12·99	17·68	23·10	29·22	36·10	43·65	51·95	70·72	92·36	116·9	144·3
9	6·115	9·555	13·76	18·72	24·46	30·94	38·22	46·23	55·00	74·98	98·10	123·7	152·9
9½	6·455	10·08	14·52	19·76	25·82	32·66	40·34	48·79	58·50	79·04	103·5	130·6	161 3
10	6·795	10·61	15·28	20·80	27·18	34·38	42·47	51·35	61·12	83·20	108·7	137·5	169·8
10½	7·135	11·14	16·04	21·84	28·54	36·10	44·59	53·90	64·20	87·36	114·1	144·4	178·3
11	7·474	11·67	16·80	22·88	29·89	37·82	46·71	56·50	67·26	91·52	119·5	151·2	186·8
12	8·154	12·74	18·34	24·96	32·60	41·27	50·94	61·63	73·35	99·84	130·4	165·0	203·7
13	8·832	13·80	19·86	27·04	35·32	44·71	55·18	66·77	79·46	108·1	140·2	178·8	220·7
14	9·512	14·86	21·39	29·12	38·04	48·14	59·43	71·90	85·58	116·5	152·1	192·5	237·7
15	10·19	15·92	22·92	31·20	40·75	51·58	63·67	77·04	91·69	124·8	162·9	206·2	254·7
16	10·87	16·99	24·45	33·28	43·47	55·02	67·92	82·17	97·81	133·1	173·8	220·0	271·7
17	11·55	18·05	25·98	35·36	46·19	58·46	72·16	87·31	103·9	141·4	184·7	233·8	288·6
18	12·23	19·11	27·51	37·44	48·90	61·90	76·41	92·45	110·0	149·7	196·2	274·4	305·7

CAPACITY OF PUMP.

If
d = diameter of barrel, in inches ; s = stroke, in inches ; n = number of acting strokes per minute ; g = gallons per hour ; then

$$g = d^2sn \div 6, \text{ approximately.}$$

Semi-Rotary Pumps.

THE ORIGINAL 'ALLWEILER' SEMI-ROTARY WING PUMP.

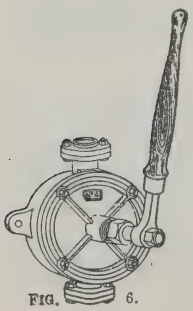


FIG. 6.

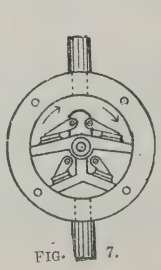


FIG. 7.

These pumps, figs. 6 and 7 (*W. H. Willcox & Co. Ltd.*), are made either double (as shown in the figs.) or quadruple acting. The working parts are of metal, and the wear and tear is reduced to a minimum ; they are claimed to be lighter and occupy less space than any other pumps of equal capacity, and are self-contained. Although fig. 6 shows a form for hand-working, these pumps are adapted for being fitted to any kind of motive power and can be employed for all general purposes.

PARTICULARS OF WILLCOX SEMI-ROTARY PUMPS.

No. of Size.	Maximum Capacity per hour, and Price. Double Acting. Galls.	Cast Iron.	Brass.	Quadruple Acting. Galls.	Cast Iron.	Brass.	Size of Suction and Delivery Pipes (inside). Ins.
0	270	£1 16 0	£2 10 0	350	£1 18 0	£2 12 0	¾
1	400	2 0 0	2 16 0	500	2 4 0	3 0 0	¾
2	540	2 8 0	3 12 0	630	2 10 0	3 16 0	1
3	670	2 18 0	5 0 0	940	3 4 0	5 4 0	1¼
4	900	3 12 0	6 0 0	1,250	3 18 0	6 10 0	1¼
5	1,240	4 0 0	7 10 0	1,420	4 10 0	7 16 0	1½
6	1,480	4 16 0	9 0 0	1,740	5 10 0	9 10 0	1½
7	1,880	6 10 0	11 10 0	2,150	7 0 0	12 10 0	2
8	2,470	8 10 0	16 10 0	2,700	9 10 0	18 0 0	2
9	3,300	11 10 0	22 0 0	3,820	12 4 0	23 10 0	2½
10	3,750	16 10 0	30 0 0	5,000	18 10 0	34 0 0	3
11	4,360	20 0 0	37 0 0	6,030	22 0 0	38 10 0	3
12	—	—	—	8,300	On application.		

The capacities in the foregoing table are based on 3 ft. suction, 3 ft. delivery, and on a full stroke; but the pumps will lift 25 ft. and force to a height of about 70 ft., according to size and speed.

'Allweiler' Semi-Rotary Wing Pumps can be supplied with rubber balls for thick liquids, etc., and with phosphor bronze balls for acids, etc.

THE BOULTON WATER ELEVATOR.

This water elevator consists of an endless band to which are fixed a series of open-ended cells, so designed as to retain water by surface tension. The band is shown in fig. 8 and the general arrangement of the apparatus in fig. 9.

FIG. 8.

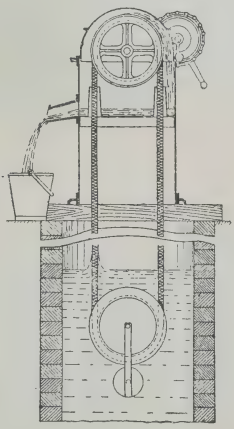


FIG. 9.

The band passes over a driving pulley at the top, carried in a suitable casing. At the lower end a weighted pulley is carried in the loop formed by the band and is submerged in the water.

When the top pulley is rotated the band is set in motion, carrying on the upward side water which is released in passing over the top pulley by centrifugal force breaking down the surface tension.

The bands are made in various sizes, both single and multi-element, ranging in output for power operation from 284 to 19,500 gals. per hour. For hand use outputs of 2·8 gals. per minute from 220 ft. deep, or 13 gals. per minute from 50 feet, can be obtained by one man.

The Boulton water elevator is suitable for dug wells of practically any depth; installations as deep as 410 ft. having already been successfully undertaken.

The efficiency is very high. From official independent test figures a band efficiency of over 95 per cent. on a well 150 ft. deep was recorded.

The headgears are made in various types to suit most purposes, both with and without force pumps, for delivering to heads above the elevator.

The elevator can also be used for handling sewage, sludge, etc., and the band and elements are practically non-corrodible.

On wells over about 30 ft. the cost is very much lower than ordinary deep-well pumps, and amongst its chief advantages are (1) ease of installation, there being no need to descend, and nothing to fix in the well; (2) simplicity—no valves, buckets, plungers, or pipes in the well; (3) it is unaffected by frost; (4) high efficiency; (5) suitable for wells where there is a big variation in water level; (6) the band is not an articulated construction, and has no pins or knuckles to wear or corrode.

STEAM PUMPS.

The Clarke Chapman Direct-Acting Feed Pumps.

(Woodeson's Patent.)

The single-cylinder direct-acting type of pump, on account of the small number of working parts and consequent small liability to break down, is very largely employed. Figs. 10 and 11, p. 1255, show Woodeson's 'Simple' Patent Direct-acting Pump (*Clarke, Chapman & Co. Ltd., Gateshead-on-Tyne*). This pump, which is extensively used, works with all steam pressures up to 400 lbs. per sq. in.

The steam chest is manufactured of the best hard close-grained cast-iron, of a suitable thickness to withstand any tendency to distort through the temperature of high-pressure steam.

The working parts of the steam chest consist of one auxiliary valve and one main valve, both of piston valve type.

The auxiliary valve is operated by the valve gear, actuated by the piston-rod crosshead. The main valve is partly moved mechanically and partly operated by steam, controlled by the auxiliary valve.

The action of the valves is as follows :—

When the main piston is nearing the end of its stroke, and the auxiliary valve commences to be moved by the valve gear, the first part of the movement closes the port through which the exhaust from one end of the main valve has been passing. The next part of the movement of the auxiliary valve opens steam to this end of the valve, and as the steam to the opposite end has not been closed, the main valve is in equilibrium. By the time that this takes place the auxiliary valve has moved so as to be in contact with the main valve, and further movement of the valve gear causes the auxiliary valve to move the main valve mechanically and so close the ports to the main cylinder. When these ports are nearly closed the auxiliary valve has closed the steam port at the end of the main valve, towards which the valves are moving, and opened an exhaust port, with the result that the main valve is moved by steam to the end of its travel and the steam cylinder ports are opened for the return stroke.

Should the main valve by any chance be delayed in being moved by steam, it would be worked mechanically during the clearance in the steam cylinder to a sufficient extent to open the steam port at the end of the cylinder towards which the piston was moving, and so forming a steam cushion to prevent the possibility of the piston striking the cover.

Advantages of the gear are :—

The main and auxiliary valves can all be mechanically actuated from the outside; this is an advantage if the pump is standing for any length of time, as it keeps the faces from getting rusted up.

During the process of the mechanical movement, the piston to which this movement is imparted is in perfect equilibrium.

Entire absence of valve setting.

These pumps are also made on the Tandem Compound principle. Great economy has been obtained with the pumps, and they are specially applicable for electric power stations and other high-class installations where economy of steam is an important consideration.

Fig. 12, p. 1255, shows a pair of these pumps fitted with automatic controlling gear, as used for Main Independent Feed-Pumps for Marine Purposes; with this automatic controlling gear there must be no dead points, and the pump must be able to start from any position, this being a special recommendation for a feed-pump for marine purposes, where, in order to derive all the benefits to be gained by using independent main feed-pumps, the speed of the feed-pumps must be automatically regulated according to the quantity of water to be dealt with, which varies with the speed of the main engines. The 'Woodeson' pump contains all the necessary features for this work.

No flat surfaces exist in any part of the water end exposed to pressure. Both suction and delivery valves are arranged in groups, and are easily accessible for overhauling. The plunger

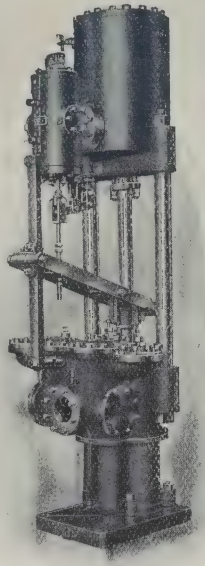


FIG. 10.

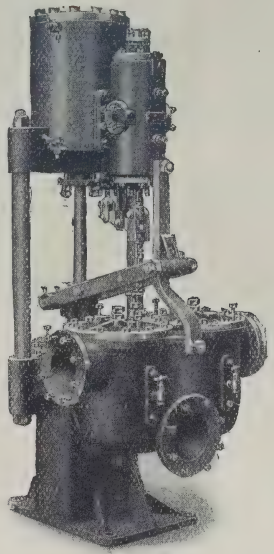


FIG. 11.

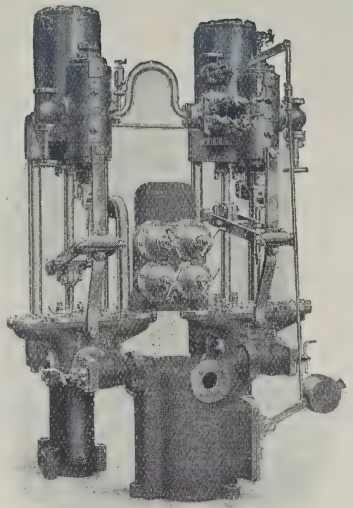


FIG. 12.

junks are of gunmetal, and are fitted with composition plunger rings, specially manufactured to deal with high pressure and high temperature.

Test taken with Clarke, Chapman Simple Type of Slow Speed Feed-Pump (Woodeson's Patent):—Diameter of steam cylinder, 10½ ins.; Diameter of water cylinder, 8 ins.; Length of stroke, 18 ins.; Water delivered per lb. of steam used, 81 lbs.; Steam used per water h.p. hour, 59·5 lbs.; Steam pressure per sq. in., 180 lbs.; Pressure in discharge pipes per sq. in., 195 lbs.; Efficiency of pump, 97 per cent.

Test taken with Clarke, Chapman Tandem Compound Type of Slow Speed Feed-Pump (Woodeson's Patent):—Diameter of h.p. cylinder, 8¼ ins.; Diameter of l.p. cylinder, 13 ins.; Diameter of water cylinder, 8¼ ins.; Length of stroke, 18 ins.; Water delivered per lb. of steam used, 110·5 lbs.; Steam used per water h.p. per hour, 36·5 lbs.; Steam pressure per sq. in., 195 lbs.; Pressure per square in. in discharge pipes, 215 lbs.; Efficiency of pump, 97 per cent.

The Warner Anthony Pump.

(*Robert Warner Pumps, Ltd., Newcastle-on-Tyne.*)

This pump is a distinct departure from the usual steam-operated gear fitted to most direct-acting steam pumps. The main difference lies in the fact that the whole of the working parts are self-contained and form a separate unit, which can be removed from, and replaced in, its valve chest by simply removing the steam chest cover, and without disturbing any adjustment. It is easily possible to shut down one of these pumps, take out the steam distribution unit, slip another unit in its place, and start up again, in five minutes. Pumps so fitted have run under adverse conditions night and day for upwards of two years without a stop.

Fig. 13, p. 1257, represents a section through a steam cylinder and valve chest with the unit in its working position in the chest. The main cylinder has an ordinary three-ported valve face on which works an ordinary slide valve B. The unit proper which bridges over this slide valve, but which is quite separate from it, consists of a small cylinder provided with faces which contact with port faces on either side of the main port face of main cylinder. Inside the cylinder A is a piston C attached to a piston rod E which passes up through the unit cylinder cover and has on its outer or upper end a circular head F, which fits into a separately formed recess in a small cover G in the top of the valve chest, which recess is so arranged that whilst the head of the piston rod can be withdrawn horizontally, no vertical motion is possible.

It will thus be seen that the usual arrangement is reversed, in that the piston C is held in one fixed position, while the cylinder A is free to move on it.

Through the lower end of the valve chest works the valve rod H attached by lever gear to the main piston rod, a fulcrum being provided by a bracket bolted to a lug on the main cylinder casting. The rod H is, like the rod E, provided with an enlarged circular head J, which works inside the hooded recess K formed in the lower end of the unit cylinder A.

Supposing now that steam is admitted to the bottom side of the main piston, the top side being open to exhaust, the sequence of operations is then as follows:

The main piston travelling upwards pulls the rod H downwards; when the rod has travelled down a certain distance, the head J engages with the lower face of the recess K and thus pulls down the cylinder A, and with it the slide valve B. At the moment when the head J engages with the lower face of the recess K, two small ports, which are arranged one on either side of the slide valve B, but which are not shown in the figures, come into register with two other small ports in the cylinder A, also not shown. When the main piston has nearly completed its stroke, and when the lower main steam port has been closed by the valve B, the head J has pulled the cylinder A so far that the small port on one side has passed over the edge of the main port face, while at the same moment the small port in the other side is put into direct communication with the main exhaust port, the result being that live steam is admitted to the lower end of the unit cylinder A, shooting it over the remainder of its travel and carrying with it the slide valve B. This action opens the upper steam port MSP 1 of the main cylinder to the live steam in the valve chest, and puts the lower main port MSP to exhaust. The same sequence of operations is then repeated, the unit in this case being pushed up instead of being pulled down.

In the sectional view the main piston is shown on its downward travel. The valve rod H travelling upwards has just come into contact with the upper face of the recess K of the unit cylinder, and is about to carry it and the slide valve up with it, so shutting off the upper main steam port MSP 1, after which the unit will be shot over by steam pressure acting at the right moment on the upper side of the fixed piston, this shooting it over and reversing the main valve B.

This system is fitted by Messrs. Warner to pumps for all duties, and is supplied by them to the British and Allied Admiralties and the principal users of this type of machinery.

The Weir Direct-Acting Feed Pump.

The Weir Patent Direct-Acting Feed Pump (*G. & J. Weir, Ltd.*), shown in fig. 14, p. 1259, is the original of a type of pump which is being largely used. Introduced at first for marine boiler feeding, its adoption by the mercantile marine, the British Navy, and numerous foreign navies, caused it afterwards to be placed on the market in a form suitable for land installations, power plants, etc.

The valve gear is positive—*i.e.* the steam valve can never be in such a position that the pump will not start immediately when steam is turned on. The only possible way in which the main

THE 'WARNER ANTHONY' PUMP.

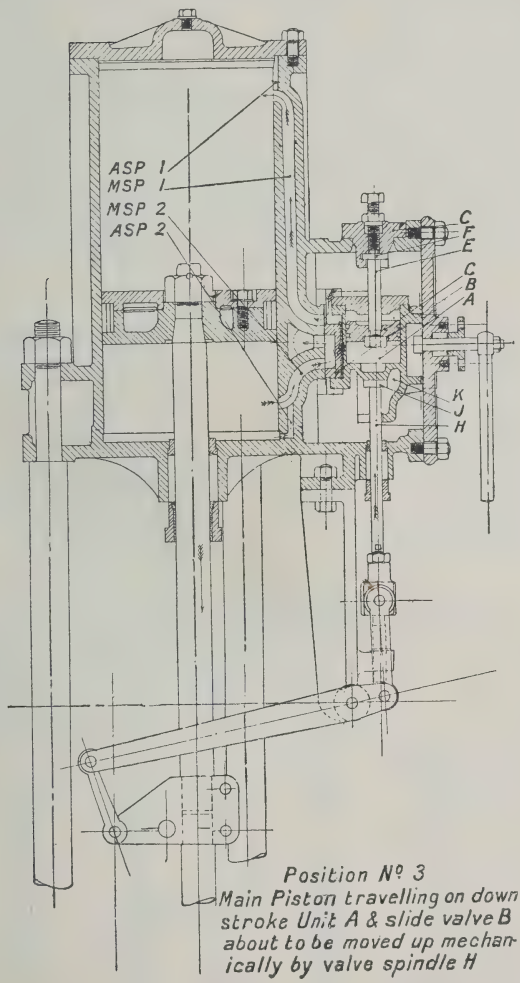


FIG. 13.

value can rest is at full travel, either for an up or down stroke of the piston. The valve arrangement also ensures constant length of stroke and certainty of action.

The steam valve is practically a D slide-valve, with a small auxiliary valve working on the back, and hence has not the same liability to leak as a piston valve. These are the only two moving parts proper in the steam chest, so that there is little opportunity for wear, and no delicate adjustments to get out of order. The steam is used expansively, and the cut-off can be regulated from the outside while the pump is working.

The water valves are of the Weir group type, providing a large area with only a small lift, thus ensuring easy working and little wear and tear. The pumps are designed to work at a moderate speed and the piston is slowed down towards the end of the stroke, enabling the valves to settle quietly in their seats. The working and maintenance of the pumps is therefore much more satisfactory than in pumps running at a high speed. In the matter of steam consumption, numerous tests and comparisons show that the Weir pump is highly economical; this is due to the fact that it has fewer ports and passages than a duplex pump, and the simple expansion arrangement enables it to be built so that the clearance is small.

The parts of each pump are of a standard size, made to gauge and interchangeable, and each complete pump is tested hydraulically and under steam before leaving the works.

Where it is desired to regulate the boiler feed automatically, a special design of float tank and regulating gear is furnished.

The Weir pumps are also supplied of the tandem compound and twin compound types.

Messrs. G. & J. Weir, Ltd., Cathcart, Glasgow, the makers, also list a line of these pumps for light service, tank duty, etc.

List of Parts.—(See fig. 14, page 1259.)

1. Steam slide valve chest.
2. Double joint.
3. Front stay.
4. Bottom spindle.
5. Valve gear levers.
6. Front stay bush.
7. Ball crosshead.
8. Main crosshead.
9. Crosshead pin.
10. Piston rod.
11. Piston body.
12. Piston rings.
13. Cylinder cover.
14. Discharge valve seat.
15. Discharge valve seat ring.
16. Suction valve seat.
17. Suction valve guard.
18. Discharge valve guard.
19. Water valves.
20. Bucket.
21. Pump rod.
22. Pump cover.
23. Valve chest covers.
24. Steam stop valve.
25. Exhaust stop valve.
26. Auxiliary valve spindle.
27. Cylinder neck ring.
28. Cylinder gland.
29. Pump neck ring.
30. Pump gland.

STANDARD SIZES AND CAPACITIES OF WEIR FEED PUMPS.

Size.	Normal Duty.		Maximum Duty.	
	Gallons per Hour.	D.S. per Min.	Gallons per Hour.	D.S. per Min.
Pump. Cyl. Stroke. Ins. Ins. Ins.				
2½ × 3½ × 5	270	30	340	38
3 × 4½ × 6	410	26	535	34
4 × 6 × 7	660	20	810	25
4 × 6 × 12	960	17	1,205	21½
5 × 7 × 12	1,450	16	1,725	19
6 × 8½ × 13	2,000	14	2,350	16½
6 × 8½ × 18	2,700	13½	3,350	16½
7 × 9½ × 21	4,000	13	4,750	15
8 × 10½ × 22	5,500	12½	6,350	14½
9 × 12 × 24	7,200	12	8,100	13½
10 × 13½ × 24	9,000	12	10,000	13½
11½ × 15½ × 24	11,250	11¾	12,000	12½
12½ × 17 × 24	13,200	11½	13,800	12
12½ × 17 × 28	14,700	11	15,400	11½
14 × 20 × 27	16,500	10	18,100	11
14 × 20 × 32	19,600	10	21,500	11

THE WEIR DIRECT-ACTING FEED PUMP.

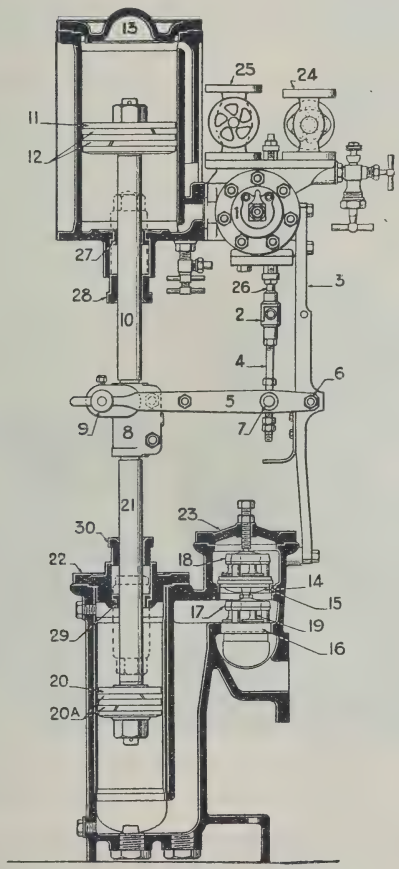


FIG. 14.

The Weir 'Turbo Feed' Pump.

(G. & J. Weir, Ltd., Cathcart, Glasgow.)

This pump, fig. 15, consists of a single pressure, multivelocity stage, impulse steam turbine, working in conjunction with a single stage centrifugal pump of special design.

The turbine casing and bearings are divided horizontally, the lower portions being embodied in the general framework or base of the set. This framework is extended at one end to form a flange, to which the pump casing is rigidly secured. By this arrangement the impeller of the pump is carried on the overhanging end of the turbine shaft.

By means of a suitable hydraulic governor, operating a balanced double beat throttle valve, the set is self-regulating for all ordinary changes in both water and steam conditions. When the latter are decidedly adverse, auxiliary nozzles may be brought into play by means of auxi-

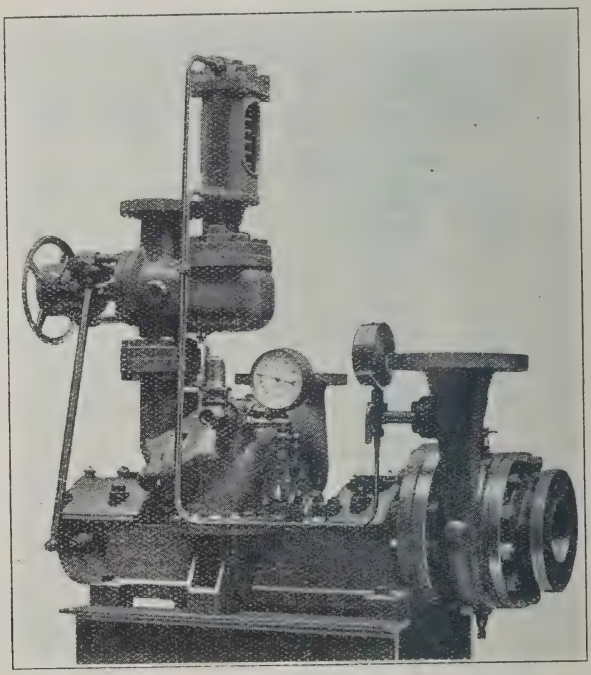


Fig. 15.—The Weir 'Turbo Feed' Pump.

liary valves in the nozzle box and the quantity of steam supplied to the turbine considerably augmented.

An emergency governor of the eccentrically-loaded ring type is provided, which, in the event of the speed of rotation exceeding a predetermined limit, actuates a trip gear which releases a spring and instantaneously closes the stop valve, and steam is completely shut off from the turbine. Thus there is no possibility of the speed of the set reaching unsafe limits in the event of an accidental failure of the supply ; and as the discharge pressure depends upon the pump speed only, the set cannot be seriously overstrained by sudden obstruction or closing of the discharge.

The turbine glands are provided with carbon packing, the bearings are of the self-oiling ring type, and the pump impeller is hydraulically balanced.

The Weir Turbo Feed Pumps are of sound mechanical construction and all parts are readily accessible for examination and overhauling. They are entirely automatic in action and are noiseless and free from vibration.

WEIR PUMPS AND AUXILIARIES

For Land & Marine Duty.

BOILER FEED PUMPS

Direct-Acting ; Steam Turbine & Electric Driven Centrifugal Types

DIRECT-ACTING & ELECTRIC TWO-THROW RECIPROCATING PUMPS

for Bilge, Ballast, Sanitary, Fresh Water, Engine Jacket and Piston Cooling ; handling Tar, Chemical Solutions, Ammoniacal Liquors, etc.

OIL PUMPS

Transfer, Cargo Oil, Fuel Oil Pressure ; Special Pumps for Refineries and Cracking Plants.

FEED HEATERS
REGENERATIVE CONDENSERS &
CLOSED FEED SYSTEMS
EVAPORATORS
DE-AERATORS
AIR COMPRESSORS
AIR PUMPS, *Etc.*

Descriptive Publications mailed on request.

G. & J. WEIR LTD.
CATHCART : GLASGOW :

F. 1260

CENTRIFUGAL AND TURBINE PUMPS

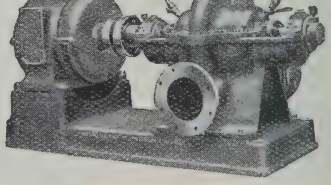
"Centrifugal" Symbol.

OF THE HIGHEST EFFICIENCY

For Mining, Boosting, Boiler Feeding, Irrigation, Coke Quenching, Slurry, Sewage, Hot-water Circulating, etc.

Other Manufactures include:—

FILTRATION & REFRIGERATION PLANTS
VACUUM PUMPS.



A motor-driven horizontally split casing "HSB" Type Pump for large quantities against heads up to 125 feet.

Write for List No. 1506

Pulsometer Engineering Co. Ltd.

11 Tothill Street, LONDON, S.W.1

Nine Elms Iron Works, READING.

F. 1261

The pumps are capable of discharging against the highest boiler pressures and are specially suitable for large units. They can be used to work in conjunction with ordinary reciprocating pumps, and are made in multi-stage types for high pressures.

A special feature of the Weir Turbo Feed Pump is its economy in steam consumption. Further, as the exhaust steam contains no oily residue, it is available for heating the feed water by direct contact, thereby utilising all the heat in the steam. In this connection Messrs. Weir have introduced a nozzle heater which has been specially designed for working in conjunction with these pumps.

The Weir Turbo-Feed Pump has been adopted by the British Admiralty and the leading Municipalities and Electric Power Corporations, including the Corporations of Cardiff, Glasgow, Greenock, Halifax, Norwich, Poplar, Salford, West Ham, Wolverhampton, Woolwich, the Clyde Valley Electric Power Co., and Treforest Electrical Consumers, Ltd.

The Pulsometer Steam Pump.

In this pump (*The Pulsometer Engineering Co., Ltd.*), figs. 16 and 17 (general view), the neck J contains a gun-metal ball I, above which is a steam pipe K. EE are suction valves placed at the base of two chambers AA. Immediately above the suction valves the two chambers com-

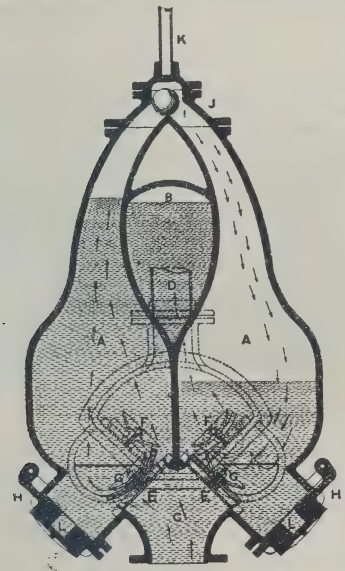


Fig. 16.

Fig. 17.

The Pulsometer Steam Pump.

municate with a ' discharge box ' through two discharge valves FF. The valves in the standard pattern of pump are of the grid type, consisting of a grid, rubber, and guard. In addition to the above valves there are three small gun-metal valves (not shown in the figure) for the admission of air, one of each being screwed into the upper part of the two water chambers and the air vessel.

Action.

Assuming one of the chambers to be open to steam and full of water, the steam entering by the steam pipe and past the ball passes into the chamber and presses upon the small surface of water exposed, depresses it without agitation (and therefore with little condensation), and drives it (as seen in the figure) through the discharge valves FF into the rising main, D. The moment the water in the chamber falls to the level of the opening in the branch leading to the discharge box, the steam blows through with a certain amount of violence, and as it is brought into intimate contact with the water in the discharge box an instantaneous condensation takes

place, and the vacuum thus formed in the emptied chamber immediately pulls the control ball over on to the corresponding seat and cuts off further admission of steam, allowing the vacuum to be completed. Water immediately enters through the suction pipe, and lifting the inlet valve rapidly fills the chamber again. A similar operation has been taking place in the fellow chamber, the period occupied by filling one chamber corresponding to that of emptying the other, and these operations continue alternately in the two chambers so long as the pump is supplied with steam and water. The function of the air valves is to introduce a small quantity of air at each stroke or 'pulsation' for the purpose of cushioning the ball when it changes its position, and for separating the steam from the water by a non-conducting film so as to prevent loss by condensation during the expulsive portion of the cycle.

The 'Pulsometer' Steam Pump will work just as well suspended from a chain or rope as when permanently fixed. Its very small size, in proportion to the quantity of water thrown, makes it available in a great many situations where no other pump can be applied. Another important feature is its ability to pump water heavily charged with mud and sand. For these reasons it is very largely used by contractors for pumping from excavations.

The 'Pulsometer' Steam Pump will, where the conditions remain constant, work day and night without attendance—in fact, as long as steam and water are supplied to it.

Boiler Pressure Required for Pumping.

What boiler pressure would be required to operate a direct-acting steam pump having an 8-in. diameter steam cylinder and a 6-in diameter water cylinder, for pumping water to a height of 100 ft. with the suction water supplied at a pressure of 10 lbs. per sq. in.?

The mechanical efficiency of a pump of the stated size, when in good working order, would be about 65 per cent. The pressure pumped against would be $100 \times 0{\cdot}433 = 43{\cdot}3$ lbs. per sq. in. due to the head, plus the pressure required to overcome friction of the water in the discharge pipe, the latter depending on the rate of pumping, and the diameter, length, and fittings of the discharge pipe. Under ordinary conditions the loss of pressure from pipe friction would be no greater than 10 lbs. per sq. in., and if assumed to be equal to the suction pressure the net resistance to movement of the water piston would be $(6 \times 6 \times 0{\cdot}7854) \times 43{\cdot}3 = 1{,}224$ lbs. With 65 per cent. mechanical efficiency of the pump, the resistance to be overcome by the steam piston would be $1{,}224 \div 0{\cdot}65 = 1{,}883$ lbs., and for overcoming that resistance the effective pressure required to be exerted on a direct-acting 8-in. diameter steam piston would be $1{,}883 \div (8 \times 8 \times 0{\cdot}7854) = 37{\cdot}46$ lbs. per sq. in. The required boiler pressure will need to be equal to the effective pressure plus the back pressure and the reduction of the boiler pressure by throttling in the steam pipe and steam passages of the pump. Allowing a back pressure of 4 lbs. and reduction by throttling of 10 lbs. per sq. in. would require a boiler pressure of $37{\cdot}46 + 4 + 10 = 51{\cdot}46$ or about 52 lbs. per sq. in.

Willcox's 'Penberthy' (Regd.) Cellar Drainer.

This is a Water Pressure Ejector (*W. H. Willcox & Co. Ltd.*) of high capacity, for use in removing water from cellars, inspection pits, stokeholes, lift pits, or wherever it is desired to keep water or other liquids from rising above a certain level. It is automatically operated by a float-controlled, quick opening and closing valve. This valve is never partially opened; it cannot leak, and allows the ejector to give its greatest efficiency by working to full capacity. The float arm is slotted where it connects to the valve lever; this prevents the ejector from operating until the water has raised the float up to the highest point, and, as the water is ejected, the arm travels down the length of these slots before the weight of the float affects the valve.

To facilitate the installation of this drainer in small holes or sumps it is made as compact as possible. The float, instead of being round, is flat on the top; this gives it more buoyancy and occupies less space than would a round float of the same diameter. To further economise space the float is built around the suction pipe. This arrangement (patented) has the advantage of being rigid, and overcomes all the trouble of a loose float. The balance of the drainer is built directly above the float; this not only keeps the whole machine in as small a space as possible, but also keeps all working parts, with the exception of strainer and float, above the water, thus preventing corrosion or the exterior parts becoming clogged up by deposits of slime, dirt, or sediment. The interior parts are prevented from coming into contact with any foreign particles by a special strainer, which is so constructed that instead of the water being drawn directly upward, it is taken in at the sides, allowing all sediment to collect under the strainer without being disturbed by the force of the suction. A foot valve is provided inside the strainer, which closes the instant the drainer ceases operating, holding all water in the pipes, all primed for starting instantly at the next filling of the sump.

The following tables show the capacities at different pressures and elevations, dimensions, pipe sizes, weights, etc.:

TABLE OF CAPACITIES OF WILLCOX-PENBERTHY CELLAR DRAINERS AT DIFFERENT PRESSURES AND ELEVATIONS, PIPE SIZES, ETC., MODEL 'R.'

Size and Model.	Pipe Sizes. Ins.		Working Head in ft.	Maximum Net Capacities from Sump, in Gallons per Hour at 10 to 100 lbs. Water Pressure.								
	Supply	Discharge		10	15	20	25	30	40	60	80	100
No. 1 Model R	¾	1	3	80	*210*	*300*	*340*	*400*	*510*	*700*	*720*	*700*
			6		90	140	*240*	*300*	*390*	*570*	*670*	*690*
			9			90	150	210	*330*	*480*	*620*	*660*
			12				100	150	220	*390*	*550*	*630*
			15					90	190	*300*	*450*	*600*
			18						120	250	*320*	*550*
No. 2 Model R	¾	1½	3	190	*390*	*550*	*620*	*750*	*910*	*1180*	*1240*	*1220*
			6		210	340	*420*	*600*	*750*	*1020*	*1140*	*1200*
			9			210	300	380	*540*	*900*	*1090*	*1150*
			12				210	300	420	*780*	*1020*	*1110*
			15					210	370	*570*	*940*	*1050*
			18						300	510	*850*	*970*

The model 'R' drainer requires 5 lbs. water pressure or more for each one foot of elevation, while a low-pressure drainer can be supplied which will work on 2 lbs. to 4 lbs. per foot of elevation.

TABLE OF CAPACITIES OF WILLCOX-PENBERTHY CELLAR DRAINERS AT DIFFERENT PRESSURES AND ELEVATIONS, DIMENSIONS AND PIPE SIZES, MODEL 'L.'

Size and Model.	Pipe Sizes. Ins.		Working Head in ft.	Actual Capacities in Gallons of Water per Hour taken from Sump, and Not Total Amount of Discharged Water.			
	Supply.	Discharge and Suction.		At 25 lbs. will elevate	At 40 lbs. will elevate	At 60 lbs. will elevate	At 80 lbs. will elevate
No. 3 Model 'L'	1	1½	3	660	1100	1440	1650
			6	520	860	1230	1440
			9		720	1050	1320
			12			840	1040
			18				760
No. 4 Model 'L'	1½	2	2	960	1600	2020	2400
			6	760	1240	1880	1980
			9		1040	1600	1800
			12			1280	1440
			18				1050
No. 5 Model 'L'	1½	2½	3	1280	2100	2700	3200
			6	1000	1620	2300	2820
			9		1260	1960	2600
			12			1560	2060
			18				1500

The above capacities are the actual capacities of water taken from sump, and not the combined discharge of operating and drainage water as given in most tables. For higher elevations than shown above, special drainers can be made.

'PENBERTHY' (REGD.) AUTOMATIC CELLAR AND SUMP DRAINER.

(New Electric Model.)

This model is the latest addition to the 'Penberthy' (regd.) Cellar Drainer, and is operated by electric current instead of by water pressure. It operates only when water accumulates, causing the float to rise and make contact by means of a special mercury type switch, and immediately ceases action when the water has been disposed of and float falls.

It is designed for fitting in a sump, which should be not less than 15 to 18 in. diameter by 2 ft. deep, and may be constructed of a piece of earthenware sewer pipe, cemented at the bottom, or of brick or concrete. A cover (constructed of almost any material) is desirable, to keep out dirt and refuse.

The pump outlet should be connected to the nearest sewer, drain, or surface outlet, and the electric lead should be plugged into the nearest plug or light socket, when the Drainer is ready for operation.

The arrangement is particularly suitable in cases (*a*) where town water pressure is not available or is very low; (*b*) where excessive quantities of seepage water have to be handled; (*c*) where discharge heads are up to 20 ft.; (*d*) where low cost of operation is an important factor.

Size No.	Pipe Connection.	Approximate Discharge Capacities per hour.				Size Motor.
		5-ft. Lift.	10-ft. Lift.	15-ft. Lift.	20-ft. Lift.	
		Galls.	Galls.	Galls.	Galls.	H.p.
1 E	1 in.	900	700	500	—	¼
2 E	1¼ ,,	2500	2000	1500	600	½

Motor must be above sump hole floor level so as to secure proper ventilation and avoid dampness.

Pump Valve Materials.

Dexine.

Dexine (*Dexine, Ltd.*) is a preparation of vulcanised indiarubber and other ingredients manipulated by a special process; it is an exceedingly tough and frictionless material which has been proved to withstand the action of oils and chemicals. It is extensively used for valves, in the place of ordinary indiarubber, and can be obtained in any degree of hardness, suitable for straight or curved guards and pumps of all descriptions and of all dimensions.

Valves made of Dexine are claimed to retain all the resiliency and natural cohesion of pure Para rubber—which enters into its composition to a certain extent—and is, in addition, rendered immune from the detrimental influence of acids, oils, alkalies, or grease. The material can be moulded to any shape, and vulcanised to any degree of hardness or flexibility to suit any particular purpose.

Dexine is also used to form the 'bucket or cup leathers' of pumps, up to 20 ins. diameter. In round or square section, or in the form of U rings, the material makes an efficient packing for compressor rods, and, having a low coefficient of friction, the wear on the material is very slight.

Hydraulic Packings.

Ram or 'U' Leathers.

A cupped leather packing (ram or 'U' leather) forms a self-tightening packing, and is very generally used, although if it is not properly made, and if the cup is not strengthened at the bend (where the greatest friction is), the leather soon wears out and fails.

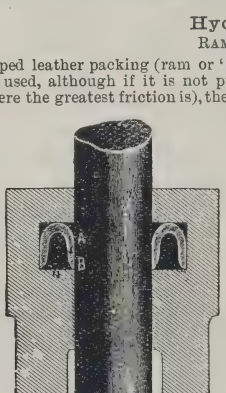


FIG. 18.

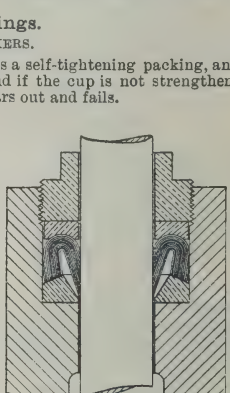


FIG. 19.

Fig. 18 shows a ram leather in its position. The water gets behind the leather and presses it tightly against the ram; the friction occurs mainly at part A, and increases as the pressure of water increases. There is not much friction at the portion between A and B, and the efficiency of the press is but little altered by making AB greater or less. It is, however, asserted by some that the distance AB is of importance, but this is on account of deterioration. The part AB is constantly in states of tension and compression, and if the leather is

not good and well made it is liable to crack. Leathers are very frequently made too deep, and this leads to their being more liable to crack and fail; a shallow leather is undoubtedly the best—it works just as well as a deep one, and in making the leather considerably less strain or pull is required to block or mould it than would be the case with a deep one.

Too much stress cannot be laid on the necessity for using leather packings of the best quality and make only. It is equally important to consider the question of tannage. The value of a tannage depends upon its suitability and adaptability for different classes of work. Low pressure work, for example, requires a different tannage and fibre from high pressure, and medium pressure a tannage and fibre between the two.

Fig. 19 shows a Patent 'SEA' Ring of compressed cotton layers supplied for packing purposes.

Pump Leathers and Packings.

'SEA' rings are specially designed with a view to a higher efficiency than may be obtained by the use of the ordinary leathers. Fig. 20 shows a double-acting plunger fitted with two 'SEA' ring cups separated by metallic junk-ring, and fig. 21 shows a 'SEA' ring collar fitted to a hydraulic or other ram. It is claimed that these ring packings will wear indefinitely where the ordinary leather would wear out and crack in a very short time. The rings are able to resist high temperatures, and are used with success for extremely high pressures. They are

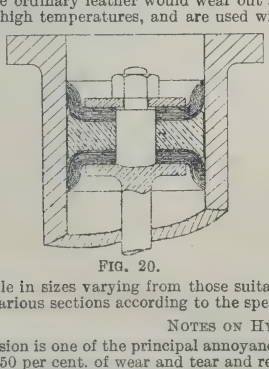


Fig. 20.

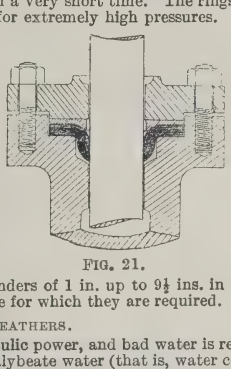


Fig. 21.

obtainable in sizes varying from those suitable for cylinders of 1 in. up to 9¼ ins. in diameter and of various sections according to the specific purpose for which they are required.

Notes on Hydraulic Leathers.

Corrosion is one of the principal annoyances of hydraulic power, and bad water is responsible for fully 50 per cent. of wear and tear and repairs. Chalybeate water (that is, water containing iron) is about the worst.

Where dirty or gritty water has to be used the leathers wear away very rapidly. When the leathers are not kept constantly under pressure, if the pressure is taken off, the water gets between the leather and the ram, and as soon as the pressure comes on again, a little gritty or dirty water passes between the leather and the ram and causes the wear of the packing; it also scaives or flutes the ram. A remedy for this consists in putting a relief valve on the pipe that delivers the water to the accumulator from the pump, and in leaving the suction valve always open. The pressure in this way is always kept on the leather, preventing leakage, and at the same time saving considerable wear and tear of the packing.

It is important and essential that a leather should be well rubbed and worked with tallow before putting in. A good plan is to fill the inner space of the leather with neatsfoot oil or sperm, about an hour before putting in. *Never use mineral oil for rams or cylinders*, it runs away quickly, through being thin, and also renders the leathers rotten and pulpy.

For lubrication, Entwistle's patent lubricant (Patent No. 16,292; 1894), which contains 50 parts stearine or tallow, or other similar substance, 20 parts soapstone or powdered mica or talc, 2 parts graphite, 10 parts neatsfoot or other oil, 10 parts powdered zinc, lead, copper, or other soft metals, and 10 parts oil of tar, has given excellent results; the particles of metal, by the motion of the ram up and down, fill up any indentations, pores, or rough surfaces there may be on the ram or in the cylinder, and being of a soft or yielding nature (not gritty) distribute themselves over the rough surfaces.

Friction of Accumulator Cup Leathers.

The loss by friction in pumping into an accumulator is a very doubtful quantity and has not been satisfactorily determined. Various formulæ have been given, but a very wide divergence exists between them, probably due to the fact that the friction varies very greatly with the smoothness of the ram—whether it is rusty or not—the nature of the leather, also as to whether it is clean or sticky, and whether the ram is stationary or sliding, and whether the ram is perfectly centrally loaded—if the load is only a very small amount out of the centre of the ram it tilts it and materially increases the friction.

The formula due to 'Hicks' gives a result which makes the friction but one-sixteenth of that given by another formula due to Professor Goodman, and even the latter formula has been questioned as giving results much too low.

CENTRIFUGAL PUMPS.

V = velocity of the periphery of fan, in feet per second. H = height, in feet, to which water is to be raised. R = revolutions per minute of fan. D = diameter of fan, in inches. W = width of fan at the periphery, in inches. d = diameter of discharge-pipe, in inches. n = number of blades. t = thickness of blades. G = gallons per minute. E.H.P. = effective horse-power required.

$$V = 8{\cdot}8\sqrt{H}, \text{ for small fans, to } 10{\cdot}5\sqrt{H} \text{ for large fans.}$$

Another formula is—

$$V = 8{\cdot}8\sqrt{H} + K, \qquad \text{where } K = 5 \text{ to } 10.$$

The last formula gives good results, varying the coefficient K with the height of lift.

$$R = \frac{V \times 230}{D} = \frac{(8{\cdot}8\sqrt{H} + K) \times 230}{D}. \qquad D = \frac{(8{\cdot}8\sqrt{H} + K) \times 230}{R}.$$

$D = 1{\cdot}5d.$ to $5d.$ $\qquad d = \frac{\sqrt{G}}{4}.$ $\qquad W = \frac{d^2}{4D - 1{\cdot}27nt}.$ $\qquad E.H.P = \frac{G \times 10 \times H \times \frac{5}{3}}{33000} = G \times H \times {\cdot}00076.$

(The fraction $\frac{5}{3}$ represents 40 per cent. efficiency.)

EXAMPLE.—What size of pump is required to lift 4,000 gallons per minute to a height of 15 feet?

First find velocity, V, of periphery of fan. Taking 6 as a mean value for K, then $V = 8{\cdot}8\sqrt{15} + 6 = 34{\cdot}08 + 6 = 40{\cdot}08$ feet per second, or, say, 40 feet per second.

Diameter, d, of discharge-pipe at 4,000 gallons per minute, G, $= \frac{\sqrt{4000}}{4} = 16$ inches.

Taking $D = 3d$ as a mean value, then $D = 3 \times 16 = 48$ inches.

Number of revolutions per minute, R, $= \frac{8{\cdot}8\ (\sqrt{15} + 6)\ 230}{48} = 192.$

Taking 6 blades (n), each having a thickness, t, of $\frac{1}{2}$ inch, then

$$\text{Width, W, of fan} = \frac{16 \times 16}{(4 \times 48) - (1{\cdot}27 \times 6 \times \frac{1}{2})} = \frac{256}{192 - 3{\cdot}81} = 1{\cdot}36 \text{ inches.}$$

The greater the diameter of the fan, the less number of revolutions it requires to make.

The speed of the water through the pipes should not much exceed 8 feet per second. In many pumps the speed is greater (10 feet per second); but this is too high for good practice.

In all cases the efficiency of a pump will be increased by a tapering enlargement of the discharge pipe to as large a diameter as convenient. Such a taper pipe should be very slowly tapered, say about 1 in 8 times its initial diameter.

PROPORTIONS OF SINGLE-STAGE CENTRIFUGAL PUMPS.

Size of Pump and Diameter of Pipes.	Nominal Horse-power of Engine for each Foot of Vertical Height from Level of Water to Top of Delivery Bend.	Maximum Quantity raised per Minute.	Diameter and Width of Pulley.	Width and Kind of Leather Belt for Lifts up to 20 Feet.	Resistance of each Yard of Pipe for maximum quantity in Ins. of Lift.
Ins.		Gallons.	Inches.		
3	·045	150	8 × 4	2 inch single	3·0
4	·078	260	9 × 4	2 „ „	2·25
5	·121	400	9 × 5	2½ „ „	1·75
6	·175	580	10 × 6	3 „ „	1·5
7	·242	800	11 × 6	3½ „ „	1·26
8	·303	1,000	11 × 7½	4 „ „	1·1
10	·484	1,600	11 × 7	4 inch double	·9
12	·700	2,300	13 × 9½	5 „ „	·75
13	·818	2,700	14 × 10½	6 „ „	·7
16	1·242	4,100	21 × 10½	7 „ „	·58
18	1·606	5,300	24 × 10	9 „ „	·5
20	1·970	6,500	30 × 11	10 „ „	·45
24	2·848	9,400	36 × 15	14 „ „	·37

SPEED TABLE OF CENTRIFUGAL PUMPS.

Giving the Number of Revolutions per Minute (including an allowance for slip of the belt) which the Pumps should make for different Lifts.

Size of Pump in Inches.	Feet Lift.							Size of Pump in Inches.	Feet Lift.						
	3	6	9	12	15	20	25		3	6	9	12	15	20	25
3	463	563	638	709	774	872	960	12	296	353	404	448	487	549	604
4	460	560	634	704	768	864	951	13	276	330	377	418	456	512	563
5	422	505	578	642	700	788	867	16	222	267	304	338	369	414	454
6	388	466	532	592	644	726	798	18	196	236	269	299	325	366	402
7	360	432	493	548	596	672	738	20	178	212	243	270	294	330	363
8	359	432	492	547	596	670	737	24	147	176	201	234	243	273	301
10	359	431	491	546	595	669	736								

The best speed to drive centrifugal pumps is a much debated question. For mine use moderate speed is preferable wherever possible, as less trouble is then experienced with both pump and motor. In other cases the higher speed pumps give good service. For 75 or 100 h.p. and upward 900 r.p.m. is satisfactory; for 30 to 50 or 75 h.p. 1,200 r.p.m., and for smaller pumps 1,500 to 1,800 r.p.m. (*H. Axford in 'Coal Age.'*)

Classification of Centrifugal Pumps.

Generally speaking, centrifugal pumps may be classed in two groups, namely :—(*a*) Without guide wheels; (*b*) with guide wheels.

(*a*) In pumps which are *not provided with guide wheels* (fig. 22), the casing is constructed

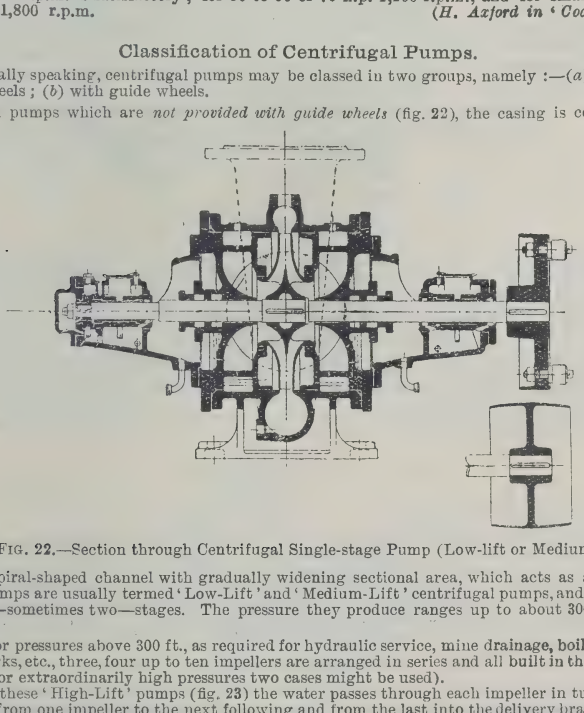


FIG. 22.—Section through Centrifugal Single-stage Pump (Low-lift or Medium).

with a spiral-shaped channel with gradually widening sectional area, which acts as a diffuser. These pumps are usually termed 'Low-Lift' and 'Medium-Lift' centrifugal pumps, and have, as a rule one—sometimes two—stages. The pressure they produce ranges up to about 300 ft. water column.

(*b*) For pressures above 300 ft., as required for hydraulic service, mine drainage, boiler feeding waterworks, etc., three, four up to ten impellers are arranged in series and all built in the common casing (for extraordinarily high pressures two cases might be used).

With these 'High-Lift' pumps (fig. 23) the water passes through each impeller in turn, and is directed from one impeller to the next following and from the last into the delivery branch of the pump through specially designed *guide wheels*, converting into pressure, without shock, the bulk of the kinetic energy transmitted to the water by the impellers. Each stage of the pump contributes equally to the total pressure. The design of the diffuser guide wheels and the blades of the impellers, i.e. the passages through which the water travels with a high velocity, is mainly responsible for the degree of efficiency attained by a centrifugal turbine pump. The steady flow

of water through the pump, with as few changes in velocity as possible, avoiding shocks and eddies with inherent friction loss, is of the greatest importance. The clearance between the rotating and stationary parts in the pump must be so dimensioned that too much water cannot flow back from one stage to the preceding one. The balancing of the axial thrust produced in multi-stage pumps is also important.

There are practically no limits as regards the pressure heads that high-lift centrifugal pumps cannot overcome (some makers, for example, have supplied pumps delivering to 3,500 ft.), as long as the quantity to be delivered is not too small. The manometric suction head should, for hydraulic reasons, not exceed about 18 ft., as otherwise the efficiency may be reduced and wear in the first stage be caused.

For hot liquids the suction water should flow from a higher position into the pump, this height being determined by the fact that no evaporation must take place in the suction pipe. (See also p. 1270.)

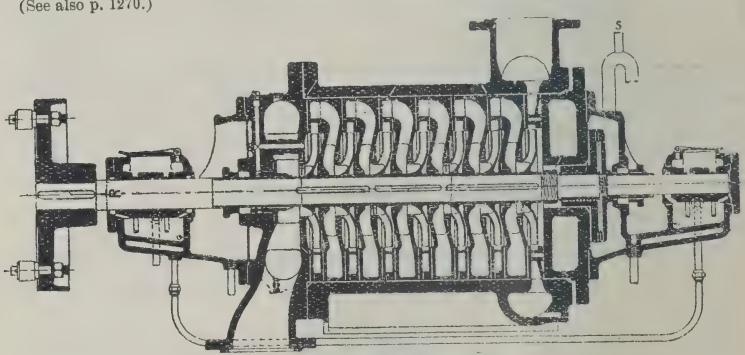


FIG. 23.—Section through Centrifugal High-lift Pump.

The characteristics of the centrifugal pump are in many respects different, even the reverse, to those of the piston pump. Thus a centrifugal pump can run with the delivery valve shut, producing no dangerous pressure. Generally speaking, at a constant speed with increase in quantity the pressure produced falls, or, in other words, a reduction in the pressure which the pump has to overcome will result in a greater quantity being delivered. Great care should, therefore, be taken in ascertaining, as accurately as possible, the head for which a pump is to be built, as otherwise the pump will possibly not come up to expectations as regards capacity; or if the head upon which the designer based his calculations is greater than the actual head against which the pump has to work, a larger quantity will be delivered by the pump and the load on the driving motor increased.

No bearings should be permitted inside the pump; they should be carried in brackets outside the casing. Stuffing-boxes should not be used as bearings.

EFFICIENCY OF CENTRIFUGAL PUMPS.

Under specially favourable conditions a centrifugal pump made by a first-class firm should attain efficiencies up to 86 per cent.

Centrifugal pumps sometimes will pick up water for a few seconds, as shown by the discharge gauge, and then lose the suction when the discharge valve is open. The reason for loss of suction generally is a large amount of air which is drawn in through the glands or suction pipe in some way. Air sometimes may be drawn in through the suction pipe, although it is entirely submerged beneath the level of the water. This is due to some extent to the vortex or whirling of the water around the suction pipe, thus forming an open passage to the air above. The cure lies in reducing the velocity of water in the suction pipe, either by adding a bell-mouth or tee, etc., or by drilling a number of holes into this pipe.

Packing of Centrifugal Pumps.

About nine-tenths of the trouble with centrifugal pumps is due to improper packing. Square flax is generally used, and this soon wears hard, cuts the shaft, leaks badly, and has to be replaced frequently at considerable expense. Soft-flax packing soaked in oil and graphite gives better results, but it lasts only a short time. A 5 in. double-suction centrifugal pump direct-connected to a 15 horse-power motor running at 1,800 revolutions per minute, supplying a barometric condenser, experienced considerable trouble owing to the packing wearing out, the pump taking air,

losing the water, and the vacuum ; this pump being packed with rawhide packing, after two months' continuous running was found not to leak. The rawhide packing is very economical, as the old packing does not have to be pulled out, and a new ring can be put in on top of the old and kept in the box till it is completely worn out, also the shaft is kept in a good condition. (*Power.*)

The 'United States' Patent Rotary Packing.

This packing is intended primarily for rotating shafts of centrifugal pumps, etc., and its principal feature is that it causes no friction whatever to the shaft, whilst providing an arrangement which will run for a considerable period without requiring renewal. It tends to increase the efficiency of the pump, and obviates any wear on the shaft.

The packing is made in two portions, one portion being for attachment to the casing, whilst the other is clamped to the shaft and rotates therewith. In this rotating portion the packing rings proper are fitted, and as these are not subjected to any friction they should last indefinitely. The two portions of the packing are always held in contact, and the friction which arises is restricted to the two surfaces (as between the fixed and the rotating portions) thus affected, but as one of these surfaces is of anti-friction metal, and provision is made for lubrication at this point, the wear is slight, and, as above stated, the packing gives considerable service before renewal of the anti-frictional part becomes necessary.

To facilitate the fitting of this patented packing, the component parts can be supplied in bisected form, when so required.

A special lubricator is furnished at slight additional cost. This is pressure operated, and maintains a feed of oil to the surfaces in contact all the time the pump is running, but ceases to operate when the pump is stopped.

It is claimed that a 20 per cent. increase in efficiency of centrifugal pumps has resulted from the adoption of this device.

The packing is absolutely air-tight, whilst it causes no friction whatever to the shaft.

Solid Matter Centrifugal Pumps.

See Vol. CCVIII. (Dec. 1920), 'Minutes of the Proceedings of the Institution of Civil Engineers,' pages 275–346, and *Engineering*, Jan. 20, 1922.

The impellers in these pumps have flexible blades.

The Pulsometer Centrifugal Pump.

(*The Pulsometer Engineering Co., Ltd.*)

This pump, fig. 24, is specially adapted for low lifts at moderate speeds. The suction is central at one side of the pump. The discharge branch can be fixed at various angles to suit the direction of the pipes. The bed, casing, and impeller are of cast iron. Gun-metal impeller, slip rings, renewable bronze sleeve on shaft, or rustless steel shaft can be supplied at an extra price. Sufficient thickness of metal is allowed in the covers where the impeller is in running contact to permit of their being bored out and slip rings fitted when wear takes place. When dealing with dirty water slip rings should be fitted in the first instance. The shaft, which is of high carbon steel and of large diameter, is of ample strength to permit of driving by the overhung pulley. The bearings are of cast iron lined with white metal, the lubrication being continuous by means of a revolving ring dipping into a large reservoir fitted with oil overflow and drain. Creeping of the oil is prevented by collars fixed on the shaft. The stuffing box is fitted with a gun-metal

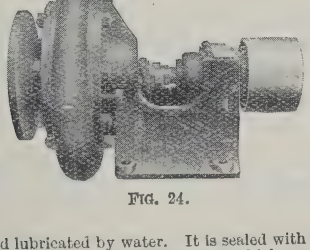


Fig. 24.

neck bush, and is cooled and lubricated by water. It is sealed with water under pressure, which prevents the ingress of air when the pump is working with a high suction lift, but the stuffing box can be sealed with grease by fitting a lubricator for semi-solid lubricant.

Centrifugal Pumps for Hot Water.

Centrifugal pumps, when designed for pumping hot water, are quite satisfactory for feeding boilers. When used for this purpose, however, the intake pipe must be about three times as large as it would ordinarily be for cold water. Sharp turns and constricted passages, both in the pump itself and in the connecting pipework, must be avoided in order to prevent the release of steam at the constricted points. This trouble generally occurs when there is a sudden demand for water. In addition, the efficiency of the pump is lowered and difficulty is also encountered in maintaining the supply of water. (See also p. 1268.) (*W. E. Warner, 'Cheap Steam,'* July 1927.)

The 'Glenfield' Tilting Disc Valve.

(*Glenfield & Kennedy, Ltd., Kilmarnock.*)

In the course of the last few years centrifugal pump design has been receiving specialised attention, with the result that an ever-increasing degree of efficiency has been obtained, with a corresponding reduction in operation costs.

In the case of valves, however, more especially of the Non-return or Reflux type, progress in design has not received the same degree of attention as that of the pumps. In the usual types of non-return valves, the head losses may, and in many instances actually do, reach altogether undue proportions.

Particularly is this so in the case of large sizes under low head operation, where, owing to considerations of capital expenditure, pipe diameters are cut down and velocities correspondingly increased, power costs due to wasted energy may amount to a considerable annual sum, especially where pumps are in continuous operation.

Glenfield and Kennedy, Ltd., Kilmarnock, with the assistance of their associates—The British Pitometer Company, Ltd., London—who specialise in flow measurements and efficiency tests, have investigated the question of such losses in circulating water plants, and the results have shown the great desirability of the use of a non-return valve designed on correct hydraulic principles with a view to eliminating turbulence, and especially to prevent the normal straight flow of the water being deflected, as every change in direction of flow, whether stream-lined or not, involves a corresponding hydraulic loss.

To conform with these conditions they have developed a non-return valve of the Tilting Disc type, which is particularly adapted for large or medium Centrifugal Pumping plants, and exceptional success has been attained by their adoption.

There is only one moving part, the door, which is made in cast steel of an improved aeroform type. The seat rings are of hard gunmetal formed to a wide 'Non-binding' seat angle. The trunnions are fixed and are of gunmetal, the door being bushed to receive them, and grease-gun lubrication is provided through the trunnions so as to eliminate grit inclusions. The outer seat is pressed into a separate ring, which takes all thrust, and is in turn held between the cast-iron body flanges.

Turbine Pumps.

The 'Pulsometer' High-Lift Turbine Pump.

(*The Pulsometer Engineering Co., Ltd.*)

In this pump, fig. 25, the axial thrust is taken up inside the pump by a balancing disc fixed on and revolving with the spindle.

The periphery of the disc moves in close contact with a seat fixed to the delivery cover. The space enclosed by the disc and its seat is supplied with water under pressure from the last cell, and this water passes out between the periphery of the disc and the seat into the chamber in which the disc revolves, and then through a connecting pipe back into the suction pipe.

There are no collars on the spindle at the bearings, so that it is free to move axially when the pump is not working, but when working the balancing disc keeps it in position. This system of balancing has the further great advantage that the stuffing box at the delivery end is exposed to suction pressure only, no matter what pressure the pump is working against.

The impellers are not balanced, and the unbalanced area of each impeller causes an axial thrust towards the suction cover. It is this thrust that the balancing arrangement successfully overcomes in the following manner:

The chamber A is supplied with water under pressure from the last impeller, through the clearance B between the spindle and the centre bush in the delivery cover.

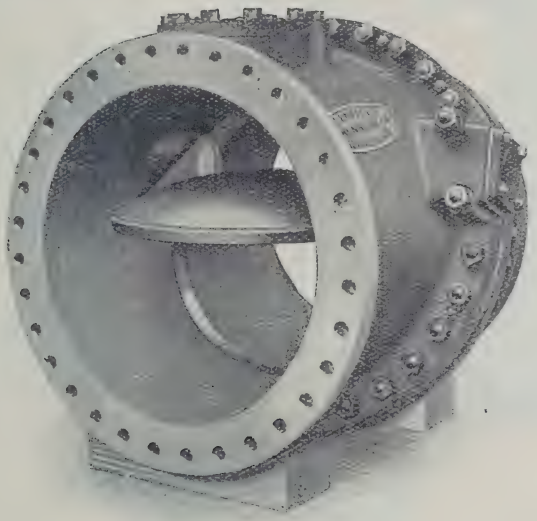
The axial thrust towards the suction causes the clearance C between the balancing disc and its seat to close up, so that more water enters the chamber A than leaves it, and the pressure in this chamber therefore increases until the axial force on the balancing disc towards the delivery cover balances the axial thrust of the impellers towards the suction cover.

As the axial thrust from the impellers increases, the clearance C closes up, and the pressure in the chamber A increases until equilibrium is again restored.

"GLENFIELD"

DROP TIGHT TILTING DISC VALVE

(Patented throughout the World).



Supplied to leading Power Stations and Waterworks Undertakings.

Universally applicable for every situation where flow —whether it be of gases, vapours or liquids—is required in one direction only.

MAKERS:

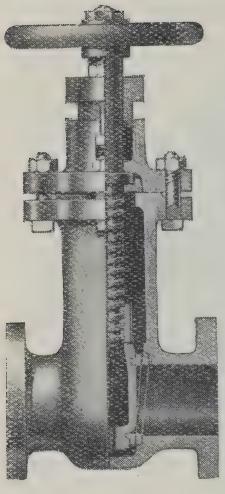
Brochure on Request

GLENFIELD & KENNEDY LTD

KILMARNOCK, SCOTLAND.

F. 1270

"GLENFIELD" VALVES



Of Every Description— For Every Service.

"Glenfield" Valves are made in the largest works in the British Empire specializing in Water Works material.

Area of Works	...	29 acres.
Productive Capacity for Valves	...	90,000 per annum.
Weekly Capacity of Foundry ...	...	400 tons.
Average number of Employees	...	2,000.

If you want the best

BUY "GLENFIELD" VALVES

ENQUIRIES SOLICITED.

GLENFIELD & KENNEDY, LTD.

KILMARNOCK.

F. 1271

As the pressure in the chamber A increases, the clearance C opens, and the pressure in the chamber A is reduced until equilibrium is restored.

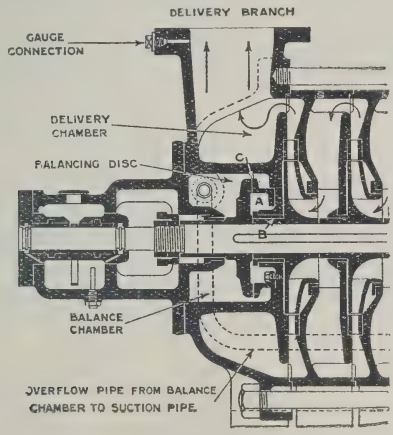


FIG. 25.

The balancing disc is made so large that there is no danger of its coming into contact with its seat even under the most unfavourable conditions of pumping, such as when the pump is running at full speed with no head.

As there is no metallic contact there is no wear, except when gritty water is pumped, and when from this cause the wear has become excessive, the disc or its seat, or both, can be replaced

FIG. 26.

in a very short time and at small cost, without dismantling the pump, it being only necessary to remove the bearing.

Fig. 26 shows a general view of one of these pumps.

THE STONE TURBINE PUMP.

(J. Stone & Co., Ltd., Deptford, S.E. 14.)

These pumps are made in seven sizes, varying in capacity from 100 to 1,500 gals. of water per min., the maximum head on the smallest size, with eight stages, being 360 ft., and with the largest size 1,600 ft.

The construction adopted is of the sectional type (fig. 27), which enables the pump to be built up to any number of stages up to the maximum. The sectional construction also possesses the very great advantage that leakage between the stages can be at once detected, and the pumps can be easily dismantled on site. This construction is usually preferred by mining engineers to the solid-casing type of pump, as with the latter it is impossible to detect leakage between the stages, and also, with the solid-casing type of pump, internal rusting is likely to cause difficulty in dismantling.

The impellers and guide vanes are usually supplied in Stone's special 'No. 1 Bronze,' which is free from zinc and has proved very successful in withstanding the action of gritty and acid mine waters. The shafts of the pumps are of large diameter, and made in nickel steel, protected, where coming into contact with the water being pumped, by renewable bronze sleeves.

Curves, figs. 28 and 29, show the characteristic curves of the pumps plotted on a percentage basis, and from which curves the performance of any individual pump, when working on any other duty than normal, can be readily seen.

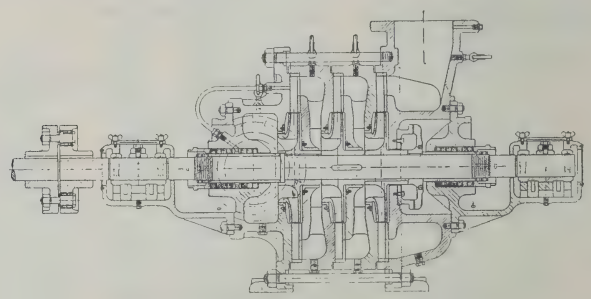


FIG. 27.

Example.—An MS. 6-2 stage pump is designed for 550 gals. per min., and 200 ft. head, with an efficiency of 72 per cent., taking 46·5 b.h.p. at coupling to drive. What would be the performance at a head of 170 ft. ?

170 ft. corresponds to 85 per cent. of the normal head, and referring to the curve marked 'Head Quantity,' for pump sizes 6 ins. and above, the curve shows that the capacity has increased to 119 per cent. of normal, or, say, 655 gals. per min. Referring to the 'Efficiency' curve for this

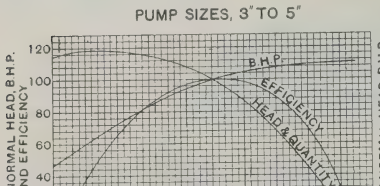


FIG. 28.

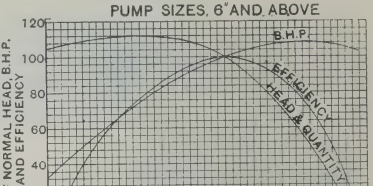


FIG. 29.

percentage increase in capacity, the curve shows that the efficiency is 95 per cent. of the normal, which is 95 per cent. of 72 per cent., or, say, 68 per cent. efficiency. From the 'Brake-horse-power' curve it will be seen that the power required to drive is increased to 106 per cent. of normal, or, say, 49·5 b.h.p.

The pumps have the advantage that neither the suction nor delivery gland is under high-pressure water, the gland at the suction end being water-sealed, and at the delivery end the balance valve reduces the pressure across the gland. By this construction the stuffing-boxes need only be packed lightly, to reduce leakage to the minimum amount; it also means less friction and absence of wear on the shaft sleeves, and consequent higher pump efficiency.

The end thrust is compensated by the standard design of balance valve. Both the balance valve and seating are made in hard bronze and are easily renewable.

The pumps are designed to give self-regulation and high efficiency over a wide range, as will be seen from the two characteristic curves, figs. 28, 29.

The outputs of the pumps at a speed of 1,450 r.p.m. are given in the 'Table of Outputs,' and in connection with the horse-powers shown the following allowances for contingencies should be made in fixing the power of the driving unit:

25% up to 3 B.H.P. 15% up to 10 B.H.P. 7½% up to 100 B.H.P.
20% „ „ 7 „ 10% „ „ 50 „ 5% above „ „

For speeds other than 1,450 r.p.m. the following rules approximately apply:

'Head' varies directly as the square of the speed; 'Capacity' varies directly as the speed B.H.P. varies directly as a product of R.P.M. and head.

TABLE OF OUTPUTS AT 1,450 R.P.M. OF STONE'S HIGH-EFFICIENCY TURBINE PUMPS.

Pump Size.	Suction Branch Diameter.	Delivery Branch Diameter.	Quantity. Galls. per Minute.	Head in Feet.	Number of Stages.						
				B.H.P. Required.	2	3	4	5	6	7	8
	Ins.	Ins.									
MS.3	3	3	100	Head in feet.	90	135	180	225	270	315	360
				B.H.P. required.	4·50	6·80	9·00	11·30	13·5	15·80	18·00
MS.4	4	4	220	Head in feet.	126	189	252	315	378	441	504
				B.H.P. required.	13	19·5	26	32·5	39	45·5	52
MS.5	5	5	340	Head in feet.	160	240	320	400	480	560	640
				B.H.P. required.	24·3	36·4	48·6	60·75	73	85	97
MS.6	6	6	550	Head in feet.	200	300	400	500	600	700	800
				B.H.P. required.	46·5	70	93	116	140	163	186
MS.7	7	7	750	Head in feet.	250	375	500	625	750	875	1000
				B.H.P. required.	77·00	115·50	154·00	192·50	231·00	269·50	308·00
MS.8	8	8	1,000	Head in feet.	300	450	600	750	900	1,05	1,200
				B.H.P. required.	117	175	233	291	349	407	465
MS.9	10	9	1,500	Head in feet.	400	600	800	1,000	1,200	1,400	1,600
				B.H.P. required.	227	340	445	568	680	795	910

The outputs given in the above table are based on cold water, with a suction lift not exceeding 20 ft.

The 'VSG' Mark III. Variable Delivery Pressure Pump.

(The Variable Speed Gear, Ltd., 6 Broadway, Westminster, S.W. 1.)

This pump is of the 'Williams-Janney' or horizontal piston type. Fig. 30 shows a vertical section through the latest design (arranged for lever control).

The 'mainshaft' is driven in one direction, and carries a rotating block or 'cylinder barrel' mounted on it by a universal joint and which is free to move axially.

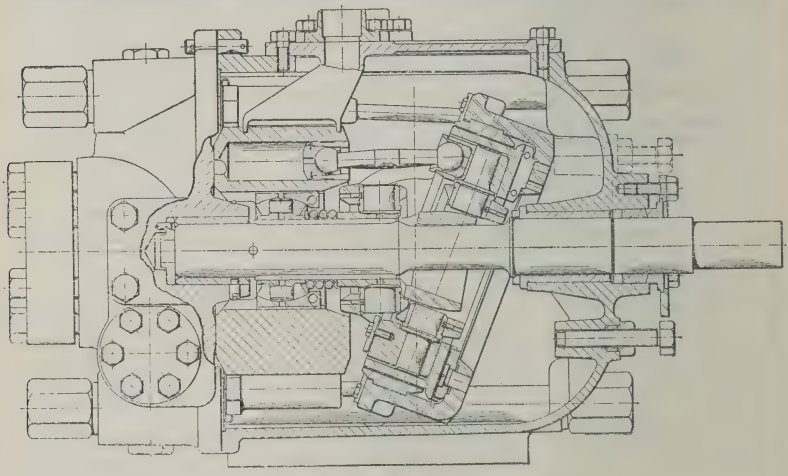


FIG. 30.

This 'cylinder barrel' contains cylinders bored parallel with the mainshaft, having ports of reduced section opposed to two kidney-shaped ports contained in the face of the 'valveplate,' which is stationary.

The 'cylinder barrel' rotates in contact with the 'valveplate' face in a state of hydraulic balance, initial contact being obtained by spring loading.

Each cylinder is fitted with a 'piston' connected by a ball-ended rod to the 'socket ring,' which in turn is connected to the 'mainshaft' by a double universal joint.

The 'socket ring' rotates in a non-rotatable but pivoted member called the 'tilting box,' 'Michell' type bearings being interposed to take the thrust and radial load. When the 'socket ring' is vertical the pump is in the neutral or no-stroke position, but any deviation from the vertical imparts stroke to the 'pistons' and causes delivery through one of the 'valveplate' ports and return through the other.

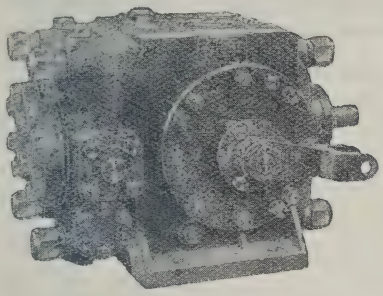
The greater the tilt the more delivery given, owing to the increased travel of the pistons. By reversing the angle of tilt the flow will also be reversed, and what was previously the delivery port will become the return port.

Lever or wheel controlled pumps can be arranged to give delivery in both directions from zero to maximum, but in the case of pumps fitted with auto-pump control, as applied to operate rams, etc., tilt is usually applied in one direction only, and the flow from these pumps is therefore not reversible.

Two distinct types of pumps can be supplied, viz. :—

Lever or Wheel controlled Pumps.	To give variable delivery at fluid pressures up to the maximum pressure at which the relief valves are set, controlled by hand or mechanical means.
Automatic Pressure controlled Pumps.	To give automatically regulated delivery over a predetermined pressure range, or alternatively at a practically constant pressure. Delivery is controlled by the pressure of the fluid in circulation acting on rams connected to the top of the tilting box opposed to springs acting on the base of the tilting box.

This pump also forms the 'A' end unit of the hydraulic transmission gear manufactured by this Company (*The Variable Speed Gear, Ltd.*), described on pp. 948-950.



"VSG Mark III" Variable Delivery Pressure Pump fitted with Lever Control.

"VSG"

VARIABLE DELIVERY PRESSURE PUMPS

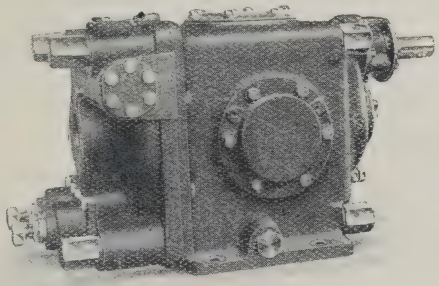
SUITABLE FOR ALL SYSTEMS EMPLOYING OIL AS THE HYDRAULIC MEDIUM

Two types of Pumps can be supplied :—

1. To give variable delivery at fluid pressures up to pre-determined maximum, controlled by hand or mechanical means.
2. To give delivery over a pre-determined pressure range, controlled automatically.

Extensively applied for operating Gun Mountings, Steering Gears of Naval and Mercantile Vessels, also Hydraulic Presses, Shears, Riveters, Control Systems, Lifts, etc.

THE VARIABLE SPEED GEAR LIMITED
6, BROADWAY, WESTMINSTER, LONDON, S.W. 1.
Telephone: Victoria 6900. *Telegrams: "Varispeed, Sowest, London."*



"VSG Mark III" Variable Delivery Pressure Pump fitted with Auto-Control and Internal Reduction Gearing.

NEW TECHNICAL BOOKS

The Strength of Shafts in Vibration

By Captain J. MORRIS, B.A., A.F.R.Ae.S. (late R.A.F.). Formerly Scholar of Corpus Christi College, Cambridge. 200 pp., Illustrated. Crown 4to, cloth ... ... *Net* **30s.**

An original work on a subject of outstanding importance to engineers and students. The results of rigorous applied mathematics reduced to relative simplicity by a suitable choice of notation and by the methods of attack adopted. The work constitutes an extensive monograph on resonant vibration in shafts, much of which is now published for the first time.

Railway and Seaport Freight Movement

Comprising Modern British and American Practice. By GEORGE BULKELEY, A.M.I.Mech.E.; Port Manager, Mombasa and Kenya Coast; formerly on Personal Staff of the General Manager, Great Western Railway, England; Author of "Mechanical Appliances for Handling Railway Traffic." With an Introduction by His Excellency Sir EDWARD GRIGG, K.C.M.G., K.C.V.O., D.S.O., M.C., Governor of Kenya and High Commissioner for Transport in Kenya and Uganda. Profusely Illustrated with Photographs and Diagrams. 240 pp. Royal 8vo, cloth ... ... *Net* **28s.**

Design of Merchant Ships and Cost Estimating

A Treatise on Ship Design and Cost Estimating, giving up-to-date methods of arriving at correct proportions, form and power to attain minimum Capital Cost with maximum service efficiency. The work is intended for the use of Naval Architects, Marine Surveyors and Consulting Engineers, Ship Draughtsmen, Students and all who are interested in the advancement of Shipbuilding and the Mercantile Marine. By ALEXANDER KARI, M.Sc. (Dunelm), A.M.I.N.A., Associate Member of N.E.C. Institution of Engineers and Shipbuilders. Third Edition. Revised. 320 pp. 112 Illustrations. 104 Tables. Also Diagrams and Charts. Royal 8vo, cloth ... ... *Net* **36s.**

Naval Architect's and Shipbuilder's Pocket-Book

Of Formulæ, Rules, and Tables, and Marine Engineer's and Surveyor's Handy Book of Reference. By CLEMENT MACKROW, M.I.N.A. Thirteenth Edition, Revised, by LLOYD WOOLLARD, M.I.N.A., Royal Corps of Naval Constructors. 744 pages. Diagrams. Fcap. 8vo, limp ... ... *Net* **21s.**

Modern Bridge Construction

By F. JOHNSTONE TAYLOR. Author of "Water Power Practice," "River Engineering," etc. 232 pp., with 118 Illustrations. Demy 8vo ... ... *Net* **15s.**

CROSBY LOCKWOOD & SON

Stationers' Hall Court, London, E.C. 4.

F. 1275

Pumps for Oils and Viscous Fluids.

The 'Stone-Paramor' Rotary Pump.

(J. Stone & Co., Ltd., Deptford, S.E. 14.)

This is a rotary pump specially designed for dealing with heavy fuel oils, lubricating oils, and all industrial viscous fluids. A special feature is that sliding contact between the inner and the outer rotor is reduced to a minimum, there being a high degree of rolling action.

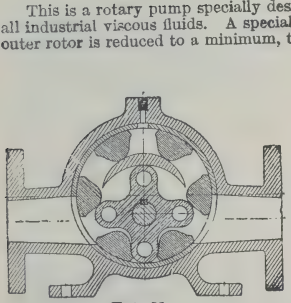


FIG. 31.

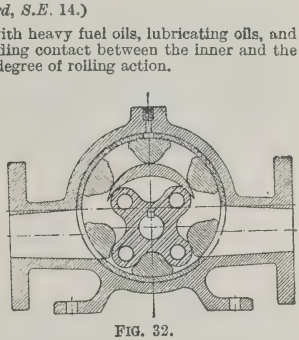


FIG. 32.

Figs. 31 and 32 show the rotors in two different positions. It will be observed that the shape of the teeth being cycloidal (as they are generated in accordance with the relative rolling action between inner and outer rotors), the teeth of the inner rotor make continuous rolling contact with the teeth or abutments of the outer rotor.

Particular attention has been given to the prevention of leakage in the suction and discharge when operating under heavy pressure. The figs. show that in any relative position of the two rotating members this point is amply provided for, so that the volumetric efficiency of the pump is very high.

In all sizes provided with 3-in. branches and above, the pumps are fitted with ball bearings carried in end brackets, so that the inner rotor is always kept in a central position.

TABLE OF CAPACITIES.

Type.	Diameter of Branches.	Capacity. Tons per Hour.	R.P.M.	
			Min.	Max.
P.R.P. 1	1½ ins.	3	500	600
,, 2	2 ,,	7½	400	550
,, 3	2½ ,,	10	350	500
,, 4	3 ,,	15	300	450
,, 5	3½ ,,	20	300	400
,, 6	4 ,,	30	250	350
,, 7	5 ,,	37	250	350
,, 8	6 ,,	50	225	300
,, 9	7 ,,	75	225	300
,, 10	8 ,,	120	200	250
,, 11	9 ,,	150	175	250
,, 12	10 ,,	200	150	200
,, 13	12 ,,	250	150	200
,, 14	14 ,,	300	125	200
,, 15	14 ,,	375	125	200
,, 16	15 ,,	450	100	150
,, 17	15 ,,	550	100	150

NOTE.—The capacities stated above represent the outputs of the pumps at minimum speed when dealing with heavy Mexican fuel oil.

The maximum speed stated can be employed on less viscous fluids and lubricating oils, but for molasses, heavy oils, tar, etc., the lower speed should be selected.

When dealing with the lighter liquids the outputs of the pumps can be increased in proportion to the speed.

The pumps are in use by many large oil companies for pumping fuel and lubricating oils, and also by many shipowners, as it is particularly adaptable for installing on board ship, also by large industrial concerns where viscous fluids have to be dealt with.

Scoop Wheels.

For raising large bodies of water through a small elevation the scoop wheel is very effective. At Amsterdam the canals are freshened by the drawing of water through them by 8 scoop wheels, discharging to the Zuyder Zee.

Their diameter is 8 m.; breadth, 3·25 m.: capacity, 300 m.^{3} each per minute; speed about 3 revolutions per minute. Floats inclined so that they enter water nearly horizontally, and leave nearly vertically. Height of lift, about 1·5 m.

The tendency to-day is to substitute the centrifugal pump for the scoop wheel even for small elevations.

Water Raising from Deep Wells by Compressed Air.

(1)

In this system, air under the necessary pressure, and of suitable volume, is admitted to the foot of a pipe immersed in the water to be lifted, as shown by fig. 33, p. 1277. By this means, alternate bands or plugs of air and water are formed in the water pipe or 'rising main,' and the air does not, as might perhaps have been anticipated, pass through the mass of water. The result of this is that the aggregate weight of the water in the rising main is insufficient to balance the column of water in the boring, and the pressure of the latter therefore produces an upward movement of the air and water column in the rising main. As the force lifting this column is the head of water at A, it follows that the height to which the water can be lifted depends upon the depth to which the pipe is immersed below the pumping level of the water in the boring: thus the higher the 'lift' required, the deeper must be the point at which the air is injected relatively to the normal water level therein, and consequently the height to which the water has to be raised determines the minimum depth to which the boring must be driven.

The following table shows the approximate horse-power required to compress 1 cubic foot of free air to different pressures per square inch:—

Locality.	Pressure in Pounds per Sq. Inch.	H.p. required to compress 1 Cub. Ft. of Air.	Locality.	Pressure in Pounds per Sq. Inch.	H.p. required to compress 1 Cub. Ft. of Air.
1. Solvay Works, Saaralben	176	0·434	5. Tunbridge Wells Waterworks	60	0·159
2. Brostowe Estate, near Friedheim	140	0·376	6. Sugar Factory, Glogan	45	0·145
3. Tunbridge Wells Waterworks	100	0·201	7. Yard Works, Zwickau	30	0·121
4. Ditto	80	0·189	8. Rossland Mines, B.C.	95	0·134

With regard to the quantity of air required to lift a given volume of water to any specified height, the following formula has been found to give correct results:—

x = cubic feet of free air per minute; A = gallons of water required per minute; B = 'lift' of water in feet;

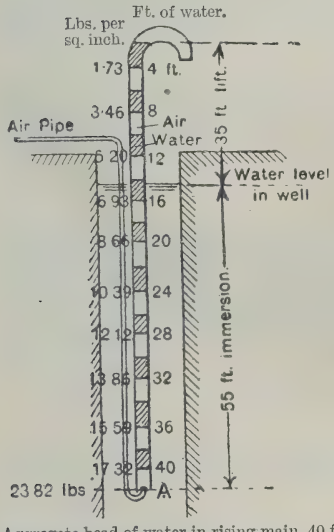
$$x = \frac{A \times B}{125}.$$

The velocity of the air should not exceed about 20 ft. per second.

In considering the design of the compressor, it is obviously important in the interests of economy to provide either 2-stage or 3-stage machines (with intercoolers) according to the amount of compression required; thus for pressures above 60 lbs. per square inch and up to 300 lbs. per square inch 2-stage machines are most suitable, and from this to 1,000 lbs. per square inch 3-stage machines are desirable. The cooling of the air is also of great importance in this respect, and care should be taken to start with an air supply at the lowest temperature obtainable.

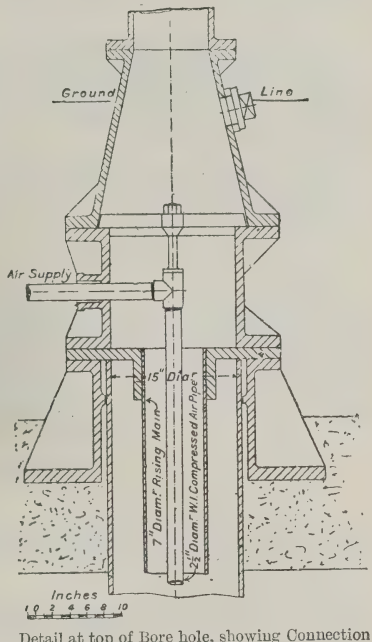
At Tunbridge Wells the compression is carried out in two stages, the first to a pressure of 25 lbs. per square inch, and the second, after intercooling, to the pressure required, which varies from 90 lbs. to 100 lbs. per square inch according to the depth of the water in the boring. The air is delivered into a large steel receiver situated in the engine-house, whence it is conveyed to the wells by means of a 4-inch cast-iron pipe. This main is formed of ordinary spigot and socket pipes with lead joints, which, however, shortly after starting the plant, showed signs of leakage adjacent to the receiver. This was traced to the expansion and contraction of the pipes due to the high temperature of the air, and was effectually prevented by fixing a cooler on the air pipe between the second stage compressor and the receiver.

There are two sets of engines and compressors, designed to deliver a sufficient volume of air to lift the desired quantity of water, if need be, over a standpipe 20 ft. high above ground level.



Aggregate head of water in rising main, 40 ft. = 17·32 lbs. per square inch.
Aggregate head in well = 55 ft. = 23·82 lbs. per square inch.

FIG. 33.



Detail at top of Bore hole, showing Connection of Air Supply.

FIG. 34.

The engines are compound, with cylinders 8 ins. and 12 ins. diameter respectively, the air-compressing cylinders being 10 ins. (first stage) and 6 ins. (second stage) diameter respectively The stroke in each case is 14 ins. The air cylinders are water-jacketed at sides and covers. The clearance in the air cylinders does not exceed 1 per cent. of the capacity. The air is drawn from outside the building, and is carried in an earthenware pipe under the floor to the inlet valves, so that it enters the compressors at the lowest possible temperature.

An intercooler (between first and second stages of compressors), formed of a series of 'Row' tubes cooled by circulating water, is fixed outside the engine-house.

The boring is 350 ft. deep. The upper portion is lined with 15-inch steel tubes, and the lower portion (150 ft.) with steel perforated tubes, 13½ ins. diameter. The rest level of the water

is about 96 ft. from the surface, and the pumping level, when drawing at the rate of 32,000 gallons per hour, is about 120 ft. from the surface, but on the cessation of pumping the water resumes its rest level very rapidly.

The water pipe or rising main is 7 ins. diameter, and is carried to within a few feet of the bottom of the boring. The air pipe was originally 1½ ins. diameter, but was subsequently increased to 2½ ins. diameter, an alteration which reduced the air pressure required from 105 lbs. to 91 lbs. per square inch, the latter figure corresponding very closely to the head of water above the bottom of the air-pipe, and thus proving the loss by friction to be reduced to a minimum.

In a trial run of 10¼ hours made soon after the plant was installed, an average of 31,402 gallons per hour was raised from this boring under a head of 133 ft., whereas an ordinary single-acting pump, such as could be fixed in a boring of this small diameter, would give no more than about 18,300 gallons per hour.

The most favourable result was obtained when the proportion of immersion of air pipe to lift of water was 3 to 1 at the start and 2·2 to 1 at the finish, the volume of air used per volume of water lifted being in the proportion of 2·69 to 1. The efficiency in this case was 36·8 per cent., calculated on the water raised proportionate to the indicated horse-power developed in steam-cylinders, and 46 per cent. calculated from the indicated horse-power shown by air-cylinder diagrams. No record of fuel consumption was made. If the proportion of immersion to lift is reduced the efficiency falls.

From a trial made with a much less immersion of air pipe than in the foregoing, and when the proportion of air used to water lifted was more than double that in the former case, the cost of fuel worked out at 1·073*d.* per 1,000 gallons for a maximum lift of 125 ft. 9 ins., the price of coal being 25*s.* 5*d.* per ton. (*W. H. Maxwell.*)

(2)

In order to enable the well to work successfully and economically, the pipe should be properly proportioned. When the pipe is too large, the water is apt to blow out in a spray and with great force, which is obviously very wasteful both of air and water. When the pipe is too small, the air is apt to rise to the surface of the water in small bubbles, and without producing the desired results. When the air pipe is outside of the water pipe, the proper relation between the diameters of the air and water pipes is as 1½ to 4, which gives a ratio of actual cross-sectional area of 6·24. When the air pipe is situated inside the water pipe, the same ratio of actual cross-sectional areas should obtain, and in this case the diameter of the water pipe is

$$D = \sqrt{\frac{a \times 6{\cdot}24 + A}{{\cdot}7854}},$$

in which a and A represent the actual internal and external areas of the air pipe. The proper areas of the air and water pipes, together with the required depth of submersion and the introduction of the air into the water pipe at the proper point, are the principal factors which determine the success of the air lift.

The water pipe should be submerged to a depth of 18 ins. for each foot lift above the lowest water level in the well; in other words, the best results are obtained when submersion r = 1·5 the lift, the latter being measured from the lowest, or the working water level, as it is called.

When the well is being worked, the water level falls below the level reached, before water is pumped out. To illustrate the foregoing rule, suppose we wish to elevate water 25 ft. above the surface of the ground, and that when drilling, the well water is obtained at a depth of 40 ft. below the surface. The lift is now 40 ft. + 25 ft. = 65 ft. Applying the foregoing rule for depth of submersion, we have 65 ft. × 1·5 ft. = 97·5 ft., or practically 100 ft., thus giving a total height of 165 ft. from the point where the air enters the water pipe to the point of discharge. After drilling to a depth of 165 ft. − 25 ft. = 140 ft., we are ready to find the working water level in the well. First put the water and air pipes into the well; then drop a float attached to a string into the well. Pass the string over a pulley, and attach a weight to the free end. The weight should be just heavy enough to take up the slack in the string, and at the same time allow the float to descend with the water level as the latter is lowered by pumping. The distance the weight rises will indicate the depth of the working level below the level first obtained in the well, and the latter level should be employed when making calculations involving lift and depth of submersion. A very simple as well as convenient arrangement for measuring the fall of the water level in the well can be made as follows: Take a pulley whose circumference is twelve times the circumference of the shaft upon which the pulley is mounted, then wind the float string around

the pulley, and fasten another string to the shaft, so that it will be wound on the shaft as the float string unwinds from the pulley. It will be seen that this arrangement reduces the travel of the string on the shaft to a convenient length, and one that can easily be measured. Suppose the water level in the well falls 10 ft. The corresponding additional depth of submersion will be 10 ft. × 1·5 ft. = 15 ft., and if the well is not deep enough it should be drilled to a sufficient depth to permit the increased depth of submersion to be obtained, and this is done in order to make the total depth of submersion equal to $1\frac{1}{2}$ times the lift, which proportion, as has been pointed out, is necessary for the best results. If it is not practicable to increase the depth of submersion after finding the working level, the height of the lift may be decreased a corresponding amount; otherwise the efficiency of the lift will be to some extent affected. If the well had been drilled deeper than necessary at the outstart, the increased depth of submersion could readily be obtained by simply lowering the pipes to the required depth. It will be seen that the lower water level in the well also determines the necessary change in the point at which the air should enter the water pipe. With a fall of 10 ft., when pumping, the air should enter the water pipe at from 38 ins. to 40 ins. from the lower end of the pipe; with a fall of 5 ft. the air should enter at from 24 ins. to 26 ins. from the lower end of the water pipe, and with a fall of from 1 ft. to 3 ft. it should enter at a point 18 ins. to 20 ins. from the lower end of the pipe. For other distances the same ratio may be used. A well put down to the depth previously determined and fitted with pipes of the sizes indicated by the rule, can be cemented over, and buildings erected over it, without fear of trouble, because the well will need no further attention, and all that is required is simply to supply it with air as long as the pipes last. A number of wells can, of course, be operated from the same compressor. The necessary machinery can be located in the engine-room, or in any convenient place, and the air pipes run to the different points of distribution.

It is estimated that 1 cub. ft. of air will raise 1 gallon of water, but in wells that are properly proportioned there will be a saving over the amount of from 40 to 47 per cent.; that is to say, the air used will be 60 per cent. of the above estimate as the maximum, and 52 per cent. as the minimum. To illustrate the application of the rule for volume of air required, if we wish to raise 100 gallons of water, we shall need 60 cub. ft. of air as the maximum, and 52 cub. ft. of air as the minimum, the required volume carrying from the maximum to the minimum according to the conditions in each individual case. The average air pressure is 60 lbs. per square inch. When starting to raise water, it requires a somewhat higher pressure, because there is a solid body of water from the surface in the well to the point at which the air enters the water pipe, and this column of water has to be moved by the first discharge of air. When the well has reached its normal working condition, the water pipe will contain a number of short columns of air and short columns of water. These short columns of air and water are discharged alternately, and in the pipe they form a continuous chain, the intermittent discharge continuing as long as the proper amount of air is supplied, and as long as there remains water to be pumped.

TABLE OF SUBMERGENCE AND EFFICIENCY.

Submergence per cent. of Length of Water-pipe.	Efficiency per cent.	Submergence per cent. of Length of Water-pipe.	Efficiency per cent.
60	50	33	25
50	40	30	22
40	30		

TABLE OF LIFT, SUBMERGENCE, AND AIR VOLUME.

Lift in Feet.	Volume of Air to 1 ft. of Water.	Submergence 60 per cent. total Water-pipe.	Submergence for special Lift.	Air-pressure Submergence 60 per cent.	Air Pressure for special Lift Submergence 60 per cent.	Cubic ft. free air per gallon per minute.	Horse-power per gallon per minute.
		Ft.	Ft.	Lbs.	Lbs.		
25	2	38	50	17	20	0·3	·0184
50	3	75	100	33	44	0·4	·0426
75	4·5	113	150	49	65	0·6	·0828
100	6	150	200	65	87	0·8	·1320
125	7·5	188	250	82	109	1·0	·1910
150	9	225	300	98	120	1·2	·2544
175	10·5	263	350	115	152	1·4	·3150
200	12	300	400	130	173	1·4	·3808

'Special submergence' (cols. 4 and 6) should be used where there is a long horizontal discharge or a great number of turns in the discharge-pipe.

TABLE OF CAPACITY AND PIPE SIZES.

Size of Well in Inches.	Water-pipe in Inches.	Air-pipe in Inches.	Gallons of Water per Minute.
4	$1\frac{1}{2}$	$\frac{3}{4}$	25
$4\frac{1}{2}$	2	1	50
5	$2\frac{1}{2}$	1	75
6	3	$1\frac{1}{4}$	100
7	$3\frac{1}{2}$	$1\frac{1}{2}$	150
8	4	$1\frac{1}{2}$	200
9	5	2	300
10	6	2	450

(*G. C. H. Fredrich.*)

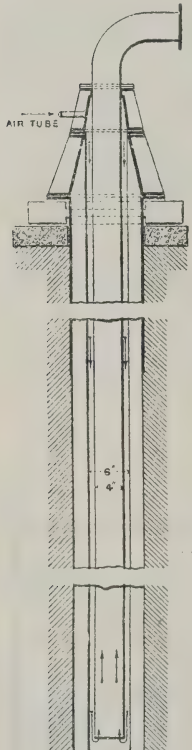


FIG. 35.

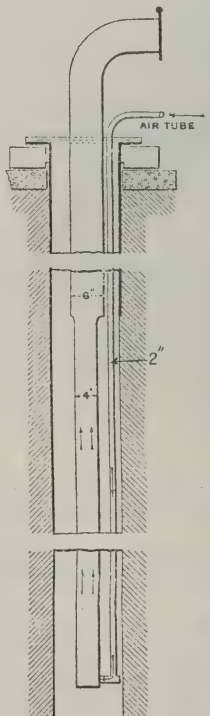


FIG. 36.

At the works of the United Alkali Company, Limited, at Preesall,* a well 450 ft. in depth and 12 ins. in diameter at the bottom is fitted, as shown in figs. 35, 36, with a lining tube going down to the rock-head, and an air tube 6 ins. in diameter, with a delivery tube 4 ins. in diameter inside it, the effective cross-sectional area of the air tube being thus the area of the annulus formed between it and the delivery tube. For working this well a small Ingersoll-Sergeant air compressor was (1906) used.

In another well the delivery and air pipes are arranged on a different plan. In this case the air pipe passes down the well outside the delivery pipe and is connected to it at the bottom, as shown in fig. 35; the air-compressing plant consists (1906) of an Ingersoll-Sergeant cross-compound air compressor, having both the air and the steam cylinders compounded. The speed is regulated by a governor in the usual manner, and also by the pressure of the air as it leaves the high-pressure air cylinder. The air cylinders are water-jacketed, and the air from the low-pressure air cylinder is delivered to an inter-cooler from which it passes to the high-pressure air cylinder. The air suction valves are carried on the piston, the air passing into the piston through a hollow tail-rod working through a stuffing box and gland on the end of the cylinder, and the delivery valves are controlled by suitable spiral springs. The steam cylinders are 20 ins. and 32 ins. in diameter, and the air cylinders 28·25 ins. and 16·75 ins. in diameter, the stroke being 24 ins. The percentage of clearance in the air cylinders is very small. The air is compressed to 35 lbs. per sq. in. in the low-pressure cylinder, and in the high-pressure cylinder to a pressure ranging between 80 lbs. and

* J. Kelly (*Proceedings of the Institution of Civil Engineers*, vol. clxiii. May 1906).

120 lbs. per sq. in., according to the depth of submersion of the delivery pipes in the well or wells under operation.

Mr. Rix, in a paper read before the Technical Society of San Francisco, concludes that, with lifts not exceeding 20 ft., 1·5 cub. ft. of air will be required for each cub. ft. of water raised; for lifts of 20 ft. to 50 ft., 2 to 3 cub. ft.; and for lifts of 50 ft. to 100 ft., 3 to 4 cub. ft. will be required. He limits the velocity of the water entering the pipes to 4 to 8 ft. per second.

Air-lift pumping systems are not advisable where the submergence is below 35 per cent., and the best condition for this type of pump is from 45 per cent. to 60 per cent. submergence. By submergence is meant the ratio of the depth of the mixing chamber below the pumping water level to the total height pumped.

With air-lift pumps the hotter the liquid to be pumped the higher will be the efficiency of the pump. It has been shown by Prof. Randall that an efficiency of 100 per cent. is possible under certain conditions when pumping water at 200 deg. Fah.

SECTION XXV

PART III

THE DEVELOPMENT OF WATER-POWER.

(Partly contributed by Arthur Surveyer, D.Eng., M.E.I.C., M.Inst.C.E.)

WATER-POWER.

The theoretical power or effective energy of a fall of water, in ft.-lbs. per min., is equal to the weight in lbs. of the volume of water flowing per min. over the fall, multiplied by the vertical height of the fall in ft.

$$\text{H.P.} = \frac{Q \times h \times W}{550} = \frac{Q \times h \times 62{\cdot}5}{550} = \frac{Q \times h}{8{\cdot}8},$$

where,

Q = discharge in cub. ft. per sec.; W = weight of 1 cub. ft. of water = 62·5 lbs.; h = net head or net fall of water in ft.; H.P. = horse-power (33,000 ft.-lbs. per min. or 550 ft.-lbs. per sec.).

Thus 1 cu. ft. of water per sec. falling through a head of 10 ft. will develop theoretically 1·134 h.p.

For practical purposes, and in order to take into account the losses in the water-wheels, it is better to adopt the following formula, which is based on a water-wheel efficiency of 80 per cent.,

$$\text{Practical H.P.} = Q \times h \div 11.$$

An efficiency of 80 per cent. is now attained even in the medium-priced wheels, and 90 per cent. is frequently reached in the high-priced wheels. Assuming an efficiency of 88 per cent., then

$$\text{Practical H.P.} = Q \times h \div 10.$$

10 tons of water (about 320 cu. ft.) falling 100 ft. develops the energy equivalent to that which can be produced through the burning of one pound of coal. That is, 10 tons, or 20,000 lbs., falling 100 ft., will develop 2,000,000 ft.-lbs. Dividing this by 778 gives 2,570 B.Th.U. of energy. Assuming a hydraulic turbine efficiency of 92 per cent., which is a fair average for a modern hydraulic turbine, the energy actually developed from this water is about 2,370 B.Th.U. This corresponds to the energy made available by the combustion of coal having a heating value of 13,000 B.Th.U. per lb. with an over-all plant thermal efficiency of approximately 18 per cent., which is perhaps fairly representative of good central-station practice although not equal to the best, which approaches 25 per cent.

Tide-Power.

A horse-power for a day of ten hours would require something like 120 tons of water falling from a height of 100 ft., so that a 500 h.p. factory, say, would need 60,000 tons of water at a 100-foot head. On the basis of 36 cub. ft. of water to the ton, there would thus be over 2,000,000 cub. ft. of water, and this would make a fair-sized pond, say, about 1,000 ft. long, 200 ft. wide, and 10 ft. deep.

The possibilities of tidal power, if it can be developed commercially, are very great.

In Great Britain the highest tides are found in the estuary of the Severn, the mean range of the spring tides at Chepstow being 42 ft., and of the neap tides 21 ft. In France the maximum range occurs at St. Malo, where it amounts to 42·5 ft. at spring tides, and about 18 ft. at neap tides. The tidal range in the Dee is 26 ft. at springs, and 12 ft. at neaps, while the mean range of spring tides around the coast of Great Britain is 16·4 ft., and of neap tides 8·6 ft.

Assuming a mean tidal range of only 20 ft. at springs, and 10 ft. at neaps, and adopting the single-basin method of development with operation on both rising and falling tides, each square mile of basin area would be capable, without storage, of giving an average daily output of approximately 110,000 h.p.-hours. In such an estuary as the Severn, where an area of 20 square miles could readily be utilised with a spring tidal range of 42 ft., the average daily output, without storage, would be approximately 10,000,000 h.p.-hours.

(*'Nature,'* June 3, 1920.)

TABLE SHOWING PRACTICAL HORSE-POWERS CORRESPONDING TO HEADS, IN FEET, AND CUBIC FEET USED PER MINUTE, OF WATER.

(1 cu. ft. per min. = 1·7 cu. metres per hr. 1 cu. metre per hr. = 0·59 cu. ft. per min.)
The practical horse-power is taken at 80 per cent. of the theoretical.

Heads, in Feet.	Cubic Feet used per Minute.	Horse-power.	Heads, in Feet.	Cubic Feet used per Minute.	Horse-power.	Heads, in Feet.	Cubic Feet used per Minute.	Horse-power.	Heads, in Feet.	Cubic Feet used per Minute.	Horse-power.
5	76	·57	14	127	2·7	28	180	7·6	42	220	14·0
6	83	·75	16	136	3·3	30	186	8·4	44	226	15·1
7	90	·95	18	144	3·9	32	192	9·3	46	231	16·1
8	96	1·2	20	152	4·6	34	198	10·2	48	236	17·1
9	101	1·4	22	160	5·3	36	204	11·1	50	241	18·2
10	107	1·6	24	167	6·1	38	210	12·1	55	252	21·0
12	118	2·1	26	173	6·8	40	215	13·0	60	264	24·0
5	152	1·15	14	255	5·4	28	360	15·3	42	441	28·0
6	167	1·5	16	272	6·6	30	373	16·9	44	452	30·1
7	180	1·9	18	289	7·9	32	385	18·7	46	462	32·2
8	192	2·3	20	304	9·2	34	397	20·4	48	472	34·3
9	204	2·8	22	320	10·7	36	409	22·3	50	482	36·5
10	215	3·3	24	334	12·1	38	420	24·2	55	505	42·0
12	236	4·3	26	347	13·7	40	431	26·1	60	528	47·9
5	228	1·7	14	383	8·1	28	541	22·9	42	662	42·1
6	251	2·3	16	409	9·9	30	560	25·4	44	678	45·2
7	270	2·9	18	434	11·8	32	578	28·0	46	693	48·3
8	288	3·5	20	457	13·8	34	596	30·7	48	707	51·4
9	307	4·2	22	480	16·0	36	614	33·5	50	721	54·8
10	323	4·9	24	501	18·2	38	630	36·2	55	758	63·1
12	354	6·4	26	521	20·5	40	646	39·1	60	792	72·0
5	342	2·6	14	574	12·2	28	812	34·4	42	993	63·1
6	376	3·4	16	614	14·9	30	840	38·1	44	1,017	67·7
7	405	4·3	18	651	17·7	32	867	42·0	46	1,040	72·4
8	432	5·2	20	686	20·8	34	894	46·0	48	1,061	77·1
9	461	6·3	22	720	24·0	36	921	50·2	50	1,081	81·8
10	484	7·3	24	751	27·3	38	945	54·4	55	1,136	94·6
12	531	9·6	26	782	30·8	40	969	58·7	60	1,188	107·9
5	418	3·2	14	701	14·9	28	992	42·0	42	1,213	77·1
6	459	4·2	16	750	18·2	30	1,026	46·6	44	1,243	82·8
7	495	5·2	18	795	21·7	32	1,059	51·3	46	1,271	88·5
8	528	6·4	20	838	25·4	34	1,092	56·2	48	1,297	94·2
9	563	7·7	22	880	29·3	36	1,125	61·3	50	1,325	100·3
10	591	8·9	24	918	33·3	38	1,155	66·4	55	1,388	115·5
12	649	11·8	26	955	37·6	40	1,184	71·7	60	1,452	132·0
5	609	4·6	14	1,022	21·6	28	1,442	61·1	42	1,766	112·2
6	668	6·1	16	1,090	26·4	30	1,493	67·8	44	1,808	120·4
7	721	7·6	18	1,156	31·5	32	1,542	74·7	46	1,849	128·7
8	771	9·3	20	1,219	36·9	34	1,589	81·8	48	1,889	137·2
9	818	11·2	22	1,278	42·6	36	1,636	89·1	50	1,929	146·0
10	862	13·0	24	1,335	48·5	38	1,680	96·6	55	2,022	168·3
12	944	17·1	26	1,390	54·7	40	1,724	104·3	60	2,142	194·5
5	799	6·1	14	1,340	28·4	28	1,893	80·2	42	2,317	147·3
6	876	8·0	16	1,431	34·6	30	1,959	88·9	44	2,373	158·0
7	956	10·1	18	1,517	41·3	32	2,023	97·9	46	2,427	169·0
8	1,011	12·3	20	1,600	48·4	34	2,085	107·3	48	2,479	180·1
9	1,074	14·7	22	1,678	55·9	36	2,147	117·0	50	2,532	191·6
10	1,130	17·1	24	1,752	63·6	38	2,205	126·8	55	2,652	220·7
12	1,239	22·5	26	1,824	71·8	40	2,252	136·3	60	2,772	251·7
5	1,027	7·8	14	1,723	36·5	28	2,434	103·1	42	2,979	189·3
6	1,127	10·2	16	1,840	44·6	30	2,519	114·3	44	3,051	203·2
7	1,226	13·0	18	1,951	53·1	32	2,610	126·4	46	3,120	217·2
8	1,299	15·7	20	2,057	62·3	34	2,681	137·9	48	3,186	231·4
9	1,381	18·9	22	2,158	71·8	36	2,761	150·4	50	3,253	246·2
10	1,453	22·0	24	2,253	81·8	38	2,835	163·0	55	3,410	283·8
12	1,593	28·9	26	2,345	92·3	40	2,908	176·0	60	3,564	323·6

TABLE SHOWING PRACTICAL HORSE-POWERS CORRESPONDING TO HEADS, IN FEET, AND CUBIC FEET USED PER MINUTE, OF WATER (*continued*).

Heads, in Feet.	Cubic Feet used per Minute.	Horse-power.	Heads, in Feet.	Cubic Feet used per Minute.	Horse-power.	Heads, in Feet.	Cubic Feet used per Minute.	Horse-power.	Heads, in Feet.	Cubic Feet used per Minute.	Horse-power.
5	1,232	9·3	14	2,067	43·8	28	2,921	123·8	42	3,575	227·2
6	1,352	12·3	16	2,208	53·5	30	3,023	137·2	44	3,661	243·8
7	1,471	15·6	18	2,341	63·8	32	3,132	151·7	46	3,745	260·7
8	1,560	18·9	20	2,469	74·7	34	3,220	165·7	48	3,823	277 7
9	1,657	22·6	22	2,590	86·2	36	3,313	180·5	50	3,905	295·5
10	1,743	26·4	24	2,703	98·2	38	3,402	195·7	55	4,092	340·6
12	1,912	34·7	26	2,815	110·8	40	3,490	211·2	60	4,277	383·8
3	1,389	6·3	10	2,536	38·4	24	3,928	142·7	38	4,944	284·3
4	1,603	9·7	12	2,776	50·4	26	4,089	160·9	40	5,071	307·0
5	1,793	13·6	14	3,000	63·6	28	4,233	179·3	42	5,193	330·0
6	1,963	17·8	16	3,208	77·7	30	4,392	199·4	44	5,316	354·0
7	2,122	22·5	18	3,403	92·7	32	4,536	219·7	46	5,424	377·6
8	2,268	27·5	20	3,588	108·6	34	4,677	240·6	48	5,353	403·3
9	2,408	32·9	22	3,760	125·2	36	4,816	262·4	50	5,665	428·7
3	1,699	7·8	10	3,103	47·0	24	4,806	174·5	38	6,048	347·9
4	1,962	11·9	12	3,398	61·7	26	5,011	197·2	40	6,206	375·6
5	2,192	16·6	14	3,679	77·9	28	5,192	220·0	42	6,357	404·0
6	2,405	21·8	16	3,924	95·0	30	5,364	243·5	44	6,509	433·4
7	2,596	27·5	18	4,161	113·4	32	5,551	268·8	46	6,657	463·5
8	2,776	33·6	20	4,388	132·8	34	5,713	294·0	48	6,801	494·0
9	2,945	40·2	22	4,601	153·2	36	5,887	320·7	50	6,942	525·3
3	2,895	13	9	5,010	68	20	7,475	226	32	9,450	458
4	3,340	20	10	5,285	80	22	7,825	261	34	9,745	501
5	3,735	28	12	5,785	105	24	8,185	297	36	10,025	545
6	4,090	37	14	6,250	132	26	8,520	335	38	10,300	593
7	4,420	47	16	6,685	162	28	8,840	375	40	10,565	640
8	4,725	57	18	7,099	193	30	9,150	415			

GENERAL FEATURES OF A WATER-POWER DEVELOPMENT.

Although the lay-out of hydraulic plants changes with every water-power site, there are certain atures which are generally found in all developments, and which can be described as follows :

1. The dam, which impounds the water, regulates the level of the upper pool, and directs the water towards the water wheels.

2. The head gates, which control the supply of water, and by means of racks permit the removal from the water supply of injurious floating and suspended matter.

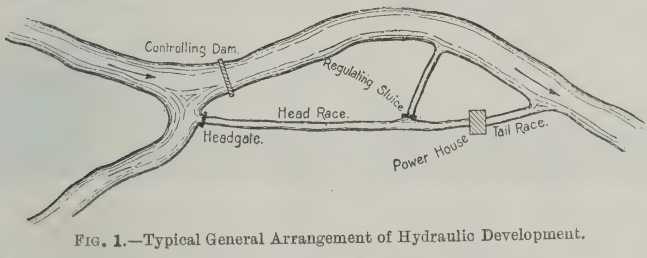


FIG. 1.—Typical General Arrangement of Hydraulic Development.

3. The head race, flume or penstock, which carries the water to the wheels.
4. The power house, which contains the hydraulic machinery.
5. The water wheels, which transform the hydraulic energy.

6. The draft tube, and the tail race, which discharge the water used in the power house into the river.

In some plants the power house forms a part of the dam, and the water is delivered to the wheels without passing through a canal or penstock and discharged back into the river directly through the draft tubes.

Fig. 1 shows a rather complicated lay-out, with a regulating canal for the head race. The regulation of the level of the upper pool is obtained either by a regulating weir, sluices in the dam, or a movable dam.

In approaching the study of any water-power site, the two first things to be determined are the head of water available at all times and the characteristic flow of the river. It is not economical to install machinery which can utilise only the minimum flow of the river, and it is considered good practice to build a plant capable of using the *characteristic flow* of the river, which can be defined as the available dependable flow during at least nine months of a low year.

The available head is determined by levelling and by the daily reading of gauges located in the upper and lower pools of the proposed development.

The *characteristic flow* is found by a series of discharge measurements combined with the daily reading of gauges extending over as long a period as economically possible.

Characteristics of the Flow of Rivers.

The discharge of rivers varies with the precipitation on their drainage basin, the evaporation, the volume of the run off, and the rapidity with which the underground as well as the surface waters reach the river.

The study of drainage basins includes the study of the discharge of streams, of the relations between the stream discharge and the water precipitation, on the drainage basin, on the flood discharges and the rains which cause them, etc.

Discharge of Rivers.

Given a river with an unchangeable bed, there is a fixed relation between the level of this river at a given point and its discharge at that point. Except in the cases of rivers carrying and depositing large quantities of gravel and sand, it is fair to assume that the bed does not change. The discharge curve of a river is obtained from actual measurements, plotted at a suitable scale on cross-section paper. The gauge heights are taken as ordinates and the discharge as abscissæ. The several points thus plotted are connected together by a curve so drawn that the algebraic sum of the squares of the errors is minimum. This curve is a parabola of the form $Q=a+bh+ch^2$. It is often found advisable to develop the discharge equation of a river, and in order to do so the parameters a, b, c can be calculated in each case, from actual measurements, by the method of the least squares.

Measurement of Quantity of Water Flowing in a Stream.

Weir Measurement.

(1)

Rectangular Notch Method.

For small streams a weir dam is used, as shown by fig. 2, consisting of a board long enough to reach across the stream with either end set in the bank. A notch is cut in the board, deep enough

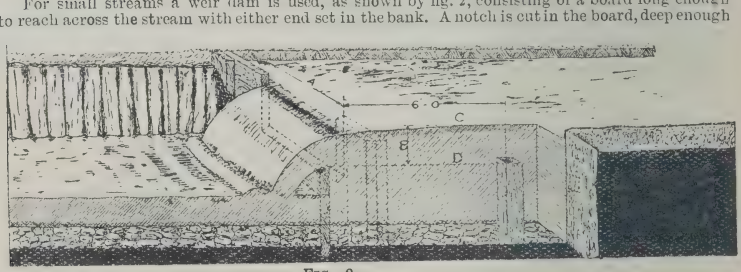


Fig. 2.

to pass all the water, and long enough to reach about two-thirds A, across the stream. The bottom and ends of the notch, D, in the board should be bevelled on the down stream side, leaving the upper edge sharp. The stake, E, should be driven in the bottom of the stream, several feet from the board, on a level with the edge of the notch D. When the water has reached its greatest depth, a measurement, B, can be made of the depth of water over the stake E. The dotted line C represents the top of the running water, and the distance between this line or level of water and the top of the stake gives the true depth or spill over the weir board.

The surface of water below the board should be at least 10 inches lower than the notch D. A part of the stream should be selected where the water can approach the notch quietly.

The weir described above is called a *contracted weir* ; if the sides of the notch are continuous with the sides of the channel it is a *suppressed weir*, or a *weir without end contractions*.

According to the different cases we have the following formulæ giving the discharge in cub. ft. per sec. :

With end contractions and a velocity of approach,

$$Q = 5{\cdot}347\ CL\ \sqrt{(H+1{\cdot}4h)^3}\ ;$$

where water has no velocity of approach,

$$Q = 5{\cdot}347\ CL\ \sqrt{H^3}$$

without end contractions but with velocity of approach,

$$Q = 5{\cdot}347\ CL\ \sqrt{(H+4/3\ h)^3}\ ;$$

where water has no velocity of approach,

$$Q = 5{\cdot}347\ CL\ \sqrt{H^3}\ ;$$

in which

L = Length of weir in ft. ;
H = Head in ft. ;
V = Velocity of approach in ft. per sec. ;
h = Head equivalent of velocity of approach = $0{\cdot}01555\ V^2$;
C = Coefficient of discharge ;
Q = Discharge in cub. ft. per sec.

VALUES OF 'C.'

For Weirs with Contracted Ends. Length of Weir in ft.

Effective Head in ft.	·66	1	2	3	5	10	19
·10	·632	·639	·646	·652	·653	·655	·656
·15	·619	·625	·634	·638	·640	·641	·642
·20	·611	·618	·626	·630	·631	·633	·634
·25	·605	·612	·621	·624	·626	·628	·629
·30	·601	·608	·616	·619	·621	·624	·625
·40	·595	·601	·609	·613	·615	·618	·620
·50	·590	·596	·605	·608	·611	·615	·617
·60	·587	·593	·601	·605	·608	·613	·615
·70	—	·590	·598	·603	·606	·612	·614
·80	—	—	·595	·600	·604	·611	·613
·90	—	—	·592	·598	·603	·609	·612
1·00	—	—	·590	·595	·601	·608	·611
1·20	—	—	·585	·591	·597	·605	·610
1·40	—	—	·580	·587	·594	·602	·609
1·60	—	—	—	·582	·591	·600	·607

For Weirs without Contracted Ends. Length of Weir in ft.

Effective Head in ft.	19	10	7	5	4	3	2
·10	·657	·658	·658	·659	—	—	—
·15	·643	·644	·645	·645	·647	·649	·652
·20	·635	·637	·637	·638	·641	·642	·645
·25	·630	·632	·633	·634	·636	·638	·641
·30	·626	·628	·629	·631	·633	·636	·639
·40	·621	·623	·625	·628	·630	·633	·636
·50	·619	·621	·624	·627	·630	·633	·637
·60	·618	·620	·623	·627	·630	·634	638
·70	·618	·620	·624	·628	·631	·635	·640
·80	·618	·621	·625	·629	·633	·637	·643
·90	·619	·622	·627	·631	·635	·639	·645
1·00	·619	·624	·628	·633	·637	·641	·648
1·20	·620	·626	632	·636	·641	·646	—
1·40	·622	·629	·634	·640	·644	—	—
1·60	·623	·631	·637	·642	·647	—	—

TABLE SHOWING DISCHARGE PER INCH OF LENGTH OVER SHARP-CRESTED, VERTICAL, RECTANGULAR NOTCH WEIRS IN CUBIC FEET PER MINUTE PER INCH OF LENGTH.

Ins.	0	1/16	1/8	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	Ins.
		·006	·01	·03	·05	07	·09	·11	·14	·17	·20	·23	·26	·30	·33	·36	
1	·40	·43	·47	·51	·55	·60	·65	·70	·74	·78	·83	·87	·93	·98	1·03	1·08	1
2	1·14	1·19	1·24	1·30	1·36	1·41	1·47	1·52	1·59	1·65	1·71	1·77	1·83	1·89	1·96	2·02	2
3	2·09	2·16	2·23	2 29	2·36	2·43	2·50	2·57	2·63	2·71	2·78	2·85	2·92	2·99	3·07	3·14	3
4	3·22	3·29	3·37	3·44	3·52	3·60	3·68	3·75	3·83	3·91	3·99	4·07	4·16	4·24	4·32	4·41	4
5	4·50	4·58	4·67	4 75	4·84	4·92	5·01	5·10	5·18	5·27	5·36	5·45	5·54	5·63	5·72	5·81	5
6	5·90	6·00	6·09	6·18	6·28	6·37	6·47	6·56	6·65	6·75	6·85	6·95	7·05	7·15	7·25	7·35	6
7	7·44	7·54	7·64	7·74	7·84	7·94	8·05	8·15	8·25	8 35	8·45	8·55	8·66	8·76	8·86	8·97	7
8	9·10	9·20	9·31	9·42	9·52	9·63	9·74	9·85	9·96	10·07	10·18	10·29	10·40	10·51	10·62	10·73	8
9	10·86	10·97	11·08	11·19	11·31	11·42	11·54	11·65	11·77	11·88	12·00	12·12	12·23	12·35	12·47	12·59	9
10	12·71	12·83	12 95	13·08	13·19	13·31	13·43	13·55	13·67	13·80	13·93	14·04	14·16	14·30	14·42	14·55	10
11	14·67	14·79	14·92	15·05	15·18	15·30	15·43	15·56	15·67	15·81	15·96	16·08	16·20	16·34	16·46	16·59	11
12	16·73	16·86	16·99	17·12	17·26	17·39	17·52	17·65	17·78	17·91	18·05	18·18	18·32	18·45	18·58	18·72	12
13	18·87	19·01	19·14	19·28	19·42	19·55	19·69	19·83	19·97	20·10	20·24	20·38	20·52	20·66	20·80	20·94	13
14	21·09	21·23	21·37	21·48	21·65	21·79	21·94	22·08	22·22	22 35	22·51	22·65	22·79	22·94	23·08	23·23	14
15	23·38	23·53	23·67	23·82	23·97	24·11	24·26	24·41	24·56	24·71	24·86	25·01	25·16	25·31	25·46	25·61	15
16	25·76	25·91	26·06	26·21	26·36	26·51	26·66	26·81	26·97	27·12	27·27	27·43	27·58	27·73	27·89	28·04	16
17	28·20	28 35	28·51	28·66	28·82	28·98	29·14	29·29	29·45	29·60	29·76	29·92	30·08	30 23	30·39	30·55	17
18	30·70	30·86	31·02	31·18	31·34	31·50	31·66	31·81	31·98	32·15	32·31	32·47	32·63	32·80	32·96	33·12	18
19	33·29	33·45	33·61	33·78	33·94	34·11	34·27	34·44	34·61	34·77	34·94	35·10	35 27	35·44	35·60	35·77	19
20	35·94	36·10	36·27	36·46	36·60	36·87	36·94	37·11	37·28	37·45	37·62	37·79	37·96	38·14	38·31	38·48	20
21	38·65	38·82	39·00	39·17	39·34	39·52	39·69	39·86	40·04	40·21	40·39	40·56	40·73	40·91	41·09	41·26	21
22	41·43	41·60	41·78	41·96	42·13	42·31	42·49	42·67	42·84	43·02	43·20	43·38	43·56	43·74	43·92	44·10	22
23	44·28	44·46	44·64	44·82	45·00	45·18	45·38	45·53	45·71	45·90	46·08	46·26	46·43	46·63	46·81	47·00	23
24	47·18	47·36	47·55	47·72	47·91	48·09	48·28	48·46	48 65	48·83	49·02	49·20	49·39	49·58	49·76	49·93	24

The readings obtained at the weir dam are used with the weir table, thus :—

The figures 1, 2, 3, etc., in the first vertical column of the weir table are the inches depth of water running over the notch of weir. The top horizontal line shows fractional parts of an inch ; the body of the table shows the cubic feet and the fractional parts of a cubic foot that will pass each minute, for each inch and fractional inch depth of water in notch. Each of these results is for one inch only of weir, as follows :—

EXAMPLE.—Suppose the notch in board is 20 inches wide and the water at the stake E 5½ inches deep. Take the column containing figure 5 and the one containing figure ½, and the square where they meet contains 5·18 cubic feet. This is the quantity of water passing for each inch in width ; but the weir is 20 inches wide, therefore this result, multiplied by 20, gives 103·6 cubic feet per minute. The same method applies to any depth from one to 25 inches, or any width.

The foregoing only applies in cases where the weir or notch is without end contractions, and discharging from still water.

(2)

'V'-Notch Method.

In this method a 'V'-shaped notch, fig. 3, is cut in the sill board, this notch being of sufficient size to allow the full volume of water to discharge through it. When the flow is steady, then

$$Q = \frac{4}{15}\, kwh\, \sqrt{2gh}, \text{ cubic feet } \textit{per second},$$

w, h, and g being in *feet*.

The value of k varies from ·59 to ·62, according to the angle α, being ·59 for a right angle (90°).

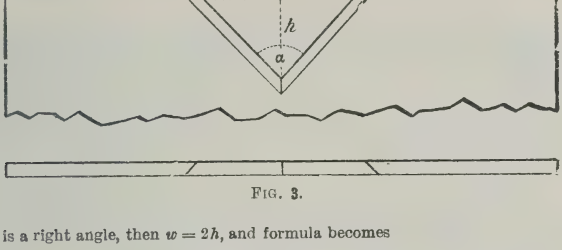


FIG. 3.

When α is a right angle, then $w = 2h$, and formula becomes

$$Q = \cdot 3h^2\sqrt{h},$$ cubic feet *per minute*,

h being in *inches*.

The advantage of the 'V'-notch method is that the ratio of the wetted perimeter of the notch to the area of outflow is the same for any value of h.

A series of 'V'-notches may be cut in the sill-board if the flow of water is great; these notches should be of equal size, and their apexes should be exactly in line, and this line should be exactly horizontal.

When a series of 'V' notches is used, there is a certain amount of mutual interference between the several jets, and in order to get complete immunity from this the notches should be seven times the head apart. If the notches are 3½ times the head apart, the error may be, but would not exceed, ·5 per cent. (*H. S. Rowell.* '*Engineering*,' *May* 2, 1913.)

WEIR TABLE FOR RIGHT-ANGLED 'V'-NOTCH (FROM ONE INCH TO THIRTY INCHES DEEP) GAUGING, GIVING DISCHARGE (Q) IN CUB. FT. PER MIN. FORMULA :—$Q = \cdot 3h^2\sqrt{h}$.

h Ins.	0	1/16	1/8	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16
1	·300	·339	·403	·461	·522	·591	·665	·742	·827	·916	1·01	1·11	1·22	1·33	1·44	1·57
2	1·70	1·83	1·97	2·12	2·28	2·42	2·61	2·78	2·96	3·15	3·35	3·55	3·76	3·98	4·21	4·44
3	4·68	4·92	5·18	5·44	5·71	5·99	6·28	6·57	6·88	7·19	7·51	7·83	8·17	8·51	8·87	9·23
4	9·60	9·98	10·4	10·8	11·2	11·6	12·0	12·4	12·9	13·3	13·8	14·3	14·8	15·2	15·7	16·3
5	16·8	17·3	17·8	18·4	18·9	19·5	20·1	20·7	21·3	21·9	22·5	23·1	23·8	24·4	25·1	25·8
6	26·5	27·2	27·9	28·6	29·3	30·0	30·8	31·5	32·3	33·1	33·9	34·7	35·5	36·3	37·2	38·0
7	38·9	39·8	40·7	41·6	42·5	43·4	44·3	45·3	46·2	47·2	48·2	49·2	50·2	51·2	52·2	53·3
8	54·3	55·4	56·5	57·5	58·7	59·8	60·9	62·0	63·2	64·4	65·6	66·8	68·0	69·2	70·4	71·6
9	72·9	74·2	75·5	76·8	78·1	79·4	80·7	82·1	83·5	84·8	86·2	87·6	89·0	90·5	91·9	93·4
10	94·9	96·4	97·9	99·4	101	102	104	106	107	109	110	112	114	115	117	119
11	120	122	124	126	127	129	131	133	135	136	138	140	142	144	146	148
12	150	152	154	156	158	160	162	164	166	168	170	172	174	176	178	181
13	183	185	187	189	192	194	196	199	201	203	206	208	210	213	215	218
14	220	222	225	227	230	233	235	238	240	243	245	248	251	253	256	259
15	261	264	267	270	273	275	278	281	284	287	290	292	295	297	301	304
16	307	310	313	316	319	323	326	329	332	335	338	341	344	348	351	354
17	358	361	364	368	371	374	377	381	384	388	391	395	398	402	405	409
18	413	416	420	423	427	431	434	438	442	446	449	451	457	461	464	468
19	472	476	480	484	488	492	496	500	504	508	512	516	520	524	528	533
20	537	541	545	550	554	558	563	566	571	575	580	584	589	593	598	602
21	606	611	615	620	625	629	634	638	643	648	653	657	662	667	671	676
22	681	686	691	696	701	706	711	716	721	725	731	736	741	746	751	756
23	761	766	772	777	782	787	793	798	803	809	814	819	825	830	836	841
24	847	852	858	863	869	875	880	886	891	897	903	908	914	920	926	932
25	938	944	950	955	961	967	973	979	985	991	997	1,000	1,010	1,016	1,020	1,028
26	1,030	1,040	1,047	1,050	1,060	1,065	1,070	1,080	1,085	1,090	1,100	1,104	1,110	1,120	1 124	1,130
27	1,136	1,140	1,150	1,156	1,160	1,170	1,176	1,180	1,190	1,196	1,200	1,210	1,220	1,224	1,230	1,240
28	1,240	1,250	1,260	1,266	1,270	1,280	1,287	1,290	1,300	1,310	1,315	1,320	1,330	1,340	1,344	1,350
29	1,360	1,366	1,370	1,380	1,390	1,395	1,400	1,410	1,420	1,426	1,430	1,440	1,450	1,456	1,460	1,470
30	1,480	1,485	1,490	1,500	1,510	1,520	1,530	1,533	1,540	1,550	1,560	1,565	1,570	1,580	1,590	1,600

FLOW OF WATER OVER SHARP-EDGED NOTCHES AND WEIRS.

(1) Flow over a triangular notch.

$$Q = 2{\cdot}48nH^{2{\cdot}47}.$$

Q = cub. ft. per sec.; n = tangent of half-included angle of notch; H = head in feet.

(2) Flow over any trapezoidal notch = flow over a rectangular weir of equal length with two end contractions, plus the flow through a triangular notch of corresponding angle.

(3) Flow over a rectangular weir with end contractions varies as—

$$Q = 3{\cdot}10\,L^{1{\cdot}02}\,H^{1{\cdot}47},$$

for all weirs up to at least 19 ft., and for head up to half length of weir, provided depth of post below sill of weir is not less than twice the head. L = length in feet.

General law for trapezoidal weirs:

$$Q = 3{\cdot}10\,L^{1{\cdot}02}\,H^{1{\cdot}47} + 2{\cdot}48nH^{2{\cdot}47}.$$

(*H. J. F. Gourley and B. Santo Crimp*, '*Proceedings Inst.C.E.*,' vol. cc. p. 388, Nov. 1915.)

See also Dixon & Macauly on 'Measurements of Discharge over a Rock-Faced Dam': Minutes of Proceedings of the Institution of Civil Engineers, vol. 220, May, 1926; and Hubert Mawson, D.Sc., on 'Applications of the Principles of Dimensional and Dynamical Similarity to the Flow of Liquids through Orifices, Notches, and Weirs'; the Proceedings of the Institution of Mechanical Engineers, No. 2, 1927.

Sluice Measurement.

(*Contributed by J. F. Maryon, M.I.Mech.E.*)

Raise the sluice gate, taking care that the level of the water above the sluice gate is constant, and take the three measurements shown, D being the height from the bottom of the sluice gate to the sill over which the water is flowing, W the width of the opening, and H the distance from

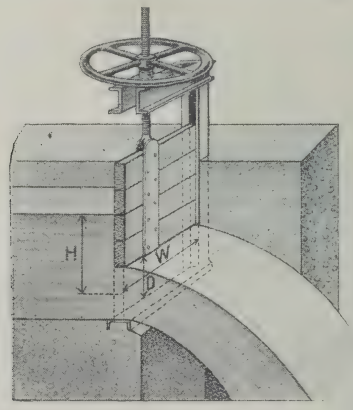


FIG. 4.

the surface of the water above the sluice gate, to a point half-way between the bottom of the sluice gate and the sill. Having obtained these figures in feet, the quantity of water passing in cubic feet per minute is as follows:—

$$Q = W \times D \times 8\sqrt{H} \times 42.$$

Measuring Open Streams.

(*Contributed by J. F. Maryon, M.I.Mech.E.*)

To measure the discharge of a stream it is necessary to obtain by soundings an accurate cross-section of the river at the point of observation. This section is subdivided into a certain number

of sections, A, B, C, D, E (fig. 5), and the mean velocity of flow accurately determined for each sub-section. We have then the relation $Q = AV$; where Q = discharge in cub. ft. per sec., A = total area in sq. ins., and V = mean velocity in ft. per sec., throughout the section.

$$Q = q_1 + q_2 + q_3 + \ldots\ldots = A_1V_1 + A_2V_2 + A_3V_3 + \ldots\ldots$$

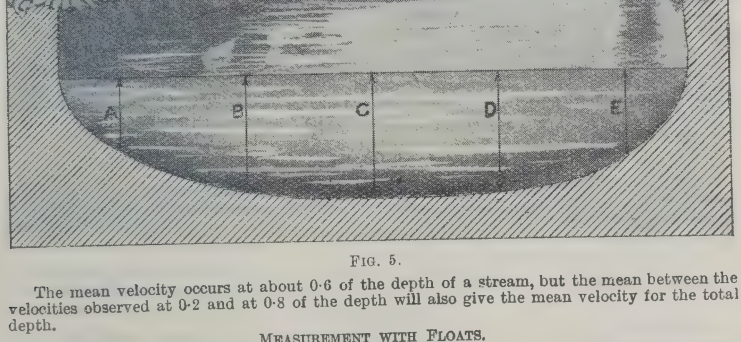


FIG. 5.

The mean velocity occurs at about 0·6 of the depth of a stream, but the mean between the velocities observed at 0·2 and at 0·8 of the depth will also give the mean velocity for the total depth.

MEASUREMENT WITH FLOATS.

In order to measure the velocity of the current in the different sections by means of floats it is necessary to lay out on the shore two parallel ranges perpendicular to the general direction of the current and distant from each other by 200 or 300 ft.

A series of floats are afterwards dropped at a certain distance above the upstream range and their course determined either by transits set up on the shore, or by sextants from a boat kept a few feet distant from each float during its downstream travel.

The passage of each float in line with the upstream and downstream range is recorded with a stopwatch and the mean velocity of the float throughout the distance calculated.

(*a*) *Measurement with Rod Floats.*

If a rod float is made to float vertically at a section of the stream so that its lower end is not more than a foot or so above the bed of the river at that point, such a float will be carried down-stream by the current at a rate corresponding approximately to the mean velocity of the current at the point considered. It is not always practically possible to fulfil this condition, and even with a complete set of hollow galvanised iron rod floats, which can be sunk at will by loading them with shot, it is found necessary to correct the observed velocity and to reduce it to the mean velocity by using the following formula, due to Francis:

$$V_m = V_r\,(1 - 0{\cdot}116\,(\sqrt{d} - x)),$$

where

V_m = mean velocity for the vertical section considered;

V_r = observed velocity of the rod float;

$$d = \frac{\text{distance between bottom of rod and river bottom}}{\text{depth of river}};$$

x is a function of $\frac{l}{d}$, l being the length of the rod and d the depth of the water.

The following shows the values of x corresponding to various values of $\frac{l}{d}$:

$\frac{l}{d}$	0·9	0·8	0·7	0·6	0·5
x	0·10	0·05	0·00	−00·5	−0·10

When $\frac{l}{d} = 0{\cdot}9$ or more, the formula is that given by Francis:

$$V_m = V_r\,(1 - 0{\cdot}116\,(\sqrt{D} - 0{\cdot}1)).$$

(*b*) *Measurement with Submerged Floats.*

It is possible by means of a surface float connected by a string to a submerged float held at a depth equal to about 0·6 of the depth of the stream at that point, to obtain directly without correction the mean velocity in a section. The surface float is generally made very flat so as not to be affected by the wind, and the sub-surface float is generally composed of two pieces of galvanised plate placed cross-wise and terminated at their upper end by hollow tubes in order to give the submerged float the necessary buoyancy.

Measurement by Meters.

The current meters used to measure the velocity of water consist in a wheel of some form so constructed that it is made to revolve by the impact of flowing water, the number of revolutions of this wheel being either recorded, or indicated to the observer. The relation between the velocity of the moving water and the revolutions of the wheel is determined by rating each meter.

Meter Rating.

To rate a meter it is necessary to drive it through still water at a uniform velocity and over a given distance. The distance being known and the time taken to cover it as well as the number of revolutions being noted, it is possible to determine the velocity per second and the number of revolutions per second for this velocity. The experiment must be made at varying speed of translation, so that a complete rating curve can be obtained for each instrument. This curve in all meters is practically a straight line. Meters are not generally reliable for the measurement of velocities of less than 0·2 ft. per sec.

(See also Section XXVII.)

DAMS.

Dams are constructions placed across a stream to impound water, to raise the level of the water upstream, or to divert a part or the whole of the river flow towards the water wheels either by means of a head-race, flumes, or pipes under pressure. Dams can be built with earth, loose stones, timber, masonry, concrete, reinforced concrete, and even with steel. They are fixed or movable according to the needs.

Dams can be built with their crest above the level of the highest waters, and provided with sluiceways, or they can be designed with spillways allowing the water to flow over their crest.

Earth Dams.

Earth dams (fig. 6) can, on account of their large base, be built on less resisting foundations than those constructed of other materials provided, of course, that the bed of the river is impermeable. The section of these dams varies according to local circumstances, and with the materials used. In general the upstream slope is either 3 to 1 or 2 to 1, and the downstream slope about 2 to 1. The centre part of earth dams is often constituted by a core of impermeable material, generally a mixture of sand and clay (two-thirds of sand and one-third of clay). Sometimes this core puddle is replaced by a concrete core wall driven into the bed of the river to an impermeable strata. In designing these concrete core walls, it is necessary to make them strong enough to resist the stresses induced by the settlement of the earth embankment.

Rock-Fill Dams.

Rock-fill dams (fig. 7) are also used on uncertain foundations, because the loose stones tumble into place when erosions take place. They are generally given a fair width at the top, with slopes varying between 1 in 1 and 2 in 1; they have the disadvantage of not being water-tight, although they can be made practically so by placing on the upstream slope an impermeable layer of sand and clay. It is not advisable to allow the water to discharge over these dams, and like the earth dams they must be provided with sluiceways and waste gates.

Timber Dams.

Timber dams (fig. 8) are either hollow or filled; the hollow dam consists of a wooden frame covered by an apron on the upstream and on the downstream side and held in place by the weight of water. The filled dams or crib dams consist of boxes made of large pieces of timber bolted together and filled with loose stones. These dams have generally a nearly vertical downstream face and an upstream face sloping 1 to 1 or 2 to 1; they are held in place partly by their own weight and partly by the weight of the water on top of them. They can be used either on solid foundations or on medium foundations; in the latter case, the upstream and downstream toes should be protected by loose stone, in order to prevent erosion.

Masonry and Concrete Dams.

These dams are used principally when the foundation is rock or hard pan, and are generally built with a practically vertical upstream face and a downstream slope of 1 to 1.

Masonry is used chiefly in the construction of storage dams provided with sluiceways, but concrete can be used either for storage dams or spillways.

Reinforced Concrete Dams.

The characteristics of these dams (fig. 9) are that they are formed by a series of vertical buttresses upon which is placed a reinforced concrete deck, the buttresses being also connected, underneath the deck, by reinforced concrete beams or slabs. These dams can be used either on a rock foundation or on uncertain ground; in the latter case they are generally built over a reinforced concrete mattress distributing the pressure over the foundation. Their upstream face is inclined at an angle of 45° and the buttresses are practically vertical on the downstream side. In the case of a spillway, the downstream side is formed in the same manner as the upstream side, by a reinforced concrete apron of the shape required by the quantity of water which is expected to flow over the crest. These dams have the advantage of requiring much less material than the

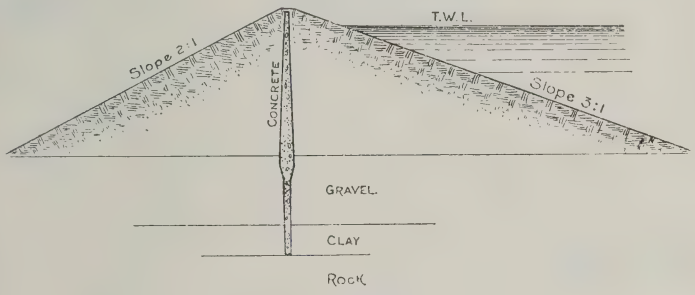


Fig. 6.—Earth Dam.

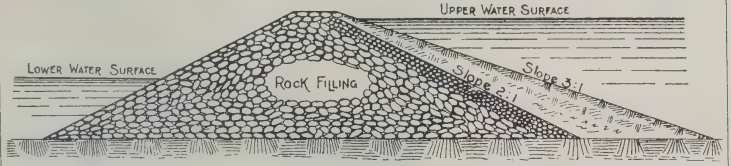


Fig. 7.—Rock-fill Dam.

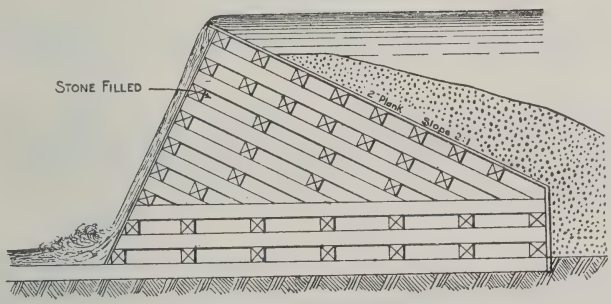


Fig. 8.—Timber Dam.

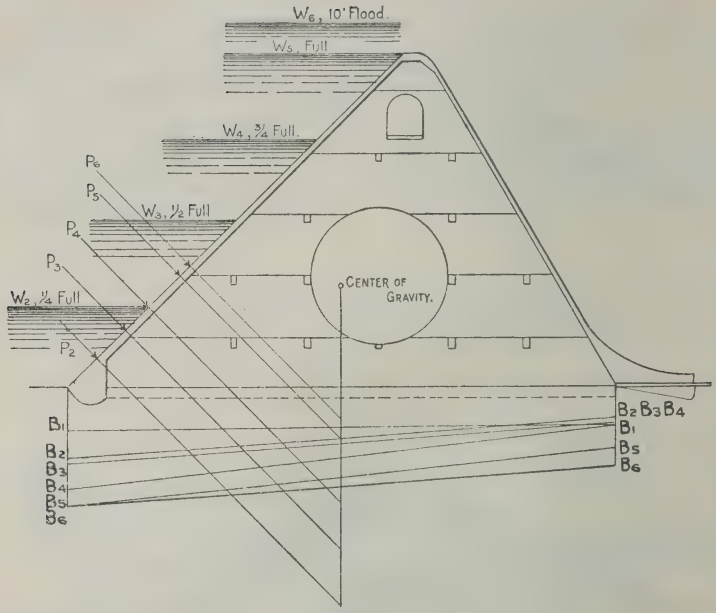


FIG. 9.—Reinforced Concrete Dam.

gravity dams, and are consequently used in places where transportation is difficult. Another advantage also is that their upstream face being inclined at 45° they resist better than the vertical face dams the ice pressures which, on account of their uncertainty, are always difficult to provide for.

Stability of Dams.

When dams are built on rock or solid foundations they can be built up from the bed of the river. If the bed is in hard pan the construction must be started below the bed of the river, and if the bed is composed of sand or gravel the dam generally rests on piles driven into the soil.

CONDITIONS OF STABILITY.

1. The weight of the dam must be sufficient to resist the water thrust tending to overturn it.
2. The dam must be so designed that there is no danger of its sliding upon its base.
3. The pressure on the downstream toe of the dam caused by the overturning moment must not exceed the resistance in compression of either the masonry or the foundations upon which it rests.
4. The resultant of all forces must be such that the upstream face of the dam is not in tension, but rather subject to a compressive stress.

Design of Dams.

Every design of dam must, of course, be tested for the foregoing different conditions, but the following rule will be found useful in arriving at a first tentative determination of the section.

Let h (fig. 10) be the greatest depth of water in the proposed reservoir at the dam. The width of the base B'C' can be taken as equal to the height h or AB', and the theoretical section of the dam would then be represented by the triangle B'AC'. In practice, however, it is necessary to give to the crest a certain thickness which can be taken as equal to one-tenth of the height. The portion AH above the highest water in the reservoir is generally made from 3 to 10 ft. When

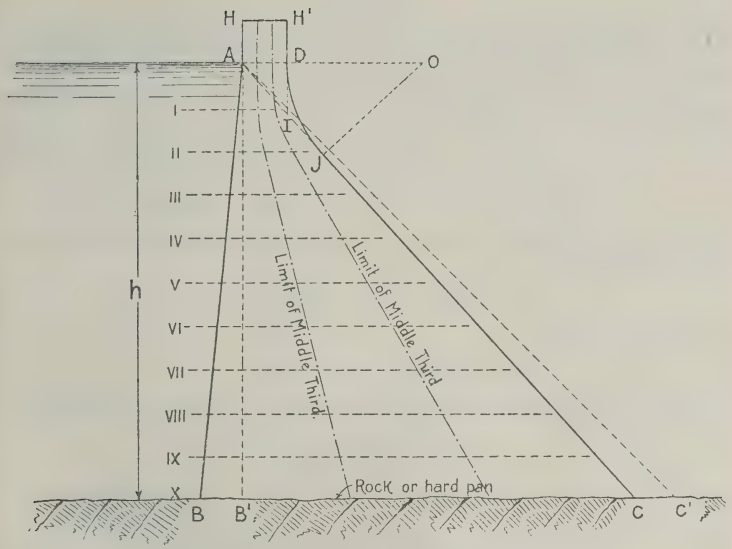


FIG. 10.—Masonry or Concrete Storage Dam.

the dam is very high it is difficult to build the upstream face vertical; it is preferable to give it a batter of 0·10 to 0·20. As it is not necessary to give to the base a greater width than the total height AB', the point C' is therefore brought back to C over a distance equal to BB'.

The downstream face is curved at the upper part into an arc tangent to the vertical H'I at D and to the inclined line AC at J. The section of the dam is therefore represented by the polygon BAHH'DJC.

DESIGN OF SPILLWAY DAMS.

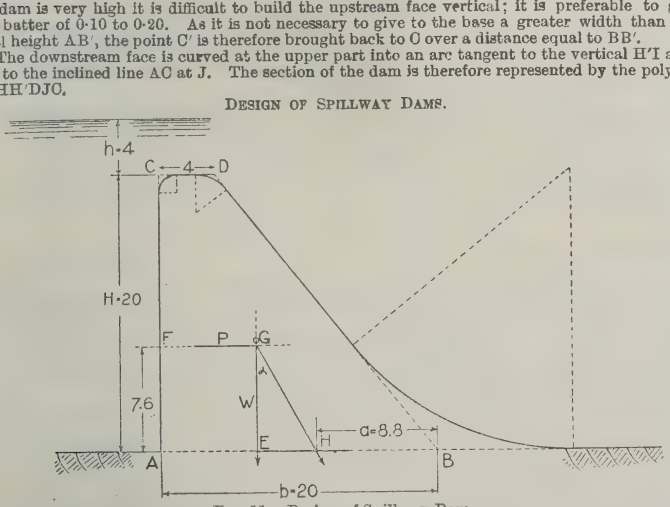


FIG. 11.—Design of Spillway Dam.

Spillway dams, or weir dams, are designed so as to allow the water to flow over their crest. The two first points to decide before beginning the design of such a dam are the elevation of the crest of the dam and the elevation of the highest possible flood.

Assuming that the elevation of the crest is 20 ft. above the bed of the river, and that the greatest flow of the river can be passed over the dam when the water level is 4 ft. above its crest, we can now proceed to design a plain concrete dam to meet these conditions, fulfilling at the same time the requirements as to stability enumerated above.

The width of the crest CD (fig. 11) can be taken as 0·2 of the height of the dam, and the width of the base b equal to the height H.

Referring to the figure we have the following:

$$A = \text{area of section of dam} = \frac{4+20}{2} \times 20 = 240 \text{ sq. ft.};$$

w = weight of 1 cub. ft. of concrete = 130 lbs.;
W = weight of section of dam 1 ft. wide = $240 \times 130 = 31{,}200$ lbs.

This force W acts vertically through the centre of gravity of the section G, whose position is determined graphically.

The water pressure P tending to overturn the dam has for value

$$P = 62{\cdot}5 \times \frac{(H+h)^2 - h^2}{2} = 62{\cdot}5 \frac{H}{2}(H+2h),$$

and in the case in question,

$$P = 62{\cdot}5 \times \frac{20}{2}(20 + 2 \times 4) = 17{,}500 \text{ lbs.}$$

This force P acts horizontally, and is applied at a height AF above the bed of the river given by the following formula:

$$AF = \frac{H}{3} \cdot \frac{H+3h}{H+2h} = \frac{20}{3} \times \frac{20+12}{20+8} = 7'{\cdot}6.$$

Taking the moments about the downstream toe of the dam B we have the following results.

$$\text{Resisting moment} = W \times EB = 31{,}300 \times 13 = 405{,}600 \text{ ft. lbs.}$$
$$\text{Overturning moment} = P \times AF = 17{,}500 \times 7'{\cdot}6 = 133{,}000 \text{ ft. lbs.}$$
$$\text{Factor of safety} = \frac{\text{Resisting moment}}{\text{Overturning moment}} = \frac{405{,}600}{133{,}000} = 3{\cdot}05.$$

A factor of safety of 3 is considered sufficient to fulfil the first condition of stability.

The second condition is that there should be no danger of the dam sliding upon its base; this condition is fulfilled if we have the following inequality:

$$\text{Weight of dam} \times \text{coefficient of friction} > \text{Total water pressure.}$$

The coefficient of friction of masonry on masonry, or masonry or concrete on rock, is 0·7, so that we have

$$31{,}200 \times 0{\cdot}7 > 17{,}500.$$

The resultant R of the forces P and W can be found graphically or algebraically by the following equation:

$$R = \sqrt{W^2 + P^2} = \sqrt{31{,}200^2 + 17{,}500^2} = 35{,}770.$$

Its direction can be found by the equation

$$\tan \alpha^\circ = \frac{P}{W} = \frac{17{,}500}{31{,}200} = 0{\cdot}56;$$

and the point of application H found thus

$$EH = AF \tan \alpha^\circ = 0{\cdot}56 \times 7{\cdot}6 = 4{\cdot}26.$$

This point of application H must fall within the middle third of the base; this condition must be fulfilled whether the reservoir created by the dam is full or empty.

It is advisable to round up the dam at the upstream corner C to prevent chipping by logs or floating ice. It is also advisable to give to the downstream face a form resembling as much as possible the probable curve of the overfalling water. Corner D should therefore be rounded up, and a concave curve built at B, so as to carry the water x away horizontally and without impact. The foregoing method can, of course, be used also in the case of storage dams.

When the proposed dam is over 30 or 40 ft. it is better to subdivide it into a certain number of sections and to test each block separately according to the method described; the resultant in each case must fall within the limits of the middle third. In the example given it has been assumed that there was no upward water pressure at the base; when infiltration and under pressure must be provided for, this new force is subtracted from the weight and combined with it before calculating the resisting moment.

Head Gates.

Head gates (fig. 12) are placed at the entrance of the head race, in order to permit the emptying of the canal and pipes in case of repairs, and in order to be able to stop quickly the admission of water in case of accident to the canal, flumes, pipes, or water wheels.

The head gates are, of course, built differently, according to the size of the opening they have to close.

Head gates for a canal consist of wooden or steel gates moving between concrete, masonry, or crib piers. The weight of the heavy gates is generally partly balanced by a counter weight, and they are raised and lowered by electric or hydraulic motor. The smaller gates are moved by hand with the help of a mechanical hoist.

The piers must be designed to withstand a pressure equal to the total head of water multiplied

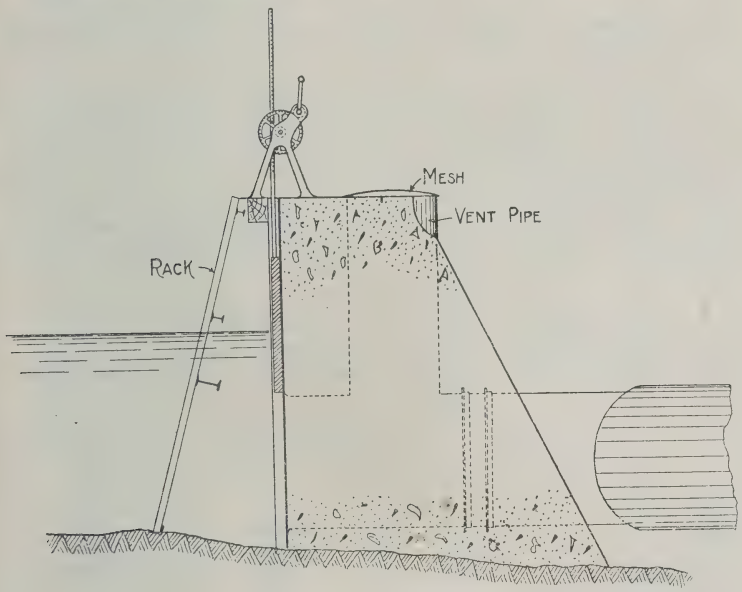


FIG. 12.—Head Gate and Rack.

by the distance between the centres of two adjoining openings. The factor of safety must be at least 3.

When pipes are used to carry the water to the turbines they are generally protected by a concrete head wall designed to resist the full hydrostatic pressure. The gates in this case are much smaller, and are operated by hand.

The force necessary to raise the gate can be calculated as follows :—The total weight of gate in water plus the total water pressure on the gate must be multiplied by a coefficient of friction which varies according to the materials used. To this weight should be added the weights represented by the mechanical losses in the transmission of power, and the grand total increased by about 13 per cent. in order to provide for the extra force necessary to start the gate.

Racks.

Racks are usually made of flat bars having sections from $\frac{1}{16}$ in. by 1 in., to $\frac{3}{8}$ in. by $3\frac{1}{2}$ ins., held together by bolts and separators. These racks may be either of large or small spacing, the former being those in which the bars are from $1\frac{1}{2}$ ins. to 4 ins. apart, and the latter where the bars are from $\frac{1}{2}$ in. to $1\frac{1}{2}$ ins. apart. The spacing is determined by the cross section of the openings of the distributor and runner buckets of the water wheels.

The racks are generally placed at an angle varying from 45° to 60° with the vertical; there should always be an operating platform at least 3 ft. wide, so placed that the attendants can work to best advantage either when cleaning or raising the racks.

The racks should be designed on the assumption that they act as a solid dam, and that all water behind may be drawn off. In such a case the racks and their supports would have to take care of the entire water pressure.

For the purpose of keeping floating *débris* from choking the gratings at the circulating water intakes at an American power station, a pipe has been installed just above water level in front of the grid. The pipe is drilled with ⅛-in. holes, and is supplied with water at 120 lbs. per square inch. The jets of water issuing from these holes wash away any leaves, sticks, and snow that might clog the grating.

Design of Head Races.

The cross-section area of a head race should be such that the water will rise to about three-quarters to seven-eighths of the height.

Wooden flumes are ordinarily rectangular and designed to carry a depth of water equal to about half the width.

Canal sections are determined by the class of material forming the perimeter and the head which it is found economical to use. The efficiency of a cross-section of a canal is greater with the value of the wetted perimeter minimum, so that a semi-circular section is the most advantageous cross-section that can be used in a channel where resistance to flow is alone considered. When the canal is lined with masonry or concrete walls it is more advantageous to make the face nearly perpendicular, and to place the batter of the wall at the back. When the canal is cut in stone or shales which will not be eroded by the flow of the water, a slope of 90° to 40° may be used. In canals dug through earth the slopes may be from 1 in 1½ to 1 in 1½. With artificial banks of earth and gravels the slope must frequently be made as low as 1 in 2.

According to Bazin, the relation between the bottom and the mean velocities in channels is the following:

$$V_b = V - 10{\cdot}87\sqrt{rs},$$

where,

V_b = bottom velocity in ft. per sec.; r = hydraulic radius;

V = mean velocity in ft. per sec.; s = hydraulic gradient or slope of channel.

The following table, calculated from the foregoing formula, gives the safe bottom velocities for different materials:

TABLE OF SAFE BOTTOM VELOCITIES.

Material of Channel.	Safe Bottom Velocity (V_b) in Ft. per Sec.	Mean Velocity (V) in Ft. per Sec.
Soft brown earth	0·249	0·328
Soft loam	0·499	0·656
Sand	1·000	1·312
Gravel	1·998	2·625
Pebbles	2·999	3·938
Broken stone, flint	4·003	5·570
Conglomerate, soft slate	4·988	6·564
Stratified rock	6·006	8·204
Hard rock	10·009	13·127

Flow of Water in Pipes, Flumes, and Channels.

The principal formula of flow in pipes, flumes and channels (see also p. 1308), is based on the formula for velocity of flow known as *Chezy's* formula, which is as follows:

$$V = C\sqrt{rs},$$

where,

V = velocity in ft. per sec.;

C = a coefficient of discharge;

A = area of cross section considered, in sq. ft.;

p = wetted perimeter in ft.;

r = hydraulic radius = $\dfrac{\text{area of section}}{\text{wetted perimeter}} = \dfrac{A}{p}$;

s = hydraulic gradient or slope of channel = $\dfrac{\text{difference of level}}{\text{distance}} = \dfrac{h}{l}$.

h = fall in ft. between points considered;

l = distance in ft. between points considered;

The formula for the flow is $Q = AC\sqrt{rs}$,
where

Q = discharge in cub. ft. per sec.

The value of the coefficient C can be determined either by Kutter's formula or Bazin's formula.

Kutter's formula is as follows :

$$C = \frac{41{\cdot}65 + \frac{0{\cdot}00281}{s} + \frac{1{\cdot}811}{n}}{1 + \left(41{\cdot}65 + \frac{0{\cdot}00281}{s}\right)\frac{n}{\sqrt{r}}}$$

where,
n = the coefficient of roughness.

Robt. E. Horton gives in 'Engineering News,' of February 24, 1916, the following category of coefficients of roughness for use in Kutter's formula :

TABLE OF 'n' FOR KUTTER'S FORMULA.

(*R. E. Horton.*)

Surface.	Perfect.	Good.	Fair.	Bad.
Uncoated cast iron pipe	0·012	0·013	0·014	0·015
Coated cast iron pipe	0·011	0·012*	0·013*	—
Commercial wrought iron pipe, black	0·012	0·013	0·014	0·015
Commercial wrought iron pipe, galvanised	0·013	0·014	0·015	0·017
Smooth brass and glass pipe	0·009	0·010	0·011	0·013
Smooth lockbar and welded 'OD' pipe	0·010	0·011*	0·013*	
Riveted and spiral steel pipe	0·013	0·015*	0·017*	—
Vitrified sewer pipe	0·010 0·011	0·013*	0·015	0·017
Glazed brickwork	0·011	0·012	0·013*	0·015
Brick in cement mortar; brick sewers	0·012	0·013	0·015*	0·017
Neat cement surfaces	0·010	0·011	0·012	0·013
Cement mortar surfaces	0·011	0·012	0·013*	0·015
Concrete pipe	0·012	0·013	0·015	0·016
Wood-stave pipe	0·010	0·011	0·012	0·013
Plank flumes :				
Planed	0·010	0·012	0·013	0·014
Unplaned	0·011	0·013	0·014	0·015
With battens	0·012	0·015	0·016	—
Concrete lined channels	0·012	0·014*	0·016*	0·018
Cement-rubble surface	0·017	0·020	0·025	0·030
Dry-rubble surface	0·025	0·030	0·033	0·035
Dressed-ashlar surface	0·013	0·014	0·015	0·017
Semi-circular metal flumes, smooth	0·011	0·012	0·013	0·015
Semi-circular metal flumes, corrugated	0·0225	0·025	0·0275	0·030
Canals and ditches :				
Earth, straight and uniform	0·017	0·020	0·0225*	0·025
Rock cuts, smooth and uniform	0·025	0·030	0·033*	0·035
Rock cuts, jagged and irregular	0·035	0·040	0·045	—
Winding sluggish canals	0·0225	0·025*	0·0275	0·030
Dredged earth channels	0·025	0·0275*	0·030	0·033
Canals with rough stony beds, weeds on earth banks	0·025	0·030	0·035*	0·040
Earth bottom, rubble sides	0·028	0·030*	0·033*	0·035
Natural stream channels :				
(1) Clean, straight bank, full stage, no rifts or deep pools	0·025	0·0275	0·030	0·033
(2) Same as (1), but some weeds and stones	0·030	0·033	0·035	0·040
(3) Winding, some pools and shoals, clean	0·035	0·040	0·045	0·050
(4) Same as (3), lower stages, more ineffective slope and sections	0·040	0·045	0·050	0·055
(5) Same as (3), some weeds and stones	0·033	0·035	0·040	0·045
(6) Same as (4), stony sections	0·045	0·050	0·055	0·060
(7) Sluggish river reaches, rather weedy or with very deep pools	0·050	0·060	0·070	0·080
(8) Very weedy reaches	0·075	0·100	0·125	0·150

* Values commonly used in designing.

TABLE OF COEFFICIENTS C FOR VARIOUS HYDRAULIC RADII (r), ACCORDING TO KUTTER'S FORMULA.

(For the same *actual* r (such as 3·28 ft., or 1 metre) the coefficient for metric units is the coefficient for British units $\times \frac{100}{181}$, or $\frac{5}{9}$, nearly.)

Slope S = ·000025 per Unit of Length, = 1 in 40,000, = ·132 Foot per Mile.

Hydraulic Radius, r, in Feet.	Coefficients n of Roughness.											
	·009	·010	·011	·012	·013	·015	·017	·020	·025	·030	·035	·040
	Coefficients C.											
·1	65	57	50	44	40	33	28	23	17	14	12	10
·2	87	75	67	59	53	45	38	31	24	19	16	14
·4	111	97	87	78	70	59	51	42	32	26	23	19
·6	127	112	100	90	81	69	60	49	38	31	26	22
·8	138	122	109	99	90	77	66	55	43	35	30	25
1	148	131	118	106	97	83	72	60	47	38	32	28
1·5	166	148	133	121	111	95	83	69	55	45	38	33
2	179	160	144	131	121	104	91	77	61	50	43	37
3	197	177	160	147	135	117	103	88	70	59	50	44
4	209	188	172	158	146	127	113	96	78	65	56	49
6	226	206	188	174	161	142	126	108	88	74	64	57
8	238	216	199	184	171	151	135	117	96	82	71	63
10	246	225	207	192	179	159	142	124	102	87	76	68
12	253	231	214	198	186	165	149	129	107	92	81	72
16	263	242	223	208	195	174	157	138	115	100	88	79
20	271	249	231	215	202	181	164	144	121	106	94	84
30	283	261	243	228	215	193	176	157	133	117	104	95
50	297	274	257	241	228	207	190	170	147	130	117	107
75	306	284	267	251	238	217	200	180	157	140	127	117
100	312	290	273	257	244	223	207	187	163	147	134	124

Slope S = ·00005 per Unit of Length, = 1 in 20,000, = ·264 Foot per Mile.

Hydraulic Radius, r, in Feet.	Coefficients n of Roughness.											
	·009	·010	·011	·012	·013	·015	·017	·020	·025	·030	·035	·040
	Coefficients C.											
·1	78	67	59	52	47	39	33	26	20	16	13	11
·2	100	87	77	68	62	51	44	35	26	21	18	15
·4	124	109	97	88	79	66	57	46	35	28	24	20
·6	139	122	109	98	90	76	65	53	41	33	28	24
·8	150	133	119	107	98	83	71	59	46	37	31	27
1	158	140	126	114	104	89	77	64	49	40	34	29
1·5	173	154	139	126	116	99	87	72	57	47	40	34
2	184	164	148	135	124	107	94	79	62	51	44	38
3	198	178	161	148	136	118	104	88	71	59	50	44
4	207	187	170	156	145	126	111	95	77	64	56	49
6	220	199	182	168	156	137	122	105	85	72	63	56
8	228	206	189	175	163	144	129	111	91	78	68	61
10	234	212	195	181	169	149	134	116	96	82	72	64
12	238	217	200	185	173	153	138	120	99	86	75	68
16	245	223	206	191	180	160	144	126	106	91	81	73
20	250	228	211	196	184	165	149	131	110	96	85	77
30	257	236	219	204	192	172	157	139	118	103	92	84
50	266	245	228	213	201	181	165	148	127	112	101	93
75	272	250	233	218	207	187	171	153	133	119	108	99
100	275	254	237	222	210	190	175	158	137	123	112	104

Slope S = ·00001 per Unit of Length, = 1 in 10,000, = ·528 Foot per Mile.

Hydraulic Radius, r, in Feet.	Coefficients n of Roughness.											
	·009	·010	·011	·012	·013	·015	·017	·020	·025	·030	·035	·040
	Coefficients C.											
·1	90	78	68	60	54	44	37	30	22	17	14	12
·2	112	98	86	76	69	57	48	39	29	23	19	16
·4	136	119	106	95	86	72	62	50	38	31	25	22
·6	149	131	118	105	96	81	70	57	44	35	30	25
·8	158	140	126	114	103	88	76	63	48	39	33	28
1	166	147	132	120	109	93	81	67	52	42	35	31
1·5	178	159	144	130	120	103	89	75	59	48	41	35
2	187	168	151	138	127	109	96	81	64	53	45	39
3	198	178	162	149	137	119	104	89	71	59	51	45
4	206	186	169	155	143	125	111	94	76	64	55	49
6	215	195	178	164	152	134	119	102	84	71	61	54
8	221	201	184	170	158	139	124	107	88	75	66	59
10	226	205	188	174	162	143	128	111	92	78	69	62
15	233	212	195	181	169	150	135	118	98	85	75	68
20	237	216	200	185	173	154	139	122	102	89	79	71
30	243	222	206	191	179	160	145	128	108	95	84	77
50	249	227	211	197	185	166	151	134	114	100	91	83
100	255	234	218	204	191	172	158	140	121	108	98	91

Slope S = ·0002 per Unit of Length, = 1 in 5,000, = 1·056 Feet per Mile.

Hydraulic Radius, r, in Feet.	Coefficients n of Roughness.											
	·009	·010	·011	·012	·013	·015	·017	·020	·025	·030	·035	·040
	Coefficients C.											
·1	99	85	74	65	59	48	41	32	24	18	15	12
·2	121	105	93	83	74	61	52	42	31	25	21	17
·4	143	125	112	100	91	76	65	53	40	32	27	23
·6	155	188	122	111	100	85	73	60	46	37	31	26
·8	164	145	131	118	107	91	79	65	50	41	34	29
1	170	151	136	123	113	96	83	69	54	44	37	32
1·5	181	162	146	133	122	105	91	77	60	49	42	36
2	188	170	154	140	129	111	97	82	64	54	45	40
3	200	179	163	149	137	119	105	89	72	59	51	45
4	205	185	168	155	143	125	111	94	76	63	55	48
6	213	193	176	162	150	132	117	100	82	69	60	53
8	218	198	181	167	155	137	122	105	87	73	64	57
10	222	201	185	170	158	140	125	108	89	76	67	60
15	228	207	190	176	164	145	131	113	95	82	72	65
20	231	210	194	180	168	149	134	117	98	85	76	68
30	235	215	198	184	172	154	139	122	103	89	80	73
50	240	220	203	189	177	158	143	126	108	94	85	78
100	245	224	208	194	182	163	148	131	113	99	90	83

TABLE OF COEFFICIENTS C FOR VARIOUS HYDRAULIC RADII (*continued*).

Slope S = ·0004 per Unit of Length, = 1 in 2,500, = 2·112 Feet per Mile.

Hydraulic Radius, *r*, in Feet.	Coefficients *n* of Roughness.											
	·009	·010	·011	·012	·013	·015	·017	·020	·025	·030	·035	·040
	Coefficients C.											
·1	104	89	78	69	62	50	43	34	25	19	16	13
·2	126	110	97	87	78	65	54	44	32	25	21	18
·3	138	120	107	96	87	73	62	50	37	30	24	21
·4	148	129	115	104	94	79	68	55	42	33	27	23
·6	157	140	126	113	103	87	75	62	47	38	31	27
·8	166	148	133	121	110	93	81	67	51	42	35	30
1	172	154	138	125	115	98	85	70	55	45	37	32
1·5	183	164	148	135	124	106	93	78	61	50	42	37
2	190	170	154	141	130	112	98	83	65	54	45	40
3	199	179	162	149	138	119	105	89	71	59	51	45
4	204	184	168	154	142	124	110	94	76	63	55	48
6	211	191	175	161	149	130	116	99	81	69	60	53
10	219	199	183	168	157	138	123	107	88	75	66	59
20	227	207	190	176	164	146	131	115	96	83	73	66
50	235	215	198	184	173	154	139	123	104	91	82	75
100	239	219	203	189	177	158	143	127	108	96	87	80

Slope S = ·0010 per Unit of Length, = 1 1,000, = 5·28 Feet per Mile.

Hydraulic Radius, *r*, in Feet.	Coefficients *n* of Roughness.											
	·009	·010	·011	·012	·013	·015	·017	·020	·025	·030	·035	·040
	Coefficients C.											
·1	110	94	83	73	65	54	45	36	27	21	17	14
·2	129	113	99	89	81	66	57	45	34	27	22	18
·3	141	124	109	98	89	74	63	51	39	30	25	21
·4	150	131	117	105	96	80	69	56	43	34	28	24
·6	161	142	127	115	104	88	76	63	48	39	32	27
·8	169	150	134	122	111	94	82	68	52	42	35	30
1	175	155	139	127	116	99	86	71	56	45	38	33
1·5	184	165	149	136	124	108	93	78	62	50	43	37
2	191	171	155	142	130	112	98	83	66	54	46	40
3	199	179	163	149	138	119	105	89	71	59	51	45
4	204	184	168	154	142	124	110	93	75	63	54	48
6	211	190	174	160	149	130	116	99	81	68	59	52
10	218	197	181	167	155	136	122	105	87	74	65	58
20	225	205	188	175	163	144	129	113	94	81	72	65
50	232	212	196	182	170	151	137	120	101	89	79	72
100	236	216	200	186	174	155	141	124	105	94	85	77

Slope S = ·01 per Unit of Length, = 1 in 100 = 52·8 Feet per Mile.

Hydraulic Radius, *r*, in Feet.	Coefficients *n* of Roughness.											
	·009	·010	·011	·012	·013	·015	·017	·020	·025	·030	·035	·040
	Coefficients C.											
·1	110	95	83	74	66	54	46	36	27	21	17	14
·2	130	114	100	90	81	67	57	46	34	27	22	19
·3	143	125	111	100	90	76	64	52	39	31	25	22
·4	151	133	119	107	98	82	70	57	44	35	29	24
·6	162	143	129	116	106	90	77	64	49	39	33	28
·8	170	151	135	123	112	95	82	68	53	43	35	31
1	175	156	141	128	117	99	87	72	56	45	38	33
1·5	185	165	149	136	125	107	94	79	62	51	43	37
2	191	171	155	142	130	112	99	83	66	55	46	40
3	199	179	162	149	138	119	105	89	71	59	51	45
4	204	184	167	154	142	123	109	93	76	63	55	48
6	210	190	173	160	148	129	115	99	81	68	59	52
10	217	196	180	166	154	136	121	105	86	74	65	58
20	225	204	187	173	161	143	128	112	98	80	71	64
50	231	210	194	181	168	150	135	119	100	87	78	71
100	235	214	197	184	172	153	139	122	104	91	82	75

To find the coefficient C, having the slope S, the hydraulic radius, *r*, and the degree *n* of roughness. Turn to the division of the Table corresponding to the given slope, S. In the first column find the given hydraulic radius, *r*. In the same line with this *r*, and under the given *n*, is the proper value of C.

To find the actual, or the greatest permissible, degree n of roughness of channel, having the slope S, the hydraulic radius *r*, and either C, or the actual or required velocity, *v*.

If the velocity is given, and not C, first find $C = \frac{\text{velocity}}{\sqrt{\text{slope}} \times \text{hydraulic radius}}$. Turn to the division of the Table corresponding to the given S, and in the first column find the given *r*. In the same line find the value given, or just obtained, for C; over which will be found the required *n*.

To find the actual or necessary hydraulic radius, *r*, having the slope S, the degree *n* of roughness, and the actual or required velocity, *v*. Assume an hydraulic radius; and from the division of the Table corresponding to the given S take out the value of C corresponding to the given *n* and the assumed *r*. Then say

$$v = \text{C so found} \times \sqrt{\text{assumed hydraulic radius} \times \text{slope}}.$$

If this *v* is the same as the given velocity or near enough to it, take the assumed *r* as the proper one. Otherwise, repeat the whole process, assuming a new *r*, *greater* than the former one if *v* is *less* than the given velocity, and *vice versa*.

To find the actual or necessary slope, S, having the dimensions of the wetted portion of the channel, the degree n of roughness, and the actual or required velocity.

Find the hydraulic radius, $r = \frac{\text{area of wet cross-section}}{\text{length of wet perimeter.}}$

Assume one of the four slopes of the Tables to be the proper one. From the corresponding division of the Table take out the value of C corresponding to the given r and n.

For slopes steeper than ·01 per unit of length, = 1 in 100 = 52·8 ft. per mile, C remains practically the same as at that slope. But the *velocity* (being = C × $\sqrt{\text{hydraulic radius} \times \textit{slope}}$) of course continues to increase as the slope becomes steeper.

Bazin's formula for the determination of C is as follows:

$$C = \frac{157{\cdot}6}{1 + \frac{y}{\sqrt{r}}},$$

where,

y = a coefficient of roughness, whose value Bazin gives as follows for different classes of channels.

Class I. y=0·109 for smoothed cement, or planed wood.
" II. y=0·290 for planks, bricks, and cut stone.
" III. y=0·833 for rubble masonry.
" IV. y=1·54 for earth channels of very regular surface or reveted with stone.
" V. y=2·35 for ordinary earth channels.
" VI. y=3·17 for exceptionally rough earth channels (bed covered with boulders) or weed-grown sides.

See also ' A New Hydraulic Formula,' by F. W. Woods, C.I.E., *The Engineer*, June 17, 1927.

Design of Pipes.

Pressure in Pipes.

The interior pressure in a pipe due to the presence of water, and tending to burst this pipe, is larger transversely than longitudinally.

The pressure in pounds per sq. in. p, determined by a head of water H expressed in feet, is:

p=pressure in lbs. per sq. in.=head in ft.×0·44,

and the diametral pressure P, tending to burst the pipe, is represented by the equation

$$P = p \times \frac{d}{2},$$

where,

d = diameter of pipe in ins.

The longitudinal tension P in the pipe is equal to

$$P' = p \times \frac{d}{4}.$$

This explains why steel riveted pipes have generally a single row of rivets at the circumferential joints and two rows of rivets along the longitudinal joints.

Design of Steel Pipes.

The thickness of metal t (expressed in ins.) required to carry a certain pressure can be determined by the following formula:

$$t = \frac{(p+p')d}{2fe} + c,$$

where,

t=thickness of steel shell in ins.;
d=inside diameter of pipe in ins.;
p=pressure of water in lbs. per sq. in.;
p'=allowance for water hammer in lbs. per sq. in.;
f=allowable tensile stress of steel in lbs. per sq. in.:
e=efficiency of riveted joint
c=thickness to be added to plate for corrosion.

The permissible unit tensile stress with ordinary open-hearth steel is assumed at 16,000 lbs. The efficiency of the joint may be taken at 55 per cent. for single riveted pipe, and at 70 per cent. for double riveted pipe. The thickness to be added to compensate for the weakening of the pipe due to corrosion is about $\frac{1}{16}$ in. In large pipes under small pressure, the thickness of metal computed with the above assumptions must still be increased to give the pipe the necessary stiffness.

Design of Reinforced Concrete Pipes.

Reinforced concrete pipes are designed under the assumption that the steel takes care of all longitudinal or transversal stresses in the pipe whilst the concrete acts as a waterproof shell. It is evident that, in this case, the steel can be reduced to the minimum thickness required.

The permissible stress per sq. in. of metal can also be taken at 16,000 lbs. per sq. in.

The area of the circles forming the transversal reinforcement can be determined as follows:

Let p = total internal pressure in lbs. per sq. in.;
d = diameter of pipe in ins.;
A_s = area of steel reinforcement per lineal foot, expressed in sq. ins.;
f = permissible stress in steel in lbs. per sq. in.;

then assuming that the concrete is to take no part of the tensile stresses, we have

$$A_s = \frac{p\,d \times 12}{2 \times f}.$$

The longitudinal reinforcement is taken with an area per lineal foot of circumference equal to half the area per lineal foot of pipe as determined above.

Spacing of Pipe Supports.

Steel and concrete pipes are not always built resting on the ground, and very often they have to be carried on concrete or masonry piers resting preferably on rock, or having a bearing surface in proportion to the load carried and to the resistance of the foundations.

The spacing of the piers is determined by considering the pipe as a circular beam resting freely on supports and carrying a uniformly distributed load consisting of the weight of the pipe plus the weight of the water. This load determines a longitudinal tension in the lower side of the pipe, and this stress added to the longitudinal tension of the pressure of water caused by the static head, plus the extra head due to water hammer, should not be allowed to exceed f, the safe tensile strength of the material.

The stress, f_b, due to bending is

$$f_b = \frac{3}{2} \cdot \frac{WL^2}{S},$$

where

L = distance between supports in ft.;
W = total load in lbs. per ft. of length;
S = section modulus of the pipe.

The longitudinal fibre stress, f_t, due to water pressure is

$$f_t = \frac{pd}{4t},$$

where,

p = pressure in lbs. per sq. in.;
d = nominal inside diameter of pipe in ins.;
t = thickness of pipe in ins.

This formula applies to steel pipe, but can be utilised in the calculation of a reinforced concrete pipe, by reducing the total area of steel reinforcement per foot length to a continuous pipe of uniform thickness. This method, however, would not take into account the resistance in compression of the concrete on the upper side of the pipe.

Adding together the stress due to bending f_b and the stress due to water pressure f_t we have a total stress,

$$f = \frac{3}{2} \frac{WL^2}{S} + \frac{pd}{4t};$$

from this we can get the distance between the supports

$$L = \sqrt{\frac{2}{3} \frac{S}{W} \left(f - \frac{pd}{4t} \right)}.$$

The permissible value of f, in a steel riveted pipe, can be taken from 8,000 to 11,000 lbs. per sq. in. according to the efficiency of the joint; in a reinforced concrete pipe 16,000 lbs. per sq. in. could safely be assumed.

Design of Wood Stave Pipes.

In Canada and in the United States extensive use is made of pipes composed of wooden staves held in place by steel wires wound continuously around the pipe, or by separate steel bands or rods connected by malleable iron shoes.

The steel is designed to take care of all the stresses and the wood forms only the shell. These pipes are very cheap, and have a greater carrying capacity than either steel or reinforced concrete pipes; they have been used for heads as high as 200 ft.

The following information has been compiled from the writings of Robert E. Horton and of Andrew Swickard.

The staves are made of California redwood, Douglas fir, and Canadian red pine. Practice varies somewhat as to the thickness of staves to be used under given pressure for a pipe of given diameter. The following considerations should govern the thickness of staves to be adopted in a given case.

1. The thicker the staves the greater stability of pipe against deformation by its own weight, and accordingly the staves must be thicker for large than for small diameter of pipes.

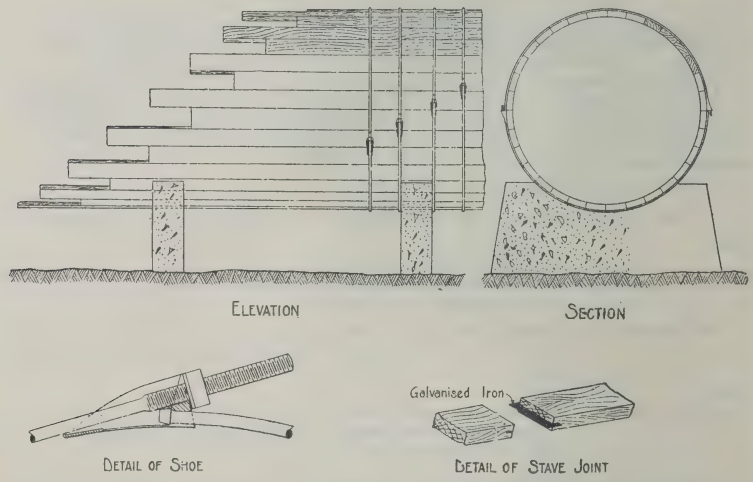


Fig. 13.—Details of Wood Stave Pipe.

2. The staves should be thick enough to prevent material loss by percolation, and accordingly should be thicker for heavy than for lighter pressures.

3. The staves should not be so thick as to prevent their being thoroughly saturated. This and considerations of economy require the use of as thin a stave as is compatible with the first two conditions named.

The following rule serves as a general guide for determining the approximate thickness of staves to be used in any case; the actual thickness of staves may be made equal to the nearest $\frac{1}{8}$ in. or $\frac{1}{4}$ in. greater thickness than shown by the rule

$$t = 1 \text{ in.} + \frac{\text{Head in ft.} + \text{Diameter of pipe in ins.}}{100}.$$

The minimum radius of bends which may be made with continuous pipe increases with the diameter of the pipe and with the thickness of the staves. An approximate rule is

$$\text{Minimum radius, in ft.} = 4 \text{ to } 5 \times (d + 4t^2)$$

where,

d = inside diameter of pipe, in ins.;

t = thickness of staves, in ins.

Wood stave pipes should be supported through an arc extending about 55° each side of the vertical axis, in order to prevent deformation of the pipe by its weight and that of the water it contains.

Stave pipes (fig. 13) are generally carried on wooden or on concrete cradles, concrete being preferable for large pipes. Joseph P. Frizell says that wooden cradles are best made of 6 in. × 8 in. or 8 in. × 8 in. timbers, spiked or drift-bolted together, of a length somewhat greater than the diameter of the pipe, and at distance apart about equal to their length. Robert E. Horton is of the opinion that the spacing of the supports is about the same for wood stave and steel pipes.

The abutting end joints of the staves are connected together by iron plates inserted in a slot sawn in for that purpose. This iron tongue should be about $\frac{3}{32}$ in. wider than twice the depth of the slot and about $\frac{1}{4}$ in. longer than the slot, allowing a projection of $\frac{1}{16}$ in. into the adjoining staves.

Creosoted wood-stave pipes put in place twenty-eight years ago were said, at the recent convention of the American Wood Preservers' Association, to be still in perfect condition. The best method of treatment for the staves was described as being the empty-cell process with 50 lbs. initial air pressure, following the creosote pressure period with several minutes' reheating at about 25° F. in excess of any previous temperature, then a vacuum of not less than 20 ins. maintained for from 30 mins. to 1 hour. By this process a satisfactory creosote penetration is obtained, and the staves come out thoroughly clean, and so dry as scarcely to soil the hand.

Area of Bands and their Spacing.

The two principal considerations governing the size and spacing of bands are as follows:—

(1) The area of section of the band per ft. length of pipe must be such as to safely withstand the water pressure plus the stress in the band due to swelling of wood when saturated.

(2) The size and spacing of bands must be such as not to produce crushing of the wood.

Taking into account both the stress due to water pressure and that due to swelling of the staves, the following formula has been derived for determining the necessary area of cross-section of metal in the bands per in. length of pipe.

$$a = \frac{pd + 200t}{2f},$$

where,

a = required area of band section per in. length of pipe;

p = pressure in lbs. per sq. in.;

d = inside diameter of pipe in ins.;

t = thickness of stave in ins.;

f = permissible unit stress in lbs. per sq. in., taken generally as 16,000 lbs.

In order to avoid dangerous crushing of the wood, the bearing pressure between bands and staves should not exceed about 800 lbs. per sq. in. of contact between the band and stave. For round bands the width of contact between the band and stave should be about equal to the radius of the band.

From these data the following formula is obtained for determining the maximum permissible spacing S in ins. between bands of given size under a given pressure.

$$S = \frac{400 \times \text{diameter of round bands in ins.}}{\text{Pressure in lbs. per sq. in.}}.$$

Shoes or Band Couplings.

The band couplings should be made of malleable iron. The weight of these shoes will vary with the size of the band and the diameter of the pipe, but the following table gives average weights for various sizes of bands, which are accurate enough for estimating purposes:

Diameter of Band, ins.	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1
Weight of Shoe, lbs.	0·35	0·40	0·65	1·00	1·75	2·75	4·00	6·00

Length of Bands.

Before the length of a band can be determined it is necessary to know how much lap-over of the rod there is in the shoe. This differs in shoes for different sizes of pipe, increasing with the increase in size of rod. However, this quantity will average about as follows for various sizes of bands :

Diameter of Band, ins. . . .	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1
Lap-over of Band in Shoe, ins. .	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	$1\frac{1}{16}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{5}{8}$

The end of the rods should be upset and threaded over a length of not less than $4\frac{1}{2}$ ins. for $\frac{3}{8}$ in., 5 ins. for $\frac{7}{16}$ in., and 6 ins. for other sizes. There are three types of head in use—button, square, and tee heads. The tee head has the advantage of keeping both ends of the rod closer to the staves, and shortens the space over which the rod is raised from the staves ; it also permits the use of a shorter shoe. The length of rods under the head should be such that, after the pipe is finally tightened, at least $1\frac{1}{2}$ ins. of thread will remain ahead of the nut. The length of the rod, excluding the head, is determined in the following manner, where the rod is in one part and 6 ins. of thread is being used :

$$\text{Total length} = (d+2t+r)\pi+k+4\tfrac{1}{2} \text{ ins.}$$

where,

d = nominal inside diameter of pipe in ins. ;
t = thickness of staves in ins. ;
r = diameter of rod in ins. ;
k = lap of rod in shoe in ins.

If the rod is in two parts, the following formula applies :

$$\text{Total length} = (d+2t+r)\pi+2k+9 \text{ ins.}$$

From 68 tests on wood-stave pipe lines with diameters from 4 to 162 ins., and velocities ranging from 1 ft. to 8 ft. per sec., it was found that with diameters from 4 to 24 ins. the average value of n in Kutter's formula (p. 1299) was 0·0109, and with diameters from 31 to 162 ins. the average value of n was 0·0124.

(*Mallett ; Flow of Water in Pipes, Vol. CCVIII (April* 1921), *Proceedings Inst. C.E.*)

Discharge of Water through Orifices.

The theoretical velocity, v, with which water flows from an orifice in the side of a vessel at a depth h from surface is the same as that of a body falling freely by gravity from height, h, viz.

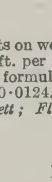


FIG. 14.

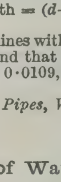


FIG. 15.

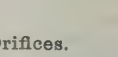


FIG. 16.

$v = \sqrt{2gh}$, but practically the converging currents produce a contraction of the jet, which assumes the form of the vena contracta, d, e, f, g, fig. 15.

h = height from surface of water to centre of orifice, in feet. v = velocity of issuing water

FIG. 17.

FIG. 18.

in feet per second. A = area of orifice, in feet. Q = cubic feet discharged per second. g = coefficient of terrestrial acceleration.

$$v = m\sqrt{2gh}. \qquad Q = mA\sqrt{2gh}. \qquad 2g = 64{\cdot}4 \qquad \sqrt{2g} = 8 \text{ practically}$$

m.
·625. Where the orifice is a thin plate, bc (fig. 14), with thickness less than $\frac{1}{2}$ smallest dimension of orifice.
·97. Orifice shape of vena contracta, d, e, f, g (fig. 15) ; A is measured at f, g.
·82. Orifice with an adjutage (fig. 16) ; O = $2\frac{1}{2}$ to 3 times least dimension at A.

m.	
·95.	Sides converging at angle of 13½° (fig. 17). m depends on diameter and angle of convergence.
1·46.	Sides diverging at angle of 5° 6′ (fig. 18). C = 9 times least dimension at A, which is of form of vena contracta. (Seldom used.)
·548.	Where 2 sluices in thin plates are so close as to interfere with each other; m being reduced from ·625.
·62.	For sluices of moderate size in lock gates, etc.
·82.	For narrow-bridge openings.
·92.	For very large sluices and bridge openings.

Other formulæ for calculating the velocity, through orifices and particularly through a Venturi tube, are given on p. 1308.

The conclusions drawn from investigations made by Messrs. *Dempster Smith* and *J. W. Walker* * are:—

(1) That above a certain critical head (about 2 ft. of water) and a critical orifice diameter (2½ ins.), the experimental values of the coefficient of contraction for different types of orifice agree closely with the values deduced from hydrodynamical theory.

(2) That the coefficient of contraction varies with the head and orifice diameter in a manner closely represented by the formula

$$C = 1 - \frac{a}{e^x} \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where a is a constant and x a function of the head and orifice diameter.

(3) That in the case of the coefficient of contraction for sharp-edged orifices

$$a = 0{\cdot}39;$$

$$x = \frac{0{\cdot}15}{d^{9}\, h^{0{\cdot}125}},$$

where h is the head causing the flow.

(4) That formula (1) may be extended to include the effect of a variation in the pipe or reservoir area. The formula for C then becomes

$$C = \frac{1}{1 + \left\{ \frac{a}{e^x - a} \right\} (1 - a^2)} \quad . \quad . \quad . \quad . \quad (2)$$

where x and a have the same significance as before and a = ratio $\frac{\text{orifice area}}{\text{pipe area}}$.

To raise a River a certain Height by a Weir.

Q = discharge of river in cubic feet per second; h_1 = height by which river is to be raised in feet; h_2 = depth of top of weir below original surface of water necessary to raise surface by height h_1; h_a = head corresponding to velocity of approach; l = length of weir in feet; m = coefficient of contraction = ·628; $2g = 64{\cdot}4$.

$$h_2 = \frac{Q}{ml\sqrt{2g(h_1 + h_a)}} - \frac{2}{3}\frac{(h_1 + h_a)^{\frac{3}{2}} - h_a^{\frac{3}{2}}}{\sqrt{h_1 + h_a}}.$$

If h_a is not taken into account, then

$$h_2 = \frac{Q}{ml\sqrt{2gh_1}} - \tfrac{2}{3}h_1.$$

* *Proceedings of the Institution of Mechanical Engineers*, 1923.

Variants from Accepted Formulæ. (See p. 1298–1307).

A formula which is simple to use and sufficiently correct for most purposes is,

$$V = \frac{fM}{D^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where f is a constant, 284, V is the velocity of flow in feet per second, M is the output in millions of gallons per day and D is the internal diameter of the pipe in inches. Strictly, the value of f is 283·678, but 284 is sufficiently accurate for most purposes.

Where the flow is in gallons per minute it can be multiplied by 1440 and divided by a million to convert it to millions of gallons per day. If preferred, when the output is stated in gallons per minute the factor 0·409 may be used as the value of f in the original formula.

The following variations of this formula are useful:

$$D = \sqrt{\frac{fM}{V}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$M = \frac{D^2V}{f} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

$$V = \sqrt{2GH} \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$H = \frac{V^2}{2G} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

where G is the acceleration due to gravity (32·16) and H the velocity head in feet.

When a Venturi tube is used, the data required are H' = Venturi differential head, in feet; H'' = velocity head for velocity in the throat of the Venturi tube, in feet; D^a = diameter of Venturi approach, in inches; D^t = diameter of Venturi throat, in inches; R = ratio of $(D^a)^2$ to $(D^t)^2$ and P = the percentage factor for converting H' into H''.

Then

$$H'' = H'P \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

$$R = \frac{(D^a)^2}{(D^t)^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

$$P = \frac{R^2}{R^2-1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

In a standard orifice made of sheet metal or similar material, the diameter of the orifice should be the same as the thickness of the sheet and the diameter of the sheet should be at least seven times that of the orifice. The sides of the orifice should each make an angle of 45° with a line drawn from the back of the orifice at right angles through the sheet. The orifice should, however, be truly cylindrical for a distance of about $\frac{1}{16}$ in. at the front of the sheet. The coefficient of discharge (c) of a standard orifice is only 0·62 of its maximum area; that of a Venturi tube may be 0·90–0·98. Hence a Venturi tube is preferable where a head must be conserved.

The differential head in a Venturi tube is ascertained by reading two manometers at D^a and D^t respectively.

To calculate the actual discharge through a Venturi tube, when D^a, D^t, c and H' are given, it is first necessary to calculate R from formula (7), then P from formula (8), H'' from formula (6), V from formula (4), and M from formula (3). This is in marked contrast with a standard orifice, for which only one formula (No. 5) is required. A similar series of formulæ is needed when the diameter of the throat or other dimensions of a Venturi tube to correspond to a given flow are required. Yet the use of these formula is often quicker than working out the single formula given in some text-books, because the latter do not supply some of the data such as the numerical value of viscous flow.

SECTION XXV

PART IV

WATER-PRESSURE ENGINES—PELTON WHEELS—TURBINES

Water-pressure Engines.

SINGLE-ACTING ENGINES

consist of single cylinder and piston. The water is admitted under the piston, raises it, is allowed to escape, and the piston falls by its own weight. Average velocity of piston should not exceed 140 feet per minute. Stroke should be as long as possible, generally from 6 to 10 feet.

Water-pressure engines are also made rotatory with single or double acting cylinders, and single and double acting rams are extensively used for working cranes, dock gates, etc., the stroke of the ram being multiplied by gearing or chain tackle. The pressure generally used is 700 lbs. per square inch.

POWER OF WATER-PRESSURE ENGINES.

$$P = 62{\cdot}42Qh \times \text{66 to } {\cdot}8 \text{ ft. lbs. per sec.}; \qquad \text{H.P.} = Qh \times {\cdot}07491 \text{ to } {\cdot}09080,$$

where,

Q = cub. ft. of water per sec., h = height of fall, in ft.

Diameter of supply-pipe = diameter of cylinder × ·41 for single cylinder engines, = diameter of cylinder × ·68 for double cylinder engines.

EXAMPLE.—A single-acting pressure engine has a pressure cylinder 50 ins. in diameter, a head of water of 132 ft., a stroke of 10 ft., and is worked 4 to 7 strokes per min.

$$Q = \frac{25^2 \times 3{\cdot}14}{144} \times 10 \times \frac{4}{60} = 9{\cdot}1 \text{ cub. ft. per sec.}$$

H.P. = 9·1 × 127 × ·09080 = 105 horse-power, with 4 strokes per min., or 184 with 7 strokes. The actual work was about 168 horse-power.

PELTON WHEELS.

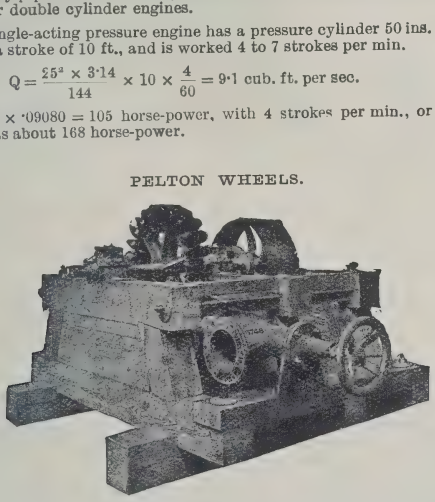


FIG. 1.

In these wheels, fig. 1, the water striking the buckets in the middle is deflected to both sides equally, and in such a way that the discharge from one bucket clears the one following. The

construction of the buckets is shown in fig. 2. An improvement of the Pelton semi-cylindrical bucket is the Doble bucket, fig. 3, each half of which is ellipsoidal in form, and has a part of the outer lip cut away, so as to avoid hitting the jet when moving under it.

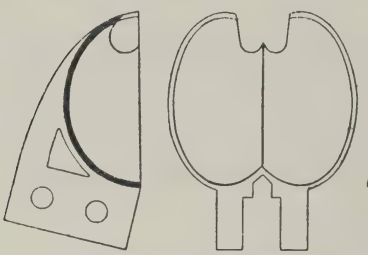


FIG. 2.—Pelton Bucket.

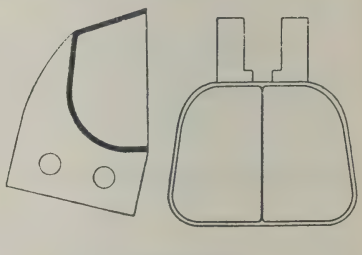


FIG. 3.—Doble Bucket.

THE ENGLISH ELECTRIC BUCKET ATTACHMENT FOR IMPULSE WATER TURBINES.

(*The English Electric Co., Ltd.*)

Fig. 4 shows the method of attachment in detail. For each bucket it consists of one tapered forged-steel bolt passing through a bush of chrome nickel steel, the external surface of which is cylindrical, whilst the interior is reamered to a taper. The bush is split longitudinally along its whole length. The holes through the disc and the lugs of the bucket are, therefore, cylindrical, so that spare buckets can be completely finished in the shops, and are interchangeable without difficulty; no reamering is necessary on site, and prolonged stoppage of the plant is obviated.

The tapered bolt is provided with nuts which, on being tightened, cause the bush to expand and tighten against the wall of the cylindrical hole; the force due to this taper is considerably greater than that exerted by the jet. At the same time, the lugs of the bucket are tightened by the nuts with absolute security against the two faces of the disc. These nuts are secured by locking washers, which are bent over both the boss of the lug and the face of the nut.

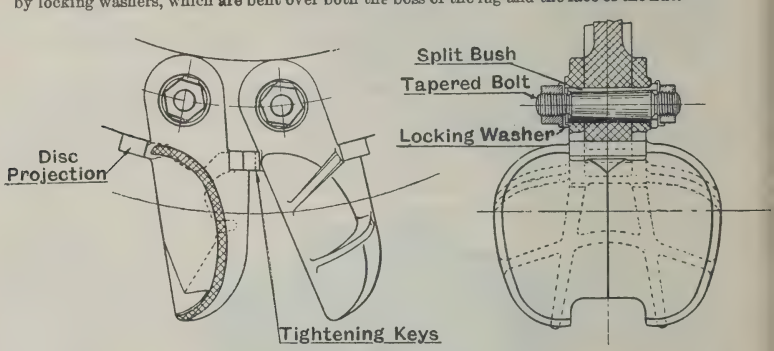


FIG. 4.

Furthermore, the back of every alternate bucket bears against a projection on the circumference of the disc, machined out of the solid metal. Between these projections each pair of buckets is finally tightened up by means of two tapered keys which when put together present parallel faces to the bucket surfaces. After final assembly the small ends of both these keys are bent to prevent any movement. By this means, the buckets are tightened under great pressure against the disc projections and the circle of buckets resists the blows of the jet as rigidly as a solid mass.

Not only is the attachment remarkably strong, but the one-bolt fastening works under better conditions than the two-bolt attachment generally used. It is extremely difficult to drill a series of pairs of holes in the cast-steel disc and buckets with the necessary accuracy to render the buckets interchangeable, and to ensure that the stresses are shared equally by each bolt. There is always the danger of one bolt bearing nothing whilst the other takes the whole of the

strain. With one bolt and one key this danger is overcome; the bolt bears the shearing, and the key the compressive stresses, thus the forces are transmitted to the disc in the best possible manner.

Proportion of Parts of Pelton Wheels.

(1.)

The area of the jet should not exceed 0·1 the projected area of the bucket, or the diameter of the jet should not exceed 0·3 the width of the bucket. The ratio of the wheel diameter to jet diameter should not be lower than 9, 12 being a common value, so that the diameter of the wheel in feet should about equal the diameter of the jet in inches.

The size of the jet depends on the amount of power desired, per unit, under the given head; the peripheral speed of the wheel varies between 0·43 and 0·47 of the theoretical jet velocity. Two and three jets are sometimes used upon one wheel, but a single jet is preferable. Sometimes two wheels are used to drive a single generator mounted between them.

The efficiency of the Pelton wheel has been found to exceed 86 per cent. Since this wheel operates by utilising the kinetic energy of the water, it is best adapted to a small discharge under a high head: its range of usefulness varies from 50 foot heads to 3,000 and even 5,000 foot heads.

A recent Gunther Pelton wheel, installed in India for electric power and lighting, develops 100 b.h.p. under 600 ft. head, and the oil-pressure governor works on to the jet deflector, the spear rod regulator being hand-controlled.

The majority of impulse wheels are now equipped with steel buckets, as the average of steel castings has been greatly improved, so that the present steel bucket will outwear the bronze bucket.

(2.)

(Contributed by W. Gunther & Sons, Oldham.)

The diameter of a Pelton wheel is fixed by the speed required. Theoretically and practically the most efficient wheel velocity is one-half the jet velocity.

$$\text{Velocity of jet in ft.-sec.} = 0{\cdot}95 \text{ to } 0{\cdot}97 \sqrt{2gH}.$$

where,

$$\text{Velocity of wheel in ft.-sec.} = V = 0{\cdot}47 \sqrt{2gH},$$

H = head in feet.

When the speed required is known or can be fixed, the diameter, D, of the wheel is given by—

where,

$$D = \frac{V\,60}{n\,\pi}$$

V = velocity of wheel in ft.-sec.; n = revolutions per min.

This is the diameter of an imaginary circle having as a tangent the line of action of the centre of the jet.

The diameter of jet required may be found as follows:—

$$\text{Diam. of jet in inches} = \sqrt{\frac{Q \times 0{\cdot}401}{\sqrt{H}}}$$

where,

Q = quantity of water in cu. ft. per *minute*; H = head in *feet*.

The ratio of the diameter of the jet to the diameter of the wheel should not normally be below 1 : 12.

The proportions of the wheel buckets are given in terms of the jet diameter, and may generally be taken as follows:—

width of bucket, 3·2 to 3·5 × jet diam.; height of bucket, 2·7 × jet diam.; depth of bucket, 1·2 × jet diam.

When two jets are used on one wheel the size of the buckets will be materially less than if one jet were used, and this often means a considerable saving. Various combinations of jets and wheels are sometimes used, e.g. two wheels keyed to one shaft, each with either one or more jets, and so on, but not more than four jets, at intervals of 90 degs., should play on one wheel, or interference takes place. In actual practice it is rarely found advisable to use more than two jets per wheel.

Pelton Wheel Regulators.

General.

The section of the nozzle is either square or circular, but more generally circular, with a needle, fig. 5, to secure a more uniform velocity in all parts of the jet; this needle can be moved back or forward by the governor, varying the area of the opening and regulating the amount of water discharged. For very high heads, where the interruption or partial suppression of flow by such an arrangement might involve danger to the pipe-line, deflecting nozzles of various kinds are employed. The movement of a regulating nozzle of this description is accomplished with comparatively little expenditure of energy, and thus the control of Pelton wheels is simple, unless affected

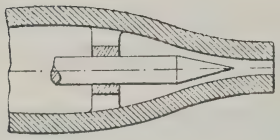


Fig. 5.

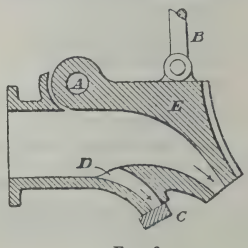


Fig. 6.

by irregularities. A deflecting nozzle, as used by Messrs. Theodor Bell & Co., is shown in fig. 6. This nozzle is rectangular in section, the upper side being formed by the piece E pivoted at A. This is moved by the governor attachment B, so as to increase or diminish the area of the nozzle at the mouth. A secondary nozzle D is closed by a cap C, which also rotates about the pivot A, and is moved through the same angle as E. As the two nozzles have the same width, a partial closure of the larger nozzle is compensated by an opening of the smaller, so that the spouting area is always a constant, and the flow through the pipe is steady, while the wheel receives that fraction of it which is required to maintain the speed constant at all loads.

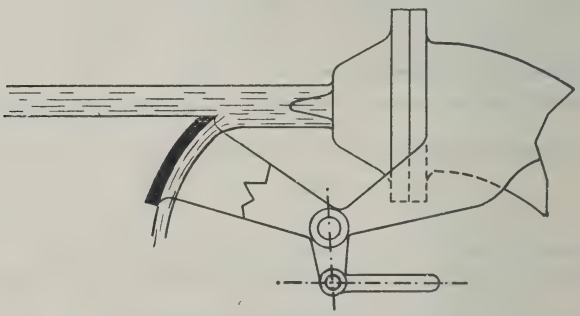


Fig. 7.—Needle Nozzle with Deflector.

The foregoing device, however, is wasteful of water even with the best of attention. Since the water hammer caused in the pipe by a sudden opening of the nozzle is less dangerous than the hammer due to the sudden interruption of the flow, a device, fig. 7, has been perfected, consisting of an automatic jet deflector and an ordinary nozzle needle, which causes the governor to act on the deflector only in case of closing, the needle being afterwards slowly closed by means of a dash-pot mechanism, whilst the deflector becomes tangential to the reduced jet. When the governor moves to increase the discharge of the nozzle, it controls simultaneously and rapidly the deflector and the needle.

Combined Spear and Deflector Regulator Controlled by Shaft Governor.

(Gilbert Gilkes & Gordon, Ltd., Sole Makers and Patentees.)

For high-falls (100 ft. and upwards), Pelton wheels or other impulse turbines may be controlled by an arrangement similar to that shown in fig. 8, as made by *Gilbert Gilkes & Gordon, Ltd.*, of Kendal.

The arrangement consists of a pendulum mounted upon the turbine shaft and coupled direct to the deflector shaft, the design of the deflector being such as to impose the least resistance to the passage of the deflector through the jet.

Movement to the spear is given by a piston moving in a hydraulic or oil-pressure cylinder fixed to the end of the branch pipe.

The position of the spear is dictated by a poppet valve controlled by levers and links connected to the deflector shaft, and so arranged that if the load be thrown off the deflector

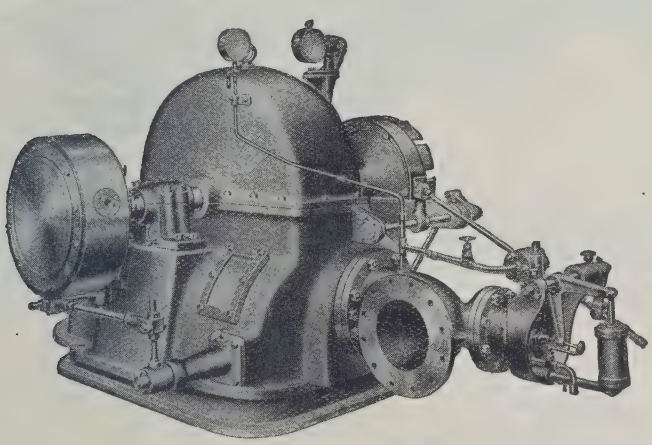


Fig. 8.

is brought instantaneously into the jet, the spear slowly following and the deflector finally taking up a position just clear of the jet. If the load is thrown on, the spear is drawn rapidly back.

When economy of water on reduced loads is immaterial, as in the case where the supply is from a stream without storage, the automatic spear-following combination is dispensed with, a hand-regulated spear being substituted.

The pendulum being mounted directly on to the shaft ensures a perfectly steady drive, saves space, and is immune from all risks due to breaking of chains, belts, etc.

As the pendulum is connected directly to the deflector without the intermission of servo-motors, etc., extreme rapidity of action is obtained, the movements being instantaneous and dead time eliminated, the results showing a momentary variation of less than 1 per cent. with full load thrown off.

The 'Seewer' Impulse Wheel Regulator

(The English Electric Co., Ltd.)

The design of this governing device for impulse turbines is based on the principle of the rapid destruction of the energy of the jet. This is obtained by placing blades in the nozzle at a slight angle to the normal direction of the water stream, so that the jet issuing from the nozzle no longer remains cylindrical, but is dispersed in a hollow conical form, thus diverting the energy of the jet from the buckets.

The blades, normally withdrawn into the hollow spearhead, fig. 9, are carried by a movable ring which is connected to the speed governor by a rod passing through the spindle of the spear. When a sudden drop in load occurs and the speed rises, the blades are pushed forward into the water stream, giving it a whirling movement which makes it lose its useful energy. Fig. 9 shows the blades advanced into the position for dispersing the jet.

Fig. 10 is a diagrammatic arrangement of this governing device for a single-jet impulse turbine.

FIG. 9.

On a sudden drop in load with its attendant rise in speed, the governor pendulum rises; point 1 being a momentary fulcrum, point 2 must fall, depressing the plunger in the distributing valve and thus allowing the oil to flow from the pump to the right-hand piston chamber of the guide-blade servomotor, and at the same time opening the exhaust from the left-hand chamber. The resultant oil pressure in the right-hand chamber, in conjunction with a spring, pushes the piston to the left and the guide blades are moved forward into the water stream by the action

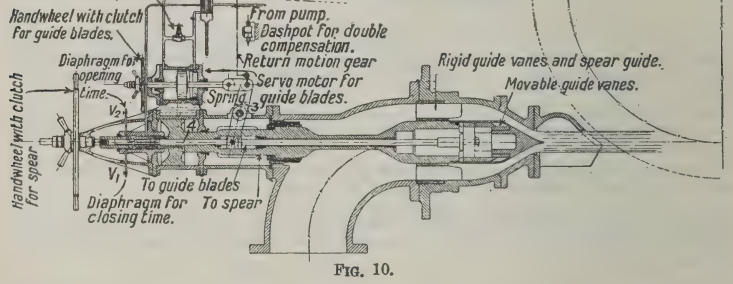


FIG. 10.

of the lever 3 on the spindle attached to the movable ring. At the same instant the rear end of the spindle 4 acts as a distributing rod and allows oil under pressure to pass into the left-hand chamber of the spear servomotor and connects the right-hand chamber to the exhaust; the spear is thereupon closed. The closing movement is controlled by the diaphragm V_1, which prevents the oil from flowing too quickly to the exhaust and thus obviates any undue pressure rise in the pipe-line.

For an increase in load the reverse operation takes place, and in this case the diaphragm V_2 controls the speed with which the spear is withdrawn and the normal operation of the jet is resumed.

In the arrangement, the strokes of the primary and secondary mechanisms—*i.e.* the blades and the spear—being always identical, no sort of cam device is required, which avoids troubles in the adjustment of governing apparatus. The spear shaft, together with the rear part of the guide blade rod, forms the distributing apparatus of the spear servomotor, thus simplifying the whole apparatus; and, being rigidly controlled by the servomotors, any danger of its getting out of order is obviated.

As the blades have no important force to sustain, they are light in design, and this fact, in conjunction with their small stroke, enables them to destroy the energy of the jet very quickly—in fact, almost instantaneously. This advantage is of the utmost importance as regards the speed variation and stabilises the running of the set considerably.

Another advantage of the governing device is the small amount of energy required, as the only effort to be overcome is that due to friction; this friction is due chiefly to the reaction of the whirling water, but is only a small percentage of the force exerted by it. Owing to this fact, the governing device requires a governor of small dimensions only, taking up less space than usual in the power house.

In case of breakdown of the governor, the governing mechanism acts as a security device, the dispersing blades being automatically pushed forward by the spring of the servomotor thus spreading the jet and avoiding accidents.

The sensitiveness of the governing device is such that good regulation is obtained without adding to the moment of inertia found in a generator of normal design; this is a consequence of considerable significance, as it is frequently found that with ordinary methods of jet control either an abnormal design of generator is called for or else a separate flywheel has to be provided.

As the results of tests taken on a 3,300 b.h.p. turbine at the Kinlochleven Power Station, running under 900 ft. pressure, it was noticed that when the full load was suddenly thrown off, the maximum speed variation was only 3·45 per cent. for a permanent speed variation adjusted at 0. When operating at fractions of full load, the speed variations on removing the load were still smaller, these qualities conducing to remarkably steady running. The pressure rise was also very low, amounting in the tests to a maximum of only 2½ per cent. for the combined movement of both the dispersing blades and the spear. For the blades alone the corresponding rise in pressure would, of course, be still less, and in fact for all practicable purposes negligible.

Notes on Regulators.

The best makes of hydraulic regulators for medium and high falls will govern within the following limits, the per cent. being that of the mean speed:

For a sudden variation of load of 25 per cent.	4 per cent.
„ „ „ 50 „	6 „
From no load to full load	10 „

The relative speed with which gates may be closed by hydraulic and mechanical regulators is greater for the hydraulic, the ratio being generally about 5 to 1 in favour of the hydraulic.

TURBINES.*

Classification of Turbines.

Action, Impulse, Tangential, or Pressureless Turbines.

In these turbines the passages are never completely filled with water and the flow of water is always under atmospheric pressure. During flow through the wheel the absolute velocity of the water is reduced as the water gives up its kinetic energy.

Reaction, or Pressure Turbines.

In these turbines the wheel passages are completely filled with water under a pressure which varies throughout the flow. During this flow through the runner, both the pressure and the absolute velocity of the water are reduced as the water gives up the energy to the wheel.

* The following books have been consulted and quoted: *Hydraulic Turbines*, by R. L. Daugherty; *Elements of Hydraulics*, by S. E. Slocum; *Water-Power Engineering*, by Daniel W. Mead; *Water-Power Plants and the Type Characteristic of Turbines*, by W. Basshuus; and *La technique de la houille blanche*, by Etienne Pacoret.

Direction of the Flow of Water.

In the radial outward flow, or centrifugal turbines (Fourneyron type, fig. 11), the guide-vanes ring is contiguous to the runner, and the water passes through the runner whilst moving away from the axis of rotation but in the plane perpendicular to it.

In the parallel, or axial flow turbines (Jonval type, fig. 12), the guide-vanes ring is superposed to the runner, and the water passes through the runner buckets whilst moving along a circular cylindrical surface having for axis the shaft of the turbine.

In the radial inward flow, or centripetal turbines (Francis type, fig. 13), the guide-vanes ring is contiguous to the runner, and the water moves towards the axis of rotation in a plane perpendicular to it.

In the mixed (radial, inward, and axial) flow turbines (American type, fig. 14) the water enters the wheel radially, in the direction of the axis, and, during its flow through the runner buckets, turns and discharges axially.

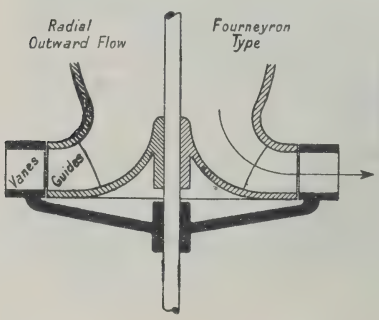


Fig. 11.

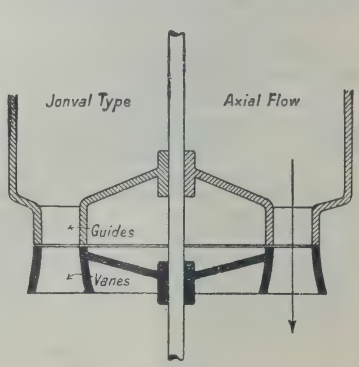


Fig. 12.

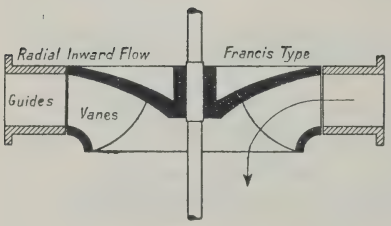


Fig. 13.

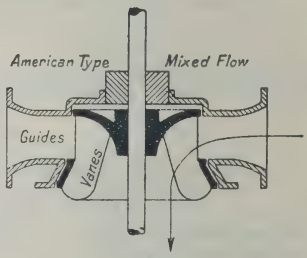


Fig. 14.

Position of the Shaft.

Turbines have either (*a*) an horizontal shaft, or (*b*) a vertical shaft; the latter are further classified as right-hand or left-hand turbines according to the direction of their rotation. If in looking down upon the wheel from above the rotation appears clockwise it is called a right-hand turbine. The reverse of this is a left-hand turbine.

Turbine builders have now reduced their production to three types, viz. the tangential or Pelton wheel, the centripetal flow or Francis turbine, and the helico-centripetal or American turbine, each of which satisfies special conditions.

Types of Turbines.

The Francis and the American turbines (both of which are of the reaction type) have superseded all the others, because they are cheaper to build, smaller, more easily regulated, and have a higher speed than the turbines of the Fourneyron or Jonval type.

The Francis Turbine.

Some of the main reasons for the pre-eminence of the Francis turbine over all other types of low and medium head turbines lie in its adaptability, since it can be designed either to pass large or small volumes of water for a given diameter of wheel, and can be constructed for low or fairly high heads, and in the relative ease with which it may be governed owing to the short stroke of its regulating apparatus, thus rendering it particularly suited for electrical work. Francis turbines may be used with either horizontal or vertical shafts, and with one, two, three, four, or even more wheels upon one shaft. The regulating device consists of a number of pivoted guide blades spaced round the circumference of the runner, and simultaneously moved to open or to close by means of a floating ring to which they are attached and which is connected to the governor by a lever and rocking shaft.

As a basis of comparison between Francis turbines, the specific speed of the runner is used. This term 'specific speed,' which was evolved by Prof. Camerer, is the speed of a runner which, operating under a fall of 1 ft. (or 1 metre), produces 1 h.p. The formula for specific speed, n_s is given by—

$$n_s = \frac{n}{H}\sqrt{\frac{N}{\sqrt{H}}}, \text{ or, } n_s = \frac{n\sqrt{N}}{H^{\frac{5}{4}}}$$

where,

n = revs. per minute; H = head in ft. (or metres); N = horse-power.

H may be in ft. or metres, and the result gives respectively the specific speed in English or metric denominations.

The speed classification, according to Camerer, is as follows:

n_s (metric) 50–120: slow speed runner; 120–250: normal speed runner; 250–400: high speed runner.

n_s = over 400 is known as an 'express' runner.

Recent progress has resulted in the construction of runners having still higher specific speeds, but these are mainly of interest in the case of very large powers under low heads, and also are only in the early stages of development. Slow-speed runners are always narrow wheels, and are largely used in spiral Francis turbines; high-speed runners have wide entrance and discharge areas.

Runners of Francis turbines are mostly made in series, geometrically identical for the same specific speed, all the dimensions varying in proportion to the diameter of the wheel.

All possible arrangements of Francis turbines may be considered under two heads—viz. 'Open-flume' and 'Cased' turbines.

Open-flume Francis Turbines.

On relatively low falls, up to about 30 ft., in dealing with large quantities of water, the Francis turbine is frequently made without casing and placed in an open flume or pit, the turbine shaft being carried through the end or retaining wall of the head water chamber. The runner is usually of a high specific speed type. This arrangement proves to be a very convenient one for direct coupling to an electric generator. Either a single runner, or two or more, may be carried on the same shaft. Fig. 15 shows a section through a twin turbine of this type, as made by Messrs. W. Gunther & Sons, Oldham, for direct coupling to an electric generator. On smaller plants the bearing farthest from the driving side is usually submerged, but in the plant shown this bearing is placed in a water-tight casing accessible from above the head water level, so that the bearing may at all times be examined.

Cased Francis Turbines.

Circumstances do not always permit of low-fall Francis turbines being placed in an open flume—*e.g.* when they are to be installed in existing buildings, or when the lay of the ground is more suited to a pipe conduit. Such turbines are then enclosed in steel or cast-iron casings, cylindrical or spiral. In fig. 15 a 500 h.p. twin turbine of Messrs. Gunther is shown, in which

GUNTHER & SONS' TURBINES.

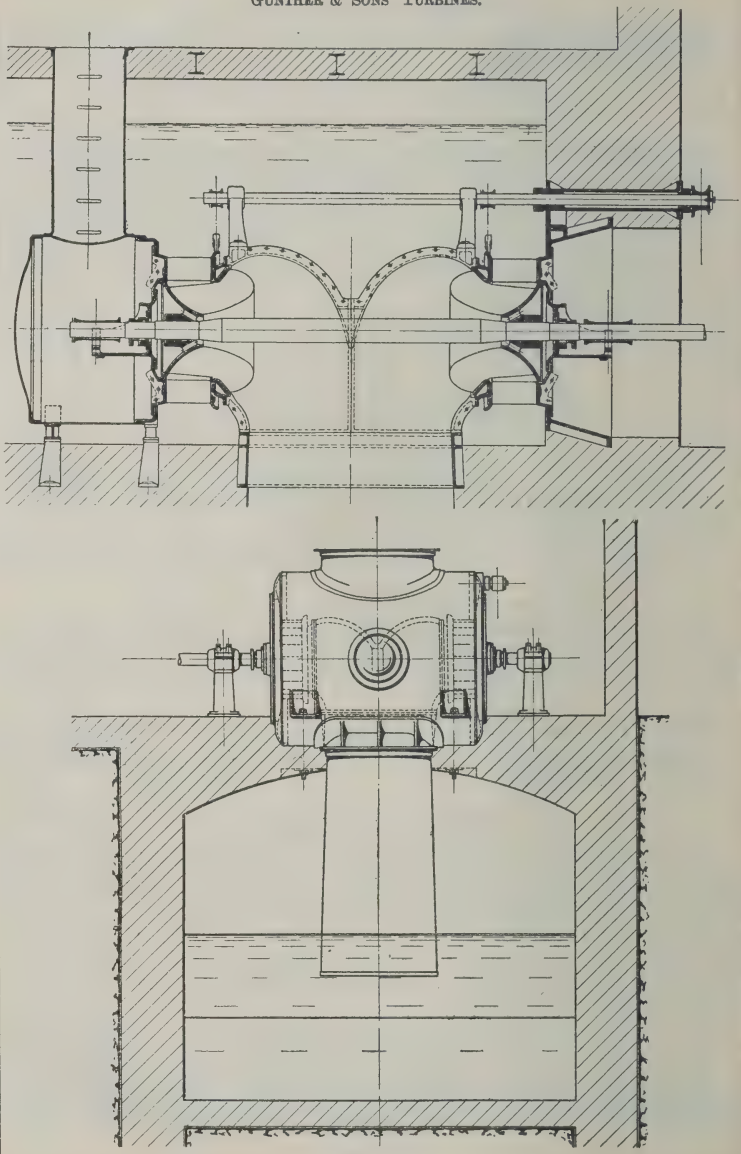


FIG. 15.

the water discharged from the two runners passes into a suction chamber formed by a double bend, and is discharged centrally into the suction tube, the whole being enclosed in a steel casing. The ring regulation of the guide ports for this arrangement of turbine may be inside or outside the casing, the more modern but more costly way being to have the regulating gear outside, and thus out of contact with the water.

One of the most valuable modern developments of the Francis turbine has been in the solution of the problem of dealing with medium heads, i.e. between about 50 ft. to 200 ft. For this range the high-speed runner would have an inconveniently high number of revolutions per minute, and a Pelton wheel may be unsuitable for several reasons—the speed may be too slow, or in order to deal with the quantity of water the Pelton wheel may become very large and heavy, or requiring perhaps two wheels on one shaft, with two or more nozzles to each. For such cases a slow specific speed Francis runner with a spiral casing of cast iron or steel may be used to advantage. In a spiral casing much higher velocities of the water may be taken, thus reducing the size and cost of the installation. The regulating arrangements are placed outside the spiral casing, and are easily accessible. A spiral Francis turbine (fig. 15, p. 1318), made by Messrs. W. Gunther & Sons, of Oldham, for driving a textile mill by means of a rope drive, has a head of 62 ft., with an output of 500 b.h.p. Regulation is effected by an oil-pressure governor.

The range of usefulness of the spiral Francis turbine naturally overlaps that of the Pelton wheel in many cases, spiral turbines having been used up to 800 ft. head or more, and the choice of a Pelton or a spiral Francis turbine is not always a rigid one.

American Turbines.

The American turbine, or mixed flow turbine, fig. 14 (p. 1316), is a modification of the Francis turbine, resulting from a demand for higher speed and power under low heads. Higher speed is obtained by reducing the diameter of the runner, greater power by increasing the capacity of the runner, making the buckets deeper and reducing the number of vanes. Since this decreased the discharge area, the vanes were curved so as to discharge axially; combined together these alterations have produced the high-speed high-capacity mixed-flow runner of to-day.

The best peripheral wheel speed for a reaction turbine varies between 0·57 to 0·87 of the theoretical spouting velocity of the water under the head considered. The efficiency of the modern reaction turbines is high, ranging between 80 and 85 per cent. under average conditions, and going up to 90 and 92 per cent. under the most favourable conditions.

(1) Crown Plate.
(2) Curb Plate.
(3) Shifting Ring.
(4) Speed Gate.
(5) Gate Bushing.
(6) Fulcrum Bolt.
(7) Link.
(8) Link Bushing.
(9) Gate Pin.
(10) Shifting Ring Pin.
(11) Hub.
(12) Rim.
(13) Bucket Vane.
(14) Main Shaft Key.
(15) Keeper Ring.
(16) Lignum - Vitæ Block.
(17) Block Holder.
(18) Adjusting Plate.
(19) Bearing Casing.
(20) Casing Head.
(21) Main Shaft.
(22) Regulating Shaft.
(23) Lever or Bell Crank.
(24) Bearing.
(25) Connecting Rod.
(26) Connecting Rod Pin.
(27) Cover Plate.
(28) Base Ring.

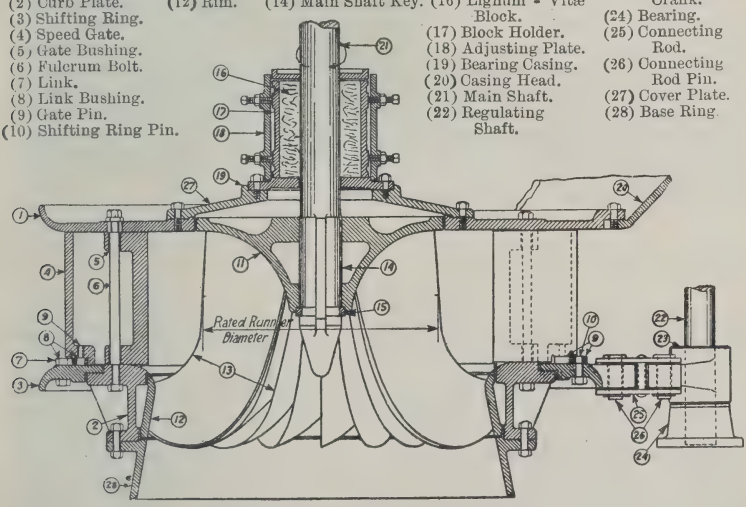


FIG. 16.—American Turbine Nomenclature.

British Turbines.

The Little Giant Double Turbine.

(*S. Howes, London.*)

The form of the wheel of this turbine is shown by fig. 17, and general views of vertical forms of complete turbines by figs. 18 and 19.

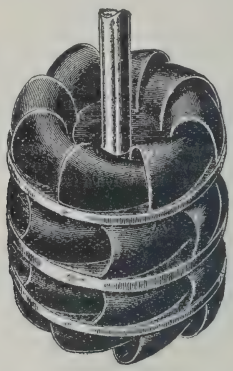


Fig. 17.—Side View of Little Giant Turbine Wheel, showing Form of Double Buckets.

The Little Giant is a double turbine having two tiers of buckets, or, strictly speaking, two turbines keyed to the same shaft, one above the other, and both running in the same case, the upper tier discharging at the top, and the lower one under the bottom of the case. The shaft is supported at the bottom end, and revolves upon an adjustable lignum-vitæ step in the bottom bridge-tree. The case is made of iron in the form of a scroll. This scroll, for larger sizes, is made of boiler-plate, which is planed parallel on the sides, and is held firmly between the cast-iron top and bottom plates by bolts. The case is supported by the bottom bridge-tree, which is bolted firmly to it. The bridge-tree is bolted firmly to the bottom of wheel-pit, upon which it rests. The gate box is made of cast iron and bolted to the case. The gate is also made of cast iron, and the surface of it, as also the ways or seat upon which it slides, are planed, thus forming a perfectly watertight joint. The pressure of water always holds the gate to its place, and the joint remains tight as the gate wears. The gates for the larger sizes of turbines, particularly if to be used under high heads, are mounted upon friction rollers.

The Division Plate.

Figs. 18 and 19 each show an iron plate passing horizontally through the turbine case, and dividing it into two compartments corresponding to the two tiers of buckets, one set being above this plate, discharging at the top of the case, the other below the plate, and discharging at the bottom; in fig. 18 the plate is shown as shut down, and in fig. 19 the plate is raised. Referring to fig. 19, the gate is shown raised clear up, or at 'full gate,' admitting the water to both tiers of buckets with the full pressure due to the head, giving the full power of both parts, or of the whole turbine. If but one-half of the power of the turbine were required, the gate could be dropped or closed down to the division plate, thus shutting the water off from the upper tier, but maintaining the full pressure on the lower buckets, which would give one-half the full power with half the quantity of water, thus obtaining as high a percentage of power at half-gate as at full-gate, which could not be done if there was only one set of buckets. If no division plates were used, the full pressure of water due to the head would be on the turbine only when the gate was full open, but when used

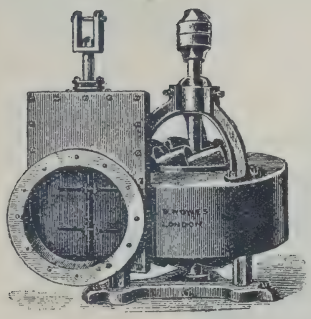


FIG. 18.—Vertical Little Giant Turbine, for joining to Iron Pipe.

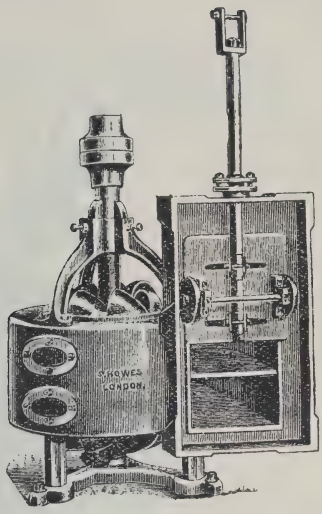


FIG. 19.—Vertical Little Giant Turbine, for joining to Wooden Pipe.

at half-gate the water, after passing through the gate-opening, would pass to both tiers of buckets but with less pressure; consequently the turbine would lose from 20 to 25 per cent. of the power in the water used.

DIMENSIONS OF VERTICAL LITTLE GIANT DOUBLE TURBINES.

Diameter of Turbine.	Diameter of Turbine Shaft.	Distance from Centre of Shaft to Centre of Gate Shaft.	Distance from Floor of Turbine Pit to Centre of Coupling on Turbine Shaft.	Distance from Floor of Pit to top of Turbine Buckets.	Space the Turbine occupies in Turbine Pit.	Size of Gate Frame for Wooden Spout.	Inside Diameter of Round Flange Gate Box for Iron Pipe.	Net Weight of Turbine.
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Lbs.
4	—	—	—	$8\frac{3}{4}$	—	—	—	—
6	$1\frac{7}{16}$	$7\frac{1}{2}$	$22\frac{1}{2}$	$12\frac{1}{4}$	$17\frac{1}{2}\times17\frac{1}{2}$	$13\frac{1}{2}\times$ 9	8	265
8	$1\frac{5}{8}$	$9\frac{3}{8}$	$26\frac{1}{4}$	15	20 $\times18\frac{1}{2}$	$16\frac{1}{4}\times10$	10	380
10	$1\frac{13}{16}$	$8\frac{7}{8}$	$29\frac{1}{4}$	$16\frac{3}{4}$	22 $\times21\frac{1}{2}$	$17\frac{1}{4}\times11\frac{1}{2}$	12	485
12	$1\frac{15}{16}$	$9\frac{3}{4}$	$33\frac{1}{4}$	$18\frac{3}{4}$	25 $\times25\frac{1}{2}$	20 $\times13\frac{1}{2}$	14	655
14	$2\frac{3}{16}$	$12\frac{1}{2}$	39	$22\frac{3}{4}$	29 $\times28\frac{1}{4}$	$24\frac{1}{2}\times15\frac{1}{2}$	18	760
16	$2\frac{7}{16}$	$13\frac{1}{2}$	$42\frac{1}{2}$	$25\frac{1}{2}$	35 $\times33\frac{1}{4}$	27 $\times16\frac{3}{4}$	20	1,140
18	$2\frac{13}{16}$	$15\frac{1}{2}$	$48\frac{1}{2}$	30	$42\frac{1}{2}\times38$	32 $\times20$	22	1,820
18*	$2\frac{15}{16}$	$15\frac{1}{2}$	$48\frac{1}{2}$	30	$42\frac{1}{2}\times38$	32 $\times20$	22	1,820
21	$3\frac{1}{8}$	18	$56\frac{1}{2}$	$32\frac{1}{4}$	45 $\times42\frac{1}{2}$	$36\frac{1}{2}\times22\frac{1}{2}$	24	2,190
21*	$3\frac{1}{8}$	18	$56\frac{1}{2}$	$32\frac{1}{4}$	45 $\times42\frac{1}{2}$	$36\frac{1}{2}\times22\frac{1}{2}$	24	2,190
24	$3\frac{7}{16}$	18	62	$37\frac{1}{2}$	$53\frac{1}{2}\times46\frac{1}{2}$	$38\frac{1}{2}\times26$	30	2,890
24*	$3\frac{7}{16}$	18	62	$37\frac{1}{2}$	$53\frac{1}{2}\times46\frac{1}{2}$	$38\frac{1}{2}\times26$	30	2,890
28	$3\frac{15}{16}$	$20\frac{3}{8}$	78	48	54 $\times58$	$52\frac{1}{2}\times28$	32	4,600
33	$4\frac{15}{16}$	$22\frac{5}{8}$	84	54	$66\frac{1}{2}\times61$	$55\frac{1}{2}\times33\frac{1}{2}$	34	5,550
33*	$4\frac{15}{16}$	$22\frac{5}{8}$	84	54	$66\frac{1}{2}\times61$	$55\frac{1}{2}\times33\frac{1}{2}$	34	6,775
38	$5\frac{1}{2}$	25	100	$67\frac{1}{2}$	78 $\times71$	69 $\times42$	37	8,000
44	6	$32\frac{1}{2}$	112	75	93 $\times78$	76 $\times66$	60	14,600

* Deep bucket.

Horizontal Little Giant Double Turbine.

(S. Howes, London.)

There may be many situations where a turbine placed upon a horizontal shaft can be used to better advantage than one on a vertical shaft. By using a horizontal turbine and placing it

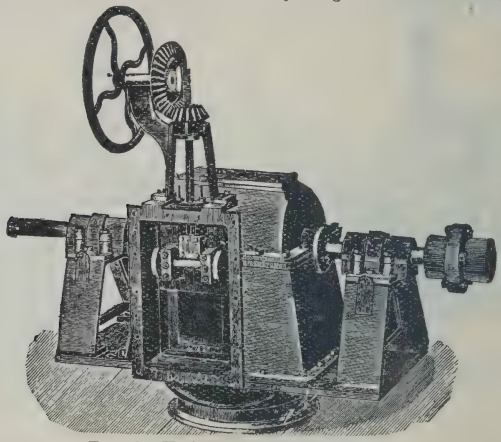


Fig. 20.—Horizontal Little Giant Turbine.

some distance above tail water, nearly the whole effect due to the head can be obtained with an air-tight discharge tube. This mode of setting a turbine (fig. 20) admits of placing a pulley directly on the turbine shaft, and belting to the line shaft.

Dimensions of Horizontal Little Giant Double Turbines.

Diameter of Turbine.	Diameter of Shaft.	Inside Diameter of Inlet Flange.	Outside Diameter of Inlet Flange.	Inside Diameter of Outlet Flange.	Outside Diameter of Outlet Flange.	Height to Centre of Shaft.	Length of Shaft (overall).	Height to top of Case.	Size of Bed Plate.
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
4	1 3/16	8	11 1/4	8	11 1/2	8 1/4	50	12	40×14
6	1 7/16	7 7/8	12	9	13	10 3/4	55 1/2	15 3/4	44×15 1/2
8	1 11/16	9 7/8	14	14	18	13	65	18 1/2	50×16 1/2
10	1 15/16	12 1/2	16 1/2	18	23	16	70	23	52×22 1/2
12	2 3/16	14	18 1/2	22	26	19	75	28	58×27 1/2
14	2 7/16	17 3/4	22 1/4	25	30	21 1/2	90	30 1/4	71×31
16	2 11/16	20	24 1/2	24×31	28×35	22 3/4	94	31	73×33 1/2
18	2 15/16	22	27 1/4	28	33	26 1/4	108	36 1/2	83×37 1/4
21	3 7/16	23 3/4	29	26×34	31×39	30	114	36 3/4	92×48
24	3 11/16	30 1/4	35 7/8	30×38	35×43	35 5/8	132	43	110×52 1/2
28	4 3/16	31 5/8	36 3/4	34×44	40×50	37 3/4	144	56 1/2	118×57
33	4 15/16	34	39	2×32	2×37	43 1/2	149	69	121×62

Size of Pipe for Little Giant Double Turbine.

Turbine 4 ins. dia.	Pipe 6 ins. dia.	Turbine 18 ins. dia.	Pipe 24 ins. dia.
„ 6 ins. „	„ 9 ins. „	„ 21 ins. „	„ 28 ins. „
„ 8 ins. „	„ 12 ins. „	„ 24 ins. „	„ 32 ins. „
„ 10 ins. „	„ 14 ins. „	„ 28 ins. „	„ 36 ins. „
„ 12 ins. „	„ 16 ins. „	„ 33 ins. „	„ 44 ins. „
„ 14 ins. „	„ 20 ins. „	„ 38 ins. „	„ 48 ins. „
„ 16 ins. „	„ 22 ins. „	„ 44 ins. „	„ 60 ins. „

Low Fall 'Gilkes-Francis' Turbines.

(Gilbert Gilkes & Gordon, Ltd., Kendal.)

The type of Francis turbine shown in fig. 21 is a medium-speed wheel (specific speed 63·0 r.p.m. in ft.-lbs. sec. units) which has been found a very useful type for small and medium powered installations. This turbine can be arranged with its shaft vertical, as shown in fig. 21, or can be fixed as a horizontal shaft turbine.

The connection of the movable guide vanes by a chain of links (as shown in the fig.) is

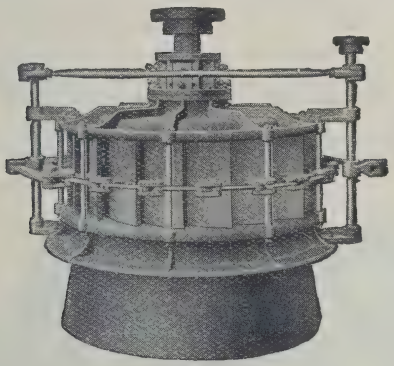


FIG. 21.

found to have a long life under ordinary working conditions, and is found to require less power than by any other method.

The vanes are so pivoted that they are as nearly as possible balanced in all positions. This fact is of great importance when it is required to control the turbine speed automatically by a governor.

FIG. 22.

All the wearing parts of the guides themselves and their connecting links are bushed with bronze and the pins are made of hard steel.

The wheel shown in fig. 22 is made of steel buckets bent to mathematically accurate curves, and then cast into an iron boss and outer skirt.

The wheel discharges to the tailwater through a cast iron flume or tail pipe, and when the turbine is arranged for a horizontal setting a cast iron discharge bend and tapered tail pipe is fitted.

For the vertical setting it is strongly recommended that the weight of the rotating parts be carried on a suitably designed ball thrust bearing arranged above the head water level. This type of bearing is much to be preferred to the lignum vitae pivot bearing under the water. The latter, though very simple and cheaper than the above water suspension bearing, may give trouble, especially if the water contains much sand or grit in suspension. The pivot is in any case difficult to inspect and costly to renew.

Medium Fall 'Gilkes-Francis' Turbine.

(Gilbert Gilkes & Gordon, Ltd., Kendal.)

Fig. 23 shows a turbine installed at San Juan Mines, in the Argentine. It is designed to develop 750 h.p. when using 7,270 cub. ft. of water per minute under a head of 70 ft. and running at 500 r.p.m.

Owing to the comparatively high speed in relation to the head, two runners are employed, each with its own casing but discharging into a central suction flume common to the two

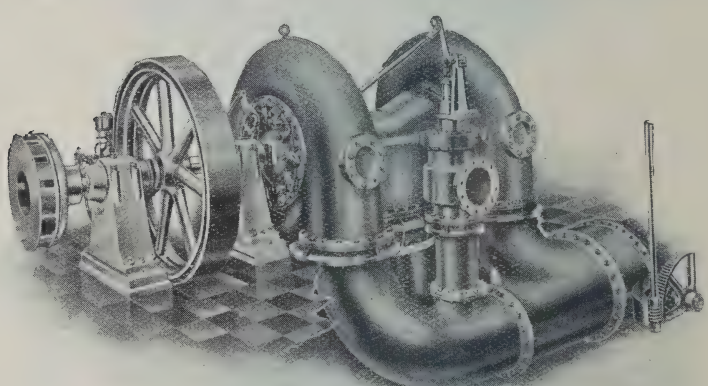


Fig. 23.

wheels, thus putting both wheels into balance and avoiding the necessity of thrust bearings. The casings are of the spiral form, which allows the kinetic energy of the water flowing through the pipes and casing to be utilised with a minimum of loss. The runners and guide blades are of the Francis type, the former being constructed with steel buckets cast into a c.i. boss and skirt.

The guide blades are of cast steel, working in phosphor-bronze bushes, and have a stem cast on which is accurately turned and carried through glands to the outside of the casing; on the end of each stem is keyed a c.i. lever, the lever being coupled together by means of bronze bushed links. Two of the stems are coupled by means of special levers to the governor. By this means all pins, joints, etc., are placed outside the turbine, where they are not affected by the action of the water and can easily be lubricated. Although this method of outside gate mechanism is rather more costly in the first place, its extra cost is soon compensated for by smoothness in running and absence of repairs. It has been known that pins acted upon by the water, assisted sometimes by electrolytic action, have required renewing after a few months' wear, entailing the shutting down of the plant for a considerable time to gain access to the inside of the turbine.

The speed of the set is controlled by a Gilkes oil pressure governor, assisted by a cast steel flywheel. In order to prevent shock on the pipe line due to the rapid shutting off of the water, an automatic by-pass valve is provided. The valve is connected direct to the governor and is so arranged that, directly any water is shut off the turbine, the valve lifts immediately, allowing

an equal quantity of water to pass through the valve, but immediately the movement of the gate ceases the valve slowly closes without shock on the pipe line. By the arrangement, the valve anticipates any rise in pressure in the pipe line. The speed of closing is regulated by an oil cataract which can be regulated to close at any desired speed.

This type of turbine is commonly adopted for falls varying from 30 ft. to 300 ft. On the higher falls a single spiral is used, discharging its water either from one side of the wheel or from both sides.

Spiral Turbines for Small Outputs.

With a view to meet the increasing demand for small but high efficiency water turbines, *Boving & Co., Ltd.*, have designed and recently placed on the market a machine which, although low in cost, nevertheless possesses a high efficiency.

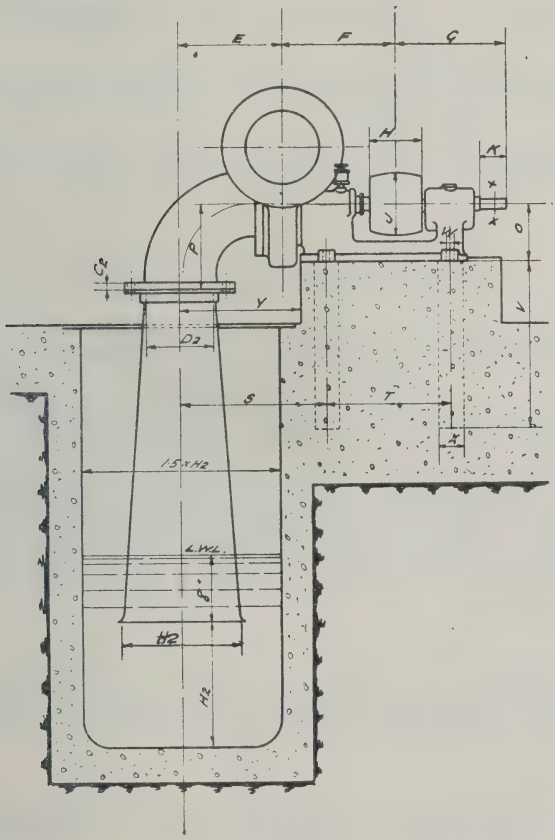


FIG. 24.

From fig. 24 it will be seen that the turbine is self-contained on its own bedplate, and consists of a specially designed spiral casing enclosing a runner of the Francis type, which is mounted on the end of the turbine shaft. The turbine is also provided with a belt pulley.

The shaft, extended at one end to accommodate a coupling, is supported on two metallic bearings, oil lubricated, and with the runner overhung; the bearings do not sustain damage should the water contain abrasive matter.

The discharge water from the runner passes through a specially designed suction bend and suction pipe into the tail race.

The foundations present no difficulties, and may be of timber blocks, steel girders, or concrete. Timber beams are frequently used with good results, as the rigid bedplate prevents the bearings from becoming out of alignment.

Another noteworthy feature is the inlet flange, which can be turned in various positions, thus simplifying the pipe line connection.

The turbine is normally arranged for the control of its speed by means of a hand-operated inlet sluice valve, but where necessary it can be equipped with the Boving patent automatic governor of the oil pressure type.

The machine described is manufactured for heads up to 60 ft., and the largest size for that head has an approximate output of 53 b.h.p. at a speed of 1,175 r.p.m. and a full-load efficiency of 82 per cent. This high efficiency was only obtained by careful design of the casing which has been evolved after a long series of tests in the laboratory.

This turbine is eminently suitable for the economic driving of mining, dairying, and similar machinery, and, owing to its unique and compact design, it can be dismantled into suitable small packages for conveyance by mule or other means where transport is difficult.

Selection of a Turbine Runner.

There are a certain number of commercial constants which are of great help to the engineer who is not a turbine designer, when attempting to select the best type of turbine to adopt for a given development and a chosen capacity per electro-generating unit.

The **Specific Speed** (N_s) is a theoretical characteristic which gives the means of comparing the relation between the conditions of speed, head and power of a runner in one installation with the same conditions relative to a runner in a totally different installation. This comparison is made after the conditions governing the design of both runners have been reduced to a common base, viz. the specific speed.

The specific speed of a runner may be defined as the speed at which any runner will run when it is reduced to such a size that it will develop one horse-power when operating under a head of unity. The numerical value, expressed in the metric system, of the specific speed of any runner may be found by first calculating the speed and power output of the runner under consideration for one metre head, and then mathematically reducing the runner in size until it will deliver one horse-power. The speed of this reduced runner when operating at its point of maximum efficiency is its rated specific speed; this speed is generally expressed in the metric system in order to adhere to an international standard. The formula is

$$N_s = 4{\cdot}45 \text{ r.p.m.} \frac{\sqrt{\text{h.p.}}}{h^{\frac{5}{4}}}$$

in which the horse-power and head are expressed in the foot-pounds system, while N_s remains in the metric system. The equivalent given in the foot-pounds system is called the 'unit speeds,' or 'type characteristics.'

$$N_u = \text{r.p.m.} \frac{\sqrt{\text{h.p.}}}{h^{\frac{5}{4}}}$$

In figuring the specific speed of any reaction turbine having more than one runner or any impulse wheel having more than one nozzle, the horse-power used in the formula, of course, is the power developed by one runner only, or one nozzle only.

For a given number of revolutions per minute (r.p.m.), and head (h), the h.p. output is proportional to the square of the specific speed; also for a given head and h.p. the r.p.m. of a turbine is proportional to the specific speed.

In applying the formula for the specific speed to turbines already built it will be found that the data of an impulse wheel will give a low specific speed, between 5 and 25, while the data on the radial inward-flow turbine and on the radial inward axial flow turbine will give a high specific speed between 40 and 450. It will be noticed that there is a gap between the impulse wheel and the reaction turbine, as values of N_s between 25 and 40 do not exist with either type; but by using more than one nozzle on one impulse wheel it is possible to increase the value of N_s, so as to cover the entire field.

The formula for the specific speed contains only the two quantities, 'speed' and 'capacity for a certain head,' but as these quantities determine the areas and angles of the turbine, and thereby again the losses which take place during the passage of the water through the turbine, it will be understood that the specific speed is inherently related to the efficiency which can be expected from the turbine.

EFFICIENCY AT THREE-QUARTER POWER OF IMPULSE WHEELS.

Specific Speed.	Efficiency.
	Per cent.
5	80
9	79
13	78
18	77
22	76

EFFICIENCY OF RADIAL INWARD-FLOW TURBINES.

Type.	Specific Speed.	Maximum Efficiency.	Efficiency at half power.
		Per cent.	Per cent.
Low speed . . .	40—100	82 at 0·75 power	76
Medium speed . .	130—220	82 at 0·75 ,,	75
High speed . . .	260—360	80 at 0·80 ,,	70
Very high speed . .	400—450	73 at 0·90 ,,	53

This table shows that the different types of radial inward turbines are of different qualities. The low-speed turbines are very favourable as to efficiency over a wide range of loads, but they are practically limited to high heads. Medium-speed turbines are also very favourable as to efficiency over a wide range of loads. High-speed turbines are favourable as to efficiency at about 0·8 load, but show a marked decrease at about half load. The extreme high-speed turbines may in some cases be tolerable as to efficiency at about 0·9 load, but at about half load the efficiency is below the limit of rational utilisation of the water. The use of such turbines is only justified for low heads, with a supply of water ample in all seasons, and where the head remains rather constant.

The curve (fig. 25, p. 1328), based on tests of higher grade turbines, shows that if one desires the best efficiency possible a specific speed should be chosen between 110 and 220.

The modern practice is towards the use of large units running at the highest speed compatible with the exigencies of economical and prudent operation, so that, although the choice of a turbine depends on the power and speed desired as well as on the head available, it would seem as if the head was in most cases the deciding factor. The following table, based on this assumption, gives the type of runner desirable in function of the head.

Heads from 500′ up.	Specific speed 5 to 25.	Type of Runner Impulse Wheel.
For heads up to 500′ . .	40 to 100	Low speed, reaction.
,, ,, ,, 300′ . .	100 to 150	
,, ,, ,, 150′ . .	130 to 180	Medium speed.
,, ,, ,, 60′ . .	175 to 250	
,, ,, ,, 30′ . .	260 to 360	High speed.
,, ,, ,, 15′ . .	400 to 450	Very high speed.

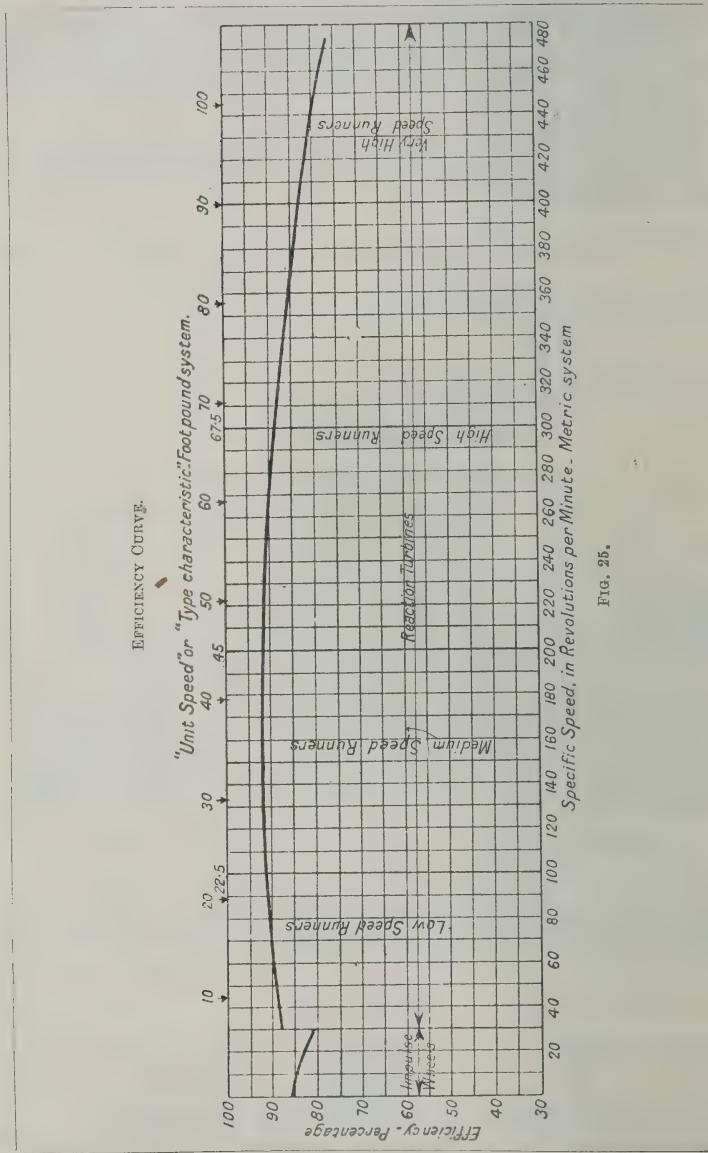

Efficiency Curve.
"Unit Speed" or "Type characteristic" Foot pound system.
Specific Speed, in Revolutions per Minute. Metric system
Efficiency - Percentage
Low Speed Runners
Medium Speed Runners
High Speed Runners
Very High Speed Runners
Reaction Turbines
Impulse Wheels

FIG. 25.

Relation between the Speed and the Discharge of a Turbine and the Head.

For maximum efficiency the peripheral speed (U_1) of any runner is a definite fraction of the spouting velocity under the given head,

$$U_1 = \phi \sqrt{2gh};$$

the best wheel speed is given by values ranging from

0·43 to 0·47 for impulse wheels,
and 0·53 to 0·57 for reaction turbines.

The discharge through a turbine is like the discharge through any orifice and varies as the square root of the head, and the coefficient of discharge has a definite value for the best value of ϕ; it follows that *the best speed and the discharge of a given turbine vary as the square root of the head* ($h^{\frac{1}{2}}$).

Relation between the Input and the Output of a Turbine and the Head.

The power available in each unit volume of water is proportional to the head, and since the quantity of water discharged varies as the square root of the head, then *the power input varies as the three-halves power of the head* ($h^{\frac{3}{2}}$).

The losses of head in any turbine vary as the squares of the various velocities concerned. Since these velocities vary as the square root of the head, their squares will vary as the first power of the head. The amount of water varies as the square root of the head, and since the power lost is the product of these two items, it follows that the *various hydraulic losses vary as the three-halves power of the head.* As the hydraulic losses vary in just the same proportion as the power input, *the hydraulic efficiency will be independent of the value of the head.*

The power output of a given turbine varies as the three-halves power of the head.

Relation between the Diameters of Stock Runners of the same Type.

When a certain type of runner has been perfected, a whole line of stock runners of that type may then be built with diameters ranging from 10 to 70 inches. All of these runners will be homologous in design, that is to say, they will have the same specific speed and each runner will simply be an enlargement or reduction of another. Having the same characteristics, that is, the same value of b and c (coefficient of discharge), they will therefore follow certain laws of proportion, which can be summed up as follows for a series of runners of homologous design:

1. The best r.p.m. will be inversely proportional to the diameter.
2. The discharge of any turbine of the series will be proportional to the square of the diameter.
3. The power of any turbine of the series is proportional to the square of the diameter.

These relations are of practical value, because if the speed, discharge, and power of any runner is known by accurate test, predictions may then be made regarding the performance of any other runner of the series.

Relation between the Diameter of a Runner and its Discharge.

Since the discharge of any runner is proportional to the square of its diameter and to the square root of the head, we may write

$$q = K_1 D^2 \sqrt{h} = \text{cub. ft. per sec.}$$

The value of K_1 depends upon the velocity of the water entering the runner and the area of the passages in the runner. The velocity depends upon the value of the coefficient of discharge c, which varies from 0·6 to 0·8, and the area depends upon the diameter, the height of the runner, the angle, and also the number of buckets and guides. To summarise, the values of K_1 vary

for the tangential water wheel, from 0·0002 to 0·0005;
for the reaction turbine, from 0·0014 to 0·0360.

Relation between the Diameter of a Runner and its Power.

Since the power of any runner is proportional to the square of the diameter and to the three halves of the head, we may write,

$$\text{h.p.} = K_2 D^2 h^{\frac{3}{2}}$$

for the tangential water wheel, $K_2 = 0{\cdot}000018$ to $0{\cdot}000045$;
for the reaction turbine, $K_2 = 0{\cdot}00012$ to $0{\cdot}0033$.

Determination of the Constants.

The constants given may be computed from theory, but for practical use they should be secured from test data. The engineer who has to select a type of turbine can determine these factors by taking the information given in the catalogues of the turbine manufacturers. The tables contained in these catalogues are supposed to be made from actual tests, and give the discharge, power, and speed of runners of various diameters under various heads. Of course, as each manufacturer usually makes several lines of runners, there will be as many distinct values of the constants as he makes types of runners.

The Draft Tube.

It is generally found advantageous to install the turbine at an elevation above the elevation of the water in the tail race, but in order to do this without any loss of head it is necessary to connect the turbine discharge with the tail race water by means of an air-tight steel or concrete tube whose efflux is below the water level. This tube was formerly known as the Jonval device, but now generally called a 'Draft Tube.'

In considering the effect of the draft tube it should be remembered that a given static head may exist either wholly in the form of pressure or pressure head, or wholly in the form of vacuum or suction or draft head, or part of the head may be in the form of pressure head and the remainder in the form of draft head.

Also, a given head may exist wholly in form of pressure or static head, or wholly in the form of velocity, or part of the head may be in form of pressure and the remainder in the form of velocity, and pressure can be converted into velocity or velocity into pressure.

The effect of a draft tube on a turbine is to remove a part of the back or counter pressure due to the pressure or weight of the atmosphere, by a hanging water-column creating a partial vacuum. This water column being held in equilibrium by the pressure of the atmosphere, its height or head cannot be greater than 34 feet.

The draft head is the vertical distance between the centre of the shaft and the tail water for horizontal wheels, and between an horizontal line passing through the centre of the runner buckets and the elevation of the tail water for vertical turbines.

It is never possible to realise an effective draft head of 34 feet, as it is necessary to subtract the various losses due to the friction of the water in the draft tube, to the change in the speed of the water between the turbine runner and the lower end of the draft tube, and to the momentum or velocity still contained in the water while leaving the draft tube.

The absolute velocity in feet per second with which the water issues from the runner buckets and enters into the draft tube, assuming its diameter to be that of the outside of the runner, should be from $\cdot 30\sqrt{2gh}$ to $\cdot 15\sqrt{2gh}$, and the velocity at the outlet of the draft tube may be $1\sqrt{2gh}$, especially if the discharge is in the direction of the flow.

When no draft tube is used, the heads corresponding to the velocities of discharge from the turbine are completely lost, whilst with a properly designed draft tube it is possible to recover about 75 per cent. of the gain in head, due to the difference between the velocity head at the turbine and the velocity head at the outlet of the draft tube.

The following table gives the greatest draft-head permissible in practice, under average conditions, and for different diameters of draft tube.

Draft Head Permissible in Practice.

Diameter in feet.	Draft Head in feet.	Diameter in feet.	Draft Head in feet.
0·5	31·0	6	17·0
1·0	29·5	7	15·4
1·5	28·1	8	14·2
2·0	26·7	9	13·2
2·5	25·3	10	12·3
3·0	23·9	11	11·4
3·5	22·6	12	10·6
4·0	21·4	13	9·8
4·5	20·2	14	9·0
5·0	19·0		

The draft head given in this table is equal to the difference of elevation between the turbine and the tail water plus the velocity head corresponding to the speed with which the water leaves the draft tube.

The draft tube should be calculated to resist the atmospheric pressure, which tends to make it collapse.

Seal Rings of Hydraulic Turbines.

Water leakage between the turbine runner and the stationary casing is an important item of loss, especially where units operate at high heads. Ordinarily, labyrinth packing is employed, which causes the water to take a very circuitous path and be subjected to close clearances. Abrasive material contained in the water, however, has a disastrous effect on such metallic packing, and is the cause of objectionable repairs. Solid matter jamming in such clearances may cause the revolving parts to bind or run out of centre.

Rubber seal rings being flexible, eliminate a great deal of the trouble encountered, reducing the wear almost to a negligible amount. They are fitted to give material contact with the shaft, the water film acting as a lubricant. Rubber of the consistency of that found in the outer casing of an automobile tire has proved the most satisfactory, after a year of experimenting by the Pelton Water Wheel Co. (America).

In most cases of putting the rings in old units, the change is quite simple and involves only the removal of the existing all-metal wearing rings or sealing rings and the substitution of the rubber rings. Usually, this is done without other modification of the unit. The present construction of the seal rings, particularly for high-head turbines, requires that the rubber shall be vulcanised directly into the metal rings. When the rubber becomes worn, however, it is unnecessary to renew the metal ring section, as it can be again used for vulcanising a new rubber facing to restore the ring to its original condition. This results in a considerable saving in cost of the seal rings over that for the initial installation.

The highest head under which rubber seal rings are now used is 350 ft., but similar rings, in process of construction for the Oak Grove turbines, will operate under a head of 850 ft. The largest rubber seal rings thus far made are 5½ ft. in diameter, but plans are being developed for making seals of this type commercially for the very large diameter low-head turbines used extensively in the eastern part of the United States. This plan contemplates the manufacturing of the rubber seals in strips, which are then secured to their ring containers.

For corrosion-proof centrifugal pumps used for pumping acids and other corrosive fluids, a material called 'ceratherm,' invented by Mr. W. Hayhurst, M.Sc., has been introduced as a lining; it is stronger than fused silica or stoneware, and does not crack even when heated to redness and plunged in cold water. It is inert as porcelain to all chemicals and acids, and pumps of this material have been used for elevating nitric, hydrochloric, and sulphuric acids, hot and cold, hot ferric chloride, copper chloride, acetic acid, and various corrosive materials used in connection with explosives, collodion manufactures, and munition recovery plant. The material is homogeneous, and can be moulded to any shape required. It contains no soluble material, and is not brittle.

A great deal of benefit is said to be derived from an installation in a turbine plant of a Western industrial concern, whereby the lubricating oil in use in three 1875 k.v.a. turbo-generators is passed through 100 ft. of 2-in. copper coil, the latter being immersed in circulating cooling water from the ocean. The effect is a decrease in temperature of this oil of 40° F.

REGULATION OF TURBINES

In many hydraulic plants, and notably in the hydro-electric plants, it is necessary that the rotating speed of the turbines should be constant, regardless of the power developed. This can be done either by a hand or by an automatic regulation of the water openings, in proportion to the power utilised. The automatic regulation is adopted in preference to the hand regulation in all important plants.

The water wheel governors are controlled by a centrifugal ball mechanism just as in the case of the steam engines, with the difference that, as the force required to move the turbine gates is large, it cannot be supplied by the centrifugal ball governors, but must be borrowed either from the hydraulic motor itself or from an exterior source.

The gates are therefore moved by a 'relay,' which is a device for transmitting energy from a source of energy independent—as to quantity—of the centrifugal governor balls but controlled by them in its application. If the relay is of 'mechanical type' the power required to operate it and the gates is transmitted, when needed, from the wheel by means of shafts, gears, friction-clutches, belts and pulleys, or other mechanical devices. In mechanical governors the flyballs may actuate pawls, friction gears, or other mechanical devices which will bring the relay into action.

If the relay is of the hydraulic type, it usually consists of a piston connected by some mechanical device to the gate rigging and moved by means of the hydraulic pressure of water taken from the penstock, or other source, or by oil supplied under high pressure from a reservoir. The pressure of the oil in the reservoir is maintained by compressed air supplied by power taken from the wheel itself. The oil thus used in moving the piston is exhausted into a receiver, from which it is pumped back into the supply reservoir. The hydraulic relay is commonly controlled by the ball governor through the medium of a small valve, which by its motion either admits the actuating water (or oil) directly to the cylinder or to a secondary piston controlling a larger admission valve.

Electrical methods of actuating the relays controlled by means of governor balls have been used to some extent but are not nearly so common as mechanical or hydraulic devices.

The best makes of hydraulic governors for medium and high falls will regulate the speed within the following limits, the percentage of variation being that of the mean speed :—

	Speed Variation. Per Cent.
For sudden variation of load of 25 per cent.	4
" " " 50 per cent.	6
From no load to full load	10

Regulation due to Flywheel Effect.*

Flywheel effect is the capacity of a rotating mass to store or supply energy. This effect is expressed at 1 ft. radius and is equal to the moment of inertia of a mass, that is the weight by the square of the radius of gyration, or pounds feet squared. Modern alternating current generator rotors are built in the shape of a common flywheel and usually have sufficient flywheel effect to obtain satisfactory regulation. The flywheel effect of the rotating parts of a turbine varies with different types and usually is not considered, but it should be carefully considered for large low-head multiplex and large diameter impulse turbines. In smaller turbines it may be regarded simply as introducing a factor of safety.

The regulation due to any given flywheel effect can be computed by means of the formula,

$$d = \frac{800{,}000 \times \text{h.p.} \times \text{T}}{\text{r.p.m.} \times \text{r.p.m.} \times \text{W}_r^2}$$

where,

d = percentage of variation in speed allowed ;
T = time of closure of governor in secs. ;
W = weight of rotating masses in lbs. ;
r = radius of gyration in ft. ;
W_r^2 = moment of inertia of the rotating masses in lbs. ft. squared.

Actually the relative change of speed (d) on account of the friction load of the turbine and generator is smaller than that given by the above formula. The effect of this friction load will vary according to the various types of turbines used, but the original formula may be modified safely as follows :—

$$d = 0{\cdot}8d.$$

Another cause which should be taken into consideration, providing the turbine is properly designed so as to give its maximum efficiency at normal or synchronous speed, is that the efficiency is reduced with either increasing or decreasing speed, which has the tendency to reduce further the relative change of speed (d).

The amount reduced varies with the type of runner, being greatest for a high-speed, high-power runner, and least for a low-speed, low-power runner, since if correctly designed, and therefore following the laws of hydraulics, the change in efficiency will be greater for a high-speed than for a low-speed runner for the same per cent. of variation in speed. The formula modifying d, for this cause, is as follows :—

$$d \text{ eff.} = \frac{d}{1 + \frac{d}{\frac{n_2}{n_o} - 1.}}$$

The value of $\frac{n_2}{n_o}$ varies with the specific speed, and we have

Specific speed	60	90	130	180	220 to 280	310 to 370	400 to 460
Value of $\frac{n_2}{n_o}$	1·8	1·7	1·6	1·5	1·4	1·3	1·2

For impulse turbines 1·8 can be used as the value of $\frac{n_2}{n_o}$. To find the speed variation for any other than a full load change, the following must be considered : If T is the regulating time for the total governor stroke or 100 per cent. load of the turbine, it would appear that for 50 per cent. load or stroke, the regulating time would be $\frac{T}{2}$, and for 25 per cent, $\frac{T}{4}$, etc.

* Adapted from *Speed Regulation of Hydro-electric Plants*, by Wm. F. Uhl.

The regulating time for the total governor stroke or the governor closing time should be adopted only after a thorough study. Sensitive governing means severe wear and tear on the gates as well as increased shocks in the penstocks, and should be avoided where possible. Many lighting plants are successfully operated where it requires 20 seconds to close the gates, and while 20 seconds may be extreme, a closing time of 5 seconds is quite satisfactory for all but the most important or special plants.

Regulation of Turbines.

(Contributed by W. Gunther & Sons, Oldham.)

Regulation of turbines is of the utmost importance with electric power stations, where two or more generators have to run in parallel. Every change of load on a turbine is accompanied at the same moment by a change in speed, Z_{off}, when load is thrown off, and Z_{on}, when load is thrown on. These momentary changes of speed are disturbing to the smooth running of the plant, and it is in the greatest interest of the buyer of a turbine that the values of Z_{off} and Z_{on} should be as small as possible. In many cases certain conditions in regard to these values are asked from the turbine maker. The *momentary* changes of speed Z_{off} and Z_{on}, and the corresponding *remaining* changes in the revolutions Y_{off} and Y_{on}, which also must be as small as possible, are generally given in per cent. of the normal revolutions. If X_{off} and X_{on} denote the respective changes of load in per cent., the following conditions are suitable for average plants:—

$$\left.\begin{matrix} X_{off} \\ X_{on} \end{matrix}\right\} \text{25 per cent.,} \qquad \left.\begin{matrix} Z_{off} \\ Z_{on} \end{matrix}\right\} \text{not more than } 3\tfrac{1}{2} \text{ per cent.,}$$

if the corresponding values of Y are

$$\left.\begin{matrix} Y_{off} \\ Y_{on} \end{matrix}\right\} \text{1 per cent. at the most.}$$

Regulating stipulations can be made more or less severe according to the value of Z required. The *desired* speed change percentages are considerations for the power consumer; for the practically *obtainable* percentages the relative pipe length is, in the first instance, the deciding factor. The term 'relative pipe length' expresses the ratio: length of pipe line / head. The greater the relative pipe length, the more difficult it is to obtain satisfactory governing, and in many cases a compromise between the desired and the practically obtainable regulating percentages is the only solution.

As a general guide, the following percentages of speed variation should be obtainable:

Plant with a Good Relative Pipe Length (2 to 3).

X_{off} per cent.	X_{on} per cent.	Z_{off} per cent.		Z_{on} per cent.		Y_{off} per cent.		Y_{on} per cent.	
		Francis.	Pelton.	Francis.	Pelton.	Francis.	Pelton.	Francis.	Pelton.
25	25	2	3	$2\frac{1}{2}$	$2\frac{1}{2}$	1	1	1	1
50	50	5	5	6	6	2	2	2	2
100	100	10	12	13	—	3	3	3	3

Plant with Unfavourable Relative Pipe Length (over 3).

X_{off} per cent.	X_{on} per cent.	Z_{off} per cent.		Z_{on} per cent.		Y_{off} per cent.		Y_{on} per cent.	
		Francis.	Pelton.	Francis.	Pelton.	Francis.	Pelton.	Francis.	Pelton.
25	25	5	$4\frac{1}{2}$	6	5	1	1	1	1
50	50	9	8	12	$9\frac{1}{2}$	2	2	$2\frac{1}{2}$	$2\frac{1}{2}$
100	100	18	15	—	—	$3\frac{1}{2}$	$3\frac{1}{2}$	4	4

With respect to the current consumer two distinctions may be made:—

(1) Electric power stations which supply current for public purposes (street lighting, trams, etc.).

(2) Private industries or private power-distributing stations with mechanical or electrical power transmission between turbine and driven machinery.

In power stations of order (1) considerable instantaneous changes of load may be experienced at any time. It is therefore necessary that the turbines driving electric generators which supply current for public purposes (especially if trams or electric railways are being supplied) be so designed that they can withstand severe working conditions imposed by load changes. In plants of order (2), similar though less severe conditions obtain, and only if an accident happens is full load thrown off suddenly.

A very close relationship exists between the pipe line and the amount of flywheel effect (Σ GD^2) of the turbine, a point not often sufficiently realised by the intending purchaser. In governed plants it is important that sufficient flywheel effect be provided, and a portion at least of the necessary flywheel effect may be incorporated in the rotor of the generator. The amount of flywheel effect required is in proportion to the horse-power, the length of pipe line, and the closing time of the governor, and is inversely proportional to the square of the number of revolutions per minute of the turbine, turbine makers having their own formulæ. If a water turbine is regulated by an automatic governor, and satisfactory regulating conditions are expected, the whole mass of the rotating system must have a minimum mass moment with respect to the axis of rotation. The necessary amount of the GD^2 required is a matter to be decided by the turbine maker, because not only is the flywheel a heavy addition to the cost, but the characteristics of the governor and the efficiency of the turbine at different loads play an important part, details of which the maker only has at his disposal.

Control of Admission of Water into Reaction Turbines.

There are three different systems of regulating the amount of water supplied to the turbine in case of a variation of the load; these are (1) the cylinder gate, (2) the register gate, (3) the wicket gate.

(1) The Cylinder Gate, fig. 26, as its name indicates, consists of a cylinder closely fitting the guides and moving in a direction parallel to the axis of the turbine, thus decreasing or increasing

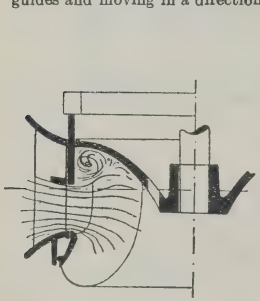


FIG. 26.—Cylinder Gate.

FIG. 27.—Register Gate.

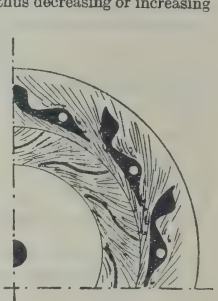


FIG. 28.—Wicket Gate.

the area left between the guide-vanes for the passage of the water. This type of gate is the simplest and cheapest but also the poorest, as the sudden contraction of the vein when passing under the partially closed cylinder gate and the sudden expansion that must take place in the bucket, produces eddying which results in decreased efficiency.

(2) In the Register Gate, fig. 27, the guide-vanes are made in two parts. The inner portion next to the runner is stationary, the outer portion is on a ring which may be rotated far enough

to shut the water off entirely if necessary. While this is more efficient than the preceding type, there is still a certain amount of eddying when the register gate is partially closed, with a consequent decrease in part gate efficiency.

(3) The Wicket Gate, fig. 28, also called the pivoted guide-vane, consists of a connection rod from the governor operating a shifting ring, which, by means of links, causes the vanes to rotate about their pivots, thus altering the area of the passages through them. This system has the working parts outside the case, which is a great advantage; its maximum efficiency is obtained before the gates are opened to the greatest extent.

Turbine Governors.

The sole efficient survivor of all the various types of governors made in the past is practically represented now by the latest forms of oil-pressure types. Oil-pressure governors as now designed are capable of giving extremely close speed regulation, the degree of regulation obtainable depending in a large measure on the hydraulic conditions under which the plant has to operate, as, although governors can be constructed which will entirely close or open the turbine gates in even as short a time as 1 second, it has been found in practice that from 2 to 3 seconds is about the maximum rate of operation which can be adopted, and even this rate is often not possible on turbines supplied through long pipe lines, owing to the dangerous water-hammer effects which would be set up. In installations of this description, relief valves can be used and quick rate of action of the governor maintained, and, on high falls, Pelton wheels provided with combined spear and deflector regulation can be successfully operated without any danger of water-hammer effects.

In practically every case it is necessary that a certain amount of flywheel effect should be provided in the plant, to assist in speed regulation and to give up or absorb power as required during the time taken by the governor to operate.

Owing to the quick rate of governor action generally adopted, it is of extreme importance that the turbine guide blades and operating gear should be of heavy and solid construction, as otherwise considerable lost motion and excessive wear will soon be set up in the gate operating gear when worked in connection with a quick-acting governor. The light construction of guide vanes and operating gear in turbines, generally known as the American type, is not suitable for obtaining efficient results and long life of working parts when used in connection with quick-acting governors.

Theoretically, speed regulation with modern quick-acting governors can be obtained to within 1 per cent. or 2 per cent. of normal with full load suddenly thrown on or off, but practical consideration of regulating gear design, governor capacity, and weight of flywheel renders such close regulation impracticable for all types of re-action turbines, although impulse turbines of the Pelton type with deflector regulation could be controlled, if necessary, within such close limits. Under ordinary conditions of power-station running, however, it is usually quite unnecessary for such fine speed regulation, and the general standard practice of governor design is to arrange for speed regulation to within 2 per cent. to 3 per cent. of normal under all ordinary working conditions, with a maximum momentary variation of 12 per cent. to 15 per cent. in the unlikely event of the whole load being suddenly thrown off.

In most standard forms of oil-pressure governors relatively small capacity oil-pumps have been used, with a reserve of power provided in the form of compressed air in the oil-receiver. This type of governor was often a source of trouble, owing to the difficulty sometimes experienced in maintaining a sufficient quantity of compressed air in the receiver, and in a number of cases to the compressed air having a detrimental effect on the oil used for operating the governor. The modern tendency is to adopt direct pumping governors without a compressed air chamber, provided with a pump of ample capacity for supplying oil to the operating cylinders at the maximum rate required.

Gilkes Oil Pressure Governor.

(Gilbert Gilkes & Gordon, Ltd., Kendal.)

This oil pressure governor (fig. 29) is the type constructed by Messrs. Gilkes for use on small and medium-size turbines, or in the larger sizes for controlling turbines of any power; it can equally be used for controlling Pelton wheels, but in Messrs. Gilkes' opinion the most suitable form of governor for controlling impulse wheels of any type is their patent shaft governor, with which much closer speed regulation can be obtained than is possible with the oil pressure type. The power of the shaft governor, however, is not enough to operate the gates on a reaction turbine.

The oil pressure governor is self-contained with its oil reservoir, oil pump, and speed-control mechanism. The base casting forms a reservoir for the oil, and a rotary pump is provided in this base which is driven directly from the turbine shaft.

This oil pump provides the oil pressure by means of which the governor operates the turbine gates. A separate drive is taken from the turbine shaft to operate the governor pendulum, which is of the Pickering type, and has great advantages in simplicity and sensitiveness over other types of pendulum. The pendulum operates upon a relay distributing valve which controls the direction in which the oil pressure provided by the pump will act.

In the upper part of the governor base casting a differential piston is fitted in the cylinder, so arranged that it can be operated in either direction according to whether the oil pressure from the pump is supplied to one side or the other of the cylinder. The piston is mounted on a piston rod which projects through the governor base at both ends. At one end it is fitted with a large hand wheel; the other end carries an eye and is arranged to operate the lever for moving the turbine gates. To this end of the piston rod is also secured the lever which controls the compensation mechanism of the governor. This mechanism is of the floating lever type; by a suitable adjustment the governor can be arranged to give either a small percentage drop of speed as the load comes on or a constant speed at all loads, or even a small percentage of rise in speed as the load increases.

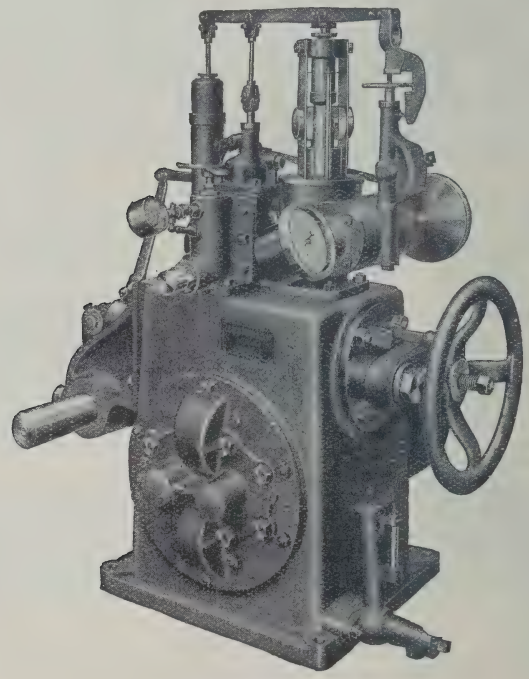


FIG. 29.

Where turbine plants have to work in parallel, the most convenient arrangement is to arrange the governor for a small drop in speed, as the load comes on, to ensure parallelism.

For isolated plants the constant speed or rise in speed characteristic is the most useful.

The governor mechanism is so arranged that the permanent speed may be varied by 5 per cent. above or below normal at will while the plant is at work. The whole governor is very accessible, and all bearings are of the continuously lubricated type, so that the governor may be run for long periods without stoppage.

TURGO IMPULSE WHEEL.

(Gilbert Gilkes & Gordon, Ltd., Kendal, Sole Makers and Patentees.)

One of the most important developments of recent years in impulse wheel design has been the introduction by *Messrs. Gilkes* of their Turgo Impulse type. Fig. 30 shows an outside view of this wheel mounted in cast iron casing and arranged for control by Messrs. Gilkes' Patent Shaft

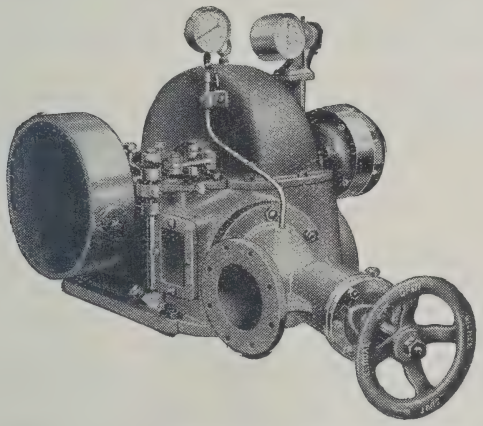


FIG. 30.—Turgo Impulse Wheel with patent Shaft Governor.

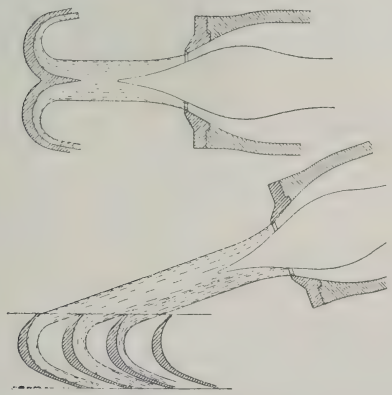


FIG. 31.—Diagram showing the difference between the principle of the Pelton and Turgo Impulse Wheels.

Governor (see p. 1313). The principle of this wheel is identical with that of the Pelton wheel in that the water gives up its power to the wheel by being freely deviated during its passage over the wheel vanes. There is no pressure in the wheel casing, and a branch pipe and nozzle piece exactly similar to that now standard with the Pelton wheel is adopted.

TURGO IMPULSE WHEELS CAPACITY TABLE (*Abbreviated*).

Diam. of Wheel in Inches.	Fall in Feet.	40	50	60	70	90	110	130	150	170	200	250	300	400	500	750
	Fall in Metres.	12·2	15·3	18·3	21·3	27·4	33·5	39·6	45·7	51·8	61·0	76·2	91·5	122·0	152·5	228·5
12	Horse-power	4·94	6·89	9·06	11·42	16·62	22·45	28·95	35·8	43·2	55·1	77·1	101·3	156·0	218	—
	Cu. ft. per min.	48·8	94·8	103·8	112	127·2	140·5	153·0	164·3	175	189·7	212·0	232	268	300	—
	Litres per sec.	39·9	44·7	48·9	52·9	60·0	66·4	72·1	77·5	82·5	89·4	100·0	109·6	126·4	141·5	—
	Revs. per min.	428	479	524	566	642	710	771	828	882	956	1070	1172	1352	1513	—
13½	Horse-power	6·5	9·08	11·93	15·05	21·85	29·6	38·15	47·1	56·9	72·6	101·5	133·3	205·0	287·0	—
	Cu. ft. per min.	110	123·0	134·8	145·6	165·0	182·6	198·5	213	227	246	275	301·0	348	389	—
	Litres per sec.	51·9	58·0	63·5	68·6	77·8	86·25	93·8	100·7	107·0	116·2	130·0	142·3	164·3	184	—
	Revs. per min.	380	425	465	503	569	630	684	735	782	848	948	1040	1200	1342	—
15	Horse-power	8·53	11·9	15·67	19·77	28·7	38·8	50·0	61·9	74·6	95·3	133·0	175·0	269	376·0	—
	Cu. ft. per min.	142·6	159·3	174·8	188·8	214	236·5	257	276	294	319	356·5	391·0	451	504	—
	Litres per sec.	67·3	75·3	82·5	89·1	101·0	111·6	121·3	130·4	138·6	150·5	168·3	184·3	213	238	—
	Revs. per min.	343	383	419	453	513	568	617	663	706	765	855	937	1082	1210	—
18	Horse-power	12·43	17·38	22·85	28·8	41·9	56·1	72·9	90·2	108·9	139·0	194·3	255·0	393·0	549·0	1010
	Cu. ft. per min.	206	230	252	272	308	341	371	398	424	460	515	563	651	727	891
	Litres per sec.	97·0	108·5	118·8	128·3	145·5	161·0	175	188·0	200·0	217·0	242·5	266·0	307·0	343·0	420
	Revs. per min.	285	319	349	377	427	473	514	553	588	638	714	781	902	1008	1234
21	Horse-power	16·95	23·65	31·1	39·3	57·1	77·0	99·2	122·8	148·2	189·2	264·5	347·5	535·0	747	1375
	Cu. ft. per min.	280	313	343	370·5	420	465	505	542	577	626	700	766	885	990	1211
	Litres per sec.	132	147·5	161·5	174·7	198	218·5	238·0	256	272	295	330	361	417	466	571
	Revs. per min.	243	273	299	323	366	405	441	473	504	546	611	669	772	864	1056
24	Horse-power	22·1	30·9	—	51·25	74·5	100·8	129·7	160·5	193·5	247	345	454	699	976	1796
	Cu. ft. per min.	366	409	448	484	549	606	659	708	754	818	915	1001	1157	1293	1585
	Litres per sec.	172·5	193	211·5	228	258·5	286	311	334	356	386	431	472	545	610	747
	Revs. per min.	214	239	262	283	321	355	386	414	441	478	535	586	676	756	926
30	Horse-power	34·5	48·2	63·4	80·0	116·5	157·2	202·5	250·5	302·0	386·0	539	709	1090	1525	2800
	Cu. ft. per min.	570	638	699	757	856	945	1029	1104	1175	1275	1427	1562	1805	2020	2470
	Litres per sec.	269	300·5	329	356	403	446	485	521	554	601	672	736	850	950	1165
	Revs. per min.	171	191	209	226	256	283	308	330	352	382	426	467	540	604	739
36	Horse-power	49·75	69·5	91·4	115·2	167·7	226·5	291·5	361·0	436·0	556	777	1020	1570	2195	4040
	Cu. ft. per min.	823	920	1008	1088	1233	1365	1482	1593	1695	1840	2058	2250	2600	2910	3560
	Litres per sec.	388	434	475	514	582	644	700	752	800	868	970	1062	1228	1372	1682
	Revs. per min.	142	159	174	188	213	236	257	276	294	319	356	390	450	504	616

Smaller wheels down to 6 ins. diameter are also made.

(*Gilbert Gilkes & Gordon, Ltd.*)

The difference in principle between the Turgo wheel and the Pelton wheel is clearly shown in fig. 31 (p. 1337). In the Pelton wheel the water is divided and discharged on both sides of the wheel, whereas in the Turgo wheel the water enters on one side and passes completely through the wheel and out at the other side.

The advantages claimed for the Turgo wheel are as follows :—

1. The specific speed is much higher than that of the Pelton wheel—that is to say, under given conditions of head and water quantity the Turgo wheel will run at a much faster speed than a Pelton wheel. The reason for this is that with a Pelton wheel the diameter of the jet must for best efficiency be not larger than about $\frac{1}{11}$th to $\frac{1}{19}$th of the mean diameter of the wheel. With the Turgo wheel a jet as large as $\frac{1}{5}$th of the diameter of the wheel can be efficiently applied to it.

2. The Turgo wheel is cast all in one piece with its central boss, wheel vanes, and external shrouding. It is, therefore, a strong type of construction and much more robust than the usual type of Pelton wheel with loose buckets separately bolted to a central disc. Moreover, owing to the fact that the wheel is shrouded externally, the windage and friction loss of the wheel is claimed to be very much less than that of the Pelton wheel.

TYPICAL HYDRO-ELECTRIC PLANTS.

St. Jérôme Hydro-Electric Municipal Plant on the North River, Canada.

This plant (fig. 32) was constructed in 1913 by the town of St. Jérôme to supply electricity to its population of 3,500; it is built on the North River, which carries the waters from the Laurentian Mountains to the Ottawa River and thence to the St. Lawrence River.

The chief features of this plant are a concrete weir, a needle dam with three openings, and a reinforced concrete pipe 8½ ft. in diameter, bringing down the water to a power house 1,000 ft. distant from the head gate. The power house consists of a semi-circular forebay of reinforced concrete communicating by means of sluices, to the three turbine pits. These turbines are of the twin open flume type, designed to have each a maximum output of 270 h.p. at 277 r.p.m. and operating under 20 foot head. The setting of the wheels is horizontal, and they are provided each with a concrete draft tube discharging under the power house proper. The building contains the generators, which are directly connected to the turbines through a cast iron wall plate provided with a stuffing box. Two of the generating units are now installed, and their electrical equipment consists of two 150 kw., 3-phase, 60-cycle, 6,600-volt alternators. The exciters are two 10 kw. units, 120 volts, 1,150 r.p.m. belt connected to the end of the generator shaft. The power is transmitted at 6,600 volts over a transmission line 4 miles long, to a sub-station in St. Jérôme and distributed through the town on a 2,200-volt primary circuit and a 110-volt secondary circuit.

The Cedars Rapids Manufacturing and Power Co.'s Development on the St. Lawrence River, Canada.

The plant (fig. 33) of the Cedars Rapids Manufacturing and Power Co. is situated on the St. Lawrence River about 27 miles from the city of Montreal. The low water flow of the river at that point is about 170,000 cub. ft. per second, and the company has been authorised by the Government to utilise 56,000 cub. ft. per second.

The Cedars Rapids have a fall of 32 ft. in two miles, and the river is approximately 3,000 ft. wide at that point. The development consists of a head-race canal about two miles in length, formed by an outer embankment built up in the river, parallel to the flow, and an inner bank placed back from the original shore line. This head-race is from 700 to 1,200 ft. wide, and is dammed at the lower end by the power house substructure.

The present length of the power house is 667 ft., extending from the outer bank towards the shore and connected to the present shore line by a rock-filled crib dam. This crib is 48 ft. high, and so placed that the work of extending the power house for the remaining 513 ft. to the new shore line for the final development may be carried on without disturbing the crib. When the power house is ready for the final development the crib and fill may be removed by dredging.

The turbines for this development have the largest dimensions of any in the world to date (although not the highest capacity output). In the design are seen all the features of recent advances, such as (1) volute casing and draft tube moulded in the concrete substructure, (2) cast iron speed ring with the vanes serving as columns to support the concrete floor, generator and thrust-bearing loads from above, (3) lignum vitæ guide bearing, (4) Kingsbury thrust bearing located above the generator, and (5) pneumatic brakes acting on the generator rotor to bring the revolving parts quickly to a standstill.

ST. JÉRÔME HYDRO-ELECTRIC PLANT, NORTH RIVER, CANADA.

(Designed by Messrs. Surveyer & Frigon.)

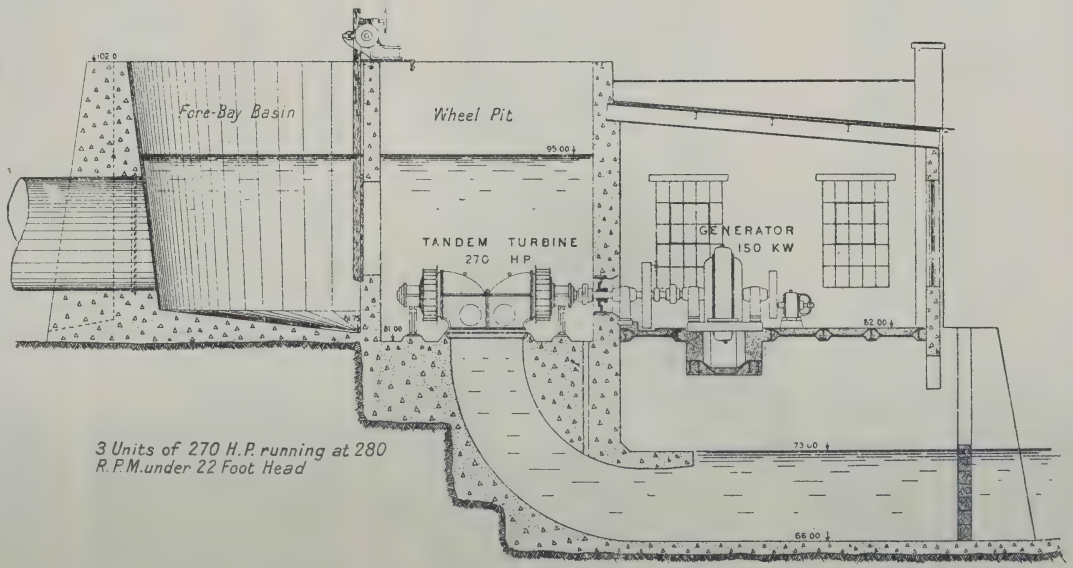


FIG. 32.—Three Units of 270 h.p. running at 280 r.p.m. under 22-foot head.

The present construction provides for ten units of 10,000 h.p. each, but 160,000 h.p. will ultimately be developed.

Each generating unit consists of a vertical American turbine guaranteed to have a maximum efficiency of at least 87 per cent. when operating under an effective head of 30 ft. running

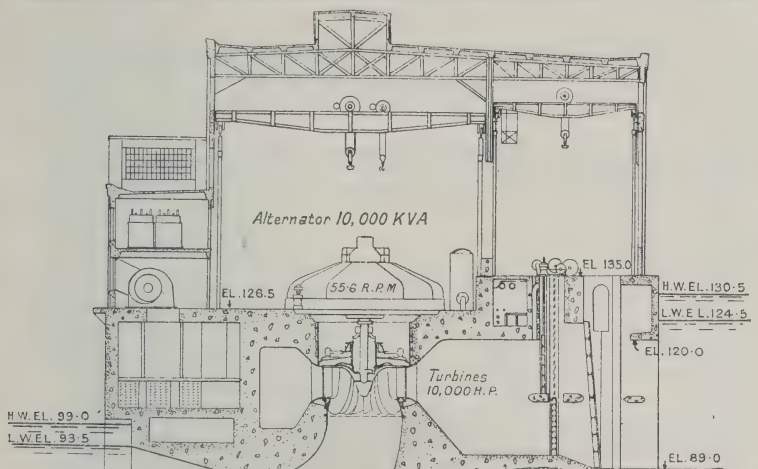


FIG. 33.—160,000 h.p. Plant on the St. Lawrence River (*Cedars Rapids Manufacturing and Power Co.*).

at a speed of 55·6 and developing between 10,000 h.p. and the maximum figure corresponding to wide open gate, provided, however, that the maximum power does not exceed 10,800 h.p. by more than 5 per cent. To each of the main turbines is connected a 10,000 kVA. 3-phase 6,600-volt 60-cycle generator of revolving field type.

In the transformer house on shore are the high-tension switches and feeders between transformers and transmission lines. Provision has been made for four transformer banks, two to be of 24,000 kVA. capacity and made up of 8,000 kVA. single-phase 6,600/110,000-volt units; the other two banks will be of 15,000 kVA. capacity, each bank having three 5,000 kVA. single-phase 6,600/66,000-volt units. The 110,000-volt banks feed transmission lines running to Messina, N.Y., where 60,000 h.p. will be taken by the Aluminium Co. of America. The remainder of the power is transmitted at 66 000 to Montreal, from the 15,000 kVA. bank.

THE GRAND'MÈRE MUNICIPAL HYDRO-ELECTRIC PLANT ON THE SHAWENEGAN RIVER, CANADA.

This plant (fig. 34) was built in 1913 by the town of Grand'Mère to supply electricity to its 5,000 inhabitants. It is located at eight miles from Grand'Mère, on the Shawenegan River.

The fall utilised is a hundred feet high, and the water is carried to the power house through a four-foot wood stave penstock, 1,100 ft. long.

The equipment consists of two 500 h.p. spiral casing water turbines, running at 600 r.p.m. and each directly connected to an alternating current generator and a direct current exciter. The generators are of 300 kW. capacity, producing electricity at 60 cycles and 2,200 volts. The current is stepped up, by transformers in the power house, to 10,000 volts, transmitted to a substation in Grand'Mère and distributed through the town on 2,200 volts primary circuit and a 110-volt secondary. The power-house floor is situated very high above the low water elevation of the Shawenegan River on account of the extreme high water level, due to the backing up of the St. Maurice River in the spring.

MUNICIPAL HYDRO-ELECTRIC PLANT AT GRAND'MÈRE, SHAWENEGAN RIVER, CANADA.

(Designed by Messrs. Surveyer & Frigon.)

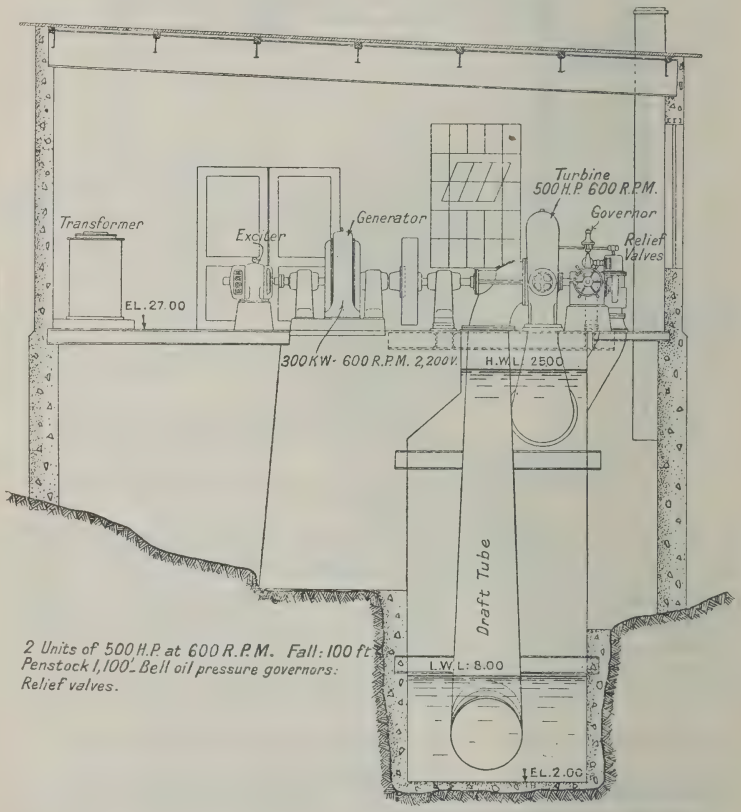


FIG. 34.

JORDAN RIVER DEVELOPMENT OF THE VANCOUVER ISLAND POWER COMPANY, LTD.

The Jordan River plant (fig. 35) of the Vancouver Island Power Company, Limited, is situated at the mouth of Jordan River, which flows into the Straits of San Juan de Fuca about 40 miles west of the City of Victoria, Vancouver Island, Canada.

This development is of particular interest on account of the high head utilised, the static head at the power house being 1,145 ft. The power house is situated on the beach near the mouth of the Jordan River, with the main floor 7 ft. 6 ins. above high tide level. Water is conveyed to the water wheels in the power house from a forebay reservoir through steel pipe lines 9,290 ft. long. The forebay, which is a small equalising reservoir formed by two earth fill dams, is about 1,152 ft. above sea level, giving a static head of 1,145 ft. at the nozzles in the power house.

The power house is composed of two buildings. The first one, completed in 1911, contains two generating units of 4,000 kW. Each of these units consists of a 4,000 kW., 2,200-volt, 3-phase, 60-cycle Allis-Chalmers-Bullcok generator, driven at 400 r.p.m. by a single Doble tangential water wheel of 6,000 h.p. which is mounted on one end of the shaft and overhangs the bearing at the end of the unit.

Water is conveyed from the terminal end of the pressure pipe through a cast-steel flanged taper pipe, which is bolted to the flanged end of the pipe. This taper pipe decreases in diameter from 30 ins. to 24 ins. at the outlet end, where it is bolted to a hand-operated, 24-in., single disc, steel body, rising stem gate valve, which is provided with a by-pass. The steel nozzle casting is bolted to this valve. The jet of water is projected on to the wheel through a Doble needle regulating nozzle, and the governing is done by a type 'Q' Lombard governor, operating the needle gear by means of an oil pressure cylinder. Surges or rams in the pipe line, caused by the quick closing of the main nozzle, are prevented by the Doble relief nozzle. This nozzle is similar to the main nozzle, but is located below it, and the stream does not hit the buckets on the wheel but discharges freely down the tailrace.

The second power house was completed in 1914; it contains a third generating unit composed of an 8,000 kW., 2,200-volt, 60-cycle, 3-phase Canadian General Electric generator, driven at 400 r.p.m. by two Doble tangential water wheels, one mounted on each end of the shaft and overhanging the bearings. The water wheels are rated at 13,000 h.p.

A separate exciter unit is provided for each main generator unit. No. 1 and No. 2 exciters are 100 kW., 125-volt, direct current generators on the same shaft with a 100-h.p. water wheel and a 150-h.p., 2,200-volt induction motor. No. 3 exciter is a 200 kW. generator direct connected to a 200-h.p. water wheel and a 300-h.p., 2,200-volt induction motor.

The transformer equipment consists of two banks of 1,450 kW. transformers (with one spare) and one bank of 3,000 kW. transformers; all of the oil-insulated, water-cooled type. They step the voltage up from 2,200 to 60,000 volts, at which pressure the current is transmitted to Victoria. The transformers are set in compartments along the back wall of the generator room.

Sluice Gates, Strainer Racks, etc.

(*James Gordon & Co., Ltd.*)

The type of sluice-gate shown by fig. 36 is usually adopted for medium or large size head races, and is made with framing and gearing suitable for single gates up to 8 to 10 ft. wide, and for wider races, frames designed for two or more gates, according to the width of the water channel.

The frames for these gates are constructed of rolled steel joists or channels, making a very simple, cheap, and extremely strong arrangement. For smaller channels the same design and arrangement can be adopted with somewhat lighter construction, or cast-iron frames fitted with cast-iron gates and suitable lifting screws or racks can be used. Strainer racks are usually constructed of flat wrought-iron bars threaded on to rods with pipe distance-pieces placed between each bar to ensure proper spacing, the wider racks being made in small sections to facilitate handling.

The water channel between strainer-rack and turbine-pit should be completely covered in, in order to prevent any possibility of foreign matter getting into the channel on the turbine side of the strainer racks.

In streams which are liable to heavy floods, and contain at times a lot of drift wood and foreign matter, it is usually advisable to put down two strainer racks, the first one protecting the intake to the head race, and being made of heavy bars with wide spaces between so as to stop all the larger material, and a fine one to protect the turbines themselves at the lower end of the race close to turbine pits or pipe intakes.

In designing head and tail races for water-power installation, special attention should be paid to their cross-sectional area, which should be ample for the quantity of water to be carried, in order to guard against loss of head when the water is running at full velocity.

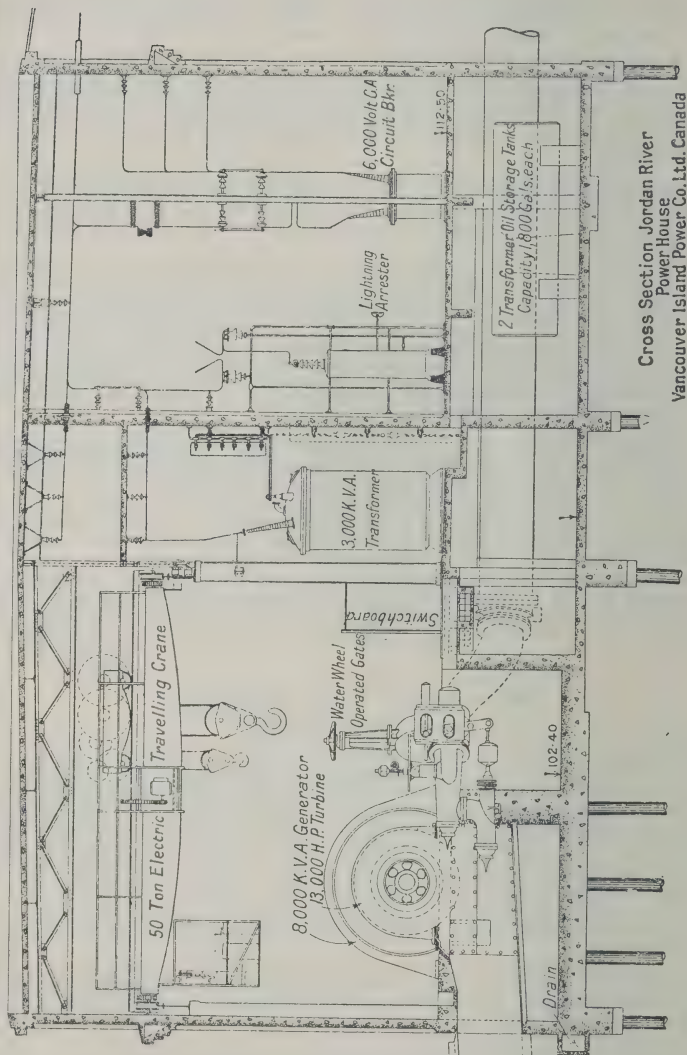
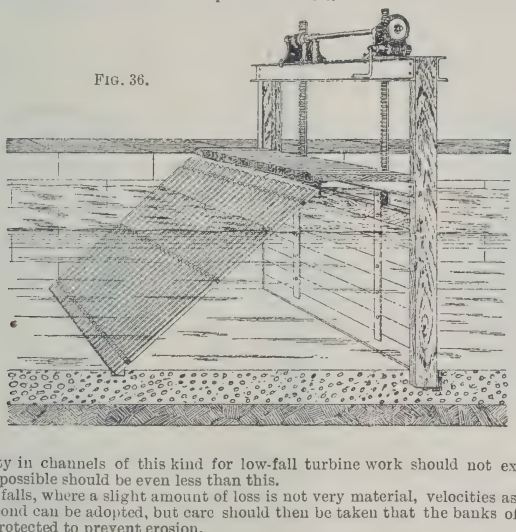

Cross Section Jordan River
Power House
Vancouver Island Power Co. Ltd. Canada
6,000 Volt CA
Circuit Bkr.
112-50
2 Transformer Oil Storage Tanks
Capacity 1,800 Gals. each
Lightning
Arrester
3,000 K.V.A.
Transformer
Switchboard
Water Wheel
Operated Gates
50 Ton Electric
Travelling Crane
8,000 K.V.A. Generator
13,000 H.P. Turbine
102-40
Drain

FIG. 35.

This is extremely important on plants for low falls, where only a few inches of head will make quite a considerable difference to the horse-power available.

FIG. 36.



The velocity in channels of this kind for low-fall turbine work should not exceed 2 ft. per second, and if possible should be even less than this.

For higher falls, where a slight amount of loss is not very material, velocities as high as 3 ft. or 4 ft. per second can be adopted, but care should then be taken that the banks of the channels are properly protected to prevent erosion.

NIAGARA FALLS INSTALLATION.

The cost of power supplied from the Niagara Falls is 3*l.* 6*s.* per horse-power per annum. The capital outlay on the plant was 45*l.* per horse-power.

GENERATION OF ELECTRICITY BY WATER POWER IN GREAT BRITAIN.

In the Annual Report of the Electricity Commission * it is stated that the total units generated by water power during the year ending March 31st, 1927, was 49,032,872 (from 33 stations), as against 45,431,048 (from 28 stations) in 1926.

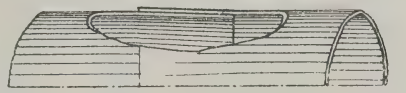
Pipe Lines.

The following table gives thicknesses for different sizes of pipes under varying pressures. The figures are based on a factor of safety of 5, which is a fair average limit to work to for all ordinary work, but in the lower end of high-pressure pipe lines the factor of safety may run up to 8, or even sometimes 9. The working pressures are given in pounds per sq. in., and are for double-riveted or welded steel pipes; the strength of single-riveted pipes is about 30 per cent. less.

Dia. Pipes in Ins.	8	10	12	14	16	18	20	22	24	27	30	33	36	42	48
Thickness.															
$\frac{1}{8}$	240	192	160	137	120	107	96	87	80	71	64	58	53	46	40
$\frac{5}{16}$	360	288	240	206	180	160	144	131	120	106	96	87	80	68	60
$\frac{1}{4}$	480	384	320	274	240	213	192	174	160	142	128	116	106	91	80
$\frac{5}{16}$	600	480	400	343	300	267	240	218	200	177	160	145	133	114	100
$\frac{3}{8}$	720	576	480	412	360	320	288	262	240	213	192	174	160	137	120
$\frac{7}{16}$	840	672	560	480	420	373	336	305	280	248	224	203	186	160	140
$\frac{1}{2}$	960	768	640	549	480	427	384	349	320	283	256	232	213	182	160

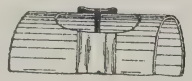
In practically every case with modern hydraulic power plants, steel piping, either riveted or welded, is used. Fig. 37, page 1346, shows types of joints which can be adopted for this class of piping. The slip joint (*a*) is suitable for low head pressures and for pipes constructed of thin

* *Generation of Electricity in Great Britain, year ending March 31st*, 1927, H.M. Stationery Office, Adastral House, Kingsway, London, W.C. 2. Price 1*s.* 6*d.* net.

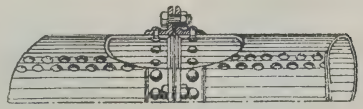


(*a*) Tapered Slip Joint for low pressures.

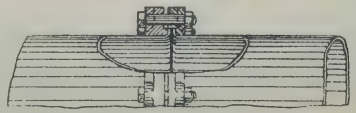
(*b*) Socket and Spigot Joint, packed with lead and yarn.



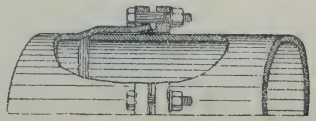
(*c*) Loose Collar Joint, packed with lead and yarn.



(*d*) Flanged Joint for riveted pipes.



(*e*) Flanged Joints for welded pipes.



(*f*) Patent Muff Joint for high pressures.

FIG. 37.

plates. The joint is made by one pipe being tapered slightly outwards and the other slightly inwards, the tapers being coated with paint, cement grout, or some similar composition, and driven into each other until they form a water-tight joint.

The socket and spigot joint (*b*) for lead and yarn packing is a very useful one for low or medium head pressures, as it has some amount of flexibility, and also allows under all ordinary conditions for any expansion or contraction in the pipes being taken up in the joints.

The collar joint (*c*) is another type of lead joint to which the same conditions practically apply.

For higher pressures ordinary forms of lead joints do not always prove satisfactory, owing to the danger of the jointing material being blown out by the pressure of the water. In these cases some form of flanged joint must be used, two of which are shown—one (*d*) with the flanges riveted on to the pipes, and the second (*e*) with the pipe ends turned up and fitted with loose flanges bolted together. The patent muff joint (*f*) is of a special type and allows for a large amount of expansion and contraction in the pipe line, and can be constructed to keep perfectly tight under the heaviest pressures which have to be dealt with.

Anchorage.—One of the most important points in the design of pipe lines is the provision for anchorage on the line itself, and especially at the foot of the line near to the power house, in order to prevent any possibility of pipe line movement being transmitted to the turbines or Pelton wheels. The pipes can be efficiently anchored in several ways, the more common forms of anchorages being to provide cast iron or steel thrust blocks, forming a foot or base on the pipe itself, which are firmly secured to heavy concrete or rock foundations; or special anchoring pipes can be provided with thrust flanges and solidly set into heavy blocks of masonry or concrete.

Expansion and *Contraction* of long lines of piping also require careful attention, and it is necessary in long straight runs of pipe to provide either a type of joint capable of taking up expansion and contraction, or to fit the pipe line with special expansion joints, which are usually constructed of cast iron or steel and of the sliding socket and spigot type.

Relief Valves.—Where the length of a pipe line is excessive in proportion to the head of water, and the plant has to deal with heavy and sudden load changes, it is necessary for the safety of the plant and pipe line to provide efficient relief valves to prevent serious water-hammer effects. These, however, should be of special design, as the ordinary forms of spring-loaded relief or safety valves are worse than useless under conditions of this kind, as if they are set to open with only a small rise in pressure the reaction of the discharging water will nearly always prevent them closing again, and if they are adjusted in the ordinary manner the pressure rises too high before they open, and the sudden closing of the valve by means of the heavy spring employed often causes worse water-hammer effects than there would be with no relief valve at all.

Relief valves for hydraulic power work should be so designed that they will open immediately with a very small rise in pressure, and in proportion to the pressure increase, the closing being effected by some form of oil cataract or dashpot which can be adjusted so that the time of closing will coincide with the time required for slowing up the moving column of water.

The best form of relief valve, however, is one which is directly connected to the gate or nozzle-operating mechanism of the power unit, and adjusted so that it will open in proportion to the decrease of gate or nozzle opening, and be closed again slowly by means of an oil dashpot or cataract. With a relief valve of this form the pressure rise is anticipated, and therefore very good results can be obtained. With a relief valve independent of the Pelton wheel or turbine rise in pressure must take place before it can come into operation.

Where Pelton wheels fitted with deflecting nozzles are used, no relief valve is generally necessary, as the quantity of water discharged remains the same for any load, or is slowly cut off by hand.

(*James Gordon & Co., Ltd.*)

Water Hammer in Pipes.

In the design of pipes (p. 1302) it was pointed out that the pipes had to be designed to resist the static head plus the head due to water hammer. The magnitude of this extra pressure depends, among other things, upon the rate of change of the velocity of the water in the pipe, and in some cases also upon the length of the pipe.

It is shown in treatises on Hydraulics that when a valve is *suddenly* closed the pressure set up due to water hammer is independent of the length of the pipe and it only depends upon the velocity of flow just before closing takes place

Let

V = velocity of flow of water in ft. per sec.;

P = pressure set up in lb. per sq. in. due to water hammer,

then,

$$P = 63{\cdot}5\ V,$$

where the elasticity of the pipe is neglected; if the elasticity is taken into account the problem becomes rather more complicated. (See Goodman's 'Mechanics Applied to Engineering,' vol. i. pp. 699–703, and vol. ii. pp. 411–413.)

The question arises—What is meant by *sudden* closing? In the references given, it is shown that when a valve is rapidly closed a compression wave traverses the pipe with a velocity of 4,720 ft. per sec., *i.e.* the velocity of sound in water. The closing is termed *sudden* if the time taken in closing the valve is equal to, or less than, the time taken by this compression wave to make the return trip of the pipe, or to traverse a distance equal to twice the length of the pipe. Thus if the length of the pipe is 6,000 ft., the time taken by the wave in traversing the pipe and returning is

$$\frac{2 \times 6{,}000}{4{,}720} = 2{\cdot}54 \text{ seconds.}$$

The pressure due to water hammer for a velocity of 5 ft. per sec. will therefore be 317 lb. per sq. in. if the time of closing be 2·54 secs. or less.

When the valve is closed more slowly than this the pressure due to water hammer will always be less than when the closing is sudden. Many expressions have been put forward to cover this case of gradual closing, but some obviously give absurd results since they give a higher pressure for gradual than for sudden closing. Even the expressions which do not err in this particular way do not all agree, largely because they are based on faulty experiments which mar the value of the results. The following expression, which is based on the assumption that the pressure set up by gradual closing, is less than that with sudden closing in the ratio:

$$\frac{\text{Pressure with gradual closing}}{\text{Pressure with sudden closing}} = \frac{P_g}{P} = \frac{T_r}{\text{Time taken in closing the valve } (T_g)}$$

where

$$T_r = \frac{2L}{4{,}720}, \text{ L being the length of pipe in ft.;}$$

then,

$$P_g = 63{\cdot}5\ V \frac{T_r}{T_g}, \text{ provided } T_g \text{ is equal to or greater than } T_r.$$

In the example given above, L = 6,000 ft.; V = 5 ft. per sec.; T_r = 2·54 secs.

The pressure set up due to water hammer will be 317 lb. per sq. in. for any time of closing less than 2·54 secs.; if the valve be uniformly closed in, say, 4 secs. the pressure due to water hammer will be, approximately, $P_g = 63{\cdot}5 \times 5 \times \frac{2{\cdot}54}{4} = 200$ lbs. per sq. in., and if closed in 5 secs. it will be 160 lbs. per sq. in., and so on.

That the above-mentioned assumption is justified is proved by some very carefully conducted experiments recently made at Soulom by Camichel, Eydoux, and Fariel. A full account of these experiments will be found in their treatise entitled 'Des Coups de Bélier,' published by The University of Toulouse, 14 Rue des Arts.

These pressures are over and above the static pressure of the water in the pipe. As a rule the pressure due to water hammer should not exceed 25 per cent. of the static pressure.

(*John Goodman, M.Inst.C.E.*)

SECTION XXV

PART V

HYDROSTATIC PRESS—HYDRAULIC RAMS—HYDRAULIC LIFTS—HYDRAULIC BOLT FORCERS—HYDRAULIC RIVETERS

The Hydrostatic Press.

P = Power or pressure applied.
l = length of lever, in inches or feet.
A = Area of ram, in square inches.
W = the weight or resistance, in lbs.
l' = length of fulcrum, in inches or feet.
a = Area of piston, in square inches.

$$\frac{Pla}{l'a} = W. \qquad \frac{Wl'a}{Pl} = A. \qquad \frac{Wl'a}{Al} = P. \qquad \frac{PAl}{Wl} = a.$$

Designing Ram Cylinders.

To Compute the Thickness of the Metal to Resist a Given Pressure.

(1.)

Rule.—Multiply the pressure per square inch in pounds by the diameter of the cylinder in inches, and divide the product by twice the estimated practical tensile resistance or *value* of the metal in pounds per square inch, and the quotient will give the thickness of the metal required.

Example.—The pressure required is 9,000 lbs. per square inch, and the diameter of the cylinder 5·3 inches; what is the required thickness of the metal or cast iron?

$$\frac{9000 \times 5{\cdot}3}{6000 \times 2} = \frac{47700}{12000} = 3{\cdot}975 \text{ ins.}$$ The *value* of the metal is here taken at 6,000.

(2.)

The following formulæ, due to F. B. Kleinhans (*Engineering News*), will be found convenient in determining the proportions of the ram cylinders:—

$$T = \frac{P\pi D_1^2}{2000 \times 4} = 0{\cdot}0003927\ PD_1^2; \qquad T_1 = \frac{PD_2}{2(S-P)};$$

$$T_2 = \frac{D_2}{2}\sqrt{\frac{P}{S}}; \qquad T_3 = \frac{D_2}{2}\sqrt{\frac{3P}{4S}}.$$

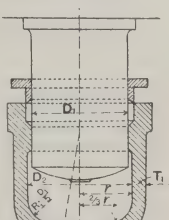


Fig. 1.

where,

T = load on rams in tons; P = water pressure in lbs. per sq. in.; S = safe fibre stress of material; D_1 = diameter of ram in inches; D_2 = diameter of cylinder in inches; T_1, T_2, T_3 = thickness of cylinder walls.

(3.)

A steel containing 0·4 per cent. of carbon is perfectly satisfactory for press, accumulator, and intensifier rams, while for pump rams a steel with a carbon content of 0·6 per cent. is generally preferred, on account of its greater hardness and durability. Stainless steel is now being extensively employed for pump rams.

Wear on the packings is considerably reduced when the ram is electro-coppered or fitted with a bronze liner, and, in consequence, the rams of machines operating at high pressures are invariably protected in this way.

In short-stroke presses the liner is carried up to a point just beyond the top of the packing—that is, almost the full length of the ram. But on long-stroke machines for baling and similar operations, where the greater part of the stroke is made at a low pressure, it is sufficient if the ram is cased for a short length only—from the end to the point at which high pressure is developed. The thickness of bronze liners varies, from $\frac{1}{4}$-in. for small rams to about $\frac{5}{8}$-in. for a ram 30 ins. diameter. The alternative to a liner is electro-coppering, the thickness of which may be anything from $\frac{1}{32}$ to $\frac{1}{8}$-in., but average requirements are met by a coating $\frac{1}{16}$-in. thick. The copper may extend just over the end of the ram or finish in a recessed joint on the wall close to the bottom. The bottom should not be coppered.

The following formula, due to Bach, for determining the thickness of the wall of a ram subjected to external pressure is in common use:

$$T = \frac{D}{2}\left(1 - \sqrt{\frac{f_c - 1{\cdot}7P}{f_c}}\right),$$

where,

T = thickness of wall in ins.; D = external diameter of ram in ins.; P = pressure in tons per sq. in.; f_c = compressive strength of material in tons per sq. in.

From this formula, a cast-iron ram 16 ins. diameter for a pressure of 2 tons per sq. in., assuming a stress of 6 tons per sq. in., gives a thickness of $2\frac{3}{8}$ ins., which is in agreement with practice.

For a ram under internal pressure the formula devised by Lamé is considered to be the most reliable.

$$T = \frac{D}{2}\left(\sqrt{\frac{f_t + P}{f_t - P}} - 1\right)$$

where,

T = thickness of wall in ins.; D = internal diameter of ram in ins.; P = pressure in tons per sq. in.; f_t = tensile strength of material in tons per square inch.

For a 16-in. ram, assuming a core diameter of 8 ins. and a stress of 2 tons per sq. in., the required thickness is 4 ins.

For low pressures and rams of small sizes the thickness of the wall as determined by the above rules, will need to be increased in order to ensure the production of generally sound and reliable castings.

A rough rule for small rams is to make the bottom flat, the end of the core semi-spherical, and the thickness at the centre equal to that of the wall. Another rule, applicable to larger rams, consists in dishing the end slightly, making the radius the same as the diameter of the ram, and the thickness about 25 per cent. greater than that of the wall.

(*Abstract of Article by H. S. C.*, '*Machinery*,' *January* 22, 1925.)

The Hydraulic Ram.

The water working the ram is supplied through the pipe S (fig. 2), and escapes through an opening at *o*, until it has gained a velocity sufficient to raise the valve or ball, B, which

suddenly stops the current, and causes an excessive pressure in the ram, R, which opens the valve or ball, C; the water is forced into the vessel and air-chamber, A, and finally through the delivery-pipe, *d*, to its destination. When equilibrium of pressure is restored between S and R, the ball, B, falls, and the operation is repeated. The ram can make as many as 200 strokes per minute, depending upon its size.

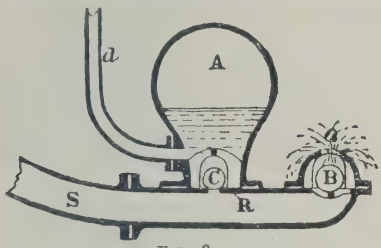


FIG. 2.

The length of the supply-pipe, S, should not be less than 5 times the height of the fall. But the pipe may be made 10 times, or more, the height of the fall.

In ordinary cases the hydraulic ram returns about 50 per cent. of the natural effect. That is, the quantity of water, q, multiplied by the height, h, of the delivery above the ram will be about 50 per cent. of the quantity of water, Q, working the ram, multiplied by the head of fall, H, in the same unit of time.

$$qh = 0{\cdot}5\mathrm{QF}. \qquad q = \frac{0{\cdot}5\mathrm{QH}}{h}. \qquad \mathrm{Q} = \frac{2qh}{\mathrm{H}}.$$

Q and q can be expressed in any unit of volume or weight. F and h can be expressed in any unit of length.

If

Q and q = cubic feet per minute;
F and h = fall and height in feet;
L = length in feet, and D = diameter in inches of the supply-pipe, S;
l = length, and d = diameter of the delivery-pipe, d,

then,

$$\mathrm{D} = \sqrt[5]{\frac{2\mathrm{Q}^2(\mathrm{L} + 5\mathrm{D})}{\mathrm{F}}}, \qquad d = \sqrt[5]{\frac{4q^2(l + 5d)}{h}}.$$

TABLE OF RESULTS OF OPERATIONS OF HYDRAULIC RAMS.

Number of Strokes.	Height of Fall.	Height of Elevation.	Water Expended.	Water Raised.	Useful Effect.
Min.	Feet.	Feet.	Cubic Feet.	Cubic Feet.	
66	10·06	26·3	1·71	·543	·9
50	9·93	38·6	1·93	·421	·85
36	6·05	38·6	1·43	·169	·75
31	5·06	38·6	1·29	·113	·67
15	3·22	38·6	1·98	·058	·35
10	1·97	38·6	1·58	·014	·18
—	22·8	196·8	·38	·029	·67

NOTES ON HYDRAULIC RAMS.

Working Fall.—Any working fall, from say, 18 ins. up to 100 ft., will do to work a ram, but the more the working fall obtained (up to about one-third of the total height the water has to be raised above the ram), the less the ram will cost to raise a given quantity, and the less driving water will be required to lift that quantity.

In sluggish streams where it is inadmissible to put a dam in the stream, it is sometimes necessary to carry the driving water several hundred yards down stream, in earthenware pipes running nearly level, before a few feet fall can be obtained; but where driving water is plentiful

it is often less costly to be content with a small working fall and have a more powerful ram, than to go to the expense of obtaining the greater fall, for the sake of enabling a smaller ram to do the work required.

Height to Force.—The height to force must be estimated from the level of the ram or bottom of the working fall.

Rising Main Pipe.—Rams will force to a distance of several miles. In one case a ram is in use forcing to a distance of three miles and to an elevation of 340 ft., with a working fall of only 10 ft.

Driving Water.—The quantity of driving water required to work a ram depends upon the amount of working fall available, the height to raise the water, and the quantity of water it is desired to raise. Rams can be made to work under favourable conditions with less than one gallon of water per minute.

In giving the quantity of water available to work a ram, it is useless to say there is 'sufficient to fill a pipe of certain size,' unless the gradient of the pipe be given at the same time, in order that the velocity of the water in the pipe may be ascertained. Unless the water be practically unlimited for the purpose required, it is preferable to make a positive measurement of the water. A ready method of measuring the flow of very small streams is to dam up the stream with a little puddle, and insert a pipe in the dam to form a spout. Then let the water flow into a bucket for, say, exactly 15, 30, or 60 seconds, and carefully measure out the contents of the bucket in pints, when the total flow in gallons per minute or hour can be calculated. If the stream be too large to measure with one spout, two or more spouts can be inserted in the dam, and each of them measured separately. If the ram required be a large one, and the volume of driving water available be too great to measure in this way, it should be gauged by means of a weir board.

Modern Hydraulic Rams.

Fig. 3 shows a modern form of ram.

There is a minimum ratio of lift to fall at which an ordinary ram will not work. This ratio is usually somewhere about 4 to 1.

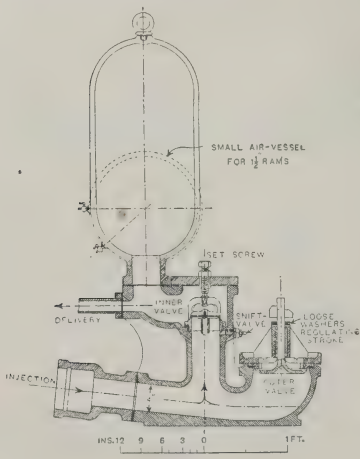


Fig. 3.

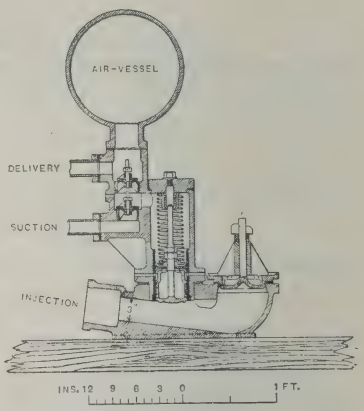


Fig. 4.

Increasing the stroke of the outer valve increases the quantity of water delivered, but reduces the efficiency, and *vice versa*.

If a ram is required to give the largest possible delivery with a limited supply of water, it is an advantage to use a large ram and give its valve a short stroke; if there is an abundance of water, it is cheaper to use a small ram with a long stroke.

A mushroom is better than a ball valve.

It has been found good practice to allow 1 sq. in. of area through the inner valve for every gallon of water to be delivered per minute. In practice a ram will not work without an air-vessel, and it must be kept properly charged with air. Certain waters in passing through the ram are found to give off air in sufficient quantity to keep the vessel charged; where, however, water has to be used which does not possess such a property, some method of supplying air must be adopted, such as a 'Snifting Valve' (p. 1201).

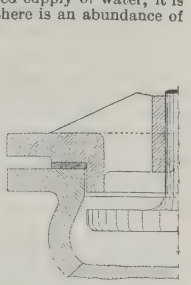


FIG. 5.

The area of the annulus between the edge of the outer valve and the body should be made rather less than that through the valve itself, so that the water should have a high speed locally and thus tend to close the valve rapidly. Fig. 5 shows the half-section of the ram body and valve, showing this contracted area.

The longer the fall pipe in relation to the fall, the better for both efficiency and capability of functioning of the ram under all ratios of lift to fall up to 24 to 1; and the shorter the pipe, the smaller the ratio above 4 to 1 at which it will work. In general it may be said that the best results are obtainable with fall pipes having a length of from 6 to 10 times the fall.

(*Anderson.*)

HYDRAULIC RAM FOR WORKING WITH DIRTY WATER.

In this type, fig. 4, water from one source of supply may be pumped by using water under a small head from another source. It is usually employed to pump clean water from a well or spring by means of dirty or contaminated water from some other source. The efficiency of the arrangement is rather low—*i.e.* from 35 to 56 per cent.

(*Anderson.*)

Hydraulic Lifts.

A speed of 200 to 400 feet per minute is usual.

For packing the ram, hat-shaped packing is preferable to the U-shaped, as it causes less friction and lasts longer. Leather packing is the best, but the most usual form is 'hydraulic' flax or hemp in stuffing boxes.

The cylinder of the lift should be absolutely plumb.

In well-constructed direct-action lifts the total friction should not be more than 5 per cent. In suspended lifts the friction varies from 15 to 20 per cent.

An air-vessel placed between the valve-box and ram cylinder will prevent any shock in the cage from the passing of the water when the valve is opened or closed, or when the water is turned into the waste-pipe.

The gear rod or rope for high-speed lifts should have a range of from 5 to 6 feet (from full pressure to full exhaust), so that the man working the lift may have proper control over it.

The cam motion opening the valve should open the latter very gradually.

The high pressure system of working hydraulic lifts is the most economical in every way, and it enables high working speed to be obtained.

Public supplies of high pressure hydraulic power are available in London, Liverpool, Hull, Manchester, Birmingham, and Glasgow.

Direct acting ram lifts are usually adopted in this country for moderate heights; the machine consists of a hydraulic cylinder having a flange cast on its upper end resting on the foundation at the top of the well bored to receive it, and a simple platform or cage moving between guides firmly screwed to the upper end of the ram working in the cylinder. The valve controlling the action of the lift is worked by a rope, or rod passing through the platform. Tappet gear automatically stops the lift at the top of its stroke, and permanent stops are also fitted to prevent overtravel. Alternative rams can be fitted to meet varying loads.

The suspended type of lift is less expensive than the direct-acting lift when the height of travel is considerable, or where there are difficulties in boring the well for the cylinder. A factor of safety of 10 to 15, to 1, on all ropes (collectively) is ample; very high factors mean, in most ordinary cases, that ropes become too large in diameter to ensure long life. The ram can be placed in the lift shaft or in the roof, or basement, either vertically, horizontally, or on an incline. Safety gear should be fitted to come into immediate action in case the ropes break. The valves are similar to those used for the direct-acting type.

The direct ram lift is becoming usual in America even in the highest buildings on account of its safety. The cylinder consists of a bored well lined with stout Artesian tubing, and failure can hardly occur, for the pipes fit closely in the bored rock. The outlet piece if well made and attached to the lining tube is not apt to fail. Balances are employed so that the long rams when out are under tension.

The guide-bars of lift should preferably be fixed to stone templates with bolts let in and secured by lead, and the bars fixed by nuts. This is preferable to wooden bricks, which are liable to shrink. The V guide-bar is the best form.

Electric push button control of lift and gates can be fitted to either type of lift.

Each elevator shaft of the 'Empire Building,' in New York, is walled up independently as high as the third floor, and is made a close fit for the car, the last 50 feet of the drop of each elevator thus forming an air-cushion. If the car fall, the air is confined in the bottom of the shaft, and, by its gradual escape, lets the elevator down easily without shock. It is stated that a car has been dropped from the twentieth floor of the building without breaking eggs placed in the car. The pressure of the air caused by a fall from the top floor, a height of 290 feet, is calculated to be about $3\frac{1}{2}$ lbs. per sq. in. The estimated proportion between the depth of the air-cushion and the length of the drop is as 1 to 6. The only precaution necessary seems to be to make the doors of the lower-floor entrances air-tight.

In hydraulic lifts trouble sometimes arises owing to water leaking from the ram glands and falling on to the ropes. Where this possibility exists the ropes should be well protected with a coating of tallow and blacklead well mixed together, otherwise the water will find its way into the interior of the rope, and by rusting the wires will rapidly weaken them. The danger in such cases lies in the fact that there is often no external evidence to show that the ropes are unsafe. In one instance the two supporting ropes of a lift broke without warning near to their anchorage to the cylinder, causing the lift car to fall the full height of the lift well, wrecking the lift and killing the attendant. On examination of the broken ropes it was found that the internal wires had been damaged by the water to such an extent that they were quite brittle and without strength, but no broken wires were noticeable at the point of failure, before the accident. The best way to prevent such failures is to design the lift so that water cannot possibly fall on the ropes, or, if this be not feasible, to fit drip trays or other appliances to keep the water off the ropes if leakage should occur.

Danish Regulations concerning the Arrangement and Working of Lifts and Hoists driven by Machinery.

The lift cage must be closed in all round, except at the entrance, either with solid sides and roof or with trellis-work having openings not exceeding 1 in. It must be lit up inside, and must contain a notice of the number of passengers allowed. From any point in its travel there must be the means of signalling if in difficulty. Except where their adoption would be attended with serious inconvenience, entrance-doors must be fitted inside the cage itself; these must not open outwards, and must be kept locked while travelling. Where they are dispensed with, at every landing the lift well must be fenced to the full height on the entrance side, with no inward projection; and the clearance between the cage and the well must nowhere exceed 2 ins. In ram lifts the cage and ram must be securely joined together. A hanging cage must be suspended by at least two ropes or chains, each of which must alone be strong enough to carry ten times the weight of the cage fully loaded. Each must also be connected with either a stopping apparatus or a brake, so arranged that the cage shall either be stopped in a fall of not more than 1 ft. or else shall be let down at a speed not exceeding 3 ft. a second. Lifts worked by a winding drum must be provided with a contrivance for preventing the rope from becoming slack in the event of the cage sticking in its descent, and at the same time for stopping the drum. Every counterbalance weight is likewise to be hung by at least two ropes or chains, each having ten times the strength necessary to carry the weight. The counterbalance weights must be securely guided, and the space in which they work must be securely fenced, so as to guard against all possible risk of accident.

See also page 829.

Hydraulic Bolt Forcers.

Fig. 6 represents Young's Hollow-ram Portable Hydraulic Bolt Forcer, constructed principally for forcing in and out coupling bolts, though it is also suitable for forcing pins, etc., in and out of machines. The apparatus consists of a hollow steel sliding ram, the ends or tails of which project through the front and back respectively of the cylinder. Inserted in this hollow ram is a steel drift, which passes right through the centre of ram; a head is forged on one end of the steel drift, and a shoe for the bolt is fitted on the other. The method of working is the same as in any ordinary hydraulic pump, viz. the fluid is forced by the pump from the reservoir into the cylinder, and the ram is gradually forced forward until it presses the shoe, on the end of the steel drift, against the bolt to be forced out. Should the bolt show no sign of moving out of coupling when the maximum hydraulic pressure has been obtained, a sharp blow is given with a hammer on the head of the drift, which transmits a shock or 'jar' to the bolt and starts it from its position, and it can then be ejected, either by pumping the ram forward or knocking it out by

hammering on the head of the steel drift; this is a valuable feature, as the bolts are generally rusted in their seats, and a sharp blow is very efficacious in starting a bolt or pin under powerful and steady pressure.

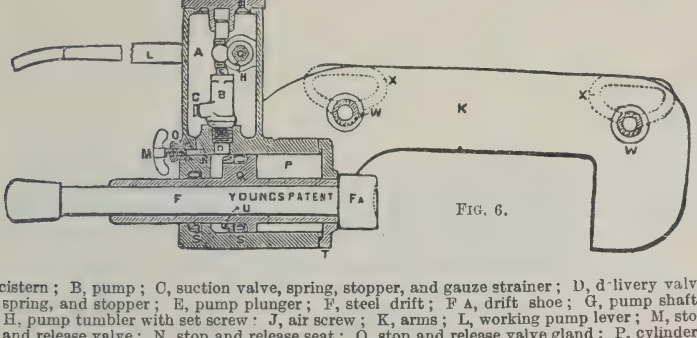


FIG. 6.

A, cistern; B, pump; C, suction valve, spring, stopper, and gauze strainer; D, delivery valve spring, and stopper; E, pump plunger; F, steel drift; F A, drift shoe; G, pump shaft; H, pump tumbler with set screw; J, air screw; K, arms; L, working pump lever; M, stop and release valve; N, stop and release seat; O, stop and release valve gland; P, cylinder; Q, hollow ram; SS, U leather for packing; T, cylinder cover; U, packing ring; V, cistern lid; W, distance pieces; X, handles.

This machine, fitted with crosshead instead of arms as shown, is largely used for forcing winch drums from their shafts.

DETAILS OF HYDRAULIC FORCERS.

Power . . . . . tons	20	20	30	40	50	75
Width of gap . . . . . inches	9	15	12	14	16	18
Run out . . . . . inches	3	3	4	5	5½	6½
Height . . . . . inches	15½	15½	16½	17	18	18½
Suitable for bolts approx. . inches diam.	2	2	2½	3	4	5
Approx. weight . . . . . cwts.	1¾	2	3	4	4½	6½

The body of the apparatus can be separated from the arms or claws by withdrawing two small sliding bolts, when the machine can then be carried in two separate pieces, the body placed first in position on the shaft, and the claws lifted up afterwards and fixed, without having to screw and unscrew any bolts. A gauge supplied with each machine indicates the pressure.

Hydraulic Riveters.

Fig. 7 represents the Tweddell hydraulic riveter (manufactured by Fielding & Platt, Ltd, Atlas Works, Gloucester), a tool which is very largely used for riveting purposes.

In this machine, by means of two rams, one inside the other, a pressure, in the first place, is brought to bear on the plates by means of an annular tool, A, encasing the moving capping die, B. When the plates are thus brought together, the capping tool heads the rivet; and after this operation is performed, the combined power of both rams (C and D) is applied on the head of the rivet, and through the shoulders of same on to the plate surrounding it. The size of work is only limited by the power of the machine, but some very good work in cold riveting, up to 1 inch diameter, has been done by these machines. An advantage of the machine is that by substituting a riveting die for the annular closing tool, A, a second and smaller power is available. This adds very much to the general use of the tool, for it is impossible to keep such a powerful machine in constant work. Excellent work had been done by this machine when using plain bolts—*i.e.*, making both heads of the rivet in the act of closing it. This machine is capable of exerting three different powers, which is a great advantage. Although these machines are made

up to 200 tons power, very little more high-pressure water is required, owing to the use of the auxiliary piston, E, which draws forward by means of the lug, F, the two main rams, C and D, the space left behind being filled with low-pressure water through check-valves, G and H, from a tank, J.

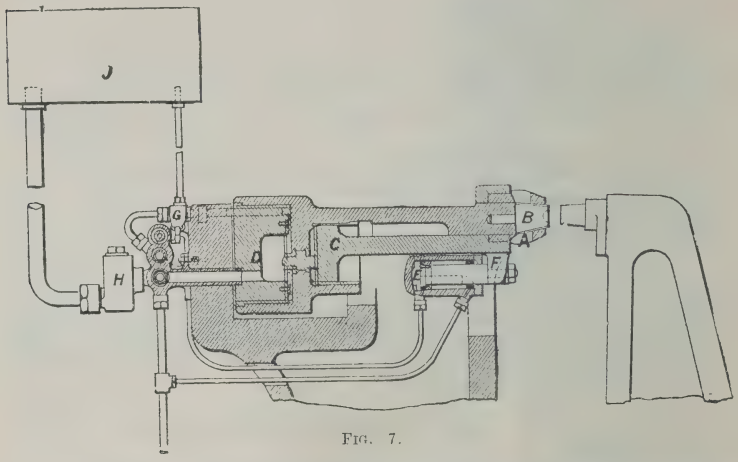


Fig. 7.

In cases where boiler-ends are placed with their flanges inwards, the machine shown by Fig. 8 is used. In this machine there is a hinged arm, A, which is first of all lowered into the boiler, and when restored to its proper position, A′, the riveting is proceeded with.

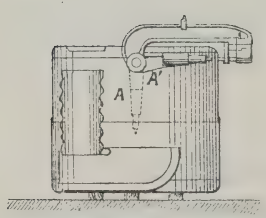


Fig. 8.

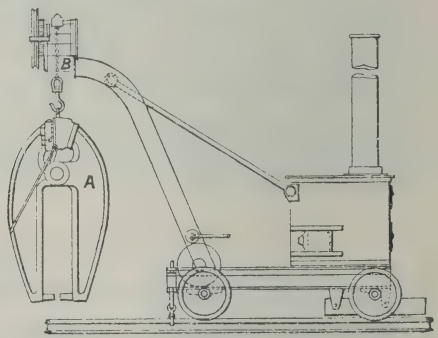


Fig. 9.

Fig. 9 shows a portable riveter, A, slung from a travelling crane. By means of the iron girder, B, this riveter is moved across the work to be riveted, whilst the crane with its rivet-heating oil or reverberatory furnace is traversed along. Riveters of this type were largely used in the construction of the Forth Bridge.

SECTION XXVI

PART I

SANITARY ENGINEERING—SEWERS—SEWERAGE—FLUSHING AND VENTILATING SEWERS AND DRAINS—SEWAGE TREATMENT AND DISPOSAL — MINISTRY OF HEALTH'S REQUIREMENTS—HOUSE DRAINAGE (pp. 1359-1382)

(Contributed by E. J. Silcock, M.Inst.C.E., F.G.S., F.S.I.)

PART II

IRRIGATION — AVAILABLE SUPPLY — EVAPORATION — ABSORPTION—EVAPORATION FROM SOILS—LOSSES FROM IRRIGATION CANALS—CAPACITIES OF CANALS—MEAN VELOCITIES—CRITICAL VELOCITY—DUTY OF WATER (pp. 1383-1398)

(Contributed by Reginald Ryves, Assoc.M.Inst.C.E., M.Cons.E.)

PART III

RAINFALL — WELLS — WATERWORKS — DAMS — FLOW OF WATER IN PIPES—WATER PIPES—WATER TANKS (pp. 1399-1432)

(With Contributions by Reginald Ryves, Assoc.M.Inst.C.E., M.Cons.E., J. H. Horobin, R. Fairfax Middleton, Assoc.M.Inst.C.E., and H. F. Rutter, M.Inst.C.E.)

SECTION XXVI

PART I

SEWERS — SEWERAGE — FLUSHING AND VENTILATING SEWERS AND DRAINS — SEWAGE TREATMENT AND DISPOSAL — MINISTRY OF HEALTH'S REQUIREMENTS — STANDARDS OF PURITY—HOUSE DRAINAGE

Contributed by E. J. Silcock, M.Inst.C.E., F.G.S., F.S.I.
(Joint author of Moore and Silcock's 'Practical Treatise on Sanitary Engineering.')

SEWERS

Rate at which Rainfall Reaches a Sewer.

C = cubic feet per second per acre reaching sewer ; c = average cubic feet of rainfall per second per acre during heaviest fall ; S = average slope of ground in feet per 1,000 ft. ; a = number of acres drained ; K = ·75 for paved streets, ·625 for ordinary cases, ·31 for suburbs with gardens, lawns, and macadamised streets.

$$C = Kc\sqrt[4]{\frac{S}{a}}$$ (*Burkli-Zeigler.*)

(For other formulæ see 'Rainfall,' **p. 1399.**)

Discharging Capacity of Sewers.

R = hydraulic mean depth in feet ; S = sine of angle of inclination ; v = velocity in feet per second.

$$v = 124\sqrt[3]{R^2}\sqrt{S}$$ (*Crimp and Bruges.*)

Discharge from the Submerged Orifice of a Culvert.

D = diameter of orifice in inches ; L = length of out-fall culvert in feet ; W = quantity discharged in cubic feet per minute ; H = head in feet.

$$D = \cdot 531\sqrt[5]{\frac{LW^2}{H}}$$ (*Eytelwein.*)

Discharge from Culverts.

The important road improvement works now in progress in the United States have shown the need for further information regarding the capacity of culverts ; the more particularly because pre-existing data have been derived almost exclusively from experiments on ordinary straight-walled metal, stoneware, or concrete ducts. Of recent years, however, considerable use has been made of corrugated metal pipes, which offer a much greater resistance to the flow than straight-walled conduits. With a view, therefore, to fix the relative merits of different types of culvert, the Bureau of Public Roads of the U.S. Department of Agriculture arranged for a very comprehensive series of tests to be made at the hydraulic laboratory of the State University of Iowa, where an ample supply of water was available, as well as facilities for the accurate measurement of heads and discharges. The work was carried out under the direction of Mr. D. L. Yarnell and Professors F. A. Nagler and S. M. Woodward, and is described in detail in Bulletin No. 1 issued by the University, Iowa City, Iowa, U.S.A. (the cost of the report is one dollar). In all

over 3,000 experiments were made, in part on pipe culverts, of concrete, vitrified clay, and corrugated metal, and in part on concrete box culverts. The pipe culverts ranged in size from 12 in. to 30 in. in diameter, and the box culverts from 2 ft. square to 4 ft. square, and, in addition, further tests were made with rectangular box culverts of other than square section. As the result of their observations the authors propose the following formulae for the discharge from pipe culverts 30·6 ft. long, with straight endwall entrances :—

Concrete pipe with bevelled lip upstream,
$$Q = 4{\cdot}61\ D^{2.18}\ H^{0.50}$$

Vitrified clay pipe with bell end up-stream,
$$Q = 5{\cdot}07\ D^{2.05}\ H^{0.50}$$

Concrete pipe with square-cornered entrance,
$$Q = 4{\cdot}40\ D^{2.09}\ H^{0.50}$$

Corrugated metal pipe,
$$Q = 3{\cdot}10\ D^{2.31}\ H^{0.50}$$

where Q = discharge in cubic feet per second ; D = diameter of pipe in ft. ; H = difference in ft. of water level between the two ends.

More general formulæ applicable to culverts of any size and length when flowing full bore are :—

Concrete pipe with bevelled lip entrance,
$$Q = \frac{A\sqrt{2gH}}{\sqrt{1{\cdot}1 + \frac{0{\cdot}026\ L}{D^{1.2}}}}$$

Vitrified clay pipe, regular bell-end upstream,
$$Q = \frac{A\sqrt{2gH}}{\sqrt{1 + 0{\cdot}023\ D^{1.9} + \frac{0{\cdot}022\ L}{D^{1.0}}}}$$

Concrete pipe, square-cornered entrance,
$$Q = \frac{A\sqrt{2gH}}{\sqrt{1 + 0{\cdot}31\ D^{0.5} + \frac{0{\cdot}026\ L}{D^{1.3}}}}$$

Corrugated metal pipe,
$$Q = \frac{A\sqrt{2gH}}{\sqrt{1 + 0{\cdot}16\ D^{0.6} + \frac{0{\cdot}106\ L}{D^{1.2}}}}$$

where A = cross-sectional area of pipe in sq. ft. ; L = length of culvert in ft.

For concrete box culverts the following formulae are proposed, in which R denotes the mean hydraulic radius :

Box culverts with rounded entrance,
$$Q = \frac{A\sqrt{2gH}}{\sqrt{1{\cdot}05 + \frac{0{\cdot}0045\ L}{R^{1.25}}}}$$

Box culverts with square-cornered entrance,
$$Q = \frac{A\sqrt{A\,g\,H}}{\sqrt{1 + 0{\cdot}4\ R^{0.3} + \frac{0{\cdot}0045\ L}{R^{1.25}}}}$$

(*Engineering*, Nov. 26th, 1926.)

Circular Sewers.

x = area of sewer ÷ the wetted perimeter ; f = inclination in feet per mile ; v = velocity of flow, in feet per minute ; a = area of flow, in square feet ; V = volume of discharge, in cubic feet per minute.

$$v = 55\sqrt{x \times 2f} \qquad V = va.$$

American Rule.

(Large Sewers.)

The following formula is used by Mr. H. R. Asserson, chief engineer of the Sewer Department of Brooklyn, for proportioning the sizes of large sewers :—

$$D = \frac{3\ \text{log. A} + \text{log. N} + 8}{10},$$

where,

D = diameter of sewer in inches ; A = the area drained in acres ; N = the distance in feet in which the sewer falls 1 ft.

(Small Sewers.)

For small sewers up to 18 ins. in diameter the formula is—

$$D = \frac{3\ \text{log. A} + \text{log. N} + 6{\cdot}8}{10}$$

Salt-Glazed Ware Pipes.

Salt-glazed ware pipes may be used up to 36 ins. diameter, but above 24 ins. they are difficult to manufacture of a cylindrical shape, and brickwork or concrete is usually adopted.

British Standard Salt-Glazed Ware Pipes.*

(No. 65—1914.) (*Abstract.*)

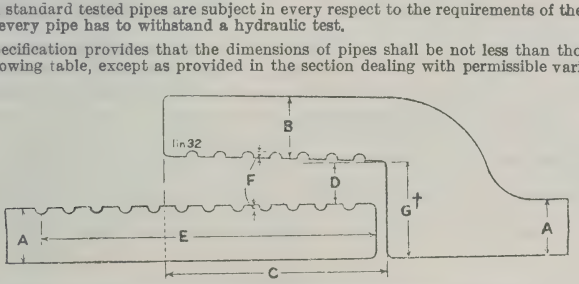
The specification divides pipes manufactured in accordance with the specification into two classes, viz. :—

British standard pipes. British standard tested pipes.

British standard pipes are pipes subject in every respect to the requirements of the specification, and a percentage are to be subjected to a hydraulic test.

British standard tested pipes are subject in every respect to the requirements of the specification, and every pipe has to withstand a hydraulic test.

The specification provides that the dimensions of pipes shall be not less than those set out in the following table, except as provided in the section dealing with permissible variation.



1	2	3	4	5	6	7
Int. Dia. of Pipe.	Minimum Mean Thickness of Barrel. A.	Minimum Mean Thickness of Socket. B.	Minimum Internal Depth of Socket. C.	Minimum Jointing Space. D.	Length of Grooving on Spigot. E=1½C.	Minimum Depth of Grooving. F.
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	In.
3	7/16	7/16	2	5/16	3	1/16
4	1/2	1/2	2	3/8	3	1/16
5	9/16	9/16	2¼	7/16	3 3/8	1/16
6	5/8	5/8	2¼	7/16	3 3/8	1/16
7	11/16	11/16	2¼	7/16	3 3/8	1/16
8	11/16	11/16	2½	1/2	3 3/4	1/16
9	3/4	3/4	2½	1/2	3 3/4	1/16
10	13/16	13/16	2¾	5/8	4 1/8	1/16
12	1	1	2¾	5/8	4 1/8	1/16
13	1 1/16	1 1/16	3	5/8	4 1/2	1/16
14	1 1/16	1 1/16	3	5/8	4 1/2	1/16
15	1¼	1¼	3	5/8	4 1/2	1/16
18	1½	1½	3	5/8	4 1/2	1/16
21	1⅝	1⅝	3¼	3/4	4 7/8	1/16
24	1¾	1¾	3½	3/4	5¼	1/16
27	1⅞	1⅞	3½	3/4	5¼	1/16
30	2	2	3½	3/4	5¼	1/16
36	2½	2½	3½	1	5¼	1/16

† Width of shoulder of socket, which must exceed the mean thickness of the barrel of the pipe by not less than the values for D given in column 5.

The sockets of pipes shall be conical, having a taper of one-sixteenth of an inch measured on the diameter per inch of length.

The length of the barrels of straight pipes shall be as follows :—

Up to and including 6 ins., 2 ft.

7 ins. to 10 ins. inclusive, 2 ft. and 2 ft. 6 ins.

12 ins. to 36 ins. inclusive, 2 ft., 2 ft. 6 ins., and 3 ft.

* By permission of the British Engineering Standards Association.

The interior of the sockets and exterior of the spigots shall have groove one-sixteenth inch deep for a length equalling one and a half times depth of socket.

The glazing shall be attained by the action of the fumes of volatilised common salt on the material of the pipes during the process of burning.

Subject to the mean thickness of the barrel and socket of a pipe being not less than that specified in columns 2 and 3 of the above table, a variation in the thickness of the barrel or socket may be allowed at any two points not exceeding the following :—

Diameter of Pipe.	Permissible Variation of Thickness.
Not exceeding 12 ins.	$\frac{1}{16}$ in.
13 ins. to 18 ins. inclusive	$\frac{3}{32}$ in.
28 ins. to 36 ins. inclusive	$\frac{1}{8}$ in.

The internal diameter of the barrel shall nowhere deviate from the standard diameter by more than the figure given in the following table :—

Standard Internal Diameter of Pipe.	Permissible Variation from Diameter.	Standard Internal Diameter of Pipe.	Permissible Variation from Diameter.
3 ins. and 4 ins.	$\frac{1}{8}$ in.	15 ins. and 18 ins.	$\frac{3}{16}$ in.
5 „ to 7 „	$\frac{3}{16}$ in.	21 „ and 24 „	$\frac{1}{4}$ in.
8 „ and 9 „	$\frac{1}{4}$ in.	27 „	$\frac{5}{16}$ in.
10 „ to 14 „	$\frac{5}{16}$ in.	30 „ and 36 ins.	1 in.

The permissible deviation of straightness of the barrels measured on the inside of the curve for all diameters of pipes shall be as follows :—

Length of Pipe.	Permissible Deviation from Straightness.
2 ft.	$\frac{3}{16}$ in.
2 ft. 6 ins.	$\frac{1}{4}$ in.
3 ft.	$\frac{5}{16}$ in.

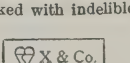
The hydraulic test shall be an internal pressure of 20 lbs. per square inch without showing signs of injury or leakage.

The pipes shall also be tested for absorption by testing pieces of the body of the pipe taken at more than 6 ins. from the end of the pipe. The test pieces shall be of whole thickness of pipe, and shall have two glazed surfaces each of which shall be not less than 8 ins. super or more than 20 ins. super. The test pieces shall be thoroughly dried at a temperature of 150°C., and shall then be immersed in cold water which shall be raised to boiling point and there maintained for one hour. The percentage of increase of weight shall not exceed the following :—

Thickness of Pipe.	Percentage of Increase of Weight.
$\frac{3}{4}$ in. and under	5 per cent.
Over $\frac{3}{4}$ in. and up to 1 in.	6
Over 1 in. and up to $1\frac{1}{2}$ in.	8 „
Over $1\frac{1}{2}$ in.	10 „

British standard pipes shall be marked with indelible black with the following :—

British standard tested pipes shall be marked :—



Concrete Pipes.

Sewer pipes constructed of concrete, either with or without steel reinforcement, are gaining favour amongst engineers, especially for the larger sizes of pipes and for sewers where numerous junctions are not required.

These pipes are made by several firms, and are constructed of a rich mixture of concrete with granite aggregate and the highest quality of cement. They are mostly made in moulds as plain cylinders with a curved joint made in the thickness of the concrete.

Recently the Stanton Iron Company have produced concrete pipes made by the Hume process, which is in essence a centrifugal system (see p. 1247) in which the ingredients are passed into rapidly revolving cylinders and are spread out into a uniform pipe of great density and strength. If desired, steel reinforcement can be placed in the revolving cylinders and be buried in the thickness of the concrete pipe. By this means the pipes can be greatly strengthened and can be made to carry high internal pressures.

The Hume Spun Pipes can be made in six and eight feet lengths, thus saving joints and labour in laying as compared with salt-glazed ware or ordinary moulded concrete pipes. The joints are made either by sockets spun on the pipe itself, or by loose collars used with plain cylindrical pipes. Square junctions and junctions inclined at 45° for branch drains can be supplied.

All concrete pipes above 18-in. diameter are best laid on a bed of concrete haunched up to the semi-diameter. This precaution is essential except in special circumstances with pipes of 24-inch diameter and upwards, whether the pipes are made of concrete or salt-glazed ware. Even allowing for the concrete bed, pipes made of concrete are less costly than salt-glazed ware pipes of the best qualities.

Egg-shaped Sewers.

The general form of the section of this class of sewers is shown by fig. 1.

When full.	$\frac{3}{4}$ths full.	$\frac{2}{3}$rds full.	$\frac{1}{2}$ full.	$\frac{1}{3}$rd full.	$\frac{1}{4}$ full.
Area of whole section . . 1·1485 d^2	·8795 d^2	·7558 d^2	·5091 d^2	·2840 d^2	·1862 d^2
Perimeter of wetted surface . . 3·9649 d	2·6455 d	2·39415 d	1·8915 d	1·3746 d	1·1055 d
Hydraulic radius . . ·2897 d	·3325 d	·3157 d	·269 d	·2066 d	·1685 d

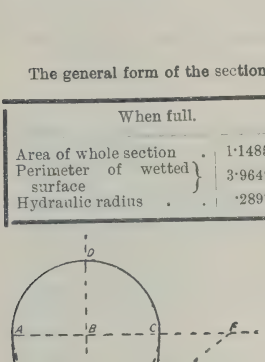


Fig. 1.

This form of sewer gives a fair hydraulic mean depth for flows varying greatly in volume. In a town sewered upon the combined system this is an advantage over the circular section, especially in the case of sewers laid in undeveloped areas where it is frequently necessary to make provision largely in excess of present requirements.

Diameter of Upper Circle = AO = d.
Conjugate Diameter = DE = $1\frac{1}{4}d$.
Radius of Invert = $\frac{1}{4}d$.
Radius of Side = FA = $1\frac{1}{4}d$.

Another advantage possessed by egg-shaped sewers is that for equal sectional areas this form is more easily entered by workmen than the circular section.

On the other hand, the egg-shaped section is not so strong to resist external pressure as the circular, and consequently if used in bad ground special precautions must be taken.

In ordinary practice, egg-shaped sewers are used for those sizes of culvert which fall between the largest circular stoneware-pipe sewers and the smallest size of circular brick culvert in which men can conveniently work.

These limits fix the maximum and minimum sizes of egg-shaped sewers at 3 ft. 6 ins. × 2 ft. 4 ins. and 2 ft. 3 ins. × 1 ft. 6 ins., respectively.

For ordinary depths the smaller sizes of egg-shaped sewers are constructed of one ring of brickwork half-a-brick thick; for the large sizes a second ring of brickwork may be introduced or, preferably, the sewer may be strengthened by placing concrete outside the brickwork.

The following table gives particulars of the ordinary sizes of egg-shaped sewers :—

Dimensions.	Sectional Area in Square Feet.		Wetted Perimeter. Lineal Feet.		Brickwork in superficial yards of half-brick thick per Lineal Yard of Culvert.	
	Total.	Up to ⅔rds of top circle.	When full.	When full to ⅔rds top circle.	One half-brick ring.	Two half-brick rings
Ft. Ins. Ft. Ins.						
2 3 × 1 6	2·58	1·70	5·95	3·59	2·72	—
2 6 × 1 8	3·19	2·09	6·61	3·99	2·99	—
2 9 × 1 10	3·86	2·54	7·27	4·39	3·24	7·33
3 0 × 2 0	4·59	3·02	7·93	4·79	3·42	7·65
3 3 × 2 2	5·39	3·54	8·59	5·19	3·54	8·06
3 6 × 2 4	6·25	4·11	9·25	5·59	—	8·60

DISCHARGE OF WATER IN CUBIC FEET PER MINUTE, IN PIPES OR SEWERS, FOR ANY LENGTH AND HEAD, AND FOR DIAMETERS FROM 1 IN. TO 10 FT.

Diameter.	Tabular No.	Diameter.	Tabular No.	Diameter.	Tabular No.	Diameter.	Tabular No.
Ins.		Ft. Ins.		Ft. Ins.		Ft. Ins.	
1	4·71	1 1	2876·7	2 7	25270·	4 1	79380·
1½	8·48	1 2	3463·3	2 8	27358·	4 2	83492·
1½	13·02	1 3	4115·9	2 9	29547·	4 3	87730·
1¾	19·15	1 4	4836·9	2 10	31834·	4 6	101207·
2	26·69	1 5	5628·5	2 11	34228·	4 9	115854·
2½	46·67	1 6	6493·1	3 0	36725·	5 0	131703·
3	73·5	1 7	7433·	3 1	32329·	5 3	148791·
3½	108·14	1 8	8449·	3 2	42040·	5 6	167139·
4	151·02	1 9	9544·	3 3	44863·	5 9	186786·
4½	194·84	1 10	10722·	3 4	47794·	6 0	207754·
5	263·87	1 11	11983·	3 5	50835·	6 6	253781·
6	416·54	2 0	13328·	3 6	53995·	7 0	305437·
7	612·32	2 1	14758·	3 7	57265·	7 6	362935·
8	854·99	2 2	16278·	3 8	60648·	8 0	426481·
9	1147·6	2 3	17889·	3 9	64156·	8 6	496275·
10	1493·5	2 4	19592·	3 10	67782·	9 0	572508·
11	1894·9	2 5	21390·	3 11	71526·	9 6	655369·
12	2356·	2 6	23282·	4 0	75392·	10 0	745038·

The last table is applicable to sewers and drains by taking the same proportion of the tabular numbers that the cross-section of the water in the sewer or drain bears to the area of the whole area of the sewer or drain.

The formula upon which the table is constructed is :—

$$Q = 2356 \frac{\sqrt{d^5}}{\sqrt{\frac{l}{h}}},$$

where,

Q = volume of discharge, in cubic feet per minute;
d = diameter of pipe in feet;
h = height of fall of the water in feet;
l = length of pipe in feet.

Application of the Table.

To compute the volume of fluid discharged, the length of the pipe or sewer, the height or fall, and the diameter being given.

Rule.—Divide the tabular number opposite the diameter of the pipe by the square root of the rate of inclination (length ÷ height); and the quotient will give the volume required.

Example.—A pipe has a diameter of 9 ins. and a length of 4,750 feet; what is its discharge under a head of 17·5 feet?

Tabular number for 9 ins = 1147·6, and

$$\sqrt{\frac{4750}{17{\cdot}5}} = \sqrt{271{\cdot}4} = 16{\cdot}47;$$

then,

$$\text{Discharge} = \frac{1147{\cdot}6}{16{\cdot}47} = 69{\cdot}67 \text{ cubic ft. per minute.}$$

Sewerage Systems.

There are two principal systems of sewerage, known respectively as the 'combined' and the 'separate,' systems. In the former, one set of sewers is provided for the removal of both the sewage proper, and also the rainwater; in the separate system two sewers are provided in every street, one for sewage and the other for rainwater. Both systems have their advantages, but these are counterbalanced by certain disadvantages.

The combined system has the advantage of simplicity, but it involves large and deep sewers with small volumes of sewage flowing in them in dry weather, and where the sewage has to be pumped and purified this system produces a larger volume of sewage than the separate system.

On the other hand, a separate system of sewers becomes very complicated, especially in large towns, and there is great risk of sewage drains being connected to the surface water sewers. The latter discharge direct into the nearest river or stream, and can therefore be made of smaller size than in the combined sewerage system, which concentrates all liquid upon one point. The drainage of all streets and yards is taken direct into the surface-water sewers. As the first washings of these surfaces are very foul, there is danger of pollution of the streams into which the surface-water sewers discharge.

In most cases neither system is carried out in its entirety, but a combination is effected with a view to adopting the best points of both.

The quantity of sewage in a strictly separate system should, if the sewers were watertight, be measured by the water consumed, and it is usually taken at from 20 to 40 gallons per head of the population draining, and the rate of flow in the middle of the day is from one and a half times to twice the mean.

Where a partially-separate system is adopted, the back roofs and back yards of the houses are usually drained to the sewage sewers, and the impervious area per head of the population varies from 10 square yards to 20 square yards.

The rainfall on this area, at the rate of from ¼ in. to 1 in. per hour, depending upon the area of the district (being smaller for large areas than for small), must be added to the sewage, to find the discharging capacity of the sewers.

In a combined system of sewers the main factor is rainfall, and the rate of fall and proportion of impervious area and size of district must be taken into consideration in determining the dimensions of the sewers.

The proportion of rainfall flowing off an urban area probably never exceeds 75 per cent. of the gross fall, and usually 30 per cent. to 50 per cent. is sufficient to provide for in ordinary circumstances.

(For discharges of catchments, including sewered areas, see 'Rainfall,' page 1399.)

As an example of practical working, the following table may be quoted, giving the basis on which sewers have for many years been designed in a North Country city with an average rainfall of 36 ins., and nearly all streets being covered with impervious paving:—

Drainage Area in Acres.	Inches of Rainfall per Hour.	Drainage Area in Acres.	Inches of Rainfall per Hour.	Drainage Area in Acres.	Inches of Rainfall per Hour.
20	1·00	75	0·47	350	0·35
23	0·95	100	0·45	375	0·34
26	0·90	125	0·44	400	0·33
29	0·85	150	0·43	425	0·32
32	0·80	175	0·42	450	0·31
35	0·75	200	0·41	475	0·30
38	0·70	225	0·40	500	0·29
41	0·65	250	0·39	525	0·28
44	0·60	275	0·38	550	0·27
47	0·55	300	0·37	575	0·26
50	0·50	325	0·34	600	0·25

Flushing Sewers and Drains.

All sewers and house drains should be laid at sufficiently good gradients to secure self-cleansing. To prevent deposit, 4-inch diameter drains should have a fall of not less than 1 in 40, 6-inch drains 1 in 60, and 9-inch drains 1 in 100. Whenever these gradients cannot be obtained special facilities should be provided for flushing, and this operation should be regularly attended to once or twice per week.

The water supply to closets, slop-sinks, etc., should be obtained from cisterns specially provided for the purpose, and not direct from the water-main, nor yet from cisterns in which the water for other domestic uses, such as cooking and drinking, is stored.

The best material for the cisterns is galvanised wrought-iron of No. 14 s.w.g. thickness.

The following table gives the capacities of flushing chambers for drains:—

CAPACITY OF FLUSHING CHAMBER REQUIRED FOR PIPES OF DIFFERENT SIZES.

Size of Pipe. Diameter in Inches.	Gradient of Pipe.	Capacity of Chamber in Gallons.
4	1 in 40	30
	„ 50	40
6	„ 60	60
	„ 100	100
	„ 200	160
9	„ 100	200
	„ 150	250
	„ 200	300
	„ 300	400
12	„ 100	300
	„ 200	400
	„ 300	500
	„ 400	750
	„ 500	1,000
	„ 600	1,500

Sewers as distinguished from drains may be laid at flatter gradients than those quoted without special flushing arrangements, but this depends upon the quantity of sewage flowing down the pipe. A sewer flowing half full will maintain itself in good order at a much flatter gradient than one flowing quarter full. It is usually found that a velocity of 2 feet to 2½ feet per second will produce self-cleansing conditions.

In practical sewage works sewers with a small flow should have the following minimum gradients :—

8 ins. diam. 1 in 75.	12 ins. diam. 1 in 150.	18 ins. diam. 1 in 250.
9 ins. diam. 1 in 100.	15 ins. diam. 1 in 200.	

Where, however, the sewers have a goodly flow of sewage the following gradients will be found to be self-cleansing :—

8 ins. diam. 1 in 175.	12 ins. diam. 1 in 400.	18 ins. diam. 1 in 800.
9 ins. diam. 1 in 250.	15 ins. diam. 1 in 600.	21 ins. diam. 1 in 1,000.

Ventilation of Sewers.

The object of the ventilation of sewers is to remove the foul emanations arising from the sewage and the fouled surfaces of the sewers above the actual liquid and to discharge them at points where they will easily become diluted with fresh air and not cause a nuisance.

Until comparatively recently it was thought that sewer air was harmful in itself and was likely, not only to impair the vitality of the subject, but also to be the direct cause of infectious disease.

Now, however, it is more generally conceded that sewer air is at least as free from bacteria as the air near the surface of a road, and that apart from any offensiveness to the sense of smell it is not dangerous to persons breathing it. The degree of offensiveness of sewer air depends more largely upon the condition of the sewer than upon the effectiveness of the ventilation.

Fresh sewage does not produce a bad smell, and modern sewers constructed with the best materials and gradients and regularly flushed do not give off foul emanations. On the other hand, sewers constructed of absorbent materials, laid to bad gradients, containing deposits of sewage solids, and in which flushing is absent or carried on at very long intervals, the sewer air becomes very foul and is liable to produce nuisance.

Formerly, the generally accepted theory was that ' sewer gas,' being lighter than air, would tend to rise up the gradient of the sewer and would be discharged at its summit.

This idea has now been demonstrated to be erroneous : there are many factors which affect the motion of air in sewers such as temperature, humidity, ebb and flow of sewage, the drag of the flowing sewage, the natural diffusion of gases, friction on the periphery of the sewer, and the action of the wind.

Of all these, by far the most important is the latter, and it dominates the other factors except on very calm days.

The system of ventilation of sewers introduced on the advice of the late Sir Robert Rawlinson many years ago, was to relieve the pressure of sewer air by means of gratings in manhole covers at ground level.

These open ventilators were fixed at intervals of about 100 yds., and if it was found that some of the ventilators were a nuisance, additional openings were to be made at shorter intervals.

It has been the almost universal experience in most towns that at certain times these surface grids caused nuisance by allowing the sewer air to escape and bad smells resulted.

A modification of the system has been effected by the introduction of tall shafts connected with the sewers alternating with openings at street level.

The tall shafts theoretically act as outlets and the openings at ground level as fresh-air inlets. It is found, however, in practice, that this order is under varying conditions reversed and it is never certain which will be the inlet and which the outlet.

The general practice now is to have all manhole covers sealed where they are situated in narrow streets near the doors and windows of houses or in other situations likely to cause nuisance, and to have vertical steel or iron columns 20 to 25 ft. in height and 6 ins. to 12 ins. diameter, fixed in the line of curb and connected to the manholes to act either as inlets or outlets : the tops of the columns are fitted with wire guards.

In some cases the shafts consist of iron socketed pipes fixed against a building and carried up above the eaves or roof ridges in such a way as to be clear of all windows, skylights, or chimneys.

In broad suburban roads or where the sewers cross agricultural land, the manhole covers are still left open for ventilation purposes.

Connecting the sewers to the flues of factory chimneys has been tried but the draught is confined to a very small area and is not nearly so effective as one would anticipate. Numerous proprietary systems of sewer ventilation by means of fans, special cowls, gas destructors, antiseptic sprays and deodorants have been put on the market from time to time, but these are generally successful only in very favourable circumstances and cannot be recommended for general use.

SEWAGE TREATMENT.

Quantity of Sewage to be dealt with.

The quantity of sewage per head of the population drained, and the effect of the admission of rainwater, have been dealt with in the preceding matter dealing with sewers.

It has now become fairly well established in practice that in towns where the sewers are upon the combined system, and where the sewage is purified, the works shall be capable of providing full treatment for a quantity equal to three times the mean dry weather flow, and that in addition to this there shall be means of partially treating a further quantity equal to three times the mean dry weather flow, any surplus beyond this to be discharged untreated into the streams.

In towns sewered on the separate system the full treatment can be restricted to a volume of twice the mean dry weather flow, and partial treatment must in that case be provided for four times the dry weather flow.

Apart from the fluctuations in volume due to the admission of storm water, the quantity varies from hour to hour during the day, and in different days of the week, owing to the greater or lesser discharge of liquid from the houses in the district.

The following diagram represents the rate at which sewage is received on an average day in dry weather at the Leeds Sewage Purification Works :—

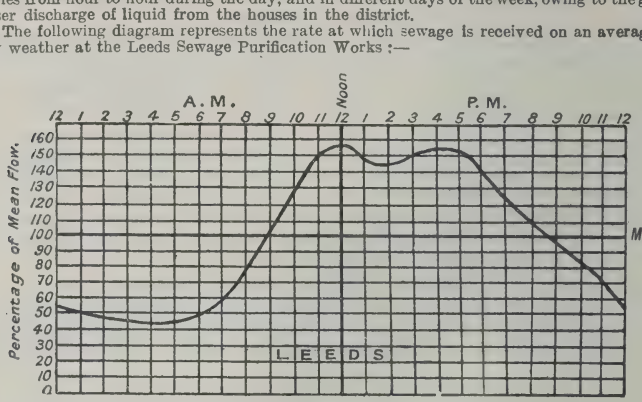


FIG. 2.

It will follow from what has been said that there are continual fluctuations in the volume of liquid to be dealt with at a sewage purification works, and such works must therefore be capable of adapting themselves to all the conditions under which they may be called upon to act.

General Outline of Treatment.

The object of all sewage purification processes is to convert the organic substances in the sewage into harmless inorganic matter. The organic matter may be in suspension or in solution, and it is not sufficient to eliminate the former, as was the case in some of the older processes, and leave the matters in solution, as these will subsequently putrefy and cause nuisance.

In average sewage of a domestic character—i.e. free from trade refuse—there are usually about 100 grains of solid matter per gallon, of which 40 grains are organic.

Grains per Gallon.	Organic.	Mineral.	Total.
Solid matter in suspension . . .	20	10	30
Solid matter in solution. . . .	20	50	70
Total	40	60	100

The solids in suspension can be reduced by subsidence, but the matter in solution must be oxidised, and this is effected by the action of bacteria.

In all cases the first step is to remove the largest and heaviest of the suspended matters, and this is done in detritus and screening tanks.

Such tanks should have a capacity of from 2 per cent. to 5 per cent. of the daily sewage flow, and should contain iron gratings for the removal of floating matters, scum boards to arrest the scum ; and they should be so designed as to reduce the velocity of flow of sewage sufficiently to allow the heaviest of the road grit, etc., to settle to the bottom of the tank.

The next step is usually to remove the lighter matters in suspension, and this is generally effected by providing tanks of suitable design : (*a*) to allow the particles to settle by gravity ; (*b*) this process is sometimes expedited by the addition of chemicals known as precipitants; (*c*) for liquefaction of the solids in suspension by bacterial action and settlement combined, as

in septic tanks; (*d*) assisting the action by presenting large surface to attract floating matter as in the Travis tank or Dibdin slate filter.

Having freed the liquid as far as possible of the suspended solids with their organic contents, the effluent from the first part of the process must be subjected to a further treatment, which has for its object the oxidisation of the matters in solution and their conversion into harmless inorganic compounds which will not putrefy or take up oxygen from the stream into which it may be intended to discharge them.

This part of the process may be effected by application to land or to some form of bacterial filter; but if the latter it is usually necessary to have a further process for the removal of the solids in suspension which are evacuated by the filter. This can be effected either by land treatment, filtration, or settlement in tanks or lagoons.

The capacity of the tanks necessary for the removal of the suspended solids depends upon the strength of the sewage and a variety of circumstances.

If the tank can be used for quiescent settlement—i.e. filled with sewage, allowed to stand, and then emptied by decanting—a capacity of two hours to four hours' sewage flow will usually be sufficient provision. The usual method of using such tanks, however, is to use them continuously by drawing off over a long weir in a thin film at the same rate as the filling; in such a case the tanks should have from six to twelve hours' capacity.

If, however, the settlement be assisted by the addition of precipitants, and the tanks are used for continuous precipitation, their capacity may be reduced by one-half.

Septic tanks vary in capacity from fifteen hours to twenty-four hours' flow, and there is tendency towards the smaller limit in present practice.

Precipitation by the use of chemicals has some effect upon the organic matter in solution as well as that in suspension, and by a sufficiently large expenditure it is possible to reduce the impurity in solution by as much as 50 per cent. The cost of doing this, however, would be prohibitive in actual practice. (See table, pages 1370 and 1371.)

The precipitants chiefly used are lime in a caustic state, 10 to 12 grains per gallon of sewage; lime and sulphate of iron or green copperas in the proportion of 6 grains to 3 grains of lime with 3 grains to 1 grain of copperas; lime and alumina in the proportion of 2 of the former to 1 of the latter, usually about 8 and 4 grains respectively; alumino-ferric, either with or without lime, proportion 10 to 7 grains lime and 5 to 3 grains alumino-ferric—if used without lime, up to 8½ grains per gallon.

Sludge.

The solid matters precipitated in the tanks produce what is known as sewage sludge, containing about 90 per cent. of water.

The final disposal of the sludge resulting from sewage purification has always been one of the chief difficulties of the problem of sewage treatment. The manurial value of the material is small, and it is a waste product of the process which must be got rid of as cheaply and as expeditiously as possible.

The methods of doing this may be briefly summarised as :—

1. Removal to sea.
2. Earth burial.
3. Cremation.
4. Drying in lagoons.
5. Drying in prepared beds.
6. Filter pressing.

The quantity of sludge per million gallons of sewage varies with the quality of the sewage, and the process adopted for the separation of the solid matter therein.

The method of estimating the quantity of sludge from a given process is as follows :—

To the average number of grains per gallon of solids in suspension in the sewage add the grains of precipitant applied, and deduct from the total the grains of solids in suspension in the tank effluent; this gives the grains of solid matter deposited from each gallon of sewage.

If

G = grains per gallon as above determined;
M = sewage per day in million gallons;
T = tons of wet sludge per day (30 per cent. water)
P = percentage of water in pressed cake;
C = tons of cake containing P per cent. water;

then,

$$T = \frac{100\,M \times G}{154} \qquad C = \frac{T}{100 - P}$$

Pressed sludge cake containing 50 to 55 per cent. of water weighs about 14 cwts. per cubic yard.

SHOWING THE PERCENTAGE REDUCTION OF OXIDISABLE ORGANIC MATTER

(All quantities are stated

Liquid used for Experiment.	Percentage reduction of dissolved Oxidisable										
	1	2	3	4	5	6	7	8	9	10	11
Raw Sewage from	Lime in solution . 3·7	Lime in solution . 5·0	Lime in solution . 10·0	Lime in solution . 15·0	Lime as Milk . 15·0	Lime in solution . 3·7 Iron sulphate . 0·3	Lime in solution . 3·7 Iron sulphate . 1·0	Lime in solution . 3·7 Iron sulphate . 2·5	Lime in solution . 3·7 Iron sulphate . 5·0	Lime in solution . 5·0 Iron sulphate . 2·0	Lime in solution . 5·0 Iron sulphate . 4·0
Metropolitan system .	20	26	29	37	30	27	27	29	33	31	33
" "	6	10	28	64	12	10	10	18	32	27	34
" "	18	21	14	19	22	12	19	25	24	26	21
" "	26	26	26	37	10	24	25	38	33	30	31
" "	10	9	10	12	11	10	12	13	27	26	24
" "	23	29	29	34	27	23	27	35	37	33	33
" "	2	8	15	29	0	0	5	17	20	12	16
" "	12	25	30	33	25	16	22	24	32	26	30
" "	7	19	21	27	10	18	9	20	22	17	14
" "	5	12	13	18	4	5	3	11	16	13	15
" "	3	12	9	18	3	10	13	15	17	14	13
" "	15	16	19	48	44	11	15	17	20	17	21
" "	12	15	21	25	17	10	16	20	22	17	20
" "	9	15	19	23	19	11	17	20	21	18	18
" "	10	14	22	19	9	8	16	14	14	16	17
" "	12	12	17	24	12	7	12	15	22	19	19
" "	15	16	16	15	9	9	11	13	17	13	14
" "	8	8	20	35	8	11	8	14	17	14	20
" "	8	7	19	34	7	10	7	13	16	13	19
" "	3	4	4	4	9	4	4	4	8	7	6
" "	4	5	5	0	7	5	5	7	5	7	7
" "	12	20	26	28	16	15	22	23	24	23	22
" "	6	9	9	14	3	5	5	7	11	9	12
Average . . .	11	15	19	25	13	11	13	18	21	18	19
Cost per annum for chemicals . .	£ 13,505	18,250	36,500	54,750	54,750	15,695	20,805	31,755	50,005	32,850	47,450
Other sources . .	..	52	..	..	..	..	..	..	..	58	61
" " . .	..	..	10	..	..	..	..	..	..	..	..

Lime taken as costing £1 per ton; Iron, £2 per ton; Alumina, £3 10s. per ton.

IN SOLUTION BY VARIOUS METHODS OF CHEMICAL PRECIPITATION.
in grains per gallon.)

Organic matter by treatment of liquid with															
12	13	14	15	16	17	18	19	20	21	22	23	24	25		
Lime in solution . 5·0 Iron sulphate . 5·0	Lime in solution . 5·0 Iron sulphate . 8·0	Lime in solution . 5·0 Iron sulphate . 10·0	Lime in solution . 10·0 Iron sulphate . 10·0	Lime in solution . 5·0 Alumina sulph. . 5·0	Lime in solution . 5·0 Iron sulphate . 5·0 Animal charcoal . 5·0	Lime in solution . 5·0 Iron sulphate . 1·0 Alumina sulph. . 5·0	Lime in solution . 7·0 Iron sulphate . 1·5 Alumina sulph. . 5·0	Lime in solution . 10·0 Iron sulphate . 2·0 Alumina sulph. . 10·0	Lime in solution . 14·0 Iron sulphate . 3·0 Alumina sulph. . 10·0	Lime in solution . 15·0 Iron sulphate . 3·0 Alumina sulph. . 15·0	Lime . . . 28·0 Iron sulphate . 6·0 Alumina sulph. . 20·0	Lime . . . 56·0 Iron sulphate . 12·0 Alumina sulph. . 40·0	Lime . . . 700·0 Iron sulphate . 100·0 Alumina sulph. . 500·0	Average.	
35	34	36	36	33	40	32	32	34	37	39	44	49	53	34	
32	36	38	36	32	32	33	36	67	46	45	40	43	58	33	
27	28	30	28	15	27	19	17	25	22	23	21	28	45	23	
32	32	36	30	26	32	30	31	30	25	36	30	30	45	30	
24	21	25	24	16	24	14	14	16	12	29	14	26	41	19	
39	36	36	40	37	38	38	39	33	37	39	44	55	62	36	
22	25	28	32	16	21	19	24	22	11	26	20	24	59	19	
34	37	41	32	24	26	28	32	29	25	31	28	43	59	30	
25	42	25	26	21	26	19	21	16	19	22	29	31	54	22	
17	21	22	21	16	22	21	25	25	25	26	25	25	45	18	
17	19	21	16	14	15	13	15	16	11	14	8	18	39	15	
20	24	25	38	24	24	25	24	26	27	38	40	46	59	27	
22	26	25	24	16	22	25	21	21	24	22	25	40	54	22	
19	22	32	19	23	18	19	19	18	18	20	19	18	48	20	
21	20	20	21	21	19	16	17	22	27	19	17	21	47	19	
21	23	23	24	15	19	17	18	22	26	25	28	39	62	21	
15	20	11	17	10	19	15	14	15	13	17	16	17	45	16	
19	26	27	31	20	23	18	23	28	29	31	30	41	58	23	
18	25	26	29	19	22	17	22	26	18	29	29	40	55	20	
8	8	10	10	0	7	7	11	12	13	12	13	29	58	10	
9	9	7	11	0	12	11	12	13	14	11	13	16	44	10	
23	26	22	23	22	20	20	20	22	21	25	22	33	59	23	
12	14	13	14	10	10	11	28	18	12	12	14	19	50	17	
18	25	25	30	18	22	20	22	24	21	26	24	31	52	..	
54,750	76,050	91,250	109,500	82,125	237,250	39,425	100,375	178,850	200,750	268,275	401,500	803,000	9,672,500	..	
..	61	69	..	..	..	..	..	..	..	..	..	..	..	..	
56	..	..	..	33	..	..	..	..	..	..	..	..	..	..	

Volume of sewage taken at 156,800,000 gallons per day. (*W. J. Dibdin.*)

Land Treatment.

The effluent from the various tank processes described may be suitably oxidised by application to land.

There are two distinct methods of using land. Where a suitable light friable soil can be obtained, filtration is the process adopted, the sewage passing downward through the soil to underdrains. If, on the other hand, the available land be all of a retentive character, which will not allow of the easy passage of water, the method adopted is to pass the tank effluent over the surface of the land by irrigation and to pick it up again in suitable catchwaters and reapply it to a fresh plot, and subsequently to a third or even a fourth plot before final discharge into the stream ; such land should not be underdrained, or only to a very slight extent.

The quantity of liquid which can be dealt with per acre of land will depend upon the character of the land, the strength of the sewage, and the degree of excellence of the preliminary treatment.

The following table gives a scale of quantities for various soils and average sewage, but great care must be exercised in applying the table :—

Soil.	Crude Sewage.		Settled Sewage.		Efficient Preliminary Treatment. Land Secondary.	
	Gallons.	Population.	Gallons.	Population.	Gallons.	Population.
Light sandy loam on gravel and sand	15,000	375	20,000	500	30,000	1,000
Sandy loam on gravel and sand	12,000	300	17,000	400	30,000	1,000
Peaty soil on gravel and sand	10,000	250	13,500	325	30,000	1,000
Sand and gravel	8,000	200	10,000	250	30,000	1,000
Gravelly loam on gravel and sand	6,000	150	8,000	200	20,000	666
Loam getting more clayey	4,500	125	6,000	150	15,000	500
Heavy loam on marl	3,000	75	5,000	130	12,000	400
Clay soil on clay	1,500	50	4,000	120	10,000	333
Stiff clay soil on dense clay	1,000	33	3,000	100	10,000	333

Artificial Bacterial Methods.

The alternative to land treatment for the final or oxidising process in sewage treatment is the use of bacteria beds. These are divided into two main classes :—

(*a*) Contact beds.
(*b*) Percolating or continuous filters.

Contact beds consist of tanks filled with material such as broken clinker, and used by filling them up with tank effluent until the filtering medium is just charged with liquid, allowing it to stand for a given period, draining off, and standing empty. The usual cycle of these operations is as follows :—

Filling	1 hour.
Standing	2 hours.
Draining	1 hour.
Aerating	4 hours.
	8 hours.

With this cycle three fillings per day are possible, and as the liquid capacity of a bed is 33 per cent. of its gross capacity, it follows that a bed is capable of treating each day a quantity of tank effluent equal to its gross capacity—i.e. when containing no filtering medium. The operations of filling and emptying may be done by hand valves or by automatic valves worked by floats. The depth of contact beds should not exceed 4 ft.

The second class of bacteria beds consists of masses of hard rough material, such as broken stone or slag of gauges varying from 3 ins. to ¼ in., over which the tank effluent is distributed in the form of spray and through which it percolates. The bed may or may not be enclosed by means of side walls, but it must have a good substantial floor, efficient means of drainage, and a proper system of distribution if good results are to be obtained. The depth of the beds varies from 4 ft. to 10 ft., depending upon the fall available and the size of the filtering medium.

The quantity of tank effluent which can be applied per square yard of surface of the bed depends upon the strength of the sewage and size of the material employed for filtering, and varies from 100 gallons per square yard per day to as much as 400 gallons, but this quantity cannot be maintained for long. A good average quantity is 150 gallons per square yard per day for a filter 6 ft. deep composed of materials averaging 2 ins. diameter; this will allow the bed to take the hourly fluctuations in the rate of flow of sewage, and for short periods it will take storm-water up to three times the figure named.

The sprinkling process can be effected by a variety of arrangements, most of which have revolving arms operating in different ways, but there are also fixed jets and reciprocating travellers for rectangular beds.

The relative merits of contact beds and percolating beds are as follows :—

1. Where there is not much fall available, single contact beds can be constructed with about 3 ft. 6 ins. of fall: double contact beds require 7 ft.

A continuous filter may be constructed with a fall of about 5 ft. 3 ins., but if a coarse filtering material is to be used this must be increased by 2 ft., or possibly more.

2. The tanks used for contact beds must be watertight, and therefore the facilities of the site for making watertight beds must be taken into consideration.

3. The weight of the material used in a continuous filter is greater than that of a contact bed, and therefore the stability of the ground must be taken into consideration.

4. Double contact beds must be provided if an equivalent degree of purification is required to that produced by a continuous filter.

5. Contact beds must be worked by automatic gear if three fillings per day are required, but may be operated by hand if two fillings are sufficient.

6. Generally speaking the gear for operating continuous filters is less liable to get out of order than that for contact beds.

7. Where the facilities for reducing suspended solids are limited, contact beds will sludge up and deteriorate much more quickly than continuous beds.

8. Percolating beds are more adaptable to fluctuating rates of flow than contact beds.

9. The first cost of percolating filters is less than that of contact beds for equal quantities of sewage.

10. Generally speaking, continuous filters give better results than contact beds.

Requirements of the Ministry of Health.

The following are believed to be the requirements of the Ministry with regard to sewage purification plants :—

Quantity of Sewage.

Quantity of sewage per head is usually reckoned at the rate of 30 gallons per day for dry-weather flow, for domestic purposes. In some cases this figure is modified to 20 gallons. If there be trade refuse to be dealt with, this must be added.

The quantity of sewage and rainwater to be fully treated at the works is :—

(*a*) Where the sewers are on the combined system, three times the domestic average plus $1\frac{1}{10}$ trade refuse.

(*b*) Where the sewers are on the separate system, twice the domestic sewage plus $1\frac{1}{10}$ trade refuse.

The quantity of sewage and rainwater to be partially treated is :—

(*a*) Where the sewers are on the combined system, three times the domestic sewage plus $1\frac{1}{10}$ trade refuse.

(*b*) Where the sewers are on the separate system, four times the domestic sewage plus $1\frac{1}{10}$ trade refuse.

Any excess over these quantities may be discharged over the stormwater overflows.

The above degrees of dilution are to be calculated on the average rate of flow throughout the 24 hours. The effect of this is that during the daytime when the average rate of flow is exceeded, the degrees of dilution will be less than those stated above, and on the other hand during the night the degree of dilution will be largely exceeded.

Storm Overflows.

Storm overflows on sewers should as far as is reasonably practicable be avoided, but where they are necessary they should be placed in such positions and with the weirs so fixed that no nuisance is likely to result. In any district where there is an active river authority the Ministry will desire to be informed of the opinion of such authority in respect of any proposed overflows.

In the absence of any special circumstances overflow weirs should be fixed so as not to come into operation until the flow exceeds six times that of the average dry-weather flow.

There should be no overflow for untreated sewage or stormwater at or near the disposal works.

Screens.

All liquid delivered at the disposal works should as a rule be passed through a screening chamber.

Storm-water Treatment.

A weir set at three times the dry-weather flow should be placed *below the screens*, and any volume passing over this weir should be dealt with in storm tanks. The tanks should be two or more in number and their total capacity should not be less than a quarter of the dry-weather flow. They should be so arranged that when they are full they will act as 'continuous flow' tanks, and that they can be readily emptied and kept empty when no liquid is passing into them.

The liquor from the tanks can be discharged without further treatment except in special cases, and the sludge should be dealt with by any of the usual methods which may be most suitable in the particular circumstances.

In cases where a sufficient area of suitable land is available for the purpose, detritus tanks followed by irrigation may be substituted for the storm tanks.

Sewage Treatment.

Detritus Tanks.—There should be two or more detritus tanks below the screening chamber. The capacity of each tank should be about $\frac{1}{100}$th of the dry-weather flow.

Septic Tanks.—Septic tanks should not be less than two in number, and their total capacity should be about equal to the dry-weather flow.

Chemical Preparation Tanks.—For *quiescent* treatment there should not be less than eight tanks, each of which should have a capacity equal to about two hours' dry-weather flow.

For *continuous flow* treatment there should not be less than two tanks, with a total capacity of at least eight hours' dry-weather flow, and in most cases a greater number of tanks will be desirable.

Settling Tanks.—For *quiescent* treatment—same as chemical precipitation tanks.

For *continuous flow* treatment there should not be less than two tanks with a total capacity of from ten to fifteen hours' dry-weather flow.

Filters.—In determining the sizes of percolating filters and contact beds the Ministry have, in order to allow for the strength of the sewage to be treated, adopted the division into 'strong,' 'average,' and 'weak' sewages recommended by the Royal Commission.

The strength of sewage should, when possible, be ascertained by analysing average samples of crude sewage taken in dry weather at frequent and regular intervals throughout seven days, and in proportion to the flow. Possibly, in the case of small works, the period for taking samples for analysis might be somewhat shorter, but the period should not be less than forty-eight hours, and Saturday and Sunday should be avoided. In every case, the daily rainfall during the period when the samples were being taken and during the preceding days should be ascertained.

The analysis should in all cases include the following items :—

In parts per 100,000 *by weight.*

(1) Ammoniacal nitrogen.
(2) Albuminoid nitrogen.
(3) Total nitrogen.
(4) Oxygen absorbed from strong permanganate in three minutes at 80 degrees Fahr.
(5) Oxygen absorbed from strong permanganate in four hours at 80 degrees Fahr.
(6) Suspended solids.
(7) Soluble solids.
(8) Chlorine.

It is also desirable that the amount of dissolved oxygen taken up during the oxidation of the ammoniacal and organic matter of the sewage should be given.

The results of analysis would of course require to be properly interpreted ; but as a rough guide it may be taken that from the figure for 'Oxygen absorbed from strong permanganate in four hours at 80 degrees Fahr.' the strength of the sewage may be very roughly classified as follows :—

'Strong' sewage	17 to 25 parts per 100,000
'Average' „	10 to 12 „ „
'Weak' „	7 to 8 „ „

In cases where the sewage cannot be analysed its strength should be estimated according to the water consumption, the flow per head, the kind of sewage system, whether water-closets are in general use, the volume and nature of trade waste, the amount of dilution by subsoil or surface water, etc.

Failing satisfactory evidence to the contrary it will be desirable to assume that the sewage is 'strong,' for the purpose of estimating the required capacity for the disposal works.

For the purpose of showing the *minimum* total cubic contents of filters required for treating three times the dry-weather flow in different cases, the tables A and B have been prepared on the data given in the Commission's Report.

The rates of filtration given by the Commission are rates which can generally be *doubled* in wet weather. Where it is proposed to deal with *three* times the dry-weather flow, the Commission say that it would generally only be necessary to provide $1\frac{1}{2}$ times the capacity of filter required for the dry-weather flow. This basis has been adopted in arriving at the figure in the Tables A and B (page 1375).

TABLE A.—CONSTANTS FOR CALCULATING THE MINIMUM CUBICAL CONTENTS OF PERCOLATING FILTERS.

Preliminary Treatment.	Strong Sewage.		Sewage of Average Strength.		Weak Sewage.	
	Coarse or Medium Material.	Fine Material.	Coarse or Medium Material.	Fine Material.	Coarse or Medium Material.	Fine Material.
Detritus tanks . . .	15	*	25	*	40	*
Septic tanks . . .	45 †	*	70	*	100	100
Settlement tanks (continuous flow) . .	45 †	*	70	*	100	100
Settlement tanks (quiescent) + ...	50 †	25	100	70	130	130
Precipitation tanks (continuous flow) . .	65	50	100	80	150	175
Precipitation tanks (quiescent)	100	65	130	130	170	200

Notes as to Filtering Material.

(*a*) A filter may be regarded as *coarse* if the material will not pass through a 1-in. sieve; as *medium* if it will pass through a 1-in. but not through a ½-in. sieve; and as *fine* if it will pass through a ½-inch sieve.

(*b*) *Coarse* material will be desirable in all cases where the liquid to be treated contains much suspended matter.

(*c*) In the cases marked * the use of fine material would not be desirable unless the circumstances were exceptional.

† If *medium* sized material were used in these cases the figures should be reduced by about 10.

TABLE B.—CONSTANTS FOR CALCULATING THE MINIMUM CUBICAL CONTENTS OF CONTACT BEDS.

Preliminary treatment.	Strong Sewage.			Sewage of Average Strength.			Weak Sewage.		
	Single Contact.	Double Contact.	Triple Contact.	Single Contact.	Double Contact.	Triple Contact.	Single Contact.	Double Contact.	Triple Contact.
Detritus tanks	—	—	25 *	—	25	†	—	38	†
Septic tanks	—	—	33	—	38	†	75	66 ‡	†
Settlement tanks (continuous flow)	—	—	33	—	38	†	75 *	66 ‡	†
Settlement tanks (quiescent) .	—	—	44	—	50	†	100	†	†
Precipitation tanks (continuous flow) . . .	—	33	†	—	50	†	133 ‡	†	†
Precipitation tanks (quiescent)	—	43	†	—	66 *	†	133 ‡	†	†

Notes to Table B.

(*a*) The beds should not be less than 2 feet 6 inches nor more than 6 feet in depth.

(*b*) The different series of beds, in double and triple contacts, should have an equal cubic contents, failing any evidence to the contrary.

(*c*) Where a blank is left in the table the particular treatment indicated would only be desirable in exceptional circumstances, as the method would not generally be economical.

(*d*) In the cases marked thus †, the particular treatment indicated would only be necessary in exceptional circumstances (e.g. where an unusually good effluent is required).

(*e*) Where three times the d.w.f. has to be dealt with in wet weather, the individual beds, in the cases marked thus ‡, should be small (to facilitate rapid filling and emptying), or the total cubic contents of the filters should be increased.

(*f*) The average liquid capacity of a cubic yard of contact bed is about 50 gallons.

* There are no data in the Commission's Report with regard to these cases.

In order to ascertain the minimum total cubic contents of the filters required, divide the dry-weather flow by the *appropriate figure* (having regard to the strength of the sewage and the kind of treatment proposed), in the tables above, and the result will be the number of cubic yards which the filters should contain for treating up to three times dry-weather flow.

In cases where a sufficient area of suitable land is used for irrigating the filtrate it will probably suffice if the total cubic contents of the filters is half that provided for by the tables, but where the filter capacity is so reduced it may be necessary in wet weather to restrict the flow to the filters to about 1½ times the dry-weather flow and to pass the remainder of tank liquor direct to the land.

Effluent Tanks or Filters.—Where the effluent from filters is not irrigated on land, tanks (with a capacity equal to about two hours' dry-weather flow and with provision for removing the sludge), or shallow straining filters, will in most cases be necessary for the purpose of preventing the suspended matters in the filtrate from passing into the river.

When the sewage, after previous sedimentation, is applied to land filtration the quantity per acre of the most suitable land shall not exceed 15,000 gallons of dry-weather flow per acre per day or the sewage from 500 persons, and surplus land up to 25 per cent. is usually required to enable the land to be rested. This area will be increased, if the land be not of the first quality.

When the sewage, after previous sedimentation, is applied to land by broad irrigation, the quantity is not allowed to exceed 4,500 gallons of dry-weather flow per acre per day, or the dry-weather sewage from 150 persons. Surplus land must also be provided in such cases from 25 per cent. to 50 per cent. for resting purposes. Where the land is less suitable, the quantity per acre may be reduced to as little as 3,000 gallons per day.

When sewage is treated by precipitation or bacterial processes and the final purification is effected by land, the tank or filter effluent may be applied at the rate of 30,000 gallons per acre per day or the sewage from 1,000 persons as a maximum on the best land.

In all cases where land of sufficient area and of a not unsuitable character can be obtained at a reasonable cost the final purification must be by land.

Activated Sludge Process.

(*Communicated by Activated Sludge, Ltd.*)

The latest development of the sewage problem is that known as the 'Activated Sludge Process,' by which purification is carried out by oxidation of the organic matter present in the sewage by means of air blown through the liquid in the presence of an accumulation of sludge.

Experiments carried out at Manchester during 1913–14 demonstrated that by continually blowing air through successive quantities of sewage, decanting the liquid, and retaining the sludge, the latter became charged with micro-organisms which had the property of increasing enormously the purification effected by the simple aeration of the sewage. The deposit obtained in this manner has now become known as 'Activated Sludge.'

The addition of activated sludge to sewage, when combined with further blowing of fine bubbles of air through the mixture in such a manner as to create and maintain a circulation of air, sludge and liquid, sufficient to prevent settlement of the sludge and keep the sludge and air bubbles equally diffused throughout the liquid, produces a very marked change in the composition of the sewage; first a clotting or clarifying action takes place, then a rapid carbon oxidation process and finally nitrification.

Upon standing, the sludge rapidly settles and separates from the liquid, which when drawn off is not liable to putrefaction and is practically free from bacteria, whilst the sludge is also rendered innocuous and is found to contain a large proportion of nitrogen in a very available form for fertilising purposes.

It will be seen from the following table, which gives the composition of the sludge, that it has a large nitrogen content, so that its value as a fertiliser is extremely high; it contains, in fact three to four times as much nitrogen as the average septic tank sludge:—

Organic and volatile matter . . .	64·70–70·40 per cent.
Minerals	35·30–29·60 ,,
Nitrogen	3·75– 7·09 ,,
Phosphoric acid	2·10– 4·20 ,,

One of the main difficulties has been to design suitable tanks, and the equipment for same, so that a regular circulation is created with a minimum expenditure of power in such a manner as to produce an equal diffusion of air and the biological sludge throughout the whole body of the

sewage in the tanks, without the creation of subsidiary eddies or deposits of sludge in pockets. This has been successfully overcome in the designs and appliances supplied by *Activated Sludge, Ltd.*, 100 *Victoria Street, Westminster, London, S.W.* 1.

A further difficulty originally was in obtaining a satisfactory settlement with a continuous flow, but, due to the researches of Mr. W. Clifford, Assoc.M.Inst.C.E., of Wolverhampton, this problem was solved by a form of tank-inlet which destroys the energy of the inlet current and eddies.

The Activated Sludge Process has many advantages; the chief of these are as follows :—

(1) The purification is rapid and perfect, and the effluent fully oxidised.

(2) Putrefaction stops at once, and the system is free from smells and flies.

(3) The plant can be installed close to buildings, so that long outfall sewers are often unnecessary.

(4) An inoffensive sludge which is a valuable manure is obtained.

(5) Only a small area of land is required.

(6) There is practically no loss of head (4 ins. to 10 ins.), and the sludge is raised automatically.

(7) Trade wastes which are alkaline, or of constant but low acidity, give no difficulty.

(8) A very small staff is needed, but it requires at least one skilled attendant.

(9) Capital costs are low owing to the simplicity of the plant, the small tank area needed (less than for sedimentation or septic tanks), and the absence of filter beds. Further, by installing the plant near a town, there may be a saving in the cost of outfall sewers.

Taken all round, the cost of installing and maintaining a plant is generally about two-thirds of the cost of septic tanks and filter beds. It should be mentioned that existing tanks can usually be converted, and in many places this has been done quite successfully at small cost.

The plants work well either with a large or small population, and would appear to be well suited for military stations. There are no real disadvantages in the system except possibly the cost of power required for the intensive aeration, which is essential for the work of the process.

A great deal of attention has been devoted to the evolution of a compressor which will give a higher efficiency for the 'diffused air' system. After much thought and experiment a compressor has now been designed which effects a great saving in h.p., and this can be seen in the latest plants.

In addition to the above a great deal of thought has also been given to the production of clean air, and this problem has been solved recently.

In many places compressed air is required for raising the sewage by means of ejectors, in which case no further compressing plant is needed.

The amount of purification of the liquid sewage depends upon the period of aeration allowed, and hence with tanks of sufficient capacity it is capable of dealing with very foul sewage which would be difficult of treatment by other processes.

The entire patents for the process are held by *Activated Sludge, Ltd.*, 100 *Victoria Street, Westminster.*

Over sixty-one permanent activated sludge plant equipments for treating either sewage or trades waste have been installed by *Activated Sludge, Ltd.*, including a few now near completion. These are situated in the following countries: Great Britain, Canada, South Africa, India, Denmark, Holland, France, and Finland.

With these plants in full operation they will be treating 30,000,000 gallons of dry weather sewage, and will serve an aggregate population of 600,000.

In addition to the foregoing the Manchester Corporation have adopted this method for treating half the flow of sewage, and the scheme has been approved of by the Ministry of Health. When the plant is in full operation it will serve a population of over 300,000.

Though the 'diffused air' method has been adopted in the U.S.A. almost exclusively, these plants, which are over twenty in number, have not been included in the above, yet the four plants at Houston, Texas, Milwaukee, Indianapolis, and Chicago will treat an aggregate D.W.F. of 329,000,000 gallons per day and serve a population of 1,745,000.

As further plants have been installed and more and more data have become available from various installations, it has become possible to make comparative tests between 'ridge and furrow' and 'spiral flow' types of tanks. A definite attempt has been made to ascertain the relative values of these two forms of design in reference to weak and strong sewages with and without trades waste. Much valuable information has resulted from the many trials.

In addition, experiments have been carried out in reference to short-circuiting and the effect of increased interfacial contact on the effective capacity of these plants. These experiments have greatly added to the detailed knowledge of the requirements of the process.

The single-unit type is generally adopted for populations up to 1,000, and the double-unit type for populations up to 6,000.

A multi-unit type of not less than 4 units is used for any populations above 6,000.

The Estuary type can be used for any population, irrespective of size, and is arranged for an aeration period of a few minutes only, a high degree of purification not being required with an outfall into large rivers or estuaries, or it can be used for preliminary treatment before final treatment on filter beds.

A general description of the above types can be found in a paper read by Mr. J. A. Coombs, A.M.Inst.M.E., A.M.I.S.E., before the Institute of Sanitary Engineers in 1923.

Standard of Purity for Sewage Effluents.

The Royal Commission on Sewage Disposal appointed in 1898 put forward the following provisional standard of purity for sewage effluents for the guidance of Local Authorities.

1. That it should not contain more than three parts per 100,000 of suspended matter; and

2. That after being filtered through filter paper it should not absorb more than

(*a*) 0·5 part by weight per 100,000 of dissolved or atmospheric oxygen in 24 hours.
(*b*) 1·0 „ „ „ „ „ 48 hours; or
(*c*) 1·5 „ „ „ „ „ 5 days.

In 1913 the Royal Commission issued their 8th Report, in which they summarised their conclusions as follows:

(*a*) The law should be altered so that a person discharging sewage matter into a stream shall not be deemed to have committed an offence under the Rivers Pollution Prevention Act, 1876, if the sewage matter is discharged in a form which satisfies the requirements of the prescribed standard.

(*b*) The standard should be either the general standard or a special standard which will be higher or lower than the general standard as local circumstances require or permit.

(*c*) An effluent in order to comply with the general standard must not contain as discharged more than 3 parts per 100,000 of suspended matter, and with its suspended matters included must not take up at 65° F. (18·3° C.) more than 2·0 parts per 100,000 of dissolved oxygen in five days. This general standard should be prescribed either by Statute or by Order of the Central Authority, and should be subject to modifications by that Authority after an interval of not less than ten years.

(*d*) In fixing any special standard the dilution afforded by the stream is the chief factor to be considered. If the dilution is very low it may be necessary for the Central Authority, either of their own initiative or on application by the Rivers Board, to prescribe a specially stringent standard, which should also remain in force for a period of not less than ten years.

(*e*) If the dilution is very great the standard may, with the approval of the Central Authority, be relaxed or suspended altogether. Our experience leads us to think that as a general rule, if the dilution, while not falling below 150 volumes, does not exceed 300, the dissolved oxygen absorption test may be omitted and the standard for suspended solids fixed at 6 parts per 100,000. To comply with this test no treatment beyond chemical precipitation would ordinarily be needed.

If the dilution while not falling below 300 volumes does not exceed 500 the standard for suspended solids may be further relaxed to 15 parts per 100,000. For this purpose tank treatment without chemicals would generally suffice if the tanks were properly worked and regularly cleansed. These relaxed standards should be subject to revision at periods to be fixed by the Central Authority, and the periods should be shorter than those prescribed for the general or for the more stringent standards.

(*f*) With a dilution of over 500 volumes all tests might be dispensed with, and crude sewage discharged, subject to such conditions as to the provision of screens or detritus tanks as might appear necessary to the Central Authority.

EDITORIAL NOTE.—The standards mentioned in (*e*) and (*f*) are not intended to apply to effluents discharged into streams which are used for drinking water or to streams the water of which, at the point of discharge of the sewage effluent, normally takes up more than 0·2 parts per 100,000 of dissolved oxygen in five days at 60° F. Up to the present time the proposals contained in the Report have not been embodied in any Act of Parliament, and have therefore no official authority, and must be treated as suggestions only.

HOUSE DRAINAGE.

House drains are usually made of salt glazed ware pipes varying from 3 ins. to 6 ins. diameter. Drains taking the discharge of water-closets should not be less than 4 ins. diameter.

Wherever possible all house drains should be laid outside the building; they should be laid as far as practicable in straight lines with easy bends, furnished with inspection chambers, and should be arranged so that they may be easily got at in case of necessity.

When it becomes imperative that drains should pass under the buildings, cast-iron pipes should be used and they should be laid in a straight line between two inspection chambers outside the building.

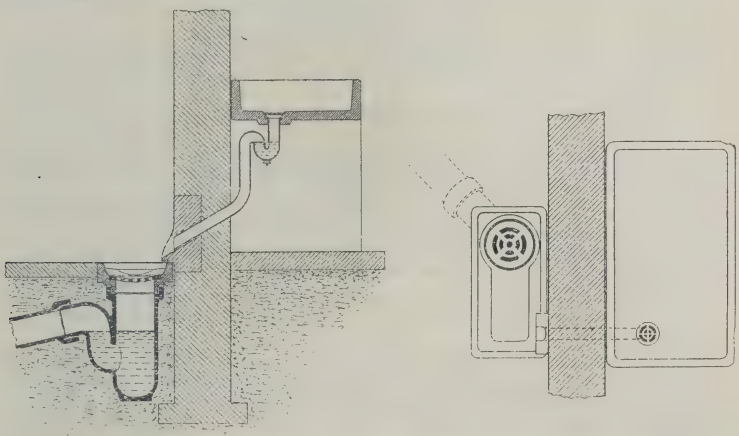


FIG. 3.—Section and Plan of Scullery Sink, with connection to Gully.

The size of drain-pipes should be sufficient to carry off sewage at the rate of forty gallons per head of the maximum population drained in six hours, and in addition rainfall at the rate of ½ in. per hour upon the area of roofs and paved yard surfaces draining to the sewers.

It was formerly considered essential that all private drains should be trapped from the sewer at a point as near as possible to the junction with the sewer, and that an inspection chamber should be built at the trap and should be provided with a fresh-air inlet.

The Local Government Board appointed a Departmental Committee to advise upon the desirability of fixing these intercepting traps near the sewers. In 1912 the Committee made their report, and stated that the case for the necessity of the intercepting trap had broken down, and that it is not only unnecessary but in many cases is harmful. It is probable, therefore, that there will be an early modification of building regulations in most towns, deleting the byelaws requiring intercepting traps between sewer and drain.

The recommendation of the Committee must not be understood to imply the uselessness of traps other than the main intercepting trap : it is just as important as ever that all inlets to drains should be trapped between the drain and the house.

The general scheme of drain trapping should be based upon the following lines :—The main drain to each house or building should be carried from the sewer without a trap or air inlet to the furthest point where a branch connection is to be made, and there a ventilating shaft should be erected and carried up well above the eaves of the house and clear of all windows, dormers, skylights, and chimneys. The end of the pipe should be fitted with a wire guard ; most cowls are useless.

Branch drains for sink wastes, bath wastes, and rain-water pipes should terminate with trapped glazed stoneware gullies placed at ground-level and paved round with a containing curb of blue brick on edge on three sides and a short paved channel at least 18 ins. long on the

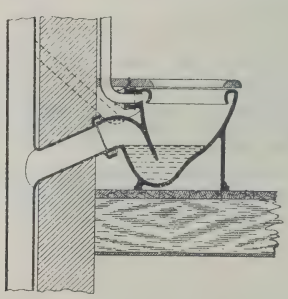


FIG. 4.—Section of Water-closet.

other on to which the waste etc. pipes should discharge. Bath wastes etc. from upper floors should be made to discharge in the open air into the cistern heads of down-pipes which are themselves cut off from the gully as above described.

Branch drains for soil pipes should be untrapped at the foot of soil pipes, and the latter should be carried up outside the external walls of the building in a straight line to above the eaves to form a ventilating column, the same precautions being adopted as in the case of the ventilation column for the end of the drain. Where there is only one soil-pipe connection it can often be arranged at the end of the main drain and so avoid the provision of more than one ventilation shaft.

When it is found inevitable that soil-pipes should be fixed inside buildings they should be of drawn lead throughout their entire length and should be carried up through the roof to give thorough ventilation.

Sinks and water-closets should, whenever practicable, be placed against external walls so that they may discharge at once into a drain outside the main wall, as shown in figs. 3 and 4.

Water-closets are best placed where they can be thoroughly ventilated directly from the external air with windows extending to the ceilings, and they should be arranged to be entered from a well-ventilated ante-room or passage with doors to shut off communication with the other portions of the house.

LONDON COUNTY COUNCIL REGULATIONS FOR WATER-CLOSETS (*Abridged*).

Building (for closet) must have one external wall abutting on a space 100 ft. square super (or if in a basement approached directly from an opening at least 5 ft. wide and 40 ft. square super) and must not be directly approached from a room used for habitation, work, storage, or preparation for food. Walls abutting on such rooms must be of brick from floor to ceiling.

A window opening directly into the external air at least 2 ft. square, and in addition at least one ventilating air brick, air shaft, or other effectual means of ventilation must be provided.

Floor of closets approached from the external air to be of hard impervious material, having a fall to the door of half inch to the foot.

Door with proper fastening to be provided.

Cisterns giving an adequate and effectual flush must be fitted to all closets. Closet cisterns must be entirely separate from cisterns containing drinking water. A proper supply of water to the cistern must be supplied. No service pipe may be allowed to be connected in any way with any part of the closet apparatus except the cistern. Downpipe and union between closet and cistern must have internal diameter of not less than 1½ in.

Closet Basin and Trap.—The basin must be of non-absorbent material of such shape that a sufficient quantity of water is held in such a manner that the soil falls direct into the water clear of the sides.

No container to be fitted under basin.

An efficient syphon trap must be fitted immediately behind or in connection with the basin. The trap must be constructed to always provide a sufficient water-seal between the basin and the drain. Traps must be ventilated where more than one closet is connected with the same soil pipe. Ventilating pipe must not be less than 2 ins. internally, and connect with the arm of soil pipe not less than 3 ins. nor more than 12 ins. from the highest part of the trap. Ventilation must be on the drain side of the water-seal, and the ventilation pipe must be carried either to the open air at a point as high as the top of the soil pipe or to the soil pipe above the highest closet.

No D trap or similar trap must be fitted.

Soil pipe must be outside buildings wherever practicable. Soil pipes inside buildings must be drawn head with wiped joints, or heavy cast iron with molten lead joints, properly caulked. Pipe to be continued without diminution and as straight as possible to a height to allow of safe outlet by open end for foul air. Height must be above highest part of roof of building, and where practicable 3 ft. above any window that lies within 20 ft. measured in a straight line from open end. Wire guard to be fitted to open end, the area of the opening in which must be equal to not less than size of pipe.

Soil pipe must not directly connect with waste of bath, rain-water pipe, or to a sink, and soil pipe must have no trap in it nor between it and the drain.

Joints.

Lead Pipe to Iron.—A brass thimble to be used—the lead wiped to open end and the other connected with the iron by a joint of molten lead, properly caulked.

Stoneware to Lead.—Brass socket or thimble wiped to lead and connected to stoneware with Portland cement.

Stoneware to Iron.—Stoneware to be inserted into socket in iron and connected with joint made with Portland cement.

Dimensions and Weights of Soil Pipes.

Internal diameter must not be less than 3½ inches.

Diameter.	Lead. Weight per 10 feet Length not less than	Iron. Weight per 6 feet Length not less than
3½ inches.	65 lbs.	48 lbs.
4 ,,	74 ,,	54 ,,
5 ,,	92 ,,	69 ,,
6 ,,	110 ,,	84 ,,

General Notes.

Water-closet basins are all made with traps attached, and this is sufficient protection against sewer air coming into the house from the soil pipe. To ensure proper ventilation of the branch pipe from the w.c. trap to the soil pipe and to prevent unsealing by syphonic action where closets are placed over one another and are connected to the same soil pipe, a small auxiliary ventilation pipe should be connected near the trap and be carried upwards at an angle of about 45° into the soil pipe, as shown in fig. 4.

Water-closet basins should be so designed as to present as large a surface of water as possible into which the excreta may fall when the closet is in use; they should have flushing appliances which effectually wash the whole of the internal surface of the basin with the minimum of water and they should have all parts easily accessible for cleansing by hand. Closets with a syphonic action are most effectual and much to be preferred to any type of valve closets. The pedestal type of closet with lift-up lid so that it can be used as a urinal is the one most in favour.

Water-closets should be fitted with waste-preventing water-cisterns of not less than 2-gall. capacity, and a 3-gall. cistern if allowed by the local waterworks regulations is preferable.

Flush pipes should be of lead or copper and should be 2 ins. diameter with a head of 6 ft. to give efficient flushing.

Urinals should not be provided at all inside private houses, and whenever they are fixed inside a building they should be made throughout of glazed stoneware with curved surfaces and as few joints as possible so as to avoid any angles in which the urine can lodge. They should be fitted with lead or copper sparge pipes capable of thoroughly washing all the surfaces liable to be soiled with urine.

Slate urinals may with advantage be painted with a mixture of common coal tar and naphtha; they should be thoroughly washed out at least once a day. Slate is not a material to be recommended for the construction of urinals; all stalls and basins should be of glazed stoneware.

The usual allowance of water for each stall in a public urinal is half a gallon per minute. The waste pipes for each stall should be 2 ins. in diameter.

White glazed bricks are preferable for facing the walls of public urinals; ordinary brick walls cemented over and painted are decidedly objectionable. The joints in the brickwork must not be thick under any circumstances, as they would present an absorbent surface.

Baths and housemaids' sinks should be of glazed porcelain fitted with trapped waste pipes which should be carried through the nearest external wall and be made to discharge in the open air over the cistern head of a downpipe: the top of the cistern head should be covered with a copper wire guard to keep out the birds, dead leaves, etc.

Kitchen and pantry sinks should be of glazed stoneware and should be fixed against an external wall, and their trapped lead waste-pipes should be carried through the wall and be made to discharge in the open air upon a paved channel leading to a trapped gully.

SECTION XXVI

PART II

IRRIGATION — AVAILABLE SUPPLY — EVAPORATION — ABSORPTION—EVAPORATION FROM SOILS—LOSSES FROM IRRIGATION CANALS — MEAN VELOCITIES — CRITICAL VELOCITY—DUTY OF WATER.

Contributed by Reginald Ryves, Assoc.M.Inst.C.E., M.Cons.E.

(Author of 'A Note on Mass Dams and Dams of Arches,' etc.)

IRRIGATION.

Available Supply of Water.

Irrigation from Large Rivers.—The discharges of the river during the seasons when the water will be required are calculated from the observed depths over weirs, from differences of water level above and below bridge openings, and by the methods hereafter to be described under 'Capacities of Canals.' Whether these supplies will be fully available or not may depend upon the water levels which can be maintained above the weir, and upon what cultivable areas are commanded by the water at these levels. In this respect the scheme must be considered as a whole, and by making suitable regulations as to the crops which may be grown on certain areas and at certain seasons the availability of the supply may be increased.

Supplies from Defined Catchments.—In the case of a supply from a defined catchment the estimates of run-off may be made by the methods described under 'Rainfall,' the necessary deductions being made when the whole of the water stored cannot be directly utilised. Out of the whole recorded range of annual or seasonal rainfalls, and corresponding runs-off, one will be chosen as that suitable as a basis for the project, and the capacity of the reservoir designed accordingly. A table drawn up for catchment areas in Nagpur, Central Provinces, India (*Strange*), gives estimated yields for monsoon rainfalls ranging from 1 in. to 60 ins., and from this table the following examples are taken.

MONSOON RAINFALLS AND ESTIMATED RUNS-OFF FOR NAGPUR.

Monsoon Rainfall in Inches.		1	5	10	20	30	40	50	60
Runs off in millions of cubic feet per square mile	Good catchment	0·002	0·116	0·999	6·970	18·33	34·85	56·69	83·63
	Aver. catchment	0·001	0·087	0·749	5·227	13·75	26·13	42·51	62·73
	Bad catchment .	0·001	0·058	0·499	3·485	9·17	17·42	28·34	41·87

Evaporation from a Water Surface.

The actual rate of evaporation from the surface of a pool of water depends mainly on four factors: the temperature of the water itself; the velocity of the wind; the degree of humidity of the air; and the temperature of the air. As regards the first, this is more or less proportional to the air temperatures if we are measuring evaporation over considerable periods of time in different places. As regards the second, the velocity of the wind, this factor does not as a rule vary so much as others if we take a period such as a year; but it sometimes varies a great deal within areas of moderate extent, and it should be taken into consideration in making estimates of evaporation. The degree of humidity varies very greatly, not only in different countries or different districts, but also in proportion to the distance from the sea and, in some cases, with

elevation. There may be a marked decrease in the humidity of the air in a distance of a few miles from a long, straight coast line, the decrease being more gradual as we pass farther inland. There are also sharp differences between places just within and those just without the path of a monsoon current. The relation between water temperature and the rate of evaporation was studied at four stations in California (*Fortier*) with the following results, the evaporation being in sixty-fourths of an inch per day. The tanks used were of galvanised iron, 22 ins. in diameter and 28 ins. deep sunk in the ground.

Water Temperature and Evaporation.—53°F., 6; 62°F., 12; 73°F., 23; 80°F., 31; 88°F., 39. The following were the results of experiments to test the rates of evaporation in air currents of different velocities, that in still air being taken as unity. (*Russell.*)

Wind Velocity and Evaporation.—Velocity 0, evaporation 1·00; velocity 5 miles an hour, evaporation 2·2; velocity 15, evaporation 4·9; velocity 20, evaporation 5·7; velocity 25, evaporation 6·1; velocity 30, evaporation 6·3. Observations taken at the Assuan reservoir show the relative importance of average month's temperatures and average month's wind velocities. The results do not, of course, aid in the establishment of a relation between actual wind velocities, actual temperatures, and rates of evaporation, but they are of practical use for purposes of comparison.

AVERAGE VELOCITIES OF WIND, AVERAGE TEMPERATURES, AND RATES OF EVAPORATION, ASSUAN, EGYPT. LATITUDE, 24°. YEAR 1912.

Month .	Jan.	Feb.	Mar.	Apr.	May	June	July	Aug.	Sept.	Oct.	Nov.	Dec.
Wind velocity, miles per hour .	3·6	4·2	6·6	5·3	4·2	3·0	3·3	3·5	1·2	1·7	3·4	3·7
Temperature °F. .	60·7°	66·7°	72·3°	80·7°	87·2°	91·4°	91·2°	92·1°	89·3°	83·3°	73·6°	62·2°
Evaporation, ins. per day	0·12	0·17	0·24	0·31	0·35	0·45	0·43	0·44	0·32	0·29	0·21	0·15

Humidity and Evaporation.

By taking a sufficient number of observations of rates of evaporation, it is possible to establish at the same time the relation between evaporation and air temperature and that between evaporation and humidity. Experiments of this kind with water at a constant temperature seem to be lacking, nor have observers, as a rule, noted the actual temperatures of the water. Results are, however, comparable in groups, such as those with small experimental tanks, and those with floating tanks in deep and shallow waters, respectively.

The following table gives the results of observations at Vehar reservoir, near Bombay, latitude 19° (*Conybeare*), and in Nagpur, 21° (*Binnie*). The values seem to be low, but this hardly affects their application in practice, as the engineer would draw the curves and fix his own zero line on the result of monthly or annual evaporations for the locality.

APPROXIMATE MONTHLY EVAPORATIONS IN FEET WITH GIVEN TEMPERATURES AND RELATIVE HUMIDITIES.

Percentage of Saturation.	Mean Temperatures, Fahr.						
	65°	70°	75°	80°	85°	90°	93°
88	0·14	0·17	0·19	0·21	0·24	0·27	0·30
85	0·19	0·22	0·22	0·23	0·27	0·30	0·33
80	0·22	0·27	0·29	0·30	0·32	0·36	0·38
75	0·24	0·29	0·33	0·35	0·37	0·40	0·42
70	0·26	0.30	0·34	0·35	0·38	0·42	0·44
65	0·28	0.31	0·34	0·36	0·40	0·44	0·46
60	0·30	0·35	0·38	0·40	0·42	0·47	0·49
55	0·32	0·37	0·42	0·46	0·50	0·52	0·54
50	0·35	0·40	0·45	0·50	0·55	0·60	0·62
45	0·45	0·50	0·57	0·67	0·71	0·75	0·78
40	0·61	0·66	0·70	0·72	0·77	0·82	0·84

The figures in the following table are taken from results of observations by the Egyptian Public Works Survey Department (*Cay*). It is clear that the rate of evaporation depends considerably upon the mean altitude of the sun, since, for air temperatures which are nearly the same, the rate of evaporation is affected by the latitude and the month. This is shown by the small tables, and the evaporation at latitude 24° in October with air temperature 83° and humidity 41 may be compared with that at latitude 27° in August with nearly the same humidity.

OBSERVATIONS BY EGYPTIAN PUBLIC WORKS DEPARTMENT.

Locality . . .	Abassia, Cairo. Lat., 30° 1′.					
Month . . .	May.	June.	July.	Aug.	Sept.	Oct.
Mean temperature of month, Fahr. .	75°	78°	81°	80°	77°	73°
Relative humidity .	47	50	55	62	70	71
Evaporation, inches .	·32	·25	·20	·22	·17	·18
Locality . . .	Asyut. Lat., 27° 11′.					
Month . . .	May.	June.	July.	Aug.	Sept.	Oct.
Mean temperature of month, Fahr. .	81°	84°	—	85°	81°	76°
Relative humidity .	26	26	—	40	46	56
Evaporation, inches	·55	·61	—	·51	·37	·27
Locality . . .	Assuan. Lat., 24° 2′.					
Month . . .	May.	June.	July.	Aug.	Sept.	Oct.
Mean temperature of month, Fahr. .	90°	90°	90°	90°	89°	83°
Relative humidity .	16	20	20	20	32	41
Evaporation, inches	·62	·60	·52	·52	·47	·35

Temperature 80°–81°F.				Temperature 89°–90°F.			
Humidity.	Evap.	Month.	Lat.	Humidity.	Evap.	Month.	Lat.
62	·22	Aug.	30°	32	·47	Sept.	24°
55	·20	July	30°	20	·60	June	24°
46	·37	Sept.	27°	20	·52	July & Aug.	24°
26	·55	May	27°	16	·62	May	24°

The effect of altitude upon evaporation was tested at four stations on the same mountain in California with the following results :—

Evaporation at Different Altitudes.—4,500 ft., 2·69 ins. ; 7,100 ft., 2·04 ins. ; 10,000 ft., 1·63 in. ; 12,000 ft., 1·61 in.

Annual Evaporation.

At stations selected as representing the chief irrigation districts of California the following were evaporations for a year (*Fortier*). At Berkeley, the cooler coast area with relatively high rainfall, 41 ins. ; at Chico, the bottom of the Sacramento Valley, 53 ins. ; at Pomona, representing Southern California, 66 ins. ; at Tulare, the San Joachin Valley, 68 ins. ; at Calexico, the Imperial Valley, 71 ins.

The evaporation at a number of representative stations is given in the following table, but the methods adopted in measuring evaporation varied considerably. At Croydon, the evaporation from a 5-in. vessel in air was more than 50 per cent. greater than that from the 12-in. vessel floating in water.

EVAPORATION AT REPRESENTATIVE STATIONS.

Locality.	Approximate Latitude.	Evaporation in Inches.	Remarks.
Whitehaven, England	54½°	29·91	Elevation 90 ft. near coast.
Bolton-le-Moors, England	53½°	25·65	Average of 10 years. Elevation 320 ft.
Croydon, England	51½°	19·95	13 years, max. 22·85, min. 15·01. A 12-in. vessel floating in a tank 4 ft. in diameter.
Kennich, Devonshire, England	50½°	20·88	Elevation 836 ft. Average of 4 years.
Dijon, France	47½°	26·11	Inland station.
Boston, U.S.A.	42½°	39·11	
Croton River, U.S.A.	41½°	24·15	
New York, U.S.A.	41°	39·21	
Tharsis Mines, Spain	37½°	60	Average of 7 years, max. nearly 65 ins.
Chicco, California	39½°	53	Galvanised iron tanks, 2 to 3 ft. in diameter, and 30 to 36 ins. deep; sunk in the earth.
Berkeley ,,	38°	41	
Tulare, ,,	36°	68	
Pomona, ,,	34°	66	
Calexico, ,,	—	71	
A station in California	—	30	Elevation 6,225 ft.
Perth, Western Australia	32°	65·8	
Rajputana, India	25° to 28°	73·8	General average.
The Rand, South Africa	26°	63	
Tulsi Lake, Bombay Presidency	19°	36·4	Floating boxes. Average of 8 years.
Red Hills, Madras, India	13°	66·9	Near the coast.

Seasonal and Monthly Losses by Evaporation.

It is often necessary to be able to estimate the losses by evaporation during particular seasons, and monthly rates of evaporation, or daily rates in individual months, are frequently recorded. For considerable areas in India the seasonal losses may roughly be reckoned as: In the cold weather 12 ins., in the hot weather 32 ins., during the monsoon 16 ins.; total, 60 ins.

In the following table, some of the rates of evaporation are comparable, but the table has been drawn up chiefly in order to show the relative rates of evaporation in different months at the same station.

MONTHLY EVAPORATIONS FROM WATER SURFACES. (Inches.)

Locality.	Approx. Lat.	Month.					
		Jan.	Feb.	Mar.	April.	May.	June.
Emdrup, Denmark (av. of 11 years)	56°	0·7	0·5	0·9	2·0	3·9	5·3
Bolton-le-Moors, England, 320 ft. (av. of 10 years)	53½°	0·64	0·95	1·59	2·59	4·38	3·84
Whitehaven, England, 90 ft.	54½°	0·95	1·01	1·77	2·71	4·11	4·25
Tulare, California	36°	1·5	2·7	3·3	3·8	7·2	10·5
Perth, Australia	32°	2·06 (July)	1·75 (Aug.)	3·20 (Sept.)	5·51 (Oct.)	7·69 (Nov.)	9·47 (Dec.)
Cairo, Egypt	30°	—	—	—	—	9·88	7·56
Assuan, Egypt	24°	—	—	—	—	19·16	17·96
Pashan, Poona, India	18°	5·27	3·92	5·27	8·10	11·78	*
Red Hills, Madras, India	13°	4·37	4·93	6·57	7·51	8·67	8·16

* Monsoon month.

MONTHLY EVAPORATIONS FROM WATER SURFACES. (Inches.)—*Cont.*

Locality.	Approx. Lat.	Month.						Year.
		July.	Aug.	Sept.	Oct.	Nov.	Dec.	
Emdrup, Denmark (av. of 11 years) . . .	56°	5·2	4·4	2·6	1·3	0·7	0·5	27·9
Bolton-le-Moors, England, 320 ft. (av. of 10 years)	53½°	4·02	3·06	2·02	1·28	0·81	0·47	25·65
Whitehaven, England, 90 ft.	54½°	4·13	3·29	2·96	1·76	1·25	1·02	29·91
Tulare, California . .	36°	11·6	9·8	7·4	5·5	3·8	2·0	68·5
Perth, Australia . .	32°	10·22 (Jan.)	8·35 (Feb.)	8·26 (Mar.)	5·21 (April)	2·27 (May)	1·80 (June)	65·79
Cairo, Egypt . . .	30°	6·35	6·95	5·09	5·49	—	—	—
Assuan, Egypt . . .	24°	16·11	16·11	14·17	10·74	—	—	—
Pashan, Poona, India .	18°	Monsoon months.			7·75	5·7	4·34	—
Red Hills, Madras, India.	13°	5·91	3·92	4·38	4·80	3·29	4·01	66·88

Losses by Evaporation and Absorption taken together.

In a great many cases the available data of losses from reservoirs are those for evaporation and absorption, or percolation, taken together. The more nearly the engineer can estimate what proportion of the total loss is due to evaporation alone, the more easily will he be able to decide as to the necessity or expediency of taking measures to reduce percolation. Fairly high losses may be tolerated from reservoirs where a considerable proportion of the percolating water reaches wells from which it can be lifted and used; and this may in some cases be an advantage when the level of the water in the reservoir is very low, or, at least, a mitigation of the disadvantage. Actual losses by percolation can be estimated when the rate of evaporation is obviously very small. Conversely, the proportion lost by evaporation may be estimated when the losses by percolation are known to be small. The losses may be measured as depths on the maximum waterspread, or as actual depths on the mean of the waterspreads at the beginning and end of each short period of observation. Thus, there are often apparent discrepancies between really similar cases, especially when the reservoirs are very shallow in parts, and when, as often happens, the maximum rate of evaporation occurs when the tank is very low.

Losses by percolation are sometimes very large. The losses from four irrigation reservoirs in the United States were as follows, measured in acre-inches or acre-feet referred to the *maximum* waterspread in acres:—(1) one year, 50·4 ins., all attributed to evaporation; (2) maximum of 2 years, 53·8 ins.; (3) maximum of 4 years, 132 ins.; (4) maximum of 2 years, 17·3 ft., ¾ of which was attributed to percolation.

At Red Hills tank, Madras, a total loss of about 7·5 ft. seems to have been indicated by one set of tests, and 8·39 by another series. An allowance of 9 ft. is now made, and is considered to be ample, a fairly liberal estimate being desirable as the water is used for the supply of the city. Tanks in Rajputana were found to lose 9·77 ft. per annum, of which 3·66 ft. was attributed to absorption.

The reservoirs of the Sheffield waterworks are estimated to have lost only about 15 to 15½ inches by evaporation and absorption, the loss at Rotherham in a neighbouring area being 22 ins.

The following were the results of observations at Red Hills Tank, Madras, the year being roughly divided into seasons which do not, however, begin and end at even approximately the same dates every year.

SEASONAL LOSSES (INCHES PER DAY) BY EVAPORATION AND ABSORPTION, RED HILLS TANK, MADRAS.

Humid Season under the modified influence of the South-West Monsoon.	North-East Monsoon and Cool Dry Season.	Hot Dry Season.
July 0·33	Nov. 0·27	Mar. 0·26
Aug. 0·32	Dec. 0·13	April 0·30
Sept. 0·38	Jan. 0·24	May 0·37
Oct. 0·27	Feb. 0·24	June 0·36
Average 0·33	Average 0·22	Average 0·31

EVAPORATION AND ABSORPTION (ALMOST ENTIRELY EVAPORATION) RECKONED ON THE TOP AREAS OF RESERVOIRS NEAR DHARWAR, BOMBAY PRESIDENCY (*Strange*).

Month.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May.	June.	July.	Aug.	Sept.	Oct.	Total.
Evaporation, ins. per day	0·16	0·16	0·16	0·33	0·67	0·67	0·51	0·31	0·16	0·16	0·16	0·17	3·65 ft.

Evaporation from the Soil.

A study of rates of evaporation from the soil is of practical importance as bearing upon estimates of the actual amounts of water required by crops under varying conditions, and the probable losses from the ground when the water is applied some time before cultivation. The importance of the subject is the greater in proportion to the extent to which the cultivation has the character of fruit growing or gardening, as distinct from the growth of large areas of grain crops. In orchards especially, it is possible so to apply the water that the loss by evaporation is considerably reduced. It is often an advantage to keep the water in distinct channels as long as possible.

CALIFORNIAN EXPERIMENTS ON EVAPORATION FROM THE SOIL.

Important series of tests have been carried out during recent years in California, and have been reported by Mr. S. Fortier, Chief of Irrigation Investigations, United States Department of Agriculture.

The experiments were carried out with tanks about 47 ins. deep and 23½ ins. diameter, sunk into the soil and filled nearly to the brim. These tanks were weighed in order to measure losses. The following tabular statement gives the results of experiments made in July and August on the evaporation from soils containing different proportions of water, measured as inches depth over the area of the surface. In the case of the initial free moisture this represented about 4½ to 6 per cent. The amounts of irrigation water added represented, respectively, very light, light, and ordinary irrigation.

WATER CONTENTS IN SOIL AND EVAPORATION (IN INCHES).

Initial moisture	2·8	2·7	2·7	4·1
Irrigation water	0	2·6	4·8	6·3
10 days' evaporation . . .	0·4	0·9	1·2	1·5

Evaporation from the soil may be much reduced by beating or crumbling the dry surface to a powder, by spreading a layer of finely divided soil over the surface, or by irrigating by means of furrows, afterwards spreading the dry soil from between the furrows by some means which will tend to crumble or powder it. In small-scale irrigation and where the cost of the water is high these considerations may be of importance. The following results were obtained in California with experimental tanks containing sandy loam, the evaporation from which when it was left uncovered after the water had been applied was compared with the evaporation after coverings of different depths of finely divided sandy loam had been added. The result of the experiment with the thinnest layer is, of course, the most important generally. During the fourteen days of the experiment the shade temperature of the air at noon was often more than 100°, while the soil temperature in the sun, at noon, varied from 120° to 140°. The figures were :—

Evaporation in hundredths of an inch per day : from uncovered soil, 72·5 ; from soil protected by a 4-inch layer, 20·6 ; 8-inch layer, 10·0 ; 10-inch layer, 2·0.

The following were the results of a twenty-one days' test.

SOIL EVAPORATION IN PERCENTAGE OF WATER APPLIED.

Days	2½	6½	10	14	17	21
Bare soil . . .	11	18	21	25	28	32
3-inch cover . .	2	5	7	9½	11½	14

A further series of experiments was carried out with tanks 47 inches deep and 23½ inches diameter.*

* See *Engineering News*, Sept. 5, 1912, but note that in figs. 2 and 3 the diagrams of evaporation in inches have been accidentally transposed.

In one set of tests the tanks were nearly filled with soil, leaving room for the covering of loose soil in each case, spread after the application of the water, while in another set of tests the tanks were nearly filled and the water applied at the bottoms of furrows of corresponding depths, the soil being afterwards spread. The following table gives the results of both experiments:

EVAPORATION AND SOIL COVERINGS.

Period after water was applied: days		7	14	21	28	21	28
		Evaporation in percentage of water applied.				Evaporation in inches.	
Surface irrigated and left		17·5	25	31·5 (29)		1·75	
Surfaces irrigated and covered with loose soil (inches)	3 ins.	4·8	9·5	13·7 (12·5)		0·75	
	6 ins.	1·5	3·2	6·0 (5·7)		0·34	
	9 ins.	1·5	1·3	4·1 (3·3)		0·22	
Surface irrigated and left		13·2	16·4	18·6	20·8		1·25
Water applied at the bottoms of furrows, depths in inches	3 ins.	9·2	12·0	14·9	16·5		0·99
	6 ins.	7·0	9·6	12·3	14·3		0·86
	9 ins.	6·1	7·5	9·9	12·0		0·72

In the column for the 21st day, the figures in brackets are calculated from the evaporation in inches, and possibly correspond to earlier readings. The other figures in the column are taken from the curve.

Relation between Evaporation from Soil and Evaporation from Water

The experiments, on the results of which the table below is based, showed what were the evaporations from soils containing different proportions of water, and from a water surface.

Percentage of Free Water.	Temperatures, Fahrenheit.				Weekly Evaporation in Inches.	
	Air in Shade.	Soil in Shade.	Soil in Sun.	Surface of Water.	Soil.	Water.
Saturated.	71°	76°	95°	77°	4·75	1·88
17·5%	76°	78°	106°	80°	1·33	1·94
11·9%	76°	78°	106°	80°	1·13	1·94
8·9%	76°	78°	108°	80°	0·88	1·94
4·8%	76°	78°	108°	80°	0·25	1·94

ANNUAL EVAPORATIONS IN INCHES DURING TWELVE YEARS, LEA BRIDGE, ENGLAND.

Water: max., 26·93; min., 17·33; mean, 22·2.
Soil: max., 25·14; min., 12·07; mean, 19·53.
Sand: max., 9·10; min., 1·42; mean, 4·65. (*Greaves.*)

Evaporation from Ice.

At Boston waterworks, U.S.A., the evaporation from ice was about 0·06 inch per day, of 7 ins. for the cold season.

Water Losses from Irrigation Canals.

No general rules can be given for estimating the rate at which water is lost from irrigation canals. Some are practically watertight, and only lose water by evaporation, while others lose large proportions of their supply by leakage.

On the Ganges Canal, at one period the water sent down was accounted for as follows:—Irrigation, 56 per cent.; loss in canal, 15 per cent.; in distributaries, 7 per cent.; in village watercourses, 22 per cent.

In the Punjab, a loss of 8 cusecs per million square feet of wetted surface has been allowed, and larger amounts in some cases. Certain canals with discharges from 400 to 450 cusecs were

found to lose about one cusec per mile. The Bari Doab Canal, when eighteen years old, lost about 12 to 14 per cent. of its discharge in a length of fifty miles.

Recently, instead of allowing 8 cusecs per million square feet of wetted area, it has been usual to allow, within the limits 4 ft. to 12 ft. depths, one cusec per foot depth per million wetted square feet.

Many canals lose a great deal of water for some time after their construction, and afterwards become reasonably watertight. The deposit of silt at a reasonable rate for some time after the canal is in service may be facilitated by suitable designing of the cross-sections and temporary adjustments of levels at the falls; but no general rules can be given as to this difficult problem.

When L is the loss from a canal, measured in the first mile, the loss up to a further distance of D miles is $L\,D^x$, where x varies from $\frac{5}{6}$ to $\frac{9}{8}$.

Losses through a Dam.

An extreme case of seepage through a gravel fill dam. The dam was 70 ft. high; upstream slope, 3 to 1; downstream slope $1\frac{1}{2}$ to 1. With the gates closed the seepage rates were approximately, during three years: with 20 ft. of water (gauge height), 60 cusecs; with 30 ft. of water, 84 cusecs; with 40 ft. of water 100 to 120 cusecs; with 50 ft. of water, 112 to 135 cusecs.

Capacities of Canals: Mean Velocity.

The discharge or capacity of a canal or river is the product of the cross-sectional area and the mean velocity. The mean velocity is usually found from Kütter's formula (see p. 1419), a coefficient of roughness, n, being chosen with regard to the nature of the soil and the probable behaviour of the canal with respect to silting and erosion. Since the formula takes account of the water slope or fall, and of the hydraulic mean radius,

$$\frac{\text{area}}{\text{wetted perimeter}}$$

every measurement or calculation of discharge by other means tends to fix the value of n for the particular conditions of the bed of the canal or stream. Small discharges may sometimes be measured directly in basins or symmetrical pools, and at notches or specially prepared weirs (pp. 1286–1290), but calculations for discharge over large weirs are themselves subject to disturbing factors and their value in the determination of n is limited.

For irrigation canals in favourable soils and working under favourable conditions the value $n = 0{\cdot}0225$ is considered suitable, and this value has been adopted in the design of very important works. For canals in less favourable conditions the value 0·025 is often adopted, and for canals distinctly below average conditions n may be taken as 0·0275. The value 0·030 may be applied to some rivers and to canals in 'defective condition,' whilst for channels which are much obstructed n may be taken as 0·035. The first two values above are those of most importance in irrigation engineering. For aqueducts of rubble masonry n may be taken about 0·017 to 0·020, and for ashlar and brickwork, 0·013.

Bazin's Formula.

Bazin's new formula is often used for channels, especially for those which, in irrigation systems, are relatively small. It is derived from the general formula

$$V = c\sqrt{RS},$$

by making

$$c = \frac{157{\cdot}6}{1 + \frac{m}{\sqrt{R}}},$$

where c is the coefficient, and V, R, and S have the values given on page 1419.

$m = 0{\cdot}109$ for smooth cemented surfaces.
$= 0{\cdot}29$ for ashlar and brickwork.
$= 0{\cdot}83$ for rubble masonry.
$= 1{\cdot}54$ for very smooth earthen channels, pitched, or partly pitched.
$= 2{\cdot}36$ for ordinary earthen channels.
$= 3{\cdot}17$ for canals encumbered with weeds or boulders.

Direct Measurement of Mean Velocities.

The Use of Floats.—Floats of various kinds have been employed for the measurement of the velocities of canals and rivers, being sent down the reach so as to maintain as far as possible their proper positions in the cross-sections, the average velocity being calculated from float speeds and depths in each strip of the reach. Double floats are often used, the lower and larger float which is a little heavier than water, being held at a determined depth below the surface by a smaller

surface float to which it is attached by a cord. Thus the velocities at different depths and at different distances from the centre-line of the stream can be measured. The essential difficulty in using these floats is that if the lower float has a specific gravity considerably greater than that of the water the surface float must be relatively large, and affect the result appreciably, while if the specific gravity of the lower float be but little greater than that of the water it is liable to be lifted towards the surface by small upward currents.

For the lower floats both cylinders and spheres have been employed; for instance, cylinders 6 ins. in diameter and 12 ins. long, or 6 to 8 ins. in diameter and 9 to 12 ins. long; and spheres 2 ins. and 3 ins. in diameter.

Various forms of surface floats have been used, including very thin discs of considerable area, intended to slip easily along the surface without seriously affecting the speed of the lower float. This form seems to be suitable in calm weather, while a deeper float of smaller surface area may be preferable when there is a breeze. Calm days are best for float observations of any kind, but if a river or canal flows with or against prevailing winds of considerable velocity observations on calm days may be to a small extent misleading.

Rod Floats.—For canals, and for reaches of rivers in which the water varies but little in depth, a good form is the rod float, which may be a tube weighted at the bottom so as to float upright, or a wooden pole similarly weighted. In the Ganges canal experiments both one-inch poles and one-inch tubes were used, the latter proving to be the better. The rods should float with their bottom ends just clear of the bed of the canal.

Current Meters.—Current meters are often used for measuring the velocity of streams and canals, and must be preferred to floats when there is no reach of sufficient regularity and length for the successful use of the latter. With a current meter the velocity at a particular cross-section may be found, and the measurement of that cross-section will then suffice, instead of the number which the irregularity of a reach may render necessary. It is not always easy in practice to keep the current meter at the desired depth, but in some cases it is held there by a rod. In a sluggish stream, the meter may be towed behind a boat, the speed of which is measured. Current meters should be calibrated, and from time to time tested, by towing them at a measured speed in still water, or, failing that, up and down the same reach of the canal or river. The mechanism of the meter can be stopped and started by a brake, on the stop-watch principle, this brake being worked by a cord from the boat. But some meters are fitted with an electrical apparatus, the wires of which form part of the tow line, and record the revolutions visibly in the boat.

The Pitot Tube.—If a tube be partly immersed, vertically, in a current of water, the lower end being suitably bent and remaining open, the water in the tube rises above the surface of the flowing water. The Pitot tube, designed on this principle, gives approximately the relation, $V = \mu \sqrt{2gh}$, where V = the velocity of the current, and h is the height of the water in the tube above the surface of the stream. Instruments which have been employed have given the results $V = 1{\cdot}7 \sqrt{h}$. A much more elaborate instrument on the same principle (*Darcy*) has also been employed, and has given good results in small channels with fairly high velocities.

Perrodil's Hydrodynamometer.—This instrument is a device for measuring the velocity of a current, by the torsion which is produced in a wire by the pressure of the water against a disc mounted on an arm at right angles to the wire. The velocity is found by the equation $V = c \sqrt{a}$, where a is the angle of torsion, and c a coefficient found by experiment, or calculated from the coefficient of torsion of the material of the wire. The instrument is very sensitive and can be used for very slow currents.

Mean Velocity in a Vertical.—The mean velocity of the water in a given vertical in the cross-section of a stream is also the actual velocity at some proportion of the total depth in that vertical. What this proportion is will depend upon the nature of the stream, and will not be the same for all the verticals. For streams of considerable width and not very variable depth at the cross-section, measurements, by means of a current meter, of the velocities at the assumed depth of mean velocity may yield a fair approximation to the mean velocity of the stream, and a closer approximation may be obtained by making preliminary tests, with rod floats for instance, to determine the depth of mean velocity in the particular case. Values given usually lie between 0·60 and 0·75, or 0·60 and 0·70 of the depth.

Relation between Mean and Maximum Velocities.—For a rough estimation of the velocity of a stream the maximum surface velocity may be measured and the mean velocity deduced from it. If V be the maximum velocity, the mean velocity is KV, where K is a coefficient usually ranging from 0·60 to 0·80. In the following table values of K are given for earth channels and rubble masonry channels.

VALUES OF 'K' IN THE EQUATION 'V MEAN = KV MAXIMUM.'

Hydraulic mean depth (feet) .	1	2	3	4	5	6	7	10	15	20
Rubble masonry channels .	·77	·79	·80	·81	·81	·81	·81	·82	·82	·82
Earthen channels	·65	·71	·73	·75	·76	·77	·78	·78	·79	·80

Critical Velocity.

The ' critical velocity ' is that mean velocity at which, for a canal of given depth, the current will neither deposit much silt nor unduly erode its bed. A determination of the critical velocity, the theory of which was evolved by R. G. Kennedy (see Min. Proc. Civil Engineers, vol. cxix.), is of great importance in the design of irrigation canals. The removal of silt from the canals is costly, and the loss of valuable fertilising material is a serious matter to the cultivators. A large amount of silting of the canals and distributaries is often inevitable, but attempts are made to lessen it by designing the canals in accordance with the principle of critical velocity.

The equation for determining the critical velocity for a given depth of water in the canal is

$$V_o = cd^m.$$

The values of c and m vary somewhat according to the nature of the silt. For fine sand-silt brought down from the hills usual values are, $c = 0{\cdot}84$ and $m = 0{\cdot}64$. The following table gives the critical velocities calculated on this basis, for different depths, and the observed critical velocities on the Shwebo and Mandalay canals, Burma. With the latter velocities there was, on the whole, a slight tendency to scour.

Critical Velocities.

Depths in feet . . .	1	2	3	4	5	6	7	8	9	10	12	15	20
$V = 0{\cdot}84\ d^m$, suitable for the Punjab . . .	0·84	1·30	1·70	2·04	2·35	2·64	2·92	3·18	3·43	3·67	4·12	4·75	5·71
Observed in Shwebo and Mandalay Canals . .	—	—	1·72	1·98	2·24	2·48	2·75	—	3·09	—	—	—	—

A canal which would be non-silting on a slope of 1 in 7,600, if drawn from the Indus in Sind, would require to have a slope of 1 in 4,200 to be non-silting if carrying the silt usually found in the Punjab (*Buckley*).

If the theory of critical velocity be accepted as regards silting and erosion up to the old limits of velocity for different soils, but be rejected when those velocities are reached, the maximum depths allowed for large canals will be correspondingly restricted. If, for instance, a maximum velocity of 3·5 ft. per sec. be prescribed, the depth will be limited to about 9 ft. On the old theory, a higher velocity would cause scour, while on the theory of critical velocity, greater depths with a limit of 3 to 5 ft. per sec. would cause silting. There is, however, no sudden change in the physical conditions, and if the theory of critical velocity holds up to 9 ft. depth it doubtless holds for considerably greater depths; and it has been accepted in the design of very important irrigation systems.

Argentina.—Adopted for canals of the Upper Rio Negro Irrigation (*R. E. Ballester*):

$$V_o = 0{\cdot}52d^{0{\cdot}44} \text{ metric},$$

the relation being, British $V_o = 0{\cdot}84d^{0{\cdot}64}$ corresponds to metric, $V_o = 0{\cdot}546d^{0{\cdot}64}$.

United States.—Working data for irrigation engineers (*E. A. Moritz*): proposed values; for fine silt or sandy mud, $V_o = 0{\cdot}464d^{0{\cdot}5}$; for somewhat coarser silt or sandy mud, $V_o = 0{\cdot}508d^{0{\cdot}5}$ and, for the same index, the coefficients: sandy, loamy silt, 0·55; coarse silt and other hard earths, 0·602.

Hydraulic Mean Depth: Mean Velocity: Scour.

(*P. A. M. Parker.*)

For:	Fine Silt.			Heavy Silt and Fine Sand.			Coarse Sand.		
There is no scour in a channel of H.M.D.	1·0	2·5	5·0	1·0	2·5	5·0	1·0	2·5	5·0
Until a mean velocity is reached of	0·4	0·7	0·9	0·9	1·5	1·75	1·75	2·25	3·0

	Small Pebbles (Size of Peas) and Gravel.			Large Pebbles (Hen's Egg) and Coarse Gravel.				Large Stones.	
There is no scour in a channel of H.M.D.	1·0	2·5	5·0	1·0	2·5	5·0	10·0	1·0	10·0
Until a mean velocity is reached of	2·25	3·0	3·5	5·0	6·0	7 0	9·0	15·0	23·0

VALUE OF c IN $V = c\sqrt{RS}$. $n = 0{\cdot}0250$. (By Kütter's formula, page 1419.)

R in Feet.	S					
	1 in 1,000. 1 in 1,000.	0·5 in 1,000. 1 in 2,000.	0·3 in 1,000. 1 in 3,333.	0·15 in 1,000. 1 in 6,667.	0·10 in 1,000. 1 in 10,000.	0·05 in 1,000. 1 in 20,000.
1·0	55·4	54·9	54·3	52·9	51·8	49·4
1·5	61·3	60·9	60·5	59·5	58·7	56·8
2·0	65·5	65·2	64·9	64·3	63·7	62·4
2·5	68·7	68·5	68·3	68·0	67·6	66·9
3·0	71·2	71·2	71·1	71·0	70·9	70·6
4·0	75·1	75·3	75·4	75·7	76·0	76·6
5·0	78·1	78·3	78·6	79·3	79·9	81·3
6·0	80·4	80·8	81·2	82·2	83·1	85·2
7·0	82·3	82·7	83·3	84·6	85·7	88·5
8·0	83·9	84·4	85·1	86·6	88·0	91·3
9·0	85·3	85·9	86·6	88·4	89·9	93·8
10·0	86·5	87·1	88·0	89·9	91·6	96·0
11·0	87·6	88·3	89·2	91·3	93·2	98·0
12·0	88·5	89·3	90·2	92·5	94·6	99·8

VALUES OF c IN $V = c\sqrt{RS}$. $n = 0{\cdot}0225$. (By Kütter's formula, page 1419.)

R in Feet.	S					
	1 in 1,000. 1 in 1,000.	0·5 in 1,000. 1 in 2,000.	0·3 in 1,000. 1 in 3,333.	0·15 in 1000. 1 in 6,667.	0·1 in 1,000. 1 in 10,000.	0·05 in 1,000. 1 in 20,000.
1·0	62·5	61·9	61·2	59·7	58·5	55·7
1·5	68·8	68·4	67·9	66·8	65·9	63·8
2·0	73·2	72·9	72·6	71·9	71·2	69·8
2·5	76·5	76·4	76·2	75·8	75·4	74·6
3·0	79·2	79·2	79·1	79·0	78·8	78·5
4·0	83·3	83·4	83·6	83·9	84·2	84·9
5·0	86·3	86·6	86·9	87·6	88·3	89·9
6·0	88·7	89·1	89·5	90·6	91·6	93·9
7·0	90·7	91·1	91·7	93·1	94·3	97·3
8·0	92·3	92·8	93·5	95·2	96·6	100·3
9·0	93·7	94·3	95·1	97·0	98·6	102·9
10·0	94·9	95·6	96·5	98·5	100·4	105·1
11·0	96·0	96·7	97·7	99·9	102·0	107·2
12·0	96·9	97·7	98·8	101·2	103·4	109·0

(For the same *actual* R (such as 3·28 ft., or 1 metre) the coefficient for metric units is the coefficient for British units $\times \frac{100}{181}$, or $\frac{5}{9}$, nearly.)

Duty of Irrigation Water.

The duty of irrigation water, or the area irrigated by a given rate of flow during the season, depends upon the nature of the crop, the initial proportion of moisture in the soil, the rate of evaporation, and the rainfall during the irrigation season. It may also depend upon custom and acquired rights and is often affected by the degree of care or of carelessness exercised by the

cultivators. All figures relating to duty are, therefore, to be accepted with due reserve for purposes of estimates. The duty reckoned on the amount passing the headworks is, of course, less than that reckoned on amounts measured out to the cultivators, and the former figures will be affected by the length of the main canal and principal branches, and the rate of loss therefrom. The duty of irrigation water may be measured in cusecs continuously for the crop; in feet; or in acres per million cub. ft. stored.

On the Jamrao Canal system, India, the duty of one cusec per 50 acres has been allowed for during the inundation season, and 100 acres during the cool season. The duty has been as high as 83 acres in the inundation season, for 'dry crops' such as millet and cotton. In the Punjab 100 acres has been allowed for in the rains, and 200 acres in the cool season.

The following duties, in acres per cusec, may be noted:—

Palked canal, Bengal Deccan, 63; Ekrak tank project, Bombay, 123; Pahakarpur canal, Bombay, 165; Kabul river canal, Bombay, 114; Lower Swat river canal, Bombay, 175; Letabo canal project, Transvaal, 145; Mooi river, Transvaal, 145.

DUTIES OF CERTAIN IMPORTANT IRRIGATIONS.

Lower Chenab: at heads of distributaries, kharif crops, 89 to 111, usually about 95; rabi crops, 217 to 284, usually about 250. At heads of canals: kharif, 73 to 95, usually about 75; rabi, 190 to 237, usually about 220.

Jamrao: kharif, 85; some branches give 90 at their heads, and one gives 100; average at heads of branches, 85.

Sind: proposed under conditions to be brought about by the Sukkur project; kharif crops, 87; rice, 43½; rabi crops about 180.

Kharif Crops: Triple Canal Project, 100; Lower Bari Canal, 109; other cases, 65 to 75, and 85.

Rabi Crops: Lower Bari Canal, 200; Lower Chenab, 208; Upper Bari Doab, 234; Lower Bari Doab, 198.

By amounts: in Kathiawar, jowari, a grain crop, requires 100,000 cub. ft. per acre, and in Jeypore State the same crop requires about 120,000 cub. ft., including losses and contingencies. In the Madras Presidency, where large areas are irrigated from numerous shallow reservoirs, the quantity, including all losses, which is stored per acre is, roughly, about 216,000 cubic ft., or nearly 5 ft. depth, for the 5 months' 'monsoon' rice crop, and 175,000 cubic ft., nearly 4 ft. depth, for a cool weather rice crop. In Central India, the storage is about 200,000 to 250,000 cubic ft. per acre.

TOTAL DEPTHS IN INCHES.

Punjab: wheat, barley, and poppy, 10·6; senji, 10·5; gram, 3·4; sugar cane, 25·3; Indian corn, 10·6; cotton, 10·6; charki, 6·3.

Punjab: the Colony Canals (H. W. Nicholson); not including preliminary watering of the ground: kharif crops: sugar-cane, 30 to 45; rice, 35 to 40; cotton, 12 to 20; maize, 12 to 18; great millet, 6 to 9; rabi crops: wheat and barley, 9 to 10; sometimes 15 for wheat; toria (*Brassica campestris*), 6 to 10; senji (*Mendicago sativa*), 15 to 20. *Bari Doab:* kharif crops, 32; rabi crops, 20.

DUTY OF IRRIGATION WATER: AVERAGES OR ESTIMATED AVERAGES: ACRES PER CUSEC.

(Duty × 14·3 = hectares per cu. metre per sec., or, $\frac{6{,}040}{\text{duty}}$ = cu. metres per day per hectare.)

Southern California, surface irrigation, 150 to 300; S. California, sub-irrigation, 300 to 500; Southern Arizona, 100 to 150; New Mexico, 60 to 80; Utah, 60 to 120; Colorado, 80 to 120; Northern Italy, 60 to 150; Madras Presidency, rice 66, 'dry crops,' 132.

The average duty of five canal irrigation systems in the Bombay Deccan, from 1894–1895 to 1900–1901, excluding two famine years, was, successively:

For *Kharif*, 70, 52, 74, 111, and 138, with a mean of 89; and for *rabi*, 60, 68, 90, 75, 85, with a mean of 76.

Costs of Irrigation Projects.

The costs of irrigation projects in India do not admit of comparison unless the nature and purpose of the project in each case be described in considerable detail. Some projects are highly productive, others are protective, and many are of an intermediate character. The conditions are very different in the United States, and, on the assumption that the works have been carried out on the narrower economic basis, the following costs may be worth noting.

COSTS OF IRRIGATION PROJECTS OF THE UNITED STATES RECLAMATION SERVICE.

Area in Thousands of Acres	216	206	164	131	129	118	103	100	72	60
Cost of water right in dollars per acre	30 to 36	22 to 30	45 to 50	55 to 66	45 to 55	22 to 30	52	30 to 35	30	45

COSTS OF IRRIGATION WITH STORED WATER, S. AFRICA.

Blijde River : per acre of irrigable area, 31*l.* 18*s.*

Calitzdorp : per acre of irrigable area, on total cost including land and roads, 120*l.* 12*s.*; on engineering cost, 110*l.* 18*s.*

Van Rynevelt's Pass : per acre of normal irrigation, 23*l.* 11*s.*; per acre of total area (irrigable in good seasons), 17*l.* 18*s.*

COST OF IRRIGATION WITH PUMPED WATER, S. AFRICA.

On the total cost of 63 pumping projects, the cost per acre was 6*l.* 17*s.* Near the coast, the cost of putting 2 ft. per acre on to the land may be taken as between 1*l.* and 2*l.* for a 50-ft. lift, and between 1*l.* 10*s.* and 2*l.* 10*s.* for a 150-ft. lift. Up country, the costs would be about 2*l.* 10*s.* and 3*l.* respectively. (*Legg.*)

COSTS OF IRRIGATION : NEBRASKA.

In Nebraska, 1920, 1 per cent. of farm lands irrigated ; average per farm, 147 acres ; capital invested, 5*l.* 4*s.* per acre ; annual cost of operation and maintenance, 6*s.* 3*d.* per acre ; average depth of irrigation water on the land, 2·4 ft.

Waste Weir Capacities.

For rates of rainfall and calculations for run-off, see ' Rainfall,' pp. 1399, 1400. For discharges of weirs, see pp. 1286–1290. The following formula is used in India for the discharges of the overflow weirs of storage tanks and reservoirs.

Discharge of clear overfall weirs :—

$$D = c\,b\,\sqrt{H^3}.$$

D = discharge in cusecs ; b = width of the crest in feet ; H = total height above the weir crest in feet, allowing for afflux ; c is a coefficient usually about 3·333, or as low as 3·25.

In ordinary practice in the Bombay Presidency, India, it is usual to assume a general rainfall over the whole of a small catchment, and to calculate the run-off accordingly, thus :—

Catchment in Sq. Miles . . .	Up to 5.	5 to 10.	10 to 25.	25 to 75.	75 to 150.	Over 150.
Run-off in ins. of rain per hour .	2·00	1·50	1·25	1·00	0·75	0·50

In using this table special circumstances have to be considered ; mountain catchments would be given larger runs-off, and for catchments wholly in the plains smaller runs-off would be assigned. (*Strange.*)

METRIC VALUES OF c IN $V = c\sqrt{RS}$. $n = 0{\cdot}0250$.

(See page 1393.)

R in Metres.	S = 1/40,000 = 0·000025	1/20,000 = 0·00005	1/10,000 = 0·0001	1/5,000 = 0·0002	1/2,500 = 0·0004	1/1,000 = 0·001	1/100 = 0·01
	c						
0·025	9	10	11	12	13	13	14
0·05	12	13	15	16	17	18	18
0·10	17	18	19	20	21	22	22
0·20	22	23	24	25	26	27	27
0·30	26	28	29	30	30	31	31
0·50	31	32	33	34	34	35	35
1·00	40	40	40	40	40	40	40
2·00	50	48	47	46	45	45	45
3·00	56	53	51	49	48	48	47
5·00	64	59	54	53	52	51	50
10·00	75	66	60	57	55	54	53
15·00	81	71	63	59	57	56	55
20·00	85	72	64	60	58	57	56
30·00	90	76	67	62	60	58	57

Note.—A curve drawn for one slope will give intermediate values for that slope and for all values of R.

Areas—Depths and Flow.

[For the full table see Buckley's 'Irrigation Pocket Book'; the figures given below suffice for the calculation of any acre-feet-time quantity, from cusecs

Cusecs.	Acre-feet per				
	Hour.	12 hours.	24 hours.	Week.	30 days.
1	0·08	0·99	1·98	13·88	59·5
2	0·17	1·98	3·96	27·76	119·0
3	0·25	2·97	5·95	41·65	178·5
4	0·33	3·97	7·93	55·53	238·0
5	0·41	4·96	9·92	69·42	297·5
6	0·49	5·95	11·90	83·30	357·0
7	0·58	6·94	13·88	97·18	416·5
8	0·66	7·93	15·87	111·10	476·0
9	0·74	8·92	17·85	124·95	535·5

Basic Economy of Dams.

Volume of Water stored per Unit Volume of Masonry in the Dam.

Poona, India, 62; Beetaloo, Australia, 79; Booton, New Jersey, 200; Cross River, New York, 610; Granite Springs, Wyoming, 716; Wachusett, Massachusetts, 1,369; Sodom, New York, 1,569; Sweetwater, California, 1,679; Assouan, Egypt, 1,980; Periar, Madras, 2,620; Roosevelt, Arizona, 7,426; Calitydorp, S. Africa, 113.

Earthen Dams: Bon Accord, Transvaal, 36·4; Blijde River, 20·2; Grassridge, S. Africa 202. Dam under construction, 1929: Saluda River, S. Carolina, 367.

Losses of Irrigated Area by Degradation.

The principal causes of losses of irrigated area, or of reduced productivity, are alkalinity of the soil and waterlogging. As regards the latter, the importance of including measures for effective drainage, as far as may be necessary, in the irrigation project, is now generally realised. The measures to be taken in order to deal with alkalinity cannot be briefly described in general terms, but are fully discussed in standard works. The extent of the two evils, taken together, in the United States may be judged from the following quotation: 'It is safe to say that 15 to 20 per cent. of the lands reclaimed by irrigation have been ruined or are endangered by seepage or alkali. Assuming that a total of 20 million acres have been irrigated within the United States, then from 3 million to 4 million acres have been affected, and possibly a million acres rendered useless.' (D. W. Murphy, 1923.)

Silting of Reservoirs.

Muchkundi: capacity, 704 million cub. ft.; catchment, 26 sq. miles; accumulated silt, in 22 years, 58 million, and in 26 years, 64 million cub. ft.; respective rates, 101,000 and 95,000 cub. ft. per sq. mile of catchment per annum. Capacity reduced 8·2 per cent in 22 years; 9·9 per cent. in 26 years.

Gokak: capacity, 908 million cub. ft.: catchment, 1080 sq. miles; accumulated silt, in 18 years, 155 million, in 19 years 174 million, and in 20 years, 189 million, or 8,750 cub. ft. per sq. mile of catchment per annum. Capacity reduced 28 per cent. in 20 years.

Lake Fife: not having sluices for silt, in the dam; at 14 ft. above sill, area of water surface: in 1883, 88·76 million sq. ft.; in 1905, 72·34 million sq. ft.; loss of capacity in 22 years, 16·33 million cub. ft., or 18 per cent.

Lake Whiteing: having sluices for silt, in the dam; no appreciable change in capacity in 5 years.

Proportions of Silt Borne by Rivers and Canals.

Reputed maximum proportions of silt to water, by weight: Rivers—Mississippi, 1/572; Nile, in August, 1/666; Indus, at Sukkur, 1/163 to 1/198, velocity 6·4 to 8·3 ft. per second; a tidal river in East Bengal, during six days, ranged from 1/455 to 1/25. The Vaal, rising, and 3 ft. 6 ins. over weir, 1/268; falling, and 1 ft. 7 ins. over weir, 1/660; Rhône, velocity 8 ft. per second 1/45; Vistula, velocity 10 ft. per second, 1/48; Kistna, June, 1/42.

Cubic feet of damp silt per million cub. ft. of water: Sirhind canal (1897, 32,600 million cub. ft. intake), 5,464, (1895, 43,000 million cub. ft. intake) 1,650, (1896, 54,600 cub. ft. intake) 2,343. Sone canals, June, 1,875; July, 3,250; Aug., 2,750; Sept., 625; Oct., 62. Guadalquivir, on successive dates, March 30 to May 2, 2,321, 775, 521, 695, 2,048, 596, 273, 222, 198.

Fluctuations of the Vaal River.

(W. Van. Warmelo.)

A table gives for eight gauging stations on the River Vaal and its tributaries the average flow for each month of the year, in one case based on 2 years' observations, in two cases 8 years', in other cases 9 to 17 years'. A rough averaging of the 8 cases gives the following months' flows, as percentages of the average 12 months' flow: Oct., 4; Nov., 10; Dec., 12; Jan., 17; Feb., 21; March, 20; April, 7; May, 2; June, 1½; July 1½; Aug., 1½; Sep., 3. For the River Vaal, at Standerton, the figures, based on 16 years' observations, were: Oct., 5·56; Nov., 14·99; Dec., 14·75; Jan., 27·15; Feb., 20·12; March, 11·49; April, 2·39; May, 1·05; June, 0·63; July, 0·53; Aug., 0·72; Sept., 0·62.

Typical South African Irrigations.

Van Rynveld's Pass: catchment, 1,477 sq. miles; average rainfall, 14½ ins.; area irrigated, max., 22,314 ac.; average, 17,000 ac. Dam, length, 1,170 ft.; crest width, 10 ft.; clear height (max.), 109 ft.; height plus deepest foundation, 152 ft.; capacity of reservoir, 64,000 acre-feet. Cost, 400,000*l.*

The Hartebeestpoort River: irrigation controlled by the Hartebeestpoort Dam; altitude of river bed, 3,827 ft.; catchment, 1,560 sq. miles; average rainfall near the dam, about 23 ins. Dam: height above river bed, nearly 190 ft.; thickness at base, 73 ft.; radii at base, downstream face, 75 ft., upstream face, 148 ft.; increasing radii to vertical top section where the upstream face radius is 240 ft.; one tangential abutment; crest and top section width, 15 ft.; both batters 1 in 5·714; freeboard above full supply level, about 22·75 ft., capacity of reservoir, at waste weir crest level, 136,241 acre-feet; capacity above outlet level, 123,200 acre-feet; waterspread at F.S.L. (waste weir crest level), 6·7 sq. miles; 9·2 sq. miles available.

Earthen Dams.

Bon Accord, Transvaal: catchment, 121 sq. miles; average rainfall, 28 ins.; storage, 400 million cu. ft.; irrigable area, about 2,750 acres; crest length, 3,476 ft.; max. height, 58 ft.; volume of the dam, 407,000 cu. yds.

Blijde River: catchment, 53 sq. miles; capacity of reservoir, 2,120 acre-feet; irrigable area, 1,066 acres; average annual rainfall, 15 ins.; crest length of dam, 1,135 ft.; max. height, 60 ft.; volume of earthwork, 148,800 cu. yds.; cost, 34,000*l.*

Saluda River, S. Carolina: dam being built (1929) for hydro-electric power; capacity of reservoir, 2½ million acre-feet; max. height, 208 ft.; max. base width, 1,150 ft.; volume, 11 million cu. yds.

The Sukkur Barrage Project (Under Construction).

Barrage 4,926 ft. long, with 66 openings of 60 ft.; gates, 63 ft. 3 ins. by 18 ft. 6 ins. in height; The Rice Canal, capacity, 12,346 cusecs; length, 87 miles; North-West Perennial Canal, 4,313 cusecs, 97 miles; South-East Perennial Canal, 2,767 cusecs, 140 miles; Rohri Perennial Canal, 10,250 cusecs, 205 miles; Eastern Nara Supply Channel, 12,200 cusecs, 15 miles; 242 miles of River Nara canalised. Totals of areas to be irrigated, acres: rice, 823,000; cotton, jowari, etc., 1,739,000; rabi crops, 3,338,000. Cultivation acreage, total, 5,900,000 annually in a total commanded area of 8,132,000 acres. Estimated cost, 1,836 lakhs; or, taking the rupee at 1*s.* 4*d.*, 12,240,000*l.*; taking the rupee at 1*s.* 6*d.*, 13,770,000*l.* Estimated annual tonnages of crops after 30 years' development: rice, 515,000: cotton, 190,000; jowari, etc., 575,000; wheat, 935,000.

By the autumn of 1929, work was at an advanced stage on 16½ spans on the right bank and 22½ spans on the left bank; the seven head regulators were nearing completion; of the 5,551,200,000 cu. ft. of excavation, in all canals, 2,471,300,000 had been completed. For details, see *The Engineer*, October 4, 1929.

Lloyd Dam, at Bhatgar, Bombay Presidency: completed October, 1928; catchment, 128 sq. miles; capacity of reservoir, 555,555 acre-feet; height above lowest foundations, 190 ft.; crest length, 5,333 ft.; 81 sluices, of which 45 are automatic; volume of masonry in the dam, 21,500,000 cu. ft.; area commanded, 834,000 acres, of which 202,000 will be irrigated annually; cost of dam, 1,260,000*l.* of the whole project, including the two main Nira canals, 106 and 100 miles long, cost 4,140,000*l.*; annual value of crops grown on the irrigated area, 2,380,000*l.*

Bhandardara Dam, impounding Lake Arthur Hill, 75 miles north-east of Bombay: altitude, 2,170 ft.; catchment, 47 sq. miles; average annual rainfall, about 220 ins.; waste weir capacity, about 56,000 cusecs; capacity of reservoir, 10,086 million cu. ft., which can be increased by 2,004 million; area 'protected,' 229,000 acres; height of dam above river bed, 270 ft.; above deepest foundation, 278 ft.; crest length, 1,663 ft., of which 600 ft. is curved to a radius of 6,000 ft.; max. width at base, 234 ft.; volume of the dam, 12 million cu. ft.; average cost, 25*s.* 3*d.* per cu. yd.; total cost of dam, rupees 84·14 lakhs, or 631,000*l.*

Dimensions of Sluices.

Lake Mentz Dam, Sundays River, South Africa.—Five free roller sluice gates, each 30 ft. wide by 25 ft. high; about 25½ tons weight; counterpoise, 53 tons; designed for water head of 26 ft. Each gate is built up of latticed bowstring girders as the span elements.

Sennar Dam, Blue Nile.—The portion called the 'sluice dam' is 606 m. (1,988 ft.) long, with 80 sluices 2·0 m. wide by 3·40 m. high to the springers (6·56 ft. × 27·55 ft.). The head regulator has 14 sluices, each 3·0 m. wide by 5·0 m. high to the springers (9·84 ft. × 16·4 ft.); seven to be provided with gates and seven (held in reserve) blocked.

Automatic Syphon Spillways: Early and Recent Examples.

Champlain Canal, U.S.A. (G. F. Stickney, 1910).—At lock No. 9 the syphon spillway was 57 ft. long as against 200 ft. if an ordinary weir had been built for the small permissible depth on sill. Each syphon had an area of 7⅞ sq. ft.; head, 10½ ft.; discharge, 160 cusecs; flared inlets. At Whitehall, on the same canal, a similar weir, 6 syphons; 16 ft. head. These syphons were formed in the body of the weir. (See *Engineering News*, October 13, 1910.)

Maramsilli Reservoir.—A syphon spillway to pass 40,000 cusecs. (See Paper No. 4602, *Proceedings Inst. C.E.*, by Powys Davies.)

Bridgeport Dam, Nevada, 1926.—Cross-section, 3 ft. by 6 ft.; air vents in syphon hoods at different elevations, so that the syphons will come into action one at a time.

Arch Dam: Measured Strains under Water Load.

Stevenson's Creek Dam, California.—A concrete dam built for the purpose of measuring strains under water load and comparing deduced stresses with calculated stresses. Founded on solid rock, in a gorge. Height 60 ft., crest length 140 ft., upstream radius 100 ft., thickness at base 7½ ft., at crest 2 ft. Tests were made at night to minimise temperature effects. At 30 ft. below crest the maximum deflection under full load, 60 ft. water depth, was 0·378 in. At the crest the deflection was downstream at the crown and upstream between the quarter points and the ends. Computed for a modulus of elasticity of 3,600,000, the stresses were: at the crown, 260 lbs. per sq. in. (theoretical 207); at the abutments, 900 to 1,000 lbs. per sq. in. (theoretical 879). Considered as assigned in part to arch resistance and in part to vertical elements resistance, the share of the latter in sustaining the water load seems to have been less than that assigned by theory. This may have been due to the formation of a horizontal crack at the base of the upstream face between the dam and its foundation, which appeared when the water had risen to 30 ft. depth. The formation of this crack is in accordance with the theory regarding the design of the foundations at and near the upstream toe of a dam, according to which the dam should not there be keyed into its foundation. [See 'Mass Dams and Dams of Arches' (R. Ryves): Madras Government Paper, 1910.]

Measured Shrinkage of an Hydraulic Fill Dam.

(*C. H. Eiffert*, '*Engineering News Record*,' *Jan.* 27, 1927.)

Huffman Dam.—Height of dam above ground level, about 62 ft.; width of crest, 32 ft.; symmetrical to central vertical plane; two berms; both slopes, from toe, 3 to 1; between berms, 2½ to 1; to crest, 1·825 to 1. Shrinkages, including settlement of foundation which is believed to have been very small: at 40 ft. below crest level, 0·40 to 0·68 per cent.; at 30 ft. below crest level, 0·31 to 0·54 per cent.; at 20 ft. below crest level, 0·44 to 0·64; at 10 ft. below crest level, 0·24 on east side and 0·73 on west side; at the crest, 0·60 on the east side and 0·49 on the west side; average, 0·51 per cent.

ADDENDUM TO 'IRRIGATION.'

Ratio of Pan Evaporation to Open Water Evaporation.

(*Meeker: Trans. Am.Soc.C.E., vol.* 90, 1927.)

Denver, Colorado field laboratory; on open prairie land at elevation 5,346 ft. Mean annual temperature, at Denver, 53 years record, 50·1° F.; mean annual precipitation 14·27 ins.; relative humidity 53 per cent.

Ground Pans.—Pans 3 ft. deep, water in pans 2·75 ft. deep; pans set 2·75 in the ground. Coefficients of reduction to open water surfaces: diameter of circular pans; 2 ft., coeff. 0·77; 4 ft., coeff. 0·84; 6 ft., coeff. 0·90; 9 ft., coeff. 0·98. Also 6 ft., tank, 2 ft. deep, 1·75 ft. in ground, coeff. 0·88; and 4 ft. dia. 10 ins. deep pan, set on timbers, depth of water 0·62 ft., coeff. 0·66.

Floating Pans.—Three feet square by 1·5 ft. deep, depth of water in pan 1·25 ft.; coeff. 0·91; circular, 4 ft. dia., 10 ins. deep, depth of water 0·58 ft., coeff. 0·92; 3·5 ft. dia., 1·83 ft. deep, depth of water 1·5 ft., 0·91 (approximate, by interpolation).

Dams Recently Built.

Pardee Dam, California, by October 1929. Curved gravity type; concrete, rock abutments; max. height (probably including foundations) 358 ft.; crest length 1,337 ft.; roadway on crest, thickness at base 241 ft.; volume 615,000 cu. yds.; foundations excavation 153,000 cu. yds.; capacity of reservoir 222,000 acre-feet, or 9,670,320,000 cu. ft., or nearly 273,830,000 cu. metres. In August 1928, 67,000 cu. yds. of concrete were placed in the dam. About 3½ miles of aerial ropeway were employed. Volume of water to volume of masonry in the dam, 582.

Bull Run Dam, Oregon, completed in the summer of 1929. Curved gravity type, 200 ft. above shallow notch foundation on rock. Concrete, the lower part in daily work layers normal to the lines of principal stress. About three-quarters of the height has an upstream batter of 1 in 25; downstream face below the curve to crest, 0·70 to 1. Water storage 31,000 acre-feet or nearly 1,350 million cu. ft., or nearly 38·1 million cu. metres. Catchment 74 sq. miles; estimated maximum flood, 21,000 cusecs.

Recent Literature.

Culverts.—'Experiments with Model Culverts made with Hume Pipes.'' By C. G. Hawes and H. S. Kahai. Bombay P.W.D. Technical Paper, 27. See 'Engineering Abstracts,' Inst. C.E., No. 306, vol. 43.

Canal Falls.—'The Evolution, Design, and Construction of Canal Falls, with a Special Description of Tando Masti Khan Fall on Rohri Canal' (a standing wave flume meter fall). By J. R. Colabawala. Bombay Engineering Congress, 18, Paper No. 123. See 'Engineering Abstracts, No. 305, vol. 43.

Government of Bombay Technical Paper No. 15. By C. C. Inglis. Is a description of the theory of the standing wave.

Regulators.—'Report on Experiments carried out at the Karachi Model Testing Station on a Model of a Flumed Regulator.' By C. G. Hawes and H. S. Kahai. Bombay P.W.D. Technical Paper, No. 31. See 'Engineering Abstracts,' No. 303, vol. 43. 'Experiments on a Model of a Canal Bifurcation' (Karachi Model Testing Station). By C. G. Hawes and H. S. Kahai. See 'Engineering Abstracts,' No. 257, vol. 45.

River Barrage.—'Les Nouveaux Barrages Mobiles de la Seine.' Rapport de Mission de M. Lekenna. Fifty pages, 21 figs. 'Annales des Travaux Publics de Belgique,' October 1930.

Reservoirs.—'Flood Absorption Effect of Lakes.' By B. D. Richards, M.Inst.C.E., M.Cons.E. Tokyo World Engineering Congress, 1929. Paper No. 233. Includes a proof of the mathematical solution of the primary problem, and practical methods of application. Reviewed in *The Surveyor*, December 18, 1930.

Floods.—'Flood Flows: a Study of Frequencies and Magnitudes.' By Allen Hazen, M.Am.Soc.C.E. Reviewed in *The Engineer*, October 31, 1930. The data are entirely American.

Irrigation.—'Water Supply and Utilisation.' By Donald M. Baker and Harold Conkling. Reviewed in *The Engineer*, December 5, 1930. Relates to engineering problems of irrigation and water supply and conservation in the arid west of the United States; also to water law.

Recent Works: Projects in Hand.

Egypt.—The Nag Hammadi Barrage (fully described in articles in *The Engineer*, commencing January 30, 1931). Contract price £1,976,555. Founded entirely on sand. Length between centres of abutment piers, 822·25 m., or nearly 2,700 ft. Total of sluiceways 1,968 ft. provided by 100 sluices of 6 m. widths and 6·1, 7·1, or 8·1 m. deep.

Spain.—The Duero Project. Progress is being made with works in this project, which includes the building of the Esla River Dam, height 300 ft. above stream bed. The estimate for the whole project is about £10,000,000.

India.—Progress of the Canvery Mettur Dam Works (see *The Engineer*, 1930, II. 262).
Perak River Hydro-electric Power Scheme (see *The Engineer*, August 22, 1930).

United States Irrigation—1890–1902: respectively—from the census reports, 1890, 1900, 1910, 1920. Number of irrigated farms: 54,136, 108,218, 159,801, 223,904. Acreages irrigated: 3,630,841, 7,536,390, 14,125,332, 18,592,888. Acres irrigated per farm: 67, 70, 88, 83.
Capital cost of irrigation system: millions of dollars—29·61, 67·77, 307·87, 676·41.
Costs per acre irrigated, dollars: 8·16, 8·89, 21·90, 36·40.

SECTION XXVI

PART III

RAINFALL — WELLS — WATERWORKS — DAMS — FLOW OF WATER IN PIPES—WATER PIPES—WATER TANKS.

RAINFALL.

(Contributed and revised by Reginald Ryves, Assoc.M.Inst.C.E., M.Cons.E.)

Rainfall in the British Isles.

For valuable data and studies relating to rainfall in the British Isles reference may be made to a paper on 'The Distribution of Mean and Extreme Annual Rainfall over the British Isles,' by H. R. Mill (*Min. Proceedings Civil Engineers*, vol. clv., 1903–1904); also to 'Variations in the English Climate during the Thirty Years 1881–1910,' by W. Marriott (*Quarterly Journal R. Met. Soc.*, July 1911). The former paper is accompanied by useful maps of rainfall distribution, and the latter by diagrams showing deviations from the mean values, month by month, for temperature, rainfall, number of 'rain-days,' and atmospheric pressure. Temperature and the number of rain-days have a considerable bearing upon water supply and sewerage schemes. Reference may also be made to a paper on 'The Abnormal Summer of 1911,' by C. Harding (*Quarterly Journal R. Met. Soc.*, January 1912).

The rainfall of each of the three divisions of the British Isles decreases from 100, 80, or 60 inches in the mountains and moorlands near the Western coasts to about 25 inches (over a few areas down to 20 inches) near the Eastern coasts of Scotland and England, and in the Midlands, and to about 35 inches on the East coast of Ireland. Over more than a third of England the rainfall ranges from 25 to 30 inches, and over a considerable total of smaller areas the fall is from 30 to 40 inches. There is a similar range, 30 to 40 inches, over about half Scotland, and a large area in the Western Highlands has a rainfall ranging from 60 to 100 inches. Over the Eastern half of Ireland the fall is from 35 to 40 inches, and over most of the Western half from 40 to 50 inches. For the period 1870–1899 the average annual falls were as follows: England, 31·62; Wales, 49·53; Scotland, 46·85; Ireland, 42·28; British Isles, 39·25. For the period 1881–1910 the average for England was 31·90.

Average, Maximum and Minimum Rainfalls.

It is generally agreed that a period of 30 to 35 years (*Binnie*) is long enough to establish for the average annual rainfall a value which will be nearly the true average, if such there be, and anyhow will be the mean of probabilities.

In the British Isles for the period 1870–1899 the rainfall of the 'wettest' year exceeded the average fall by 43 per cent., and that for the 'driest' year was 29 per cent. below the average (*Mill*). The following, expressed in percentages of the averages, are the figures for the different parts of the British Isles and for the whole area, taking in every case the wettest and driest years for the area referred to, period 1870–1899:—

England, 146 and 70; Wales, 147 and 75; Scotland, 147 and 72; Ireland, 133 and 69; British Isles, 143 and 70.

For the period 1881–1910 (*Marriott*) the percentage figures for England are 131 and 76·7. This period seems, however, to have been somewhat abnormal. The actual rainfalls compare thus:

England.	Maximum.	Average.	Minimum.
Period 1870–1899	46·98	31·62	22·83
Period 1881–1899	41·78	31·90	24·45

This suggests that although the 30 years' period may be satisfactory as regards the average annual rainfall it may not include a fall approximating to the maximum, nor one approximating to the minimum.

For the period 1881–1915 the results (inches) are as follows :—

	England and Wales.	Scotland.	Ireland.	British Isles.
October–March . .	19·61	29·2	24·28	23·42
April–September . .	15·62	21·12	19·02	17·99
Year . . .	35·23	50·32	43·30	41·41

At stations in the British Isles the following annual rainfalls in percentages of the average have been recorded : 58·8, 59·8, 61, 60·5, 63·6, and 64·5, showing defects from 35½ to 41 per cent. of the averages.

Mr. John Glasspoole, from 40 stations in the British Isles, for the 58 years, 1870–1927, furnishes a diagram showing the means of extremes for any period up to 12 months : for four months, min. 11, max. 71 ; for 6 months, min. 23, max. 95 ; for 8 months, min. 36, max. 114 ; for 12 months, min. 63, max. 154 ; these percentages based on the average rainfalls at the individual stations.

Sir Alexander Binnie considered the true mean rainfall to lie between the period of 33 and 35 years. From 14 stations scattered over the world he arrived at the following results :—

1. The percentage of years, in which the rainfall is above the mean rainfall, is 45·8.
2. The percentage of years, in which the rainfall is below the mean rainfall, is 54·2.
3. The average fall of wet years is 1·19 of the mean rainfall.
4. The average fall of dry years is 0·83 of the mean rainfall.
5. The average fall of the three consecutive dry years is 0·76 of the mean rainfall.
6. The maximum rainfall is 1·52 of the mean.
7. The minimum rainfall is 0·59 of the mean.
8. The proportional number of periods of three consecutive dry years in 100 years is 20·3.

The following figures are maximum annual rainfalls (for whole countries) expressed as a percentage of the average annual fall in each case (*Binnie*) :

Norway, Denmark, Holland and Belgium, 148 per cent. ; France, 161 ; Italy, 159 ; Switzerland, 147 ; Germany, 139 ; Bavaria and Austria, 144 ; Russia, 166 ; India, 162 ; Australia, 156 ; North America and Canada, 141 ; typical observations for the world, 151.

Rainfalls in India.

Normal Rainfalls, June–September. Inches.

Burma : 83·8 ; Bay Islands, 67·3 ; Lower Burma, 118·4 ; Upper Burma, 34·3. *Assam :* 61·1. *Bengal :* 60·9. *United Provinces :* 36·1 (East, 36·1 ; West, 36·1). *Bihar and Orissa :* 45·5 ; Orissa, 47·8 ; Chota Nagpur, 43·7 ; Bihar, 44·8. *Punjab :* 15·7 ; East and North, 19·9 ; South-west, 8·3. *Kashmir :* 6·2. *North-West Frontier Province :* 5·0 ; Baluchistan, 2·4. *Sind :* 4·7. *Rajputana :* 18·1 ; West, 10·7 ; East, 21·7. *Bombay :* 37·9 ; Gujerat, 22·4 ; Deccan, 21·9. *Central India :* 33·8 ; West, 28·5 ; East, 39·1 ; Berar, 27·4. *Mysore :* 15·5. *Malabar*, 74·2. *Central Provinces :* 40·5 ; West, 41·9 ; East, 45·0 ; Konkan, 88·8. *Hyderabad :* 26·7 ; North, 29·7 ; South, 24·5. *Madras :* 26·3 ; South-east, 10·8 ; Deccan, 16·3 ; Madras Coast, North, 22·4.

Normal Rainfalls, October–July. Inches.

Burma : 26·03 ; 7·98 in October, 11·1 in May. *Assam :* 33·05 ; 8·74 in April, 12·00 in May. *Bengal :* 21·83 ; 5·42 in October, 8·51 in May. *Bihar and Orissa :* 11·06 ; 3·65 in October, 3·00 in May. *United Provinces :* 5·42. *Punjab :* 5·47. *North-West Frontier Province :* 6·30. *Sind :* 1·40. *Rajputana :* 2·30. *Bombay :* 4·63. *Central India :* 3·65. *Central Provinces :* 6·05 ; 2·05 in October. *Hyderabad :* 6·23 ; 2·39 in October. *Mysore :* 15·02 ; 5·56 in October ; 2·59 in November, 4·20 in May. *Madras :* 23·59 ; 8·06 in October, 6·69 in November, 2·59 in December, 3·13 in May.

Notable Months' Normal Rainfalls. Sub-Divisions.

October : Bombay, Deccan, 3·11. *November :* Madras, South-east, 9·39. *December :* Madras, South-east, 4·72. *January :* Kashmir, 2·90. *February :* Kashmir, 2·63. *March :* Kashmir, 3·98. *April :* Kashmir, 3·36 ; Malabar, 3·43. *May :* Lower Burma, 14·73 ; Malabar, 8·71.

Rainfalls Elsewhere in the British Empire.

South Africa.—At 550 ft. above King William's Town, in the Amatola Mountains, head-waters of Buffalo River, average annual rainfall for 39 years to December 1925, 73·99 in.

Blijde River, 15 ins. Calitzdorp, less than 12 ins. Bon Accord, Transvaal, 28 ins., 21 ins. November-February. Van Ryneveld's Pass, Sunday River, about 14·5 ins.

The following are actual inches of annual rainfall, maximum, average and minimum, for a period of 24 years, in the Madras Presidency, 7 stations :—

70, 47, 31—78, 40, 17—48, 29, 9—60, 50, 17—43, 28, 17—165, 117, 87—182, 131, 95.

The Driest Group of Three Years Running.

Experience has shown that the most serious shortage in water supplies usually occurs as the result of the defect of rainfall in the driest group of three years running, and this consideration is usually one of the factors in the design of water supplies and irrigation works. The principle is to be accepted in all cases, inasmuch as there is, for every place, some period which is that towards the end of which the effects of a drought will probably be most severely felt.

The estimated average annual rainfall for the three years' period is sometimes put at 80 per cent. of the general average, though it is doubtful whether so high a percentage could be relied upon in any country. Sometimes it is put at 75 per cent. At Cape Town the average rainfall for the driest three years is 73 per cent. of the general average. In any case it must be remembered that the available run-off in the three years, as will be shown later, may be a much smaller percentage of the average available run-off. If, out of 50 inches of rain, 15 are lost by evaporation and absorption, we have 35 inches available. With a 40 inch, or 80 per cent. fall, we should have 25 inches available, or about 71 per cent., and the loss in the dry year might be greater than the average.

Run-off and Losses.

The available supply and the losses by evaporation and absorption are often expressed as percentages of the rainfall, but the amount of run-off or of loss bears no definite relation to the amount received by the catchment. The yield of a catchment area may better be reckoned as proportionate to the rainfall less some constant quantity which may be called the 'handicap' (*Lapworth*). This method is not, however, applicable to all cases, though in most cases it is suitable for the purposes of preliminary plans and may often give quite accurate results. The following are 'handicaps' for different areas (*Lapworth*):

Average in Great Britain, 10 to 12 inches; Thames Valley, 16 inches; parts of Dartmoor, nearly 18 inches; North-Eastern and Eastern parts of the United States, about 16 to 20 inches; in California, 23 inches or more; in India, up to 30 inches; in South Africa, about 25 inches. The mean annual loss from 29 streams in England ranged from 28 inches (loss) to a *gain* of 2 inches. Average about 14 inches; range of 18 of the 29, 11 inches to 17 inches. Small losses, or, in extreme cases, gains, may be the result of precipitation not recorded by rain gauges. In some catchments the dews are very heavy; in some there is often heavy condensation on trees and herbage, especially heather, during fogs or drifting mists.

It is necessary to find the particular 'handicap' in every case, since its amount necessarily depends upon topographical and geological features, upon the area of the catchment and the manner in which the water is collected. Further, it must be noted that the method is by no means generally applicable, and notably fails when the effect of one year upon the next is strongly marked. In the Ajamenta Valley, Gold Coast Colony, the following, in inches, were year's falls and the corresponding yields (*Hunter*):—

38·8, 1·800—59·9, 8·280—47·7, 2·540—42·4, 0·140—30·8, 0·560—29·6, 0·004—40·2, 0·006.

Again, if we take the figures for the Sweetwater catchment basin, California, we find that the following comparisons may be made (*Folwell*):—

A year's rainfall of 21 inches yielded 12 per cent., but a fall of 23·4 inches yielded only 9 per cent., while another year's fall of over 23 inches yielded less than 0·4 per cent. A fall of 17·2 inches yielded 4 per cent., but falls of 18 to 19 inches yielded less than a quarter per cent. The average fall of nearly 21 inches yielded less than 6 per cent., but an actual year's fall of 20 inches yielded 8 per cent.

The yield from the Coolgardie catchment, in a period of six years (*Palmer*) ranged from 0·20 per cent. to 3·5 per cent., while the rainfall ranged only from 19 to 33¼ inches.

In a period of 10 years the worst discharge of the Guadalquivir (Spain) was 9 per cent. of the rainfall, and in the best year it was 50 per cent. In the Karroo, S. Africa, a yield of 2 or 3 per cent. would be expected from a year's fall of 8 to 10 inches, and from a fall of 20 inches the yield would be about 8 or 10 per cent. (*Legg*).

Generally, in dry countries with a small rainfall the ratio of evaporation to the rainfall, or to a part of the rainfall, increases rapidly with a decrease in the actual amount of the fall. Conversely, in wet countries with a fairly large proportion of rainy or cloudy days the effect of relatively dry conditions or of scanty rainfall is not felt to nearly the same extent. In some cases it is found that the rainfall to be considered is that of only a part of the year, and it is then necessary to study maximum and minimum rainfalls, and the worst three years, as applying only to this season. In Natal about ⅔ of the rainfall occurs in the wet season, September to April. In Southern Rhodesia, about ⅓ from November to March. In other cases, especially when the area does not feel the full effects of any of the seasonal rains, it is necessary to examine the rainfall records in detail for the worst three years, and to eliminate all falls of less than some predetermined amount (single falls or day's rain). Next, the yield from each of a series of falls should be estimated as, in each case, a proportion depending partly upon the amount of the fall and partly upon the interval which had elapsed since the previous fall. The maximum percentage yield will be assigned to continuous rain on soaked ground and soaked subsoil. In dry areas it may be necessary to adopt such a method of estimating the yield, when, owing to the nature and distribution of the rains, no general rule can be applied to the whole rainfall of the year, or to that of a season. For a rough estimate it may be sufficient to eliminate small or

isolated falls and to calculate the yield of the remainder as a diminishing proportion of the remaining rainfall, less some constant. In some conditions the proportion need not be a diminishing one.

Hartebeestpoort Dam (near): altitude over 3,800 ft.—Monthly rainfalls and evaporations (approximate), inches: January, rainfall 4·99 and evaporation 6·51; and in that order, February, 4·11 and 5·65; March 3·31 and 5·07; April 0·80 and 5·13; May 0·69 and 5·10; June 0·06 and 4·35; July 0·27 and 4·91; August 0·73 and 6·60; September 0·48 and 7·91; October 2·26 and 8·41; November 4·66 and 7·63; December 3·98 and 7·40; the year 26·34 and 74·67.

Absorption and Evaporation.

While it is possible, by the exercise of judgment, to form some idea as to the extent of losses by absorption for a particular area, it is clear that the large number of factors involved makes it impossible to lay down any general rules. Under some circumstances all the rain runs off, under other circumstances it is all absorbed, and there is every conceivable intermediate condition.

A few points are, however, worth noting. Clay soils, after the surface is wet, absorb water only very slowly. Some light soils are absorbent up to a given rate of absorption and rainfall in excess of that rate runs off. If the water absorbed by the soil percolates to places where it is available for use, the loss by evaporation from the soil will largely depend upon the average depth of the subsoil water. A dusty surface to a great extent prevents evaporation from the soil.

In practice, the loss from evaporation and percolation taken together is found by gaugings of the streams from the catchment, or, provisionally, is based upon data obtained from similar catchments, with such modifications as seem to be desirable owing to differences in the conditions. The estimate of the loss by evaporation and absorption, obtained from one set of data, should agree with the estimate of total yield based upon another set of data, or upon general considerations. References to evaporation from free water surfaces and from soils will be found under 'Irrigation,' pp. 1383, etc.

Dr. Walter Leather gives the following formula for estimating evaporation from water surfaces:—

$$E = 2{\cdot}0\ (\log t - 1{\cdot}74) + 0{\cdot}33\ (\log D - 1{\cdot}00) + 0{\cdot}36\ (\log W - 0{\cdot}125)$$

where,

E = Evaporation in millimetres for 24 hours; D = Dryness = 100 − (humidity at 8 A.M.); W = Mean wind velocity for the 24 hours in miles per hour; t = Mean temperature in degrees F. for the 24 hours.

Rainfall Quantities and Rates.

One inch of rain is equal to:—

2,323,200 cubic feet per square mile; 14,485,000 gallons per square mile;
3,630 cubic feet per acre; 22,633 gallons per acre.

One inch of rain per hour is at the rate of:—

645 cubic feet per second on a square mile, nearly 1 (1·01) cubic foot per second on an acre;
241,419 gallons per minute on a square mile;
377 gallons per minute on an acre.

I. Table showing Equivalent Rainfall in Cubic Feet per Minute.

Areas in Acres. (A_1)	Rainfall in Decimals of an Inch in 24 Hours. (r)								
	0·1	0·2	0·3	0·4	0·5	0·6	0·7	0·8	0·9
1	0·2521	0·5042	0·7562	1·1083	1·2604	1·5125	1·7646	2·0170	2·2687
2	0·5041	1·0084	1·5125	2·2166	2·5208	2·0250	3·5290	4·1340	4·5370
3	0·7563	1·5126	2·2687	3·3249	3·7812	4·5370	5·2930	6·0510	6·8060
4	1·0084	2·0168	3·0249	4·4332	5·0416	6·0500	7·0580	8·0680	9·0740
5	1·2605	2·5210	3·7812	5·5416	6·3020	7·5620	8·8230	10·0800	11·3430
6	1·5126	3·0252	4·5374	6·6499	7·5610	9·0750	10·5870	12·1000	13·7200
7	1·7646	3·5294	5·2937	7·7583	8·8820	10·5870	12·3500	14·1100	15·8600
8	2·0166	4·0336	6·0499	8·8666	10·0800	12·1000	14·1100	16·1300	18·1500
9	2·2687	4·5378	6·8061	9·9749	11·3400	13·6120	15·8800	18·1500	20·4200
10	2·5208	5·0416	7·5625	11·0833	12·6000	15·1250	17·6400	20·1700	22·6900

Cubic feet per minute = $A_1 r \times 2{\cdot}521$.

II. Table showing Equivalent Rainfall in Cubic Feet per Minute.

Area in Square Miles (A_2)	Rainfall in decimals of an inch in 24 hours. (r)								
	0·1	0·2	0·3	0·4	0·5	0·6	0·7	0·8	0·9
1	161·3	322·6	484·0	645·3	806·6	968·0	1129·3	1290·6	1452·0
2	322·6	645·3	968·0	1290·6	1613·3	1936·0	2258·6	2581·3	2904.0
3	484·0	968·0	1452·0	1936·0	2420·0	2904·0	3388·0	3872·0	4356·0
4	645·3	1290·6	1936·0	2581·3	3226·6	3872·0	4517·3	5162·6	5808·0
5	806·6	1613·3	2420·0	3226·6	4033·3	4840·0	5646·6	6453·3	7260·0
6	968·0	1935·9	2904·0	3872·0	4839·9	5808·0	6776·0	7743·9	8712·0
7	1129·3	2258·6	3388·0	4517·3	5646·6	6776·0	7905·3	9034·6	10164·0
8	1290·6	2581·3	3872·0	5162·6	6453·3	7744·0	9034·6	10325·3	11616·0
9	1452·0	2904·0	4356·0	5808·0	7260·0	8712·0	10164·0	11616·0	13068·0
10	1613·3	3226·6	4842·0	6453·3	8066·6	9680·0	11293·0	12906·0	14523·3

Cubic feet per minute = $A_2 r \times 1613{\cdot}7$.

Flood Discharges.

For every area, however small or however large, there is some period of time at the end of which the effect of continuous rain reaches the maximum. What this period is depends upon all the circumstances, the size of the area in question, the nature of the rainfall, the physical character of the catchment, and the nature of the vegetation or of artificial works all affect the result.

In most cases the maximum capacity of the catchment for surface water is reached only for short periods after the heaviest rushes of rain, and the amount of such storage affecting the result is the average over a relatively long period, and cannot be estimated with any degree of accuracy. A period longer than the critical period will give a lower rate of discharge, because the rate of rainfall will be less, and a period shorter than the critical period will give a discharge lower than the maximum, because the water from the more distant parts of the catchment will not have time to reach the outlet. It is usually necessary to have recourse to empirical formulæ, or formulæ partly empirical and partly rational, and the engineer has to exercise judgment in choosing one coefficient affecting the whole quantity or several coefficients affecting different factors. In some large river basins the effects of short storms over several tributary valleys may, by the simultaneous arrival of the resulting floods, yield a larger discharge than that due to the lower rate of rainfall for the critical period corresponding to the whole area. In such cases the purely empirical formula may be better than the rational formula, because the engineer makes a note of every real coefficient established by experience, and all these coefficients are strictly comparable.

Further, in the case of a steep, rocky valley, the river of which drains flat uplands, or glens under dense jungle, the heavier rainfall for the shorter 'period of total contribution' for the main valley may give a larger flood discharge than the smaller rainfall for the longer period of total contribution for the whole catchment. Even in normal river basins, areas projecting beyond the average distance of the watershed of the distant parts of the catchment may be eliminated, the loss in area being more than made up for by the gain in rate of rainfall for the reduced period of total contribution (*Ryves*).

The following formulæ are used in calculating flood discharges of rivers and, in some cases, are also employed for quite small catchments.

Ryves' formula—

$$Q = CM^{\frac{2}{3}}.$$

Q = discharge in cu. ft. per sec.; M = area in square miles, C is a coefficient usually ranging from 450 to 1,000, according to the nature and situation of the catchment; rainfall data are also taken into consideration, by comparisons with established results.

This formula is largely used in Southern India. (Discharge in cu. metres per sec. = the same coefficient $\times \frac{15}{1{,}000} \times$ (area in sq. kiloms.) $^{\frac{2}{3}}$).

For a considerable range of catchment areas the coefficient 1,000 is a good provisional figure; it applies to many parts of the tropics in hilly country. For small catchments in such districts the discharge may correspond to higher values, up to 2,000 for the maximum in an indefinite period of time. But the site of the Gatun dam, catchment 168 sq. m., the maximum flood, estimated from records, corresponded to a coefficient of 1,525.

Dickens' formula—

$$Q = 825\ M^{\frac{3}{4}}.$$

This is the form in which the equation is usually presented, but it is doubtless advisable to vary the coefficient as in the case of the Ryves' formula.

The Dickens' formula is probably more particularly suited to the conditions of Bengal and other parts of Northern India, but both formulæ may be usefully employed in other countries.

The formula $Q = CM^{\frac{3}{4}}$ has been found to apply, for *ordinary floods* in the New England and Appalachian catchments, with a coefficient C = 130 to 200; in Eastern Middle State river basins, with a coefficient from 60 to 100; and for Western tributaries of the Mississippi, with coefficients from 12 to 50 (*Fanning*).

Chamier's formula—

$$Q = 640 \times R \times C \times \frac{M^{\frac{3}{4}}}{M}.$$

R = average rate of greatest rainfall in inches per hour for such duration as will allow of the flood-water flowing to the outlet from the farthest extremity of the catchment. C = coefficient of surface discharge, giving the proportion of rainfall that may be expected to flow off the surface, M = area of catchment in square miles.

This formula seems to have given good results, and by comparing the calculated discharges for many catchments with those actually observed it should be possible to arrive at some idea of the figures for the rates of flow, from watershed to outlet, upon which the period of total contribution depends, and from which, therefore, the rate of rainfall is derived. Since this factor cannot, in comparison of results, be disentangled from the factor C, it will take longer to extend the use of the formula than it does in those cases in which the exercise of judgment and the results of practice focus on to a single coefficient.

In order that the length of the catchment might be taken into consideration the following formula was proposed:

Burge's formula—

$$Q = C \cdot \frac{M}{(L)^{\frac{2}{3}}},$$

where L is the length of the main river or stream in miles; C is usually taken as 1,300.

A modification of this formula has been proposed, as follows (*Jackson*):—

$$Q = C \times \frac{B}{L} 100\ M^{\frac{3}{4}},$$

where L is the extreme length and B the extreme breadth of the catchment area.

It may be suggested that when it is possible to make a fair estimate of the period of total contribution, and, therefore, of the maximum rate of rainfall, such an estimate should form the basis of the calculation for maximum flood discharge, and when no such estimate can be made an empirical formula with a single coefficient should be used.

Storm-water Discharge of Sewers.

The same principles that govern the discharges of large river basins hold in the case of the storm-water discharge of sewers, but it is usually much easier to estimate the average rate of flow from the most distant parts of the catchment (measured in time) and, therefore, to estimate the period of total contribution, the rainfall for which can be found from local records.

The following are formulæ employed:

Bürkli-Ziegler's formula—

$$Q = R \times C \times \sqrt[4]{\frac{S}{A}}.$$

Q = discharge in cubic feet per second per acre.
R = average rainfall in cubic feet per second per acre 'during heaviest rainfall.'
C = 0·75 for paved streets; 0·31 for rural districts; average, 0·625.
A = area of district drained in acres.
S = general fall of area per thousand.

It is clear that R will depend upon the period of total contribution which will have to be found in each case in order that the formula may be applied. The factor C would, as regards unpaved areas at any rate, also depend upon the period of total contribution.

Kuichling's formula—

$$Q = A \times a \times t \times (b - c \times t).$$

A = area drained, in acres; a, b, c, empirical constants which vary for different localities; t = time required for the concentration of storm water at the outlet.

This formula seems needlessly complicated, since if t be known the rainfall can be found and a single modifying coefficient could be applied to it, as well as one to be applied to the quantity as a whole, if considered necessary.

Lloyd-Davis' formula—

$$Q = \left(60{\cdot}5 \times \frac{60}{Tc} \times R\right) \times Ap.$$

Q = cubic feet per minute; Tc = time of concentration in minutes (time of flow through the longest sewers) R = rainfall in inches during Tc; Ap = percentage of impermeable area in acres.

Having found the period of total contribution, the maximum rate of rainfall for that period must be found from local rainfall records. Thus, a curve drawn for records of heavy falls for a district in Southern India gave the following results (*Ryves*):—

Heaviest fall in 30 mins., 2·0 ins.; in 60 mins., 3·0 ins.; in 85 mins., 3·8 ins.; in 104 mins., 4·2 ins.; in 134 mins., 4·8 ins.; in 150 mins., 4·9 ins. Such curves are not necessarily drawn for the heaviest 'spot' falls, which are usually due to cyclones traversing a very narrow path. The maximum spot rainfall may be much greater than the maximum for the storm-water area of a large town. Even when the maximum rainfall does affect the whole area it is not always worth while to design the works for this fall. For instance, in some places cyclones are rare, and the maximum falls may greatly exceed the largest intervening falls, for periods of many years, being perhaps 4 or 5 times greater. The same principle may hold with respect to large areas, and the cost of works to deal with another fall such as that which occurred in East Anglia in August 1912 or with floods such as those experienced in Paris in 1913, may be so great that it is better to risk the chance of a similar visitation.

Heavy Single Falls of Rain over Considerable Areas in the British Isles.

London, whole area, 2·5 ins. in 24 hours.
Lake District, 714 sq. miles, 5·63 ins.
Eastern Ireland, 1,160 sq. miles, 4·76 ins. } single fall in less than 3 days.

Aug. 1912.

East Anglia	18	sq. miles, over	8	ins. in about	24	hours		
"	"	238	"	7	"	"	24	"
"	"	705	"	6	"	"	24	"
"	"	1,039	"	5	"	"	24	"
"	"	3,463	"	3	"	"	24	"

Before 1912 the maximum fall for 24 hours recorded in Norfolk was 4·48 ins.

Heavy Station Rainfalls in the British Isles.

Inches in 24 hours: ‡Bruton, Somerset, 1917, 9·56; Cannington, Somerset, 1924, 9·4; Loch Quoich, Inverness-shire, 8·40; Seathwaite, 8·03 and 7·00; Ben Nevis, 7·74 and 7·29; Rosthwaite, 6·94; Angerton Hall, 6·70; Lyme Regis ('in the night,' July), 3·58 ins.; Ballycumber, 3·8 ins. For shorter periods:—Norwich, 0·52 in. in about 5 mins.; Dover, 0·44 in. in 12 mins.; Margate, 0·51 in. in 20 mins.; Hellingley, Sussex, 1·21 ins. in 50 mins., and 0·73 in. in 12 mins.; Oxted, 2·14 ins. in 24 hrs.; East Grinstead, 1·80 ins. in 1½ hrs.

Banstead, 3·59 ins. in about 90 mins.; Wallington, 2·77 ins. in 65 mins., and 3·50 ins. in 60 mins.; Wimbledon, 1·58 in. in 80 mins.; Croydon, 1·85 in. in 2 hours 48 mins., and 0·8 in. in 15 mins.; Ponders End, 1·05 in. in 19 mins.; Birmingham, 1·04 in. in 27 mins.; Burnham, 1·73 in. in 40 mins.; a fall of 1·2 in. in 15 mins. seems to be the quarter-hour record for the British Isles.

Sheffield, August 4, 1922, 2·45 ins. in 18 hours, 1·04 in. in 2½ hours, and nearly 1·0 in. in 30 mins. Great Britain, February 11, 1927: at 10 stations the falls exceeded 2 ins. in 1 hour; Balham High Road, 5 ins. in 1 hour (report of L.C.C. main drainage committee).

Heavy Station Rainfalls, Canada and United States.

Toronto, 0·5 in. in 10 mins.; and 0·6 in. in 15 mins.; New York, 1·2 in. in 10 mins.; Philadelphia, 1·6 in. in 20 mins.; Chicago, 0·97 in. in 7 mins.; Taylor, Texas, September 1921, 23 ins. in 24 hrs.; July 1916, at 12 stations in the U.S., more than 10 ins. in 24 hrs.

Heavy Station Rainfalls in or near the Tropics.

Central America, 0·6 in. in 10 mins.; 1·1 in. in 30 mins.; 1·7 in. in 60 mins. Panama Isthmus, 6 ins. in 95 mins.; 4·9 ins. in 60 mins.; 5·86 ins. in 60 mins. Philippines, 27 ins. in 24 hours, also (1913) 48 ins. in 48 hours, and 24 ins. in 8 hours (same storm). Manila, 17 ins. in 15 hours. Near Johannesburg, 3·2 ins. in 15 mins. Near Durban, 15·6 ins. in 15 hours; 17·6 ins. in 24 hours; 6 ins. in 3 hours. Mauritius, 18 ins. in 24 hours, and the mean of 15 stations 9·86 ins. in 24 hours; 99 ins. in 6 days; 24 ins. in 24 hours. Ceylon, 21 ins. in 24 hours; 8 ins. in 4 hours.

In 2 days: Santa Cruz, Teneriffe, 12·58 ins.
In 24 hrs.: Newlands, Cape Peninsular, South Africa, 16·5 ins.
In 12 hrs.: Beaumont, Texas, 13·54 ins.
In 8 hrs.: Santa Cruz, Teneriffe, 3·47 ins.
In 4½ hrs.: Beaumont, Texas, 12·76 ins.

Heavy Station Rainfalls in India.

Madras, 4·95 ins. in 3 hours; 2 ins. in 18 minutes; 3 ins. in 60 minutes, 10, 11, 12, 13, 18, and 20 ins. in 24 hours. Madras Presidency (South Arcot), 7, 8, 9, and up to 11·3 ins. in 24 hours. Central Provinces (Talgaon), 7·75 ins. in 6 hours. West Coast, Bombay Presidency, 8 ins. in 2 hours. Bombay Town, 7 to 9 ins. in 24 hours. Sholapur, 10 ins. in 7 hours. Calcutta, Alipore Observatory, May 6, 1928, 3·25 ins. in 40 mins. (record intensity for this station or 4·88 ins. per hour); the whole fall during this storm being 3·39 ins.

In 24 hours: Karwar, Konkan, 13 ins., June; Hukitala, Orissa, 15 ins., July; Burdwan, Bengal, 10 ins., June; Sangor Island, 10 ins., June; Kyankpyu, 9 ins., June. In 48 hours: Bombay, 17 ins., June. In 72 hours: Cherrapanji, 31 ins.

Forests and Rainfall.

A point which is often overlooked is the effect of trees in increasing the total precipitation considerably beyond that recorded by rain gauges. A large proportion of the rime which collects on the twigs of trees in frosts afterwards reaches the ground as water, and, in climates such as those of the British Isles, the total amount of water deposited on the twigs from fogs and drifting clouds is considerable, and most of it reaches the streams or underground storage, or at least replaces losses from subsequent rainfall.

Of more importance, however, to hydraulic engineers is the effect of woodlands in modifying the run-off. The rush of water from bare hillsides is exchanged for the slower delivery from the matted carpet of the woodland, losses by evaporation may be much diminished and the melting of snow usefully retarded. In catchments from which flood waters are largely lost, woodlands may increase the available run-off by extending the period of surface flow. The maximum floods of rivers are reduced, and the lowest summer flow increased. Woodlands are usually much more effective than minor vegetation, such as gorse and heather, in preventing the soil from being carried from the land into an open reservoir.

To protect a reservoir from silting, it may be unnecessary to plant large areas, the silt being arrested by suitable planting of narrow belts of woodland, or by the protection of natural growth, along the margins of the streams.

Some engineers consider that in the case of small reservoirs the shelter afforded by a belt of trees along the margins is of value in reducing the amount of scour of the banks caused by wave action. Afforestation over considerable areas in large river basins would, in many cases, reduce the amount of silting in navigable rivers and estuaries.

A matter which does not receive sufficient attention in connection with hydraulic engineering is the effect of judicious planting or woodland conservation over small areas. A narrow belt of woodland along the foot of a slope will arrest the soil brought down by rains from the hillside. The encouragement of dense vegetation along the bottom of a narrow valley or glen may check the rate of flood discharge to a useful extent. The planting of suitable trees along ridges, and for a little way down the slope facing the rain bearing and damp winds, will produce the maximum of certain desired effects, in proportion to the area occupied. Suitable tree and bush growths in swampy areas and around their margins will increase their effect in checking flood discharge, and may prevent these areas from contributing large quantities of silt to the streams during very heavy rains. Areas of soft, cultivable soil liable to denudation may similarly be protected. Generally, a country which is, in the ordinary English sense of the words, ‘well-timbered’ is, from the point of view of the hydraulic engineer, a favourable country; and in the development of new lands the future effects of a proposed agricultural policy should be considered from this point of view, and in consultation with hydraulic engineers.

Rain Gauges.

The smallest diameter for a serviceable rain gauge is 5 ins., and is the size generally used. Trials with gauges of various diameter ranging from 1 in. to 2 ft. have shown that if they are set perfectly level, and observed with great care, exactly the same rainfall has been registered by all of them.

The standard height above ground for rain gauges is 1 ft.

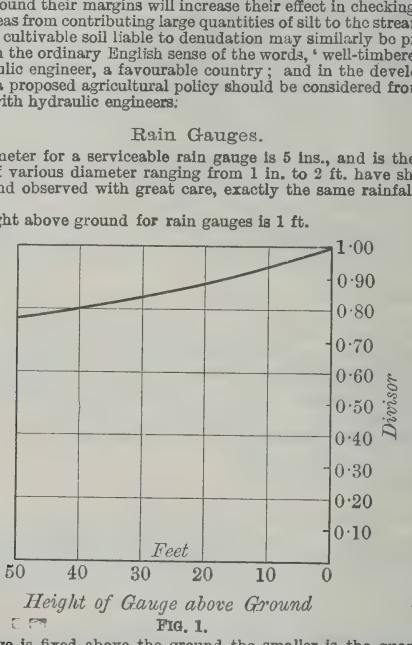


Fig. 1.

The higher a gauge is fixed above the ground the smaller is the quantity of rain caught owing to the eddies carried by the wind about any prominent upstanding object. The curve diagram (fig. 1), given in Beardmore’s Hydrology, gives the divisor to be used for increasing the

quantity of rain observed at any given height above ground level, so as to make the observation equal to that at ground level.

It is more important to have rain gauges placed at a uniform height above the ground than to have them of uniform diameter.

Available Power from Rainfall.

In terms of *average* values of rainfall, catchment area, and head, the available power can be expressed :—

$$\text{H.P.} = 0{\cdot}00828\, f\, m\, h\,;\ \text{E.H.P.} = 0{\cdot}0073\, f\, m\, h\,;\ \text{kW} = 0{\cdot}0062\, f\, m\, h\,;$$

where

f = average rainfall in inches ; m = catchment area in sq. miles ; h = average head in ft.

Example.—The catchment of the Lochaber water-power scheme is 303 sq. miles ; the average annual rainfall is 73 in. ; and the average head of water available near Fort William is 742 ft. What will be the theoretical power ?

$$\text{H.P.} = 0{\cdot}00828 \times 73 \times 742 = 136{,}000 \text{ H.P.},$$

and, as 1·0 H.P.-year = $1 \times 8{,}760 \times 0{\cdot}746 = 6{,}535$ kW-hours, the output (theoretical) would be

$$6{,}535 \times 136{,}000 = 888{\cdot}76 \textit{ million kW-hours.}$$

(*W. T. Taylor*, '*The Electrical Review*,' *October* 12, 1923.)

WELLS.

The back of the steining should be puddled to a depth of 12 or 15 feet, so as to make the water pass through a considerable amount of soil before entering the well.

The ground around should be puddled, sloped away from the well, and paved, and the lining carried up about 2 feet above the ground level, to keep all dirty water out of the well. The mouth should be covered over, and the water drawn by a pump rather than a bucket.

Form of Wells.—Generally circular, but when there is no steining or a large water-yielding surface is required, as in headings driven from the shaft, a rectanguiar, elliptical, or other form may be used. Ordinary wells have a clear diameter of from 4 to 5 feet, but when very deep may require to be made large enough at the top to allow of reductions as the depth increases, still leaving room enough below for any pumping machinery.

Temperature in deep wells increases about 1° F. for every 55 feet in depth below 60 feet, at which depth it is pretty uniform at 50°.

Tubing for boreholes is sometimes of cast iron, the lengths being held together by wrought-iron collars fitting over the ends and flush with the outside of the tubes.

The approximate average cost (pre-war prices) in England may be taken at 1*s.* 3*d.* per foot for the first 30 feet, 2*s.* 6*d.* per foot for the second 30 feet, and so on, advancing 1*s.* 3*d.* per foot for every additional 30 feet in depth. This does not include cost of tubing, conveyance of plant and tools or working in rock of unusual hardness. (See also Artesian Wells, page 1409.)

STEINING.

This is lining the well, chiefly to keep the sides from falling in. Where bricks are not available stone may be used, and even timber, but the latter soon decays. Iron cylinders are also employed.

Brick Steining.—The bricks should be hard, well burnt, and well shaped. Malm paviors are very good ; if stocks are used, they should be very carefully picked. The bricks are either laid dry, called '*dry steining*,' or in cement, or good hydraulic mortar, where strength is required or to keep out water.

In cement work cast-iron curbs are sometimes built in at suitable intervals, to facilitate the execution of the well, and also to strengthen the work.

In dry steining one or two courses set in cement are introduced for strength at intervals varying mostly from 5 to 12 feet.

The bricks are laid flat, breaking joint. The thickness of the steining varies with the depth and diameter of the well and the firmness of the ground ; from 9 inches for diameters up to 9 feet, to 14 inches for diameters up to 14 feet, with an extra half-brick in very bad or loose soils. The best method of laying them is in half-brick rings, with headers at intervals, tying them together. Courses of headers are objectionable, unless the diameter of the well is large.

To exclude land or other springs not under very great pressure, concrete or clay puddle is introduced behind the steining ; clay puddle is the more certain to be watertight. When the bricks are laid dry, the swelling of the earth behind will gradually fill up the spaces between them. It is very essential that the steining should be well backed up by working as close as possible to the sides of the excavation ; and wherever, from any irregularity in the excavation, this is not convenient, the space should be well filled in either with puddle or brickwork ; the latter is preferable, especially where a slip of earth has taken place, in which case the brickwork should be laid in cement. The principal object is to prevent the steining failing by being forced in at one part and out at the other by the pressure of the ground.

The ground round a well should be waterproofed for 30-feet radius of a circle about the well. This ensures that no water can even reach the back of the steining, which has not passed through a considerable bulk of earth.

Well-Sinking.

There are two methods of sinking and steining wells, the first adapted to dry or firm, and the second to wet or loose, ground.

First Method.—The excavation being sunk to a safe depth, a curb or ring of timber is placed at the bottom, formed in two or more thicknesses and lap-jointed to equalise its strength, cut circular on the inside but not on the out, of a breadth sufficient to receive the brick-lining of the well, and not less than 3 inches thick. On this, truly concentric with a central plumb-line, the brick lining is built up to the surface, the earth or a puddling of clay being punned in hard round the outside as the work rises. The excavation is now resumed in the centre, leaving enough earth undisturbed under the curb to support the work done. When a safe depth is again attained, the earth under the curb also is often removed in four equidistant places, the duty of support devolving temporarily on the intermediate portions left as piers; and from a broad central footing of timber 4 raking struts are set up and firmly fixed, to support the curb in the points from under which the earth was taken. This being done, the piers of earth are dug away, and the bottom is then, as before, made level around as a bed for a second curb; and thus the work proceeds in stages down to the required depth, the projecting angles of the curbs outside, by their penetrating the soil, contributing to the support of the brickwork in its progressive extension downwards. Sometimes the curbs, instead of being propped from below, are suspended by cross-timbers from the mouth of the well by iron rods.

Second or Older Method.—The surface of the ground having been levelled, or an excavation made to a safe depth, an iron curb with a sharp lower edge, or a wood curb shod with iron, with an outer diameter equal to that of the brickwork cylinder, is laid on the surface or in the excavation, and as the brick steining is built upon it, it sinks by its own gravity, the bricklayer continuing building up at the surface, while the digger inside descends, the soil being hoisted out as fast as excavated. When from the swelling of the ground the descending work becomes earth-bound, the process has to be recommenced at its base to a smaller diameter.

Iron Cylinders are sunk as just described for brick cylinders. They are mostly inserted in order to continue the well when the brick steining has been interfered with by running sands or the irruption of water. Until the water-yielding stratum has been passed through, the sinking of the cylinders is carried on with the assistance of the ordinary boring tools. The cylinders are generally of cast iron, made up of 5 or 6 feet lengths, bolted together by means of internal flanges, leaving the exterior surface perfectly smooth. The bottom length has, instead of a lower flange, a cutting edge to facilitate the descent. The joints are made watertight, either with gasket soaked in tallow or by being caulked with iron cement. Wrought-iron cylinders used for steining are jointed with angle and T iron by rivets countersunk on the outside, so as to leave a smooth surface. When the cylinders become earth-bound they are loaded, or else pressed down by jacks or rammed down by wooden 'dollies.'

The Number of Bricks and Quantity of Brickwork in Wells and Cylindrical Sewers for each Foot in Depth or Length.

Diameter of Well, in Feet.	Half-Brick Thick.			One Brick Thick.		
	Number of Bricks.		Cubic Feet of Brickwork.	Number of Bricks.		Cubic Feet of Brickwork.
	Laid Dry.	Laid in Mortar.		Laid Dry.	Laid in Mortar.	
1·0	28	23	1·6198	70	58	4·1233
1·3	33	27	1·8145	80	66	4·7124
1·6	38	31	2·2089	90	74	5·3015
1·9	43	35	2·5035	102	82	5·8905
2·0	48	41	2·7979	112	92	6·4795
2·3	53	44	3·0926	122	100	7·0686
2·6	58	48	3·3870	132	108	7·6577
3·0	68	57	3·9760	154	126	8·8357
3·6	79	65	4·5651	174	142	10·0139
4·0	89	73	5·1541	194	159	11·1919
4·6	100	82	5·7432	214	176	12·3701
5·0	110	90	6·3322	234	192	13·5481
5·6	120	98	6·9213	254	209	14·7263
6·0	130	107	7·5103	276	226	15·9043
6·6	140	115	8·0994	296	242	17·0825
7·0	150	123	8·6884	316	260	18·2605
7·6	160	131	9·2775	336	276	19·4387
8·0	170	140	9·8665	358	292	20·6167
8·6	180	148	10·4556	378	308	21·7949
9·0	191	156	11·0446	398	326	22·9729
10·0	212	174	12·2227	438	360	25·3291

Quantity of Water in Wells.

To ascertain the quantity of water in a well, take half the circumference (in the clear) and multiply by half the diameter; multiply the result by the depth, which gives the cubic measures, then reckon 6 gallons and 1 pint to the foot cube.

To Increase the Yield from a Well already Executed.

1. The water-level in the well may be lowered by pumping, etc., by which the hydraulic depth for the inflow is increased.

2. The bore-hole may be deepened, thus probably leading to denser strata, through which there would be less percolation, and thus more yield to the well.

3. Headings may be driven laterally from the well; the best level for them is just above the lowest level to which the water is ordinarily pumped; in plan they should be straight lines through the well.

4. Dynamite or other explosive is lowered down to the bottom of the bore-hole, and then exploded by electricity.

Protective Distance from Drains.

A well should not be closer to a drain than 4 feet for every foot in depth; thus a 20-foot well should be at least 80 ft. from any possible source of corruption. Wells 100 ft. deep, if with water-tight sides, may be relied upon as safe from leakage contamination, as this depth of soil is capable of removing a greater amount of organic matter that may exist in the water.

Prof. Frankland gives 200 yards from a cesspool, cemeteries, etc., as the nearest safe distance; in Italy, 100 yards; in France and Austria, 200 yards. But much will depend on the dip and nature of the strata.

Artesian Wells.

In the best modern practice, artesian wells are lined with stout steel-socketed wrought-iron or steel pipes, to a point at least lower than the lowest water level. In chalk bores the pipes are lightly driven at least 10 feet into the chalk—or further, if necessary—the projection of the sockets being very effectual in securing the shutting off of the upper, and perhaps undesirable, waters from the lower.

No pipe should be less than $\frac{1}{4}$ inch thick, from 3 to 6 inches diameter, up to $\frac{3}{8}$ inch thick for pipes of 10 to 16 inches.

When the water level is considerably below surface, the yield of a well is measured by the capacity of the largest pump that can be put down the bore tube. Deep-bore pumps consist simply of a plain barrel hung in the water by a wrought-iron rising main pipe of slightly greater diameter. The bucket is hung by rods inside the rising main, and these project through a stuffing-box in an outlet casting on the top of the main, and are actuated by a crank or other means to give a stroke of 18 to 24 inches in ordinary cases. A plunger is added in the outlet casting for forcing to considerable heights above surface, and a balance lever should be connected to the cross-head by links.

The best water-bearing strata in England is the Lincolnshire (Oolitic) limestone, which contains many spouting copious wells; notably one at Sleaford, only 6 inches diameter, which overflows at 30,000 gallons per hour 3 feet above surface. The chalk is very free bearing in many parts, but capricious and occasionally nearly dry. The Keuper marl yields good brewing water at Newark, Burton, Leicester, etc., but not very copiously, and apt to be very small in amount and capricious in character. The Keuper sandstone is a free yielding stratum, and gives good soft water. The Permian sandstone is also a fair and at times a copious water-bearing rock and so also is the magnesian limestone.

The Lower Greensand sometimes yields copiously, and, as a rule, the more copious the supply, the better the quality. It is at times ferruginous. The Lias clay is apt to produce saline water; so also is the London clay. The Upper Tertiary beds of the London basin below the London clay often yield fair but slightly alkaline supplies.

The carboniferous rocks vary greatly, and often give copious supplies, and but few rocks are quite dry. As a rule, the more copious yields are all within 200 feet of the surface. Deeper boxes yield less and less. Water has been obtained from the Lower Greensand in Windsor Forest at 1,243 feet below surface, the quantity being small, the quality good, and the water rising to about 230 feet above sea level, or overflowing the surface nearly 8 feet.

For borings in rock having a surface outcrop near the site of the boring, the water level will usually be little or nothing above the nearest watercourse crossing such outcrop, but where borings penetrate an impermeable surface, and reach a rock having a distant outcrop, the water level will be determined by such outcrop, and, though lower than the receiving surface, will be higher than a distant draining point, if any.

It is by no means unusual for well borers to pass through water and declare it to be absent. They will also continue boring into strata that are hopeless as water-bearing strata under the impression that they have not yet passed through the upper stratum. Thus in places the Gault way rests upon the Kimmeridge clay, and it is difficult to distinguish one clay from the other. Well boring should only be carried out under educated inspection, and regular samples of everything passed through should be kept, and records of the water level every morning and of any change during the day. Small changes in the material and variations of the water level are often very instructive and significant to the hydrogeologist.

Wells in the lower chalk formation are frequently failures, owing to failure of the workmen to recognise the evidences about them.

While there can be no good done by carrying wells to excessive depths, it is a common experience to find wells that have been sunk only to their high-water mark horizon, so that when the water level sinks the well is dry. This is a frequent occurrence in the lower chalk, in which there is a very wide difference between the highest and the lowest water levels. Wells sunk in high-water periods invariably run dry when the lower-water period approaches.

For the varying strata of the British Isles, and the usually small depths required, the ordinary system of boring with chisels and rods appears to be as cheap as any other. For depths beyond 500 feet the rope is often used. No one system can be adhered to, but rods, ropes, and the diamond must be employed, according to the requirements of each case.

More recent practice is to bore by means of the rotary chilled steel shot system. A plain thick steel pipe is notched at its lower end and rotated upon the rock to be pierced. A quantity of chilled steel shot is fed down the pipe. This gets under the lower end of the pipe by way of the notches, and, rolling on the rock under heavy pressure, comminutes by the intensity of the rolling crushing action. Water forced down removes the groundaway rock, and good cores are obtained. Several feet per day may be bored in hard limestone. The method involves no loss of costly diamonds.

To calculate the rate of flow of an artesian well, a simple plan is to lower a bottle of aniline fluid to a depth of, say, 500 feet, and then electrically explode a cap to burst the bottle. The time required for the fluid to appear at the surface gives an accurate gauge as to the velocity of flow. It is claimed that this method gives results as accurate as a weir. The diameter of the pipe being known, the rate of flow readily follows. (*W. H. Booth.*)

WATERWORKS.

(Revised by R. Fairfax Middleton, Assoc.M.Inst.C.E.)

MASONRY DAMS.

Rankine gives for low masonry walls to retain water that the thickness at the base should be about $\frac{4}{10}$ of the height either for rectangular walls or for those of trapezoidal form where the thickness at the top is $\frac{1}{4}$ that at the base.

Another rule for low dam walls is: Thickness at bottom 0·7 height; thickness at middle 0·5 height; thickness at top 0·3 height.

For high masonry dams Molesworth gives H = height of dam in ft.; x = any depth below

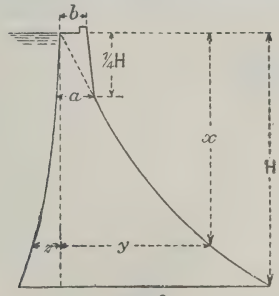


FIG. 2.

the water surface in ft.; y = offset from vertical line to outer face of dam at any depth x; z = ditto ditto to inner face; b = width of dam at top in ft.; a = width of dam at 1/4H from top in ft.; P = limit of pressure on masonry in tons per sq. ft.

$$b = 0{\cdot}4a; \quad y = \sqrt{\frac{{\cdot}05\,x^3}{P + {\cdot}03\,x}}; \quad z = \left(\frac{{\cdot}09\,x}{P}\right)^4;$$

y at any section must not be less than 0·6 x, fig. 2.

In modern practice a common typical section is triangular, the inner face being given a slight batter, and the back a batter which will generally approximate to 1 in 1½ but depends upon the weight of the masonry and the exact form of the wall.

The dam must be so proportioned that the resultant thrust acts within the middle third of the base, both with the reservoir full and empty; this provision should also apply to any horizontal section of the dam. The theoretical apex of the triangle must be high enough to allow, if necessary, for the water overtopping the wall, also for wave action.

EARTH DAMS.

The top width may be anything from 6 ft. to 20 ft., depending on the height of the dam and the freeboard.

The inner slope will generally be about 3 to 1 for gravel or other good material, but with clays it may not stand at this and must then be made flatter.

The outer slopes may be as steep as 2 to 1 depending on the angle at which the material will stand; the outer slope should be well drained. The inner slope must be protected from wave action; this is generally done by laying stone pitching or concrete slabs on the surface of the bank.

Watertightness is generally obtained by means of a core of puddled clay in the centre of the bank, the puddle being carried right down through the solid ground below the bank and well bonded into the impervious material on which the reservoir is built.

The puddle wall should generally be about 4 ft. thick at the top and both sides batter outwards about 1 in 40 to ground level, below which the thickness of the puddle should be quickly reduced to about 6 ft. and carried down at this as far as necessary. This applies to good puddle; the thickness must be increased if the puddle is of poor quality.

The puddle is sometimes replaced by a concrete wall; care must, however, be taken to avoid the possibility of unequal strains upon such a wall, as these might lead to cracking and possible failure.

Another method is to fill the trench in the solid ground with concrete and to build on this a wall of puddle. In this case the puddle must be well tongued into the concrete so as to ensure a watertight joint.

The François Cementation Process (*The François Cementation Co., Ltd.*) has been successfully employed for the injection of dam foundations to render them watertight, thus considerably reducing the amount of required excavation and the necessity of masonry aprons: e.g. the Mazoe Dam, in South Africa, (*A. H. Krynauw, Engineering Society of South Africa, May* 18, 1918), also Brownhill Reservoir Dam, for the Batley Corporation, and Frankley Reservoir Dam, for Birmingham Corporation.

The same process is employed with equal ease and efficiency to seal off leakages making through existing dams and reservoirs: e.g. Stobshiel Reservoir, Scotland; Upper Booth Dam, near Halifax; Leighton Reservoir, Leeds; and Camarasa Reservoir, Spain.

See also 'Shafts,' Section XXXIX.

Lining Distribution or Service Reservoirs.

In small reservoirs watertightness may be obtained by covering the whole of the bottom and sides of the reservoir with a layer of puddle. The puddle in this case should be covered with concrete to protect it from damage. In reservoirs of any size, however, the cost of the method would be prohibitive.

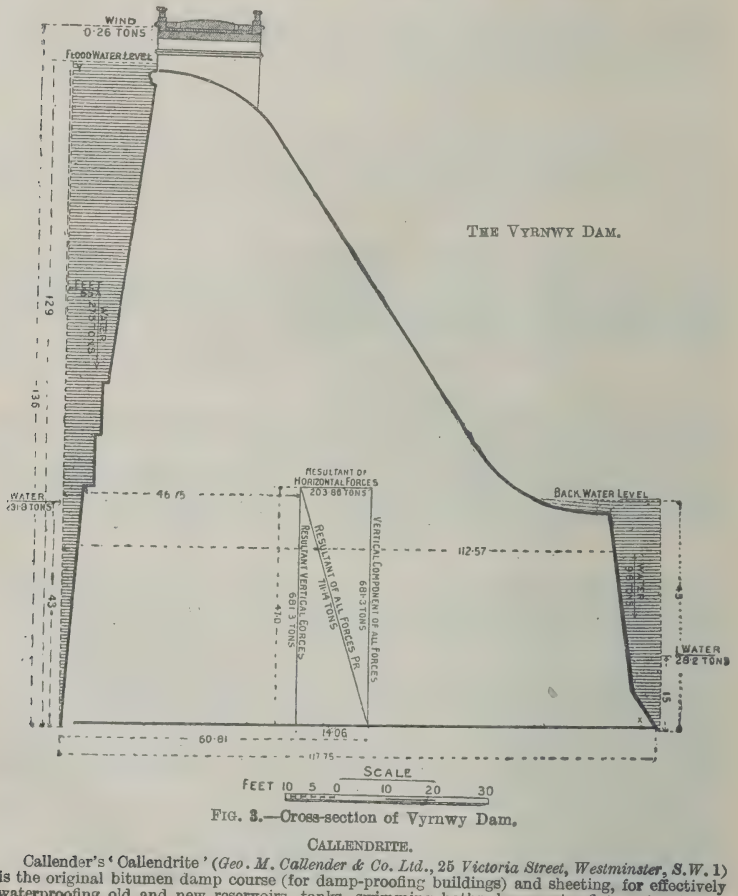


FIG. 3.—Cross-section of Vyrnwy Dam.

CALLENDRITE.

Callender's 'Callendrite' (*Geo. M. Callender & Co. Ltd., 25 Victoria Street, Westminster, S.W.* 1) is the original bitumen damp course (for damp-proofing buildings) and sheeting, for effectively waterproofing old and new reservoirs, tanks, swimming baths, basements, flat roofs, etc., as

invented by Callender's over fifty years ago. It is used by the Admiralty, the War Office, the Office of Works, the London County Council, and other bodies.

The sheeting—for all engineering work—is made in standard rolls, 24 ft. long, 6 ft. wide. Damp course in rolls 24 ft. long, to all the usual wall widths. When necessary, Callender's Patent Impervious Wall Ties and Patent Joining Irons should be employed. Price of Callendrite (including free delivery for fair order) averages 3½*d.* per foot super for damp course and 5*d.* per foot super for sheeting. Inclusive price given, with guarantee of efficiency for laying and fixing sheeting by Messrs. Callender's own workmen.

THE CROTON DAM.

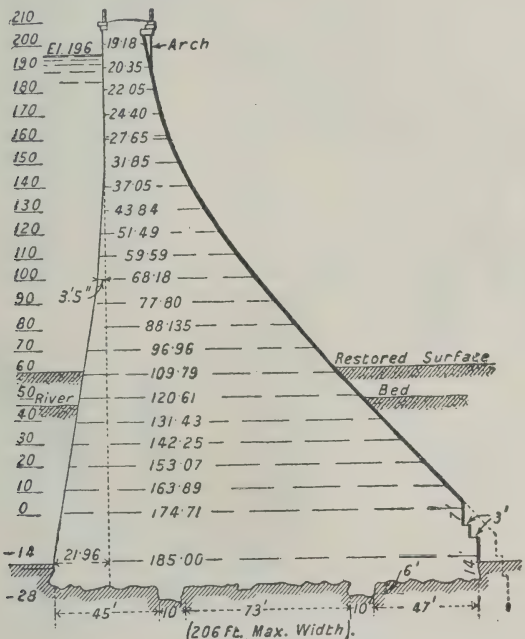


FIG. 4.—Cross-section of Croton Dam.

THE VYRNWY DAM.

The Vyrnwy Dam (fig. 3) is 1,350 feet long on the top, the plan being straight. The maximum height above the foundation is 136 feet. The profile adopted is designed to resist the water pressure and also to form a waste-weir; the front face of the wall is, therefore, made to conform to the curve described by water in overflowing, and to deflect it into the basin in front of the dam. The foundation is laid on a clay slate rock. The dam is built of 'Cyclopean rubble,' the stones varying in weight from 2 to 8 tons. For the lower part of the dam the stones are set in mortar composed of 2 parts washed sand, 1 part Portland cement; in the main portion of the work, instead of plain sand, a mixture of 1 part of the latter with 2 parts pulverised quarry rock was used for making the mortar, 2½ parts of the mixture being mixed with 1 part Portland cement. The area of the section, shown by fig. 3, is 8,972 square feet. With the reservoir full and the wind blowing with a force of 60 lbs. per square foot, the maximum stress on the masonry is 6·36 tons per square foot. Where the bed rock has the lowest elevation there are 26 drains in a length of 198 feet of the wall; these drains are 9 to 12 inches square, and lie on the rock near such places where leakage is apt to occur; they are 25 feet from the front face and 30 feet from the back face, and connect with a central tunnel 4 feet high by 2½ feet wide, which traverses the foundation longitudinally at an elevation of 46·3 feet above the base of the cross-section. By means of a cross-tunnel leading down-stream any water that may filter into the foundation above the elevation of the drains is discharged at the front face of the dam.

THE ASSUAN RESERVOIR.

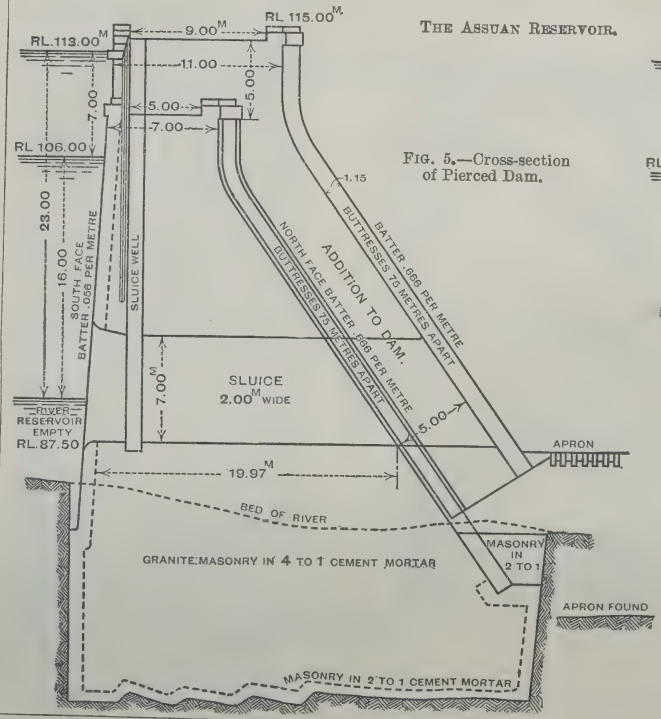


FIG. 5.—Cross-section of Pierced Dam.

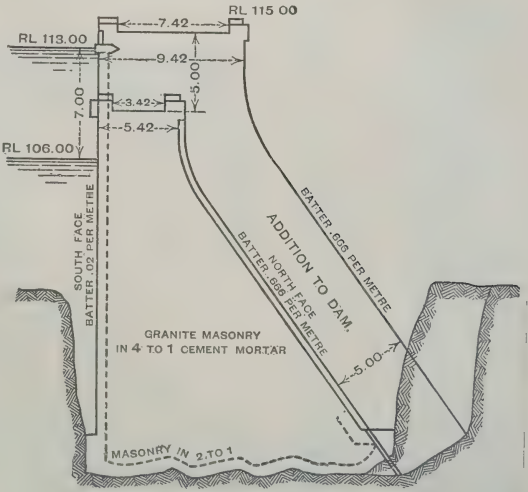


FIG. 6.—Cross-section of Solid Dam.

Mean High Flood Level at Aswan 92.80.
Mean Low Water Level.....85.10.

Contents of Reservoir-Original Dam 980,000,000M3
" " " Heightened Dam 2,300,000,000M3

THE ASSUAN DAM.

The Assuan Dam, fig. 6, is a granite rubble masonry structure founded on a similar rock which is traversed by dykes of softer igneous material. It is situated at the head of the first cataract of the Nile.

As originally constructed it was just under two kilometres in length, the top level of the structure was RL. 110·00, the water being held up to a level of 106·00.

It is pierced for the greater part of its length by sluices which are closed by steel gates, most of them working on the Stoney principle. There are 180 sluices in all, of which 22 are at a level of 100·00 and 18 at 96·00; these sluices are 3½ metres high by 2 metres wide. Of the others, which are 7 metres high by 2 metres wide, 75 are at RL. 92·00 and 65 at RL. 87·50; 30 of the lowest sluices are lined with cast iron, the rest smooth-faced granite ashlar.

The original dam was finished in the summer of 1902.

It was found that the rush of water through the sluices was such as to erode what had appeared to be solid rock, and it was therefore necessary to build masonry aprons along practically the whole length of the pierced dam.

Between 1907 and 1912 the dam was raised so as to impound water to a level of 113·00 metres, and it was at the same time thickened by a blanket wall five metres thick on the downstream side.

The original reservoir impounded 980,000,000 cubic metres of water, or 216,000,000,000 gallons.

The heightened reservoir contains 2,420,000,000 cubic metres, or 533,370,000,000 gallons.

The reservoir is used for irrigation purposes.

GENERAL ARRANGEMENT OF RESERVOIRS.

In reservoirs filled by gravity a waste weir must be provided to carry off flood water, and should be so designed that the maximum flood will not cause the water to rise more than 18 inches over the lip of the weir.

The waste weir channel should be built in solid ground: it is generally constructed in a flight

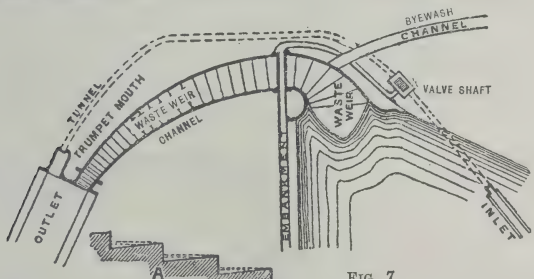


FIG. 7.

of steps, so as to check the velocity of descent; the risers vary from 1 ft., which is best, to about 5 ft., and the tread should, if possible, be at least double the riser. The best form is a series of weirs and pools, or tumbling bays, or flights of steps with long landings or pools at intervals, the drop into pool being as small as possible. The pools may be formed as shown at A (fig. 7): depth of pool = drop, length of pool = drop × 10.

Fig. 7 shows a tunnel round the flank, also the valve shaft, byewash, and other accessories.

The tunnel and byewash are used for passing storm waters during construction, the former being used afterwards for supply purposes and the latter for passing turbid water in times of storm.

Storage Reservoirs.

The capacity of storage reservoirs is arrived at from the formula

$$C = \frac{500}{\sqrt[3]{R}},$$

where,

C = number of days' storage, R = available rain in inches, as explained under heading 'Gathering Grounds.'

Messrs. W. J. E. Binnie and H. Lapworth have lately suggested the following formula for calculating storage:—

$$s = 15\frac{(n-d)^2}{r^{1\cdot85}},$$

where,

s = inches of storage required to supply (constantly from a stream) n inches per annum; d = the extreme minimum dry-weather flow reduced to inches per annum; r = mean run-off, also in inches per annum.

The storage required in gallons is equal to $s \times$ area of gathering-ground in acres $\times$ 22,610.

The formula is empirical, and was arrived at by a study of the data of seven areas of supply.

GATHERING GROUNDS.

In arriving at the water available from an area the average rainfall on the area must be found from rainfall records; ⅔ of this is then generally taken as the average for the three driest consecutive years and then a deduction must be made for evaporation, generally 14 to 16 inches; the remainder, subject to special conditions, will be the available rain; ⅓ of this often has to be given in compensation.

The above applies to ordinary cases in the United Kingdom. For a study of run-off under various climatic conditions, see 'Rainfall' (p. 1399).

On the moorland gathering-grounds supplying Sheffield, rainfall averages 40 ins. a year, two-thirds of which is reckoned on as available; one-third goes as compensation to millowners, etc., and one-third to inhabitants. Every 1,000 acres is said to give an average of 1,250,000 gallons per diem. On the Leeds moorland gathering-grounds ¼ cubic feet per second, or 134,600 gallons per diem per 1,000 acres, is the usual minimum dry-weather discharge; the formation is millstone grit, rainfall 26 ins. to 45 ins., of which 14 ins. is allowed for evaporation and 6 ins. as compensation to millowners.

FILTRATION.

(Contributed by H. F. Rutter, M.Inst.C.E.)

Filtration should be preceded by sedimentation. The researches of Dr. A. C. Houston prove that prolonged storage not merely reduces enormously the ordinary and harmless bacterial contents of river waters, but that such storage is a complete protection against danger from pathogenic microbes in the case of water initially infected with such germs.

The area of sand surface needed per million gallons per diem varies considerably, according to the character of the water and the fineness of the filtering medium. Yields of from one million to two million gallons per acre are common values. Modern filters have four feet of water over the sand, two feet nine inches of sand supported by eight inches of graduated shingle. Care should be taken that the uppermost layer of shingle should consist of stones less than half the size of a pea, so as to avoid the danger of the sand being drawn through the shingle. The means of collecting the water should cover the whole floor of the filter and should not consist of transverse pipes placed at intervals. A satisfactory and inexpensive method is to cover the floor with transverse rows of bricks laid dry end to end with a space of 4½ inches between each two rows. These are then completely covered with bricks laid dry at right angles to the rows. A series of 4½ in. by 3 in. channels are thus formed, which lead the filtrate to the central channel made below the floor.

A Deacon meter on the outlet pipe forms a useful means of checking and recording the rate of flow of the filter. Float gauges should be provided to show the relative levels of the water over the sand and in the outlet pipe. The difference between these levels indicates the head under which the filter is working. This head may vary from one inch when the filter is started slowly after cleaning, up to 2 or 3 feet.

Means should be provided for drawing the raw water from above the sand to waste, in case it be required to clean a filter immediately which has become clogged. A quarter to half an inch of sand with the matter arrested thereon is removed by flat shovels. This sand can be washed and spread again either at once, or after the depth of the sand has been reduced by five or six successive cleanings. The cleanings may be at intervals of from one week to four months, according to the character of the water, and the amount of algal growth in the water, which latter condition may materially affect the life of the filter. The removal and cleaning of the main body of the sand is needed at very rare intervals. Filters are in use yielding water of perfectly satisfactory bacteriological character in which the main body of sand has not been changed for forty years.

When a filter is to be cleaned the water should not be drawn down in the sand to a level more than 12 inches below the surface of the sand. This leaves the surface quite hard enough to support the weight of the cleaners, and prevents the sand becoming unduly compacted.

All sensible suspended matter is arrested on the surface of the sand of a filter, and the largest elimination of bacterial life takes place at or near the surface. The bacterial contents are, however, still further reduced progressively in the passage of the water through from five to ten inches of the depth of the sand.

THE 'TORRENT' FILTER.

(The Pulsometer Engineering Co., Ltd., Reading.)

With this plant (fig. 8) the filtration is effected by passing the water under pressure through a certain thickness of suitable filtering medium, arrangements being made so that the entire filtering medium can be thoroughly cleaned.

The construction is very substantial, all parts are accessible, and the various valves for operating and cleaning the filter are brought together at one point for convenient handling. There are no mechanically operated components either inside or outside the filter, so that there are no parts to wear or keep in adjustment.

The filter is made either cylindrical or square (the latter when it is necessary to economise space or when the filter has to be brought in parts into an existing building through ordinary doors). A strainer plate is fixed near the bottom of the filter casing, and this plate carries

layers of graded filtering medium. The unfiltered water admitted to the upper part of the filter percolates through the filtering medium, where the matter in suspension is retained, and reaches the space below the strainer plate in a clear and bright condition. The bottom of the filter casing is connected to the filtered water main.

The cleaning of the filter is effected by means of air agitation and a reverse flow of filtered water. If there is sufficient pressure in the filtered water main, say equal to 10 to 15 ft. head,

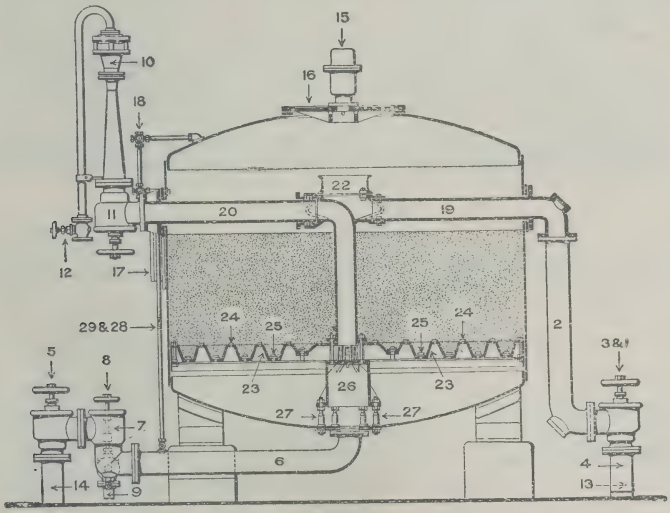


FIG. 8.

a reverse flow from the filtered water main can be used; but if there is not sufficient head in this main, or if the filtered water reservoir is below the level of the filter, it is necessary to employ a centrifugal pump to force the washing water through the filter.

The agitation (by compressed air) is effected by means of a separate system of air nozzles in the strainer plate, through which the air is equally distributed over the whole area of the filter bed. The air nozzles being entirely separate from the strainers make it possible to use the air simultaneously with the reverse flow of water during the process of cleaning, if found

LIST OF COMPONENT PARTS.

1. Unfiltered water inlet valve.
2. Unfiltered water inlet and wash-out bend.
3. Wash-out valve.
4. Wash-out pipe.
5. Filtered water outlet and wash water inlet valve.
6. Filtered water and wash water bend.
7. Patent wash water regulating valve.
8. Drain valve.
9. Drain pipe.
10. Air blower.
11. Blower valve.
12. Steam stop valve to air blower.
13. Unfiltered water pipe.
14. Filtered water pipe.
15. Automatic air escape valve.
16. Manhole cover.
17. Differential mercury gauge.
18. Water gauge glass.
19. Internal inlet and wash-out pipe.
20. Internal air pipe to strainer plate.
22. Internal wash-out and inlet bell mouth.
23. Strainer plate.
24. Air nozzles.
25. Strainers.
26. Stool for strainer plate.
27. Studs for strainer plate stool.
28. Pressure pipe from underside of filter medium.
29. Pressure pipe from above filter medium.

desirable. The air main is carried above the filter, so that there is no risk of its becoming water-logged, which might cause obstruction to the passage of the air.

A steam blower may be used for supplying the compressed air, but if steam is not available, a small air compressor, with or without an air receiver, can be employed.

The air causes a violent ebullition in the filter and has the effect of raising the filtering medium, so that the various particles are free to move, and in knocking against one another dislodge all the matter which adheres to their surfaces. Whilst the air agitation is going on an upward flow of water carries away, through the wash-out pipe, the impurities which have been dislodged by the action of the air.

This system of cleaning effects a considerable saving in the amount of wash-water used as compared with that necessary when water only, or water and mechanical agitators, are used.

The time taken to clean the filter may be from 10 to 20 mins.—depending on the nature of the water and the time the filter has been at work since the last cleaning. Reference to fig. 8, p. 1417, will show that as the nozzles are distributed equally all over the strainer plate and the air has free access to all of them, the cleaning process is as thorough at the sides of the filter as it is in the middle.

A differential gauge is fitted to show the head lost in the filter. This gauge gives an indication of when the filter requires cleaning, because the difference in pressure between the top and bottom of the filter increases as the impurities accumulate on the filtering medium.

Pollution by Birds.

It has been found in Western America that a grid of wires spaced from 40 ft. to 50 ft. apart each way and suspended about a foot above the water surface, will stop seagulls from alighting on the surface of potable water reservoirs and polluting them.

MAINTENANCE AND RENEWAL OF WATERWORKS.

In the Metropolitan area, 'pumping' constitutes, on an average, 43 per cent. of the entire maintenance expenditure. During the last thirty years 4,408,846*l.* has been expended out of revenue on pumping operations and on the maintenance and renewals of the pumping-engines and machinery, which during that period have supplied the Metropolis with 1,745,718 million gallons of water for all purposes; thus the cost of pumping alone has been, on the average, 2*l.* 10*s.* 7¼*d.* per million gallons, or ·000606*d.* per gallon.

The cost of repairs and renewals of water-mains and pipes amounted to 2,964,127*l.*, or an average of 1*l.* 14*s.* per million gallons.

The average maintenance cost per mile of water-main or pipe-way (adopting the analogy of a railway as a means of conveyance) has been 24*l.* 5*s.* 5*d.*

The cost per million gallons for filtration is 0·37*l.* (*Price-Williams.*)

Standard of Purity of Water.

Total hardness in practice varies from 0 to 30 degrees; 5 degrees is generally considered best.

Water containing more than the following quantities of impurities should be looked upon with suspicion: Chlorine, 1 part per 100,000; Nitrites, 0·2 part per 100,000; Nitrates, 0·3 part per 100,000; Ammonia, 0·1 part per 100,000; Organic Carbon, 2 parts per 100,000; Organic Nitrogen, 3 parts per 100,000; Albumenoid Ammonia, 0·01 part per 100,000.

WATER TESTING.

Simple and ready tests for rough analysis.

For *Sewage.*—Four drops of a solution of potash (Condy's Fluid) to a small glass of water. The colour turns pale or yellow if decomposed organic matter is present.

Lead.—Six drops of sulphuric acid in small glass of water. White precipitate will be found if lead is present.

Zinc.—Six drops of ferro-cyanide of potassium to small glass of water. Green colour if zinc is present.

Copper.—Eight drops of ammonia in small glass of water. Water turns blue if copper is present.

CHEMICAL PURIFICATION OF WATER.

Water is sometimes purified by chemical means: there are several methods, of which the most important are (1) by the addition of chloride of lime and (2) by submitting the water to the action of ozone.

By these methods the impurities are not removed, but the water is sterilised and all the bacteria which it contains are destroyed.

Reservoirs in which vegetable growth or algæ has become profuse have been cured by dragging a bag of sulphate of copper behind a boat to and fro over the area of the reservoir. A very minute fraction of this poisonous salt will effectually destroy all growth, and it is said that the salt cannot be detected in the water after the process is complete. Apparently the organic matter has absorbed it; but as the salt is extremely poisonous great care must be taken that too much is not used.

COST OF CHLORINATING WATER.

At the Walton works the cost of chlorinating a million gallons of water is $5 \cdot 77d$.

(*Stilgoe.*)

Action of Water on Lead.

Peaty water from moorlands has a solvent action on lead pipes, due to acidity. It varies, in terms of a sulphuric acid equivalent, from 1 to 44 parts in 100,000. Peaty grounds of small incline are particularly acid, and such water is dangerous on account of its action on lead pipes. According to Mr. Ackroyd the solvent action would not occur if the acidity did not exceed ·5.

Diameter of Pipes.

The diameters of pipes are usually calculated by either Eytelwein's or Kütter's formula.

Eytelwein's formula is

$$V = \sqrt{(11704\,RS) + \cdot 01698} - \cdot 1303$$

where,

V = velocity in feet per sec.; R = Hydraulic mean radius in feet = Area ÷ wet perimeter = D ÷ l for circular section of pipe, where D = diameter of pipe in feet; S = sine of inclination = $h \div 4$, where h = head in feet; l = length in feet.

An adaptation of this formula which gives good average results is

$$Q = 4 \cdot 7\, D^2 \sqrt{\frac{Dh}{l}}$$

where,

Q = discharge in cub. ft. per min., and D is in inches.

This formula gives rather high results in the case of small pipes with flat gradients, and somewhat low results for large pipes with steep gradients.

Kütter's formula is

$$V = \left[41 \cdot 6 + \frac{1 \cdot 811}{n} + \frac{\cdot 00281}{S} \right] \div \left[1 + \left(41 \cdot 6 + \frac{\cdot 00281}{S} \right) \frac{n}{\sqrt{R}} \right] \sqrt{RS}.$$

where V, R, and S have the meanings stated above, and n is Kütter's coefficient for roughness, which varies from ·009 to ·017, ·013 being a value which is frequently taken.

For new asphalted cast-iron pipes, Barnes (*Inst. C.E. Proc.*, ccxi., Jan. 1922) gives the following formula:—

$$Q = 47 \cdot 087\, d^{2 \cdot 769}\, i^{0 \cdot 529}$$

where,

Q = discharge in cub. ft. per sec.; d = diameter in ft.; i = hydraulic gradient = $\left(\frac{\text{height}}{\text{length}}\right)$.

Incrustation and Corrosion of Iron and Steel Pipes.

High-water velocity is favourable to corrosion and incrustation. Cast-iron being taken to rust at the rate of 100, wrought-iron may be taken as 129, and steel at 133. Some waters, as chalk-well waters, treated by Clark's process, will simply cause a clean layer of carbonate of lime inside a pipe. Upland waters, especially if containing peat acid, cause rust. Old red-sandstone water neither rusts nor causes deposit. Greensand water causes rust. Filtered water is better than unfiltered. Angus Smith composition (see Index) is not a permanent protector. Founders do not as a rule exercise proper care in dipping pipes; the pipes ought to be well brushed out, by passing through them a revolving steel brush *from both ends*, so as to get the brushing effect each way, and they should be then at once heated and dipped. Dipping should be vertical, and the pipes ought to again be brushed by the revolving steel brush when immersed in the dip, without which treatment perfect contact may be difficult to secure between the iron and the composition. It is undoubtedly at the pinholes, caused by minute air bubbles, that the first encroachments of corrosion are made upon the pipe.

According to H. F. Rutter, pipes of cold-blast iron are far superior to those of hot-blast, as regards resistance to corrosion.

Tracing Leakages.

To identify and trace the source of underground water, such as leakages from canals, reservoirs pipes, etc. :—

Fluorescin.—Mix fluorescin with the supposed source of the leakage. One grain is sufficient to colour 100 tons of water.

Indigo.—One grain of indigo dissolved in sulphuric acid will colour a ton of water.

The coloration is more easily detected by placing the water in a very long test-tube and looking through the latter endways.

Waste Water Detection.

(The Palatine Engineering Co., Ltd.)

The detection of invisible leakages between the reservoir and the consumer is usually done by meters of the 'Deacon' type, similar to fig. 9. A large recording meter is fixed on the main

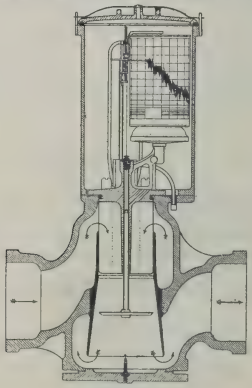


FIG. 9.

at the reservoir. The supplied area is divided into districts of about 2,000 inhabitants and arranged so that the whole supply comes from one direction, and passes through a Deacon meter. Fig. 10 is a plan of the area showing valves.

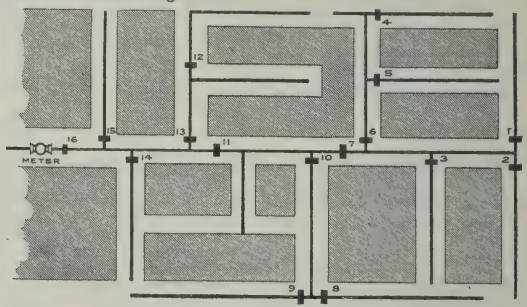


FIG. 10.

A typical chart for twenty-four hours in an uncontrolled district is shown in fig. 11. The minimum flow during the night is 2,000 gallons per hour (represented by the shaded portion), or 48,000 gallons per day. In the absence of night supplies for trade purposes, this water is leaking through defects. The water used or misused (represented by the cross-hatched portion) is 29,775 gallons, making a total supply of 77,775 gallons per day.

To locate this waste a chart is fitted in which the time scale is multiplied by four. During

the night the inspector proceeds to the end of the district and closes No. 1 valve, and the other valves in succession, at about 15-minute intervals, noting the time and the location of the valve.

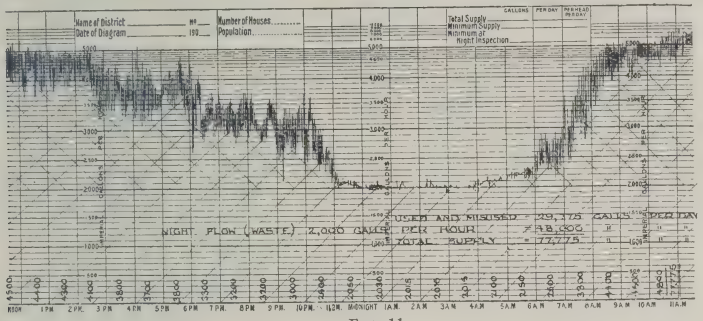


FIG. 11.

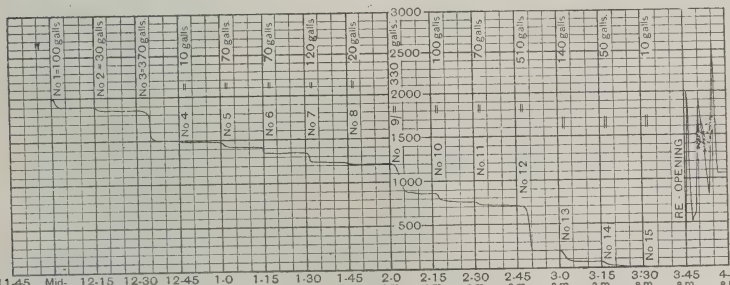


FIG. 12.

The result is shown in fig. 12, the rate of flow in each portion before isolating being clearly recorded. There is an indication of large defects in portions supplied through valves Nos. 3, 9, and 12. These are afterwards traced by a form of stethoscope and examination of fittings. In some of the areas where this system is carried out the consumption, in gallons, per head, per day, is:—Bath, 22; Birkenhead, 19·92; Frimley and Farnborough, 13·24; South Staffs Waterworks Company, 17·49. Trade requirements are measured through separate meters and are not included in these figures.

Hardness of Water.

A water is said to be *hard* when it is difficult to obtain a lather with soap; in washing, soap dissolves in pure (soft) water, but with a hard water it combines with the calcium and magnesium present, and forms an insoluble precipitate. Before a lather can be obtained it is necessary to add sufficient soap to precipitate all the soluble calcium and magnesium salts present in the water. About ½ oz. of soap is required to remove 1° of hardness from 10 gallons of water.

Formerly, the hardness of a sample of water was measured by ascertaining how much soap must be added before a lather could be obtained. Later, the hardness was measured in *degrees*, one degree being equivalent to one grain of calcium carbonate per gallon of water. At the present time the hardness is preferably expressed in terms of the salts actually causing it, and the proportion of each of these in the water.

Temporary hardness is that which is removed by boiling the water. It is due to calcium bicarbonate and/or magnesium bicarbonate, which are decomposed when the water is heated, calcium carbonate and/or magnesium carbonate being precipitated and so rendered inert to soap.

Permanent hardness is that which remains after boiling. It is chiefly due to calcium and/or magnesium sulphate and/or chloride, but may also be caused by any other soluble salts in the water. The calcium and magnesium salts combine with soap and form insoluble compounds. Soluble salts act differently and precipitate the soap as soap, because it is insoluble in a solution of salt. In most permanently hard waters the proportion of soluble salts other than those of

calcium and magnesium is so small that it may be ignored, but in softening sea-water or other saline water it must be taken into consideration.

Acid water also decomposes soap and so may be regarded as hard water.

Zero hardness is a term applied to water which is devoid of hardness. Most softened waters retain two or three degrees of hardness and are not softened to zero.

Softening Water.

Hard water may be softened by (i) rendering the hardness-causing substances insoluble, or (ii) by removing them. The substances causing *temporary hardness* may be rendered insoluble by (*a*) heating the water—usually to about 150° F., but in all cases by boiling it (212° F.). The heating decomposes the bicarbonates present, carbon-dioxide passing off as gas, and the mono-carbonates being left suspended in an insoluble and inert form in the water.

(*b*) Adding some substance which will decompose the bicarbonates without forming a product which can effect soap. Any alkali soluble in water can be used; the commonest being lime, caustic soda, and baryta. Lime is the cheapest of these substances. *Lime* combines with half the carbon dioxide in the bicarbonate and with it forms calcium carbonate which is insoluble and inert to soap.

$$XCO_3.H_2CO_3 + CaO = \underbrace{XCO_3 + CaCO_3} + H_2O$$

Soluble bicarbonate. Lime. = Insoluble carbonates. Water.

Baryta acts precisely like lime and forms insoluble barium carbonate.

Caustic soda decomposes the bicarbonate, precipitating the insoluble mono-carbonate of lime and/or magnesia, and forming soluble sodium carbonate. The latter is useful if the water is permanently hard, but may be objectionable in water used for industrial or scientific purposes.

Substances causing temporary hardness may be removed from hard water by passing it through a porous mass of zeolite such as 'Permutit.' This is a sodium alumino-silicate which gives up its sodium in exchange for an equivalent amount of lime or magnesia in the water.

$$Na_2Z + XCO_3H_2CO_3 = XZ + 2NaHCO_3$$

Sodium zeolite Soluble bicarbonate Altered zeolite Soluble bicarbonate

The soluble sodium bicarbonate is useful in removing an equivalent of permanent hardness, but may be objectionable in water used for some purposes.

Permanent hardness can be removed from water by:

(1) Distillation, *i.e.* boiling the water and condensing the steam. This is the only method for fully purifying water rich in soluble sodium or potassium salts (*e.g.* sea water).

(2) Precipitating any soluble calcium or magnesium salts by the addition of a soluble carbonate, sodium carbonate (soda) being generally used.

$$XSO_4 + Na_2CO_3 = X\,CO_3 + Na_2SO_4$$
$$XCl_2 + Na_2CO_3 = X\,CO_3 + 2NaCl$$

Hardness-forming salt. Soda. Insoluble carbonate. Soluble salt.

This method leaves a soluble sodium salt in solution; in most instances it does no harm, but it is objectionable in water used for some purposes. This method softens acid water by neutralising the acid and so rendering it inert to soap.

If caustic soda is used for softening a water having both temporary and permanent hardness, sodium carbonate will be formed first from the action of the 'temporary hardness' and this will be available for removing an equivalent amount of permanent hardness.

$$XCO_3H_2CO_3 + 2NaOH = XCO_3 + Na_2CO_3 + 2H_2O$$

Temporary hardness. Caustic soda. Insoluble carbonate. Sodium carbonate. Water.

$$YSO_4 + Na_2CO_3 = YCO_3 + Na_2SO_4$$

Permanent hardness. Sodium carbonate. Insoluble carbonate. Sodium sulphate.

Hence, for every 100 parts of calcium carbonate (present as bicarbonate) causing temporary hardness, sufficient sodium carbonate will be formed to remove 136 parts of the calcium sulphate causing permanent hardness. For this reason, when a water has both temporary and permanent hardness it is usually cheaper to use caustic soda than lime for removing the temporary hardness. Incidentally caustic soda is much less troublesome to use as it is readily soluble in water. An excess of caustic soda must be avoided as it may cause embrittlement and leakage at the fittings.

(3) Removing the soluble calcium or magnesium salts by passing the water through a porous mass of zeolite such as 'Permutit.'

$$Na_2Z + XSO_4 = XZ + Na_2SO_4$$

Sodium zeolite. Hardness-forming salt. Altered zeolite. Sodium sulphate.

The advantage of using a zeolite for softening water is that it acts automatically so long as any unaltered zeolite remains; and there can be no 'mistake' due to adding too much or too little reagent. On the other hand, when all the zeolite is altered, it has to be re-activated by treatment with a solution of common salt.

The periodic renewal of zeolite is necessary with most waters containing permanent hardness, as the sodium bicarbonate liberated by the zeolite combines with an equivalent amount of calcium and/or magnesium sulphate or chloride, and precipitates the corresponding carbonate in the pores of the zeolite which are gradually filled with it.

Softening by a zeolite also has the advantage of producing a water of zero hardness in many instances where the addition of 'chemicals' leaves about 3° of permanent hardness.

For further information on the Treatment of Feed-water for boilers and the softening of water see pp. 1624-6 and 1627-38. For information on the formation of *scale* in boilers see p. 1623.

Corrosion of Pipes by Water.

Water-pipes may be corroded by impurities in the water passing through them (see p. 1623).

TABLE SHOWING VELOCITY IN FEET PER SECOND, AND SUPPLY IN GALLONS PER MINUTE, FOR LONG PIPES FLOWING FULL.

(See note, page 1425.)

Diameter of Pipe in Inches.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Diameter of Pipe in Inches.
	Head of Water divided by Length of Pipe.										
	$\frac{1}{1000}$		$\frac{2}{1000}$		$\frac{3}{1000}$		$\frac{4}{1000}$		$\frac{5}{1000}$		
¾	·173	·05	·278	·08	·363	·10	·436	·13	·502	·14	¾
¾	·212	·11	·336	·17	·436	·22	·522	·27	·600	·31	¾
¾	·278	·32	·436	·50	·562	·64	·670	·77	·770	·88	¾
1	·336	·69	·522	1·07	·670	1·37	·798	1·63	·911	1·86	1
1¼	·388	1·24	·600	1·91	·770	2·45	·911	2·90	1·04	3·31	1¼
1½	·436	2·00	·670	3·08	·856	3·93	1·02	4·67	1·16	5·44	1½
1¾	·481	3·00	·736	4·60	·938	5·86	1·11	6·94	1·27	7·91	1¾
2	·522	4·26	·798	6·51	1·02	8·29	1·20	9·81	1·37	11·17	2
2½	·600	7·64	·911	11·62	1·16	14·76	1·37	17·45	1·56	19·85	2½
3	·670	12·30	1·02	18·66	1·29	23·64	1·52	27·92	1·73	31·73	3
4	·798	26·03	1·20	39·23	1·52	49·63	1·79	58·53	2·04	66·46	4
5	·911	46·48	1·37	69·79	1·73	88·14	2·04	103·84	2·31	117·80	5
6	1·02	74·64	1·52	111·66	1·92	140·83	2·26	165·77	2·56	187·95	6
7	1·11	111·09	1·66	166·07	2·09	209·22	2·46	246·10	2·79	278·90	7
8	1·20	156·92	1·79	234·11	2·26	294·70	2·65	346·45	3·01	392·46	8
9	1·29	212·71	1·92	316·87	2·41	398·58	2·83	468·35	3·21	530·36	9
10	1·37	279·16	2·04	415·34	2·56	522·08	3·01	613·22	3·40	694·24	10
11	1·45	356·95	2·15	530·42	2·70	666·39	3·17	782·42	3·59	885·56	11
12	1·52	446·66	2·26	663·07	2·83	832·63	3·33	977·29	3·76	1105·8	12
15	1·73	793·27	2·56	1174·7	3·21	1473·2	3·76	1727·8	4·26	1956·7	15
18	1·92	1267·5	2·83	1873·4	3·55	2347·2	4·16	2751·4	4·71	3110·0	18
24	2·26	2652·2	3·33	3909·2	4·16	4891·4	4·88	5727·8	5·51	6470·3	24
30	2·56	4698·8	3·76	6911·3	4·71	8638·9	5·51	10109·7	6·22	11415·0	30

	Head of Water divided by Length of Pipe.										
	$\frac{6}{1000}$		$\frac{7}{1000}$		$\frac{8}{1000}$		$\frac{9}{1000}$		$\frac{1}{100}$		
¾	·562	·16	·618	·18	·670	·19	·720	·21	·770	·22	¾
¾	·670	·34	·736	·38	·798	·41	·856	·44	·911	·46	¾
¾	·856	·98	·938	1·08	1·02	1·17	1·09	1·24	1·16	1·33	¾
1	1·02	2·08	1·11	2·27	1·20	2·45	1·29	2·63	1·37	2·79	1
1¼	1·16	3·69	1·27	4·04	1·37	4·36	1·47	4·67	1·56	4·96	1¼
1½	1·29	5·91	1·41	6·46	1·52	6·98	1·63	7·47	1·73	7·93	1½
1¾	1·41	8·79	1·55	9·68	1·66	10·38	1·78	11·10	1·89	11·79	1¾
2	1·52	12·41	1·66	13·56	1·79	14·63	1·92	15·65	2·04	16·61	2
2½	1·73	22·04	1·82	23·30	2·04	25·96	2·18	27·75	2·31	29·45	2½
3	1·92	35·21	2·09	38·43	2·26	41·44	2·41	44·29	2·56	46·99	3
4	2·26	73·68	2·46	80·36	2·65	86·61	2·83	92·51	3·01	98·60	4
5	2·56	130·52	2·79	142·30	3·01	153·31	3·21	163·69	3·40	173·70	5
6	2·83	208·16	3·09	226·84	3·33	244·32	3·55	260·81	3·76	277·10	6
7	3·09	308·76	3·37	335·60	3·62	362·21	3·87	386·57	4·09	408·45	7
8	3·33	434·35	3·62	473·09	3·90	509·38	4·16	543·49	4·41	575·83	8
9	3·55	586·82	3·87	639·02	4·16	687·85	4·44	733·83	4·71	777·50	9
10	3·76	767·92	4·09	833·58	4·41	899·75	4·71	959·88	4·99	1016·9	10
11	3·97	979·39	4·32	1066·1	4·65	1147·2	4·96	1223·7	5·26	1298·4	11
12	4·16	1222·9	4·53	1331·8	4·88	1432·0	5·18	1522·7	5·51	1617·6	12
15	4·71	2159·7	5·12	2349·8	5·51	2527·4	5·87	2694·9	6·22	2853·8	15
18	5·18	3426·1	5·65	3737·0	6·08	4019·3	6·48	4284·7	6·86	4536·5	18
24	6·08	7145·4	6·61	7769·1	7·11	8351·6	7·58	8900·4	8·02	9421·1	24
30	6·86	12601·3	7·50	13772·4	8·02	14720·4	8·54	15684·5	9·04	16599·0	30

TABLE SHOWING VELOCITY IN FEET PER SECOND, AND SUPPLY IN GALLONS PER MINUTE, FOR LONG PIPES FLOWING FULL (*continued*).

Diameter of Pipe in Inches.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Diameter of Pipe in Inches.
	Head of Water divided by Length of Pipe.										
	$\frac{2}{100}$		$\frac{3}{100}$		$\frac{4}{100}$		$\frac{5}{100}$		$\frac{6}{100}$		
$\frac{3}{8}$	1·16	·33	1·47	·42	1·73	·50	1·96	·56	2·18	·62	$\frac{3}{8}$
$\frac{1}{2}$	1·37	·70	1·73	·88	2·04	1·04	2·31	1·18	2·56	1·31	$\frac{1}{2}$
1	1·73	1·98	2·18	2·50	2·56	2·94	2·90	3·33	3·21	3·68	1
$1\frac{1}{4}$	2·04	4·15	2·56	5·22	3·01	6·13	3·40	6·94	3·76	7·68	$1\frac{1}{4}$
$1\frac{1}{2}$	2·31	7·36	2·90	9·24	3·40	10·85	3·85	12·27	4·26	13·59	$1\frac{1}{2}$
2	2·56	11·75	3·21	14·73	3·76	17·28	4·26	19·57	4·71	21·60	2
$2\frac{1}{2}$	2·79	17·43	3·50	21·84	4·09	25·53	4·63	28·95	5·12	31·98	$2\frac{1}{2}$
3	3·01	24·53	3·76	30·72	4·41	35·99	4·99	40·67	5·51	44·93	3
4	3·40	43·39	4·26	54·35	4·99	63·55	5·63	71·79	6·22	79·27	4
5	3·76	69·11	4·71	86·39	5·51	101·10	6·22	114·15	6·86	126·01	5
6	4·41	144·20	5·51	179·73	6·44	210·17	7·27	237·17	8·02	261·70	6
7	4·99	254·50	6·22	317·08	7·27	370·57	8·20	418·02	9·04	461·08	7
8	5·51	404·30	6·86	504·05	8·02	588·82	9·04	663·96	9·97	732·18	8
9	5·99	598·62	7·50	749·83	8·71	870·78	9·82	981·68	10·83	1082·4	9
10	6·44	840·68	8·02	1046·8	9·36	1222·0	10·55	1377·3	11·63	1518·3	10
11	6·86	1134·1	8·54	1411·6	9·97	1647·4	11·24	1856·4	12·38	2046·2	11
12	7·27	1482·3	9·04	1844·3	10·55	2152·0	11·89	2424·5	13·10	2665·9	12
15	8·02	2355·3	9·97	2928·7	11·63	3416·1	13·10	3838·9	14·43	4239·6	15
18	9·04	4149·7	11·24	5157·0	13·10	5998·2	14·75	6769·5	16·25	7457·0	18
24	10·07	6653·8	12·38	8184·8	14·43	9539·1	16·25	10738·1	17·89	11826·3	24
30	11·63	13664·4	14·43	16958·3	16·81	19754·1	18·92	22228·8	20·83	24474·3	30
	13·10	23993·0	16·25	29828·1	18·92	34732·5	21·28	39073·2	23·43	43011·1	
	Head of Water divided by Length of Pipe.										
	$\frac{7}{100}$		$\frac{8}{100}$		$\frac{9}{100}$		$\frac{1}{10}$		$\frac{2}{10}$		
$\frac{3}{8}$	2·37	·68	2·56	·73	2·74	·79	2·90	·83	4·26	1·22	$\frac{3}{8}$
$\frac{1}{2}$	2·79	1·42	3·01	1·53	3·21	1·64	3·40	1·36	4·99	2·54	$\frac{1}{2}$
1	4·09	8·34	4·41	9·00	4·71	9·60	4·99	10·17	7·27	14·82	1
$1\frac{1}{4}$	4·63	14·77	4·99	15·89	5·32	16·94	5·63	17·95	8·20	26·13	$1\frac{1}{4}$
$1\frac{1}{2}$	5·12	23·50	5·51	25 27	5·87	26·95	6·22	28·54	9·04	41·50	$1\frac{1}{2}$
2	5·57	34·79	5·99	37·41	6·38	39·89	6·76	42·23	9·82	61·35	2
$2\frac{1}{2}$	5·99	48·87	6·44	52·54	6·86	56·01	7·27	59·29	10·55	86·08	$2\frac{1}{2}$
3	6·76	86·18	7·27	92·64	7·74	98·73	8·20	104·50	11·89	151·53	3
4	7·50	137·72	8·02	147·20	8·54	156·85	9·04	165·99	13·10	239·93	4
5	8·71	284·34	9·36	305·50	9·97	325·41	10·55	344·32	15·26	498·17	5
6	9·82	500·86	10·55	537·99	11·24	572·99	11·89	606·13	17·18	876·11	6
7	10·83	795·20	11·63	854·02	12·38	909·42	13 10	959·72	18·92	1389·3	7
8	11·76	1175·3	12·63	1262·1	13·44	1343·7	14·22	1421·2	20·52	2051·3	8
9	12·63	1648·4	13·56	1765·8	14·43	1884·3	15·26	1992·7	22·02	2874·7	9
10	13·44	2221·3	14·43	2384·8	15·36	2538·6	16·25	2684·5	23·43	3871·0	10
11	14·22	2900·4	15·26	3113·5	16·25	3314·2	17·18	3514·5	24 76	5051·3	11
12	14·96	3691·7	16·06	3962·7	17·09	4217·9	18·07	4459·7	26·04	6425·9	12
15	15·66	4601·1	16·81	4938·6	17·89	5256·2	18·92	5557·2	27·25	8004·7	15
18	17·63	8091·3	18·92	8683·1	20·13	9240·3	21·28	9768·3	30·63	14059·7	18
24	19·41	12830·1	20·83	13766·8	22·16	14648·5	23·42	15484·0	33·70	22273·1	24
30	22·59	26545·4	24·24	28477·6	25·79	30296·3	27·25	32018·8	39·17	46016·9	30
	25·41	46642·4	27·25	50029·4	28·99	53218·5	30·63	56238·9	44·0	80770·7	

TABLE SHOWING VELOCITY IN FEET PER SECOND, AND SUPPLY IN GALLONS PER MINUTE, FOR LONG PIPES FLOWING FULL (*continued*).

Diameter of Pipe in Inches.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Diameter of Pipe in Inches.
	Head of Water divided by Length of Pipe.										
	$\frac{3}{10}$		$\frac{4}{10}$		$\frac{5}{10}$		$\frac{6}{10}$		$\frac{7}{10}$		
½	5·32	1·52	6·22	1·78	7·02	2·01	7·74	2·22	8·42	2·41	½
¾	6·22	3·17	7·27	3·71	8·20	4·18	9·04	4·61	9·82	5·01	¾
1	7·74	8·89	9·04	10·37	10·19	11·69	11·24	12·89	12·20	14·00	1
1¼	9·04	18·44	10·55	21·52	11·89	24·25	13·10	26·66	14·22	29·00	1¼
1½	10·19	32·48	11·89	37·88	13·39	42·67	14·75	47·01	16·01	51·02	1½
1¾	11·24	51·57	13·10	59·98	14·75	67·70	16·25	74·57	17·63	80·91	1¾
2	12·20	76·21	14·22	88·82	16·01	99·99	17·63	110·14	19·13	119·48	2
2½	13·10	106·64	15·26	124·54	17·18	140·18	18·92	154·37	20·52	167·45	2½
3	14·75	188·42	17·18	219·03	19·33	246·46	21·28	271·34	23·08	294·29	3
4	16·25	298·28	18·92	347·32	21·28	390·73	23·43	430·11	25·41	466·42	4
5	18·92	617·47	22·02	718·66	24·76	808·21	25·25	824·14	29·55	964·31	5
6	21·28	1085·4	24·76	1262·8	27·84	1419·8	30·63	1561·7	33·21	1692·6	6
7	23·43	1720·4	27·25	2001·2	30·63	2248·9	33·70	2474·8	36·53	2682·4	7
8	25·41	2539·4	29·55	2953·2	33·21	3319·2	36·53	3651·1	39·59	3957·0	8
9	27·25	3557·6	31·67	4134·9	35·60	4647·9	39·17	5113·0	42·44	5540·9	9
10	28·99	4789·7	33·70	5568·3	37·87	6256·8	41·65	6881·1	45·13	7456·4	10
11	30·63	6246·9	35·60	7262·3	40·01	8161·0	44·00	8974·3	47·67	9724·4	11
12	32·20	7947·9	37·41	9234·4	42·00	10365·4	46·24	11411·6	50·10	12364·4	12
15	33·70	9892·2	39·17	11504·2	44·00	12923·0	48·38	14209·8	52·40	15391·1	15
18	37·87	17380·0	44·00	20192·7	49·42	22679·2	54·33	24932·9	58·85	27009·8	18
24	41·65	27524·2	48·38	31972·0	54·33	35903·4	59·72	39465·9	64·52	42641·5	24
30	48·38	56839·1	56·18	66005·2	63·07	74102·4	69·32	81442·6	75·07	88204·4	30
	Head of Water divided by Length of Pipe.						Note.				
	$\frac{8}{10}$		$\frac{9}{10}$		$\frac{1}{1}$						
½	9·04	2·59	9·63	2·76	10·19	2·92					½
¾	10·55	5·38	11·24	5·73	11·89	6·06					¾
1	13·10	15·00	13·93	15·98	14·75	16·92					1
1¼	15·26	31·14	16·25	33·14	17·18	35·04					1¼
1½	17·18	54·76	18·28	58·28	19·33	61·61					1½
1¾	18·92	86·83	20·13	92·40	21·28	97·68					1¾
2	20·52	128·21	21·84	136·42	23·08	144·20					2
2½	22·02	179·67	23·43	191·16	24·76	202·05					2½
3	24·76	315·71	26·34	335·86	27·84	354·95					3
4	27·25	500·29	28·99	532·18	30·63	562·22					4
5	31·67	1033·7	33·70	1099·9	35·60	1162·0					5
6	35·60	1815·6	37·87	1931·1	40·01	2040·2					6
7	39·17	2876·1	41·65	3058·2	44·00	3230·8					7
8	42·44	4242·3	45·13	4510·7	47·67	4764·9					8
9	45·50	5940·0	48·38	6315·4	51·10	6671·1					9
10	48·38	7993·0	51·44	8499·2	54·33	8975·8					10
11	51·10	10423·5	54·33	11081·3	57·38	11704·2					11
12	53·70	13252·8	57·08	14088·5	60·29	14880·0					12
15	56·18	16501·3	59·72	17540·4	63·07	18525·6					15
18	63·07	28146·2	67·04	30767·7	70·80	32492·7					18
24	69·32	45811·5	73·67	48690·4	77·80	51416·9					24
30	80·44	94508·9	85·49	100438·1	90·26	106052·8					30
	90·26	165708·0	95·92	176090·0	101 26	185902·0					

The values in the foregoing table are calculated from the formula

$$V = 140\sqrt{rs} - 11\sqrt[5]{rs};$$

where V = velocity in feet per second, r = *hydraulic mean depth* (= area of cross section divided by wet perimeter = for full pipes, ¼ diameter) in feet, s = sine of inclination, or total fall divided by total length.

For pipes of greater diameter than above divide the proposed diameter by 4, then twice the velocity opposite to the quotient in the table will be the velocity required, or the corresponding supply multiplied by 32 will be the approximate supply in gallons per minute.

Velocity in Feet per sec. = $8{\cdot}025\sqrt{H}$ ($8{\cdot}025 = \sqrt{2g}$)	Head in Feet (H)	Lbs. Press. per sq. in.	Diameter of Bore in Inches and Area of Bore in Feet.													
			½ in. ·00136	⅝ ·00306	1 ·00545	1¼ ·0085	1½ ·01227	2 ·02182	2½ ·03408	3 ·04909	3½ ·0668	4 ·0872	5 ·1363	6 ·1963	7 ·2672	8 ·349
			THEORETICAL DISCHARGE (Area in feet × Velocity = Cubic feet per Sec. × 6·227													
8·025	1	·4335	·0681	·1529	·2719	·4247	·6131	1·090	1·703	2·453	3·338	4·357	6·811	9·809	13·35	17·44
11·35	2	·8670	·0963	·2162	·3851	·6007	·8669	1·542	2·408	3·469	4·721	6·163	9·632	13·87	18·88	24·67
13·90	3	1·300	·1180	·2648	·4717	·7356	1·062	1·888	2·950	4·248	5·782	7·537	11·80	16·99	23·13	30·22
16·05	4	1·734	·1362	·3058	·5438	·8494	1·226	2·180	3·405	4·906	6·675	8·714	13·62	19·62	26·70	34·89
17·95	5	2·170	·1523	·3419	·6090	·9498	1·371	2·438	3·808	5·485	7·464	9·744	15·23	21·94	29·86	39·01
19·66	6	2·601	·1668	·3745	·6671	1·040	1·502	2·670	4·171	6·008	8·176	10·67	16·68	24·03	32·71	42·73
21·23	7	3·034	·2802	·4046	·7205	1·124	1·622	2·884	4·506	6·490	8·832	11·53	18·02	25·95	35·33	46·16
22·70	8	3·468	·1926	·4324	·7703	1·201	1·733	3·084	4·817	6·938	9·442	12·33	19·27	27·75	37·77	49·34
24·08	9	3·901	·2043	·4587	·8157	1·274	1·839	3·270	5·107	7·359	10·01	13·07	20·43	29·43	40·06	52·33
25·38	10	4·335	·2154	·4835	·8612	1·343	1·939	3·447	5·385	7·759	10·55	13·78	21·54	31·02	42·22	55·17
26·62	11	4·768	·2258	·5080	·9030	1·411	2·035	3·619	5·651	8·134	11·09	14·48	22·58	32·57	44·32	57·90
27·80	12	5·200	·2359	·5297	·9434	1·471	2·124	3·776	5·900	8·498	11·56	15·10	23·59	33·98	46·25	60·43
28·93	13	5·635	·2458	·5523	·9819	1·534	2·213	3·935	6·148	8·845	12·06	15·74	24·58	35·41	48·19	62·96
30·03	14	6·069	·2548	·5728	1·017	1·591	2·294	4·077	6·366	9·171	12·49	16·32	25·48	36·70	49·93	65·24
31·08	15	6·502	·2638	·5922	1·055	1·645	2·375	4·219	6·580	9·484	12·93	16·88	26·38	37·93	51·68	67·55
32·10	16	3·936	·2724	·6116	1·088	1·699	2·452	4·360	6·810	9·812	13·35	17·43	27·24	39·24	53·41	69·78
33·09	17	7·369	·2804	·6306	1·122	1·752	2·528	4·491	7·014	10·12	13·78	17·96	28·08	40·45	55·01	71·87
34·05	18	7·813	·2884	·6488	1·153	1·803	2·600	4·622	7·209	10·40	14·16	18·49	28·89	41·56	56·60	73·96
34·98	19	8·236	·2969	·6670	1·186	1·853	2·673	4·796	7·412	10·68	14·56	19·01	29·69	42·77	58·21	76·04
35·89	20	8·670	·3046	·6888	1·218	1·900	2·742	4·876	7·616	10·97	14·93	19·49	30·46	43·87	59·72	78·02
36·75	21	9·102	·3115	·7008	1·246	1·947	2·808	4·992	7·787	11·24	15·30	19·98	31·20	44·95	61·14	79·88
37·64	22	9·537	·3189	·7176	1·276	1·993	2·875	5·112	7·982	11·50	15·66	20·45	31·94	46·01	62·61	81·79
38·52	23	9·970	·3264	·7344	1·306	2·040	2·942	5·232	8·169	11·77	16·03	20·93	32·69	47·09	63·86	83·71
39·31	24	10·40	·3336	·7490	1·334	2·080	3·003	5·340	8·343	12·02	16·35	21·35	33·37	48·06	65·41	85·46
40·13	25	10·84	·3405	·7645	1·360	2·124	3·065	5·451	8·512	12·26	16·69	21·79	34·06	49·05	66·76	87·22
43·95	30	13·01	·3732	·8384	1·491	2·329	3·359	5·973	9·327	13·44	18·30	23·89	37·32	53·76	73·15	95·57
46·76	34	14·74	·3964	·8919	1·586	2·478	3·574	6·355	9·911	14·30	19·47	25·42	39·71	57·19	77·88	101·7
47·48	35	15·17	·4026	·9058	1·610	2·516	3·629	6·453	10·06	14·52	19·77	25·81	40·32	58·08	79·03	103·2
50·76	40	17·34	·4307	·9671	1·722	2·686	3·878	6·895	10·77	15·52	21·11	27·56	43·07	62·04	84·45	110·3
53·83	45	19·51	·4556	1·025	1·822	2·847	4·107	7·303	11·40	16·43	22·38	29·21	45·63	65·73	89·50	116·9
56·75	50	21·68	·4828	1·086	1·931	3·018	4·353	7·739	12·09	17·42	23·71	30·96	48·35	69·65	94·79	123·9
59·52	55	23·84	·5046	1·135	2·018	3·153	4·548	8·088	12·64	18·20	24·78	32·35	50·53	72·79	99·06	129·4
62·16	60	26·01	·5276	1·184	2·110	3·290	4·750	8·443	13·16	19·01	25·86	33·75	52·76	75·87	103·4	135·1
67·14	70	30·35	·5692	1·281	2·277	3·557	5·131	9·123	14·25	20·53	27·96	36·83	57·00	82·11	111·8	146·0
71·78	80	34·68	·6090	1·367	2·436	3·799	5·484	9·751	15·23	21·94	29·86	38·98	60·92	87·74	119·4	156·0
76·13	90	39·02	·6461	1·451	2·584	4·029	5·817	10·34	16·16	23·28	31·66	41·34	64·61	93·06	126·7	165·5
80·25	100	43·35	·6810	1·529	2·719	4·247	6·131	10·90	17·03	24·53	33·38	43·57	68·11	98·09	133·5	174·4
113·5	200	86·70	·9632	2·162	3·852	6·007	8·671	15·42	24·08	34·69	47·21	61·63	96·32	138·7	188·8	246·7
139·0	300	130·1	1·180	2·648	4·717	7·357	10·620	18·88	29·50	42·49	57·82	75·47	118·0	169·9	231·3	302·2
160·5	400	173·4	1·362	3·058	5·438	8·494	12·260	21·80	34·05	49·06	66·76	87·14	136·2	196·2	267·0	348·9
179·5	500	216·8	1·523	3·419	6·090	9·498	13·710	24·38	38·08	54·85	74·64	97·44	152·3	219·4	298·6	390·1
196·6	600	260·1	1·668	3·745	6·671	10·40	15·020	26·70	41·71	60·08	81·76	106·7	166·8	240·3	327·1	427·3
212·3	700	308·5	2·802	4·046	7·205	11·24	16·220	28·84	45·06	64·90	88·32	115·3	180·2	259·5	353·3	461·6
227·0	800	346·8	1·926	4·324	7·703	12·01	17·340	30·84	48·17	69·38	94·42	123·3	192·7	277·5	377·7	493·4
240·8	900	390·2	2·043	4·587	8·157	12·74	18·390	32·70	51·07	73·59	100·1	130·7	204·3	294·3	400·6	523·3
253·8	1000	433·5	2·154	4·835	8·612	13·43	19·390	34·47	53·85	77·59	105·6	137·8	215·4	310·2	422·2	551·7

For intermediate heads, multiply $\sqrt{H}$ by gals. per sec for 1 ft. head corresponding to pipe of 95 ft. Gals. per sec. $= \sqrt{95} \times 2{\cdot}453$ (gals. per sec. for 3 in. pipe

Diameter of Bore in Inches and Area of Bore in Feet.

9	10	11	12	13	14	15	16	18	20	22	24	30	36	42	48	Head in Feet (H).
·4418	·5454	·6599	·7854	·922	1·069	1·227	1·396	1·767	2·18	2·64	3·142	4·909	7·068	9·621	12·57	

IN GALLONS PER SECOND.
galls. in a cu. ft.) = Galls. per second as below.

22·08	27·25	32·98	39·25	46·07	53·42	61·3	69·8	88·3	109·0	131·9	157·0	245·3	353·2	480·8	627·9	1
31·22	38·54	46·64	55·51	65·16	75·55	86·7	98·7	124·9	154·2	186·6	222·0	346·9	499·5	679·9	888·1	2
38·24	47·21	57·12	67·98	79·80	92·52	106·2	120·8	153·0	188·8	228·5	271·0	424·9	611·8	832·7	1087·6	3
44·15	54·51	65·95	78·49	92·14	106·9	122·6	139·5	176·6	218·0	263·8	314·0	490·6	716·4	961·5	1255·8	4
49·37	60·95	73·74	87·76	103·0	119·5	137·1	156·0	197·5	243·8	295·0	351·1	548·5	789·8	1075·1	1404·2	5
54·08	66·76	80·77	96·14	113·9	131·9	150·2	170·9	216·3	267·0	323·2	384·5	600·8	865·2	1177·6	1538·1	6
58·41	72·11	87·25	103·8	121·9	141·3	162·2	184·6	233·7	288·5	349·0	415·4	649·0	934·5	1272·0	1661·4	7
62·44	77·09	93·27	111·11	130·8	151·1	173·3	197·3	249·8	308·4	373·1	444·0	693·8	999·0	1359·8	1776·1	8
66·23	81·76	98·92	117·7	138·2	160·3	183·9	209·3	264·9	327·0	395·8	470·9	735·9	1059·6	1442·3	1883·8	9
69·81	86·19	104·3	124·1	145·7	168·9	193·9	220·6	279·2	344·7	417·2	496·4	775·9	1116·9	1520·3	1986·0	10
73·37	90·47	109·6	130·3	153·1	177·4	203·6	231·7	293·2	361·9	438·0	521·2	814·4	1172·6	1596·3	2085·0	11
76·48	94·41	114·2	136·0	159·6	185·1	212·4	241·7	305·9	377·6	457·0	543·8	849·7	1223·5	1665·4	2175·2	12
79·78	62·27	119·1	141·7	166·4	192·9	221·3	252·0	318·8	393·5	476·2	566·8	885·5	1275·1	1735·7	2267·0	13
82·65	101·9	123·4	146·8	172·5	199·9	229·4	261·2	330·4	407·7	493·4	587·2	917·5	1321·1	1799·3	2349·4	14
85·51	105·5	127·7	151·9	178·4	206·9	237·5	270·2	342·0	422·1	511·0	607·9	950·0	1368·0	1862·1	2432·0	15
88·31	109·0	131·9	157·0	184·3	213·7	245·3	279·0	353·2	436·1	527·7	627·9	981·2	1412·8	1923·0	2511·7	16
91·16	112·4	136·0	161·8	190·1	220·3	252·8	87·6	364·0	449·1	543·5	647·0	1011·7	1456·4	1982·1	2589·1	17
93·70	115·6	139·9	166·4	195·6	226·6	260·1	295·9	374·5	462·2	559·3	665·8	1040·2	1497·6	2039·7	2663·3	18
96·36	118·8	143·8	171·1	201·0	232·9	267·3	304·2	385·0	475·2	575·5	684·5	1068·2	1539·9	2096·3	2737·6	19
98·74	121·9	147·5	175·5	206·1	238·9	274·2	312·0	394·9	487·6	590·0	702·1	1097·0	1579·6	2150·1	2808·3	20
101·2	124·8	151·1	179·8	211·2	244·7	280·9	319·6	404·5	499·2	604·1	719·2	1123·5	1617·7	2203·1	2876·8	21
103·7	127·8	154·7	184·1	216·2	250·5	287·5	327·2	414·1	511·2	618·7	736·3	1150·4	1656·5	2254·9	2944·8	22
106·1	131·0	158·4	188·2	221·3	256·4	294·3	334·9	423·8	523·2	633·2	753·6	1177·3	1695·3	2307·6	3014·0	23
108·2	133·5	161·6	192·3	227·7	263·7	300·4	341·8	432·6	534·1	646·3	769·1	1201·6	1730·3	2355·3	3076·2	24
110·4	136·3	169·9	196·2	230·4	267·1	306·6	348·8	441·5	545·1	659·6	784·9	1226·4	1766·0	2403·8	3140·0	25
121·1	149·3	180·1	215·0	252·6	292·7	336·0	382·5	483·9	597·3	723·3	860·3	1344·0	1935·5	2634·8	3441·0	30
128·8	158·9	192·4	228·8	268·8	311·4	357·4	406·9	514·8	635·5	769·6	915·3	1430·1	2059·2	2803·1	3660·8	34
130·8	161·3	195·4	232·4	272·9	316·2	363·0	413·2	522·7	645·8	781·4	929·4	1452·2	2090·9	2847·5	3717·2	35
139·6	172·4	208·6	248·2	291·4	337·8	387·8	441·2	558·4	689·3	834·4	992·7	1551·8	2233·8	3040·7	3971·4	40
148·1	182·9	221·1	263·0	308·9	357·9	410 8	467·7	591·6	730·3	884·4	1052·0	1643·0	2366·4	3221·4	4407·6	45
156·9	193·5	234·3	278·7	327·3	379·3	435·3	495·6	626·9	773·9	937·2	1114·7	1742·0	2507·7	3413·7	4458·2	50
164·0	202·6	244·9	291·2	312·1	396·4	454·9	517·9	655·2	808·8	979·4	1165·0	1820·0	2620·7	3567·5	4660·0	55
171·0	210·9	255·5	303·9	356·9	413·8	474·9	540·4	684·0	844·2	1021·9	1215·8	1901·1	2735·9	3724·1	4864·1	60
185·0	228·1	276·1	328·5	385·9	447·1	513·2	584·2	739·1	912·3	1104·8	1314·1	2053·0	2956·3	4024·3	5255·7	70
197·5	243·8	296·0	351·1	412·1	477·8	548·4	624·0	789·8	975·1	1180·0	1404·2	2194·0	3159·2	4300·3	5616·6	80
209·4	258·6	312·9	372·3	437·1	506·8	581·7	661·8	837·7	1033·9	1251·5	1489·0	2327·7	3350·7	4561·0	5957·1	90
220·8	272·5	329·8	392·5	460·7	534·2	613·1	697·6	883·0	1090·1	1319·2	1569·8	2452·9	3531·9	4807·6	6279·2	100
312·2	385·4	466·4	555·1	651·6	755·5	867·1	986·6	1248·8	1542·0	1865·7	2220·2	3469·0	4995·0	6799·2	8880·5	200
382·4	472·1	571·2	679·8	798·0	925·2	1062·0	1208·3	1529·4	1888·2	2085·0	2719·1	4248·6	6117·5	8327·2	10876·0	300
441·5	545·1	659·5	784·9	921·4	1068·4	1226·3	1395·2	1765·9	2180·3	2638·4	3139·6	4905·8	7063·7	9615·2	12558·0	400
493·7	609·5	737·4	877·6	1030 3	1194·5	1371·1	1559·9	1974·5	2437·8	2950·0	3510·5	5485·0	7898·0	10751·0	14042·0	500
540·8	667·6	807·7	961·4	1138·6	1318·5	1501·9	1708·8	2162·9	2670·4	3231·5	3845·4	6008·0	8651·5	11777·0	15381·0	600
584·1	721·1	872·5	1038·4	1219·0	1413·3	1622·2	1845·7	2336·2	2884·3	3490·4	4154·1	6490·0	9344·7	12720·0	16614·0	700
624·4	770·9	932·7	1111·1	1303·2	1510·9	1734·2	1973·1	2497·5	3084·0	3731·4	4440·4	6938·0	9990·0	13598·0	17761·0	800
662·3	817·6	989·2	1177·4	1382·2	1602·5	1839·4	2092·7	2648·9	3270·4	3957·6	4709·4	7358·7	10596·0	14423·0	18838·0	900
698·1	861·9	1042·9	1241·1	1457·0	1689·2	1938·9	2206·0	2792·2	3446·5	4171·8	4963·5	7758·9	111.9·0	15203·0	19857·0	1000

required diameter. Example:—Required ga's. per sec. passing through a 3-in. pipe, under head and 1 ft. head) = 9·746 × 2·453 = 23·906 gals. per sec. (*J. H. Horobin.*)

TABLE GIVING HEAD NECESSARY TO DELIVER A GIVEN NUMBER OF GALLONS OF WATER PER MINUTE THROUGH VARIOUS-SIZED PIPES.

Gallons per Minute.	Diameter of Pipes in Inches.																		
	1	2	3	4	5	6	7	8	9	10	12	14	15	16	18	20	22	24	30
	Factors.																		
15	1·02																		
20	1·64	·051																	
55		·392	·051																
80		·823	·108																
120			·243	·057															
180			·549	·131															
200				·160	·052														
300				·361	·118														
350					·161	·064													
500					·329	·132	·061												
700						·259	·120												
750						·297	·137	·0706											
900							·198	·101											
1,000							·244	·125	·069										
1,150								·180	·100										
1,300								·212	·117	·069									
1,600									·178	·105									
2,000										·164	·066								
2,200										·199	·080								
2,800											·129	·050							
3,000											·141	·0688	·0487						
3,500												·0954	·0677	·0495					
4,000												·122	·0867	·0627					
4,500													·1108	·0804	·0446				
5,000													·135	·0981	·0544				
5,300															0630				
6,500															·0927	·0546			
7,000															·107	·063	·040		
8,500																·094	·060	·0374	
10,000																	·081	·0516	
10,600																		·0550	·0194
12,000																		·0744	·0243
14,500																			·0356

To find required head multiply length of pipe in yards by factor in table for given pipe and quantity of water required; the result is the head in feet.

EXAMPLE.—Head in feet required to pass 1,000 gallons of water per minute through 30 yards of pipe. In table under 8, and in line with 1,000, will be found ·125; then ·125 × 30 = 3·75, which is feet head required.

The figures in the table are true for plain, smooth, straight pipes. Additional head must be allowed if pipes are at all of a rough nature or broken in contour internally. For each right-angle bend having a radius of about twice the diameter of pipe, and passing the average quantities given in table, about 1½ ins. of additional head must be allowed.

(*T. Box.*)

TABLE SHOWING GALLONS OF WATER PER MINUTE CORRESPONDING TO SPEED OF WATER THROUGH PIPES, IN FEET PER SECOND.

Diameter of Pipe.	Speed of Water through Pipes. Feet per Second.									
	1	5	6	7	8	9	10	11	12	13
Ins.	Gallons per Minute.									
1	2·04	10·2	12·2	14·2	16·3	18·3	20·4	22·4	24·4	26·5
2	8·16	40·8	48·9	57	65·2	73·4	81 6	89·7	97·8	106
3	18·19	94·9	109	127	153	163	181 9	200	218	236
4	32	162	195	227	260	293	324	357	390	421
5	50	250	300	350	400	450	500	550	600	650
6	73	365	440	514	558	661	735	808	882	955
7	100	500	600	700	800	900	1,000	1,100	1,200	1,300
8	130	650	780	910	1,040	1,170	1,300	1,430	1,560	1,690
9	165	825	970	1,155	1,320	1,485	1,650	1,815	1,980	2,145
10	204	1,020	1,224	1,428	1,632	1,836	2,040	2,244	2,448	2,652
12	294	1,472	1,736	2,030	2,325	2,620	2,945	3,210	3,515	3,800
14	400	2,000	2,400	2,800	3,200	3,600	4,000	4,400	4,800	5,200
15	461	2,305	2,766	3,227	3,688	4,149	4,610	5,051	5,532	5,993
16	520	2,600	3,120	3,640	4,160	4,680	5,200	5,720	6,240	6,760
18	662	3,313	3,975	4,638	5,300	5,963	6,626	7,288	7,950	8,613
20	820	4,100	4,920	5,740	6,560	7,380	8,200	9,020	9,840	10,660
22	990	4,950	5,940	6,930	7,920	8,910	9,900	10,890	11,780	11,270
24	1,175	5,875	7,020	8,242	9,420	10,957	11,775	12,952	14,130	15,307
27	1,491	7,455	8,946	10,437	11,928	13,419	14,910	16,401	17,882	19,373
30	1,837	9,187	11,025	12,862	14,700	16,538	18,375	20,212	22,050	23,887
33	2,227	11,137	13,365	15,592	17,820	20,047	22,275	24,502	26,730	28,957
36	2,625	13,125	15,750	18,375	21,000	23,625	26,250	28,875	31,500	34,125
39	3,110	15,550	18,660	21,770	24,880	27,990	31,110	34,210	37,320	40,430
42	3,600	18,000	21,600	25,200	28,800	32,400	36,000	39,600	42,200	46,800
45	4,125	20,625	24,750	28,875	33,000	37,125	41,250	45,375	49,400	53,625
48	4,720	23,600	28,320	33,040	37,760	42,480	47,200	51,920	56,640	61,360
51	5,308	26,540	31,848	37,156	42,464	47,772	53,080	58,388	63,696	69,904
54	5,490	29,700	35,640	41,580	47,520	53,460	59,400	65,340	71,280	77,220
60	7,365	36,825	44,190	51,555	58,920	66,295	73,650	81,015	88,380	95,745

(*Gwynnes Pumps, Ltd.*)

METROPOLITAN WATER BOARD STANDARD SOCKETS (Table, page 1430)

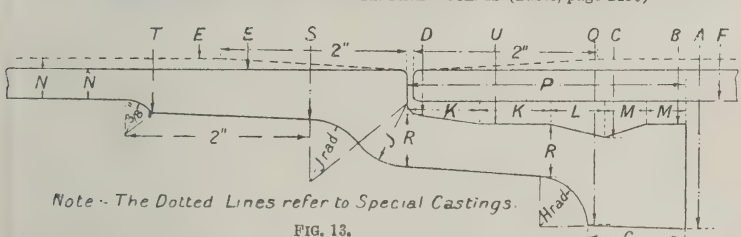


FIG. 13.

The joints of the above standard pipes Fig. 13 are made by caulking a ring of lead wire ¼ in. in diameter to the bottom of the socket, then running in molten lead for the remainder of the depth, and afterwards caulking or 'setting up.' Alternatively, lead wool may be caulked in for the whole depth

METROPOLITAN WATER BOARD.

STANDARD SOCKETS FOR STRAIGHT AND SPECIAL CAST IRON PIPES.

(See page 1429 for reference.)

Size of Pipe.	A	B	C	D	E*	E†	F	G	H	J	K	L	M	N*	N†	P	Q	R	S	T	U	Length of Pipes over all.				Weight of Pipe and Socket.			Weight of Pipe and Socket, 3 ft. shorter in Length.			Lbs. weight of Lead in each Joint.
																						Ft.	In.	Ft.	In.	Cwt.	Qrs.	Lbs.	Cwt.	Qrs.	Lbs.	
2	$5\frac{3}{8}$	$3\frac{1}{8}$	$3\frac{7}{16}$	$2\frac{7}{8}$	2	2	$2\frac{3}{8}$	$1\frac{1}{16}$	$\frac{9}{16}$	$1\frac{1}{16}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	3	$5\frac{1}{4}$	$\frac{9}{16}$	3	$2\frac{7}{8}$	3	6	3	—		0	2	0	—			3
3	$6\frac{3}{4}$	$4\frac{1}{4}$	$4\frac{9}{16}$	4	3	$2\frac{7}{8}$	$3\frac{3}{4}$	$1\frac{1}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{3}{8}$	$\frac{1}{2}$	3	$6\frac{5}{8}$	$\frac{5}{8}$	$4\frac{1}{8}$	4	$4\frac{1}{4}$	9	3	—		1	0	14	—			$4\frac{1}{2}$
4	$8\frac{1}{8}$	$5\frac{3}{8}$	$5\frac{1}{16}$	$5\frac{1}{8}$	4	$3\frac{7}{8}$	$4\frac{7}{8}$	$1\frac{5}{16}$	$\frac{11}{16}$	$1\frac{3}{16}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{9}{16}$	3	8	$1\frac{1}{16}$	$5\frac{1}{4}$	$5\frac{1}{8}$	$5\frac{3}{8}$	9	3	—		1	3	0	—			$5\frac{1}{2}$
5	$9\frac{1}{8}$	$6\frac{3}{8}$	$6\frac{11}{16}$	$6\frac{1}{8}$	5	$4\frac{7}{8}$	$5\frac{7}{8}$	$1\frac{3}{16}$	$\frac{11}{16}$	$1\frac{3}{16}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{9}{16}$	3	9	$1\frac{1}{16}$	$6\frac{1}{4}$	$6\frac{1}{8}$	$6\frac{3}{8}$	9	3	—		2	0	14	—			$6\frac{1}{2}$
6	$10\frac{1}{8}$	$7\frac{1}{2}$	$7\frac{13}{16}$	$7\frac{1}{4}$	6	$5\frac{7}{8}$	7	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$3\frac{1}{4}$	$10\frac{3}{8}$	$\frac{3}{4}$	$7\frac{3}{8}$	$7\frac{1}{4}$	$7\frac{1}{2}$	9	$3\frac{1}{4}$	—		2	3	14	—			8
7	$11\frac{1}{4}$	$8\frac{1}{2}$	$8\frac{13}{16}$	$8\frac{1}{4}$	7	$6\frac{7}{8}$	8	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$3\frac{1}{4}$	$11\frac{3}{8}$	$\frac{3}{4}$	$8\frac{3}{8}$	$8\frac{1}{4}$	$8\frac{1}{2}$	12	$3\frac{1}{4}$	9	$3\frac{1}{4}$	4	1	7	3	1	7	9
8	$12\frac{3}{8}$	$9\frac{5}{8}$	$9\frac{15}{16}$	$9\frac{3}{8}$	8	$7\frac{3}{8}$	$9\frac{1}{4}$	$1\frac{5}{16}$	$\frac{13}{16}$	$1\frac{5}{16}$	$\frac{7}{8}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{9}{16}$	$\frac{11}{16}$	$3\frac{1}{4}$	$12\frac{3}{4}$	$\frac{13}{16}$	$9\frac{1}{4}$	$9\frac{3}{8}$	$9\frac{5}{8}$	12	$3\frac{1}{4}$	9	$3\frac{1}{4}$	5	1	21	4	0	21	10
9	$13\frac{7}{8}$	$10\frac{5}{8}$	$10\frac{15}{16}$	$10\frac{3}{8}$	9	$8\frac{7}{8}$	$10\frac{1}{4}$	$1\frac{5}{16}$	$\frac{13}{16}$	$1\frac{5}{16}$	$\frac{7}{8}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{9}{16}$	$\frac{11}{16}$	$3\frac{1}{4}$	$13\frac{3}{4}$	$\frac{13}{16}$	$10\frac{1}{4}$	$10\frac{3}{8}$	$10\frac{5}{8}$	12	$3\frac{1}{4}$	9	$3\frac{1}{4}$	6	0	7	4	2	21	11
10	$15\frac{1}{4}$	$11\frac{3}{4}$	$12\frac{1}{16}$	$11\frac{1}{4}$	10	$9\frac{7}{8}$	$11\frac{1}{4}$	$1\frac{3}{8}$	$\frac{7}{8}$	1	1	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$3\frac{1}{2}$	$15\frac{1}{8}$	$\frac{7}{8}$	$11\frac{5}{8}$	$11\frac{1}{2}$	$11\frac{3}{4}$	12	$3\frac{1}{2}$	9	$3\frac{1}{2}$	7	2	0	5	3	0	14
12	$17\frac{3}{8}$	$13\frac{7}{8}$	$14\frac{3}{16}$	$13\frac{5}{8}$	12	$11\frac{7}{8}$	$13\frac{3}{8}$	$1\frac{7}{16}$	$\frac{15}{16}$	$1\frac{1}{16}$	1	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{11}{16}$	$\frac{13}{16}$	$3\frac{1}{2}$	$17\frac{1}{2}$	$\frac{15}{16}$	$13\frac{5}{8}$	$13\frac{3}{8}$	$13\frac{7}{8}$	12	$3\frac{1}{2}$	9	$3\frac{1}{2}$	9	3	7	7	2	0	17
14	20	16	$16\frac{5}{16}$	$15\frac{3}{8}$	14	$13\frac{7}{8}$	$15\frac{1}{2}$	$1\frac{1}{2}$	1	$1\frac{1}{8}$	1	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{3}{4}$	$\frac{7}{8}$	$3\frac{1}{2}$	$19\frac{7}{8}$	1	$15\frac{5}{8}$	$15\frac{3}{4}$	16	12	$3\frac{1}{2}$	—		12	2	0	—			$19\frac{1}{2}$
15	21	17	$17\frac{5}{16}$	$16\frac{3}{8}$	15	$14\frac{7}{8}$	$16\frac{1}{2}$	$1\frac{1}{2}$	1	$1\frac{1}{8}$	1	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{3}{4}$	$\frac{7}{8}$	$3\frac{1}{2}$	$20\frac{7}{8}$	1	$16\frac{5}{8}$	$16\frac{3}{4}$	17	12	$3\frac{1}{2}$	—		13	1	7	—			$20\frac{1}{2}$
16	$22\frac{1}{2}$	$18\frac{1}{4}$	$18\frac{9}{16}$	18	16	$15\frac{7}{8}$	$17\frac{5}{8}$	$1\frac{9}{16}$	$1\frac{1}{16}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{13}{16}$	$\frac{15}{16}$	$3\frac{3}{4}$	$22\frac{3}{8}$	$1\frac{1}{16}$	18	$17\frac{5}{8}$	$18\frac{1}{4}$	12	$3\frac{3}{4}$	—		15	1	21	—			28
18	$24\frac{1}{2}$	$20\frac{1}{4}$	$20\frac{9}{16}$	20	18	$17\frac{7}{8}$	$19\frac{3}{8}$	$1\frac{9}{16}$	$1\frac{1}{16}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{13}{16}$	$\frac{15}{16}$	$3\frac{3}{4}$	$24\frac{3}{8}$	$1\frac{1}{16}$	20	$19\frac{5}{8}$	$20\frac{1}{4}$	12	$3\frac{3}{4}$	—		17	1	0	—			31
20	$26\frac{3}{8}$	$22\frac{3}{8}$	$22\frac{11}{16}$	$22\frac{1}{8}$	20	$19\frac{7}{8}$	$21\frac{3}{4}$	$1\frac{5}{8}$	$1\frac{1}{8}$	$1\frac{5}{16}$	$1\frac{1}{8}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{7}{8}$	1	$3\frac{3}{4}$	$26\frac{3}{4}$	$1\frac{1}{8}$	$22\frac{1}{8}$	22	$22\frac{3}{8}$	12	$3\frac{3}{4}$	—		20	2	21	—			34
21	$28\frac{1}{4}$	$23\frac{1}{2}$	$23\frac{13}{16}$	$23\frac{1}{4}$	21	$20\frac{7}{8}$	$22\frac{7}{8}$	$1\frac{11}{16}$	$1\frac{3}{16}$	$1\frac{3}{8}$	$1\frac{1}{4}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{15}{16}$	$1\frac{1}{16}$	4	$28\frac{1}{8}$	$1\frac{3}{16}$	$23\frac{1}{4}$	$23\frac{1}{8}$	$23\frac{1}{2}$	12	4	—		23	1	0	—			39
22	$29\frac{1}{4}$	$24\frac{1}{2}$	$24\frac{13}{16}$	$24\frac{1}{4}$	22	$21\frac{7}{8}$	$23\frac{7}{8}$	$1\frac{11}{16}$	$1\frac{3}{16}$	$1\frac{3}{8}$	$1\frac{1}{4}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{15}{16}$	$1\frac{1}{16}$	4	$29\frac{1}{8}$	$1\frac{3}{16}$	$24\frac{1}{4}$	$24\frac{1}{8}$	$24\frac{1}{2}$	12	4	—		24	1	7	—			$40\frac{1}{2}$
24	$31\frac{5}{8}$	$26\frac{5}{8}$	$26\frac{15}{16}$	$26\frac{3}{8}$	24	$23\frac{7}{8}$	26	$1\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{7}{16}$	$1\frac{1}{4}$	$\frac{5}{8}$	$\frac{7}{16}$	1	$1\frac{1}{8}$	4	$31\frac{1}{2}$	$1\frac{1}{4}$	$26\frac{3}{8}$	$26\frac{1}{4}$	$26\frac{5}{8}$	12	4	—		28	1	7	—			44
30	$38\frac{3}{8}$	$32\frac{7}{8}$	$33\frac{3}{16}$	$32\frac{5}{8}$	30	$29\frac{7}{8}$	$32\frac{1}{4}$	$1\frac{7}{8}$	$1\frac{3}{8}$	$1\frac{9}{16}$	$1\frac{1}{4}$	$\frac{5}{8}$	$\frac{7}{16}$	$1\frac{1}{8}$	$1\frac{1}{4}$	4	$38\frac{1}{4}$	$1\frac{3}{8}$	$32\frac{5}{8}$	$32\frac{1}{2}$	$32\frac{7}{8}$	12	4	—		39	2	14	—			$54\frac{1}{2}$
33	$41\frac{3}{4}$	36	$36\frac{5}{16}$	$35\frac{3}{4}$	33	$32\frac{7}{8}$	$35\frac{3}{8}$	$1\frac{15}{16}$	$1\frac{7}{16}$	$1\frac{5}{8}$	$1\frac{1}{4}$	$\frac{5}{8}$	$\frac{7}{16}$	$1\frac{3}{16}$	$1\frac{5}{16}$	4	$41\frac{5}{8}$	$1\frac{7}{16}$	$35\frac{3}{4}$	$35\frac{5}{8}$	36	12	4	—		45	3	7	—			60
36	$44\frac{3}{4}$	39	$39\frac{1}{16}$	$38\frac{3}{4}$	36	$35\frac{7}{8}$	$38\frac{3}{8}$	$1\frac{15}{16}$	$1\frac{7}{16}$	$1\frac{5}{8}$	$1\frac{1}{4}$	$\frac{5}{8}$	$\frac{7}{16}$	$1\frac{3}{16}$	$1\frac{5}{16}$	4	$44\frac{5}{8}$	$1\frac{7}{16}$	$38\frac{3}{4}$	$38\frac{5}{8}$	39	12	4	—		49	3	7	—			65
42	$51\frac{5}{8}$	$45\frac{1}{8}$	$45\frac{7}{16}$	$44\frac{5}{8}$	42	$41\frac{7}{8}$	$44\frac{1}{2}$	2	$1\frac{1}{2}$	$1\frac{11}{16}$	$1\frac{1}{4}$	$\frac{5}{8}$	$\frac{7}{16}$	$1\frac{1}{4}$	$1\frac{3}{8}$	4	51	1	$44\frac{7}{8}$	$44\frac{3}{4}$	$45\frac{1}{8}$	12	4	—		61	0	21	—			75
48	$57\frac{5}{8}$	$51\frac{1}{8}$	$51\frac{7}{16}$	$50\frac{7}{8}$	48	$47\frac{7}{8}$	$50\frac{1}{2}$	2	$1\frac{1}{2}$	$1\frac{11}{16}$	$1\frac{1}{4}$	$\frac{5}{8}$	$\frac{7}{16}$	$1\frac{1}{4}$	$1\frac{3}{8}$	4	57	$1\frac{1}{2}$	$50\frac{7}{8}$	$50\frac{3}{4}$	$51\frac{1}{8}$	12	4	—		69	2	7	—			85

* Straight Pipes. † Special Castings.

WEIGHTS OF LEAD PIPE REQUIRED BY WATER COMPANIES

Company.	Weights per Yard, of Lead Pipe.						Minimum Depth of Underground Pipes.	Weight, per Yard, of Warning Pipe.		
	⅝ In.	½ In.	⅜ In.	¼ In.	1 In.	1¼ In.		½ In.	¾ In.	1 In.
	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.		Lbs.	Lbs.	Lbs.
Metropolitan Water Board	5	6	7½	9	12	16	2 ft. 6 ins.	3	5	7
Colne Valley.	5	7	9	11	16	—	3 ft.	—	—	—
Croydon .	4	5	7	9	12	16	2 ft.	—	—	—
East Surrey .	5	6	8	10	14	—	—	—	—	—
,, when pressure exceeds 300 ft.	6	7	9	11	16	—	—	—	—	—
South Essex.	—	6	7½	9	12	16	1 ft. 6 ins.	—	—	—
Sutton . .	—	6	7½	9	12	16	2 ft. 6 ins.	3	5	7
West Surrey.	4	5½	—	9	14	20	3 ft.	—	—	—

Sizes of Lead Pipe are Internal. 1¼ is the usual Minimum Size for Closet Pipe, and 9 lbs. per Yard Minimum Weight.

Comparative Value of Cast-Iron, Wrought-Iron, and Steel Drain Pipes.

Extra heavy cast-iron soil pipes in a number of buildings in nearly all cases had the cast-iron vents evenly pitted on the inside, but in the house drains and branch wastes the lead-caulked joints had opened up more or less. Black steel vents showed scaling to a considerable depth after only 10 years' use. Galvanising was observed to be less of a protection against corrosion on steel pipes than on wrought-iron pipes. The galvanised coating does not adhere so firmly to the smooth surface of steel pipes as to the comparatively rougher surface of wrought-iron pipes. The galvanising on steel pipes showed signs of disappearing within 10 to 20 years. The investigation furnished ample evidence in favour of genuine wrought-iron pipe as a far more durable material for house drainage purposes than steel pipe.

(*W. P. Gerhard, Proc. of the American Soc. of Mech. Engineers.*)

Stoneware Drain Pipes (p. 1360) have the advantage over all the foregoing, in not corroding at all.

Unbalanced Pressures on Bends, etc.

Where any considerable pressure has to be sustained by pipes of large size the unbalanced pressures on bends, etc., must be resisted, horizontally by strutting or tying, vertically by strutting or anchoring. The unbalanced pressure on any bend may be found thus :—

Unbalanced pressure on back of bend = the sum of the areas of the ellipses shown in the elevation × the pressure per unit of area, or A × B × ·7854 × 2P = unbalanced pressure, fig. 14.

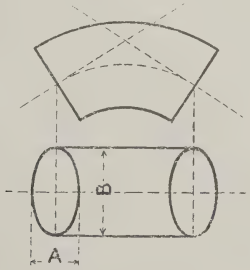


FIG. 14.

WATER TANKS.

Water tanks for house supplies should be covered from the light in order to prevent vegetable growth, but they should be ventilated. A good form of ventilator is an ordinary house sewer air inlet with the mica flaps removed. These should be placed, one or more, according to the size of the tank, at each end of the tank, standing a foot or two above the cover. At one end their openings, which are lateral, should face to, say, the south-west, and at the other end in the opposite direction, so as to set up a flow of air.

The cheapest tanks, to be durable, are excavated partly in the ground, the removed earth being banked against the walls built above ground and rammed at an inward slope so as to afford support against the water pressure. The walls may be of brick or of concrete, rendered internally in good cement mortar. The banks of spoil should be neatly trimmed and sodded or grass sown, and in this way a tank may be almost hidden. Where economy of material is an object, a tank is best circular, since the bank outside can be harder rammed without danger of turning the walls inwards.

These 'tanks' should not be confounded with 'cisterns.'

SECTION XXVII

METERING AND MECHANICAL TESTING APPLIANCES.

METERING OF WATER, AIR, GAS, STEAM, AND COAL.
Contributed by J. L. Hodgson, B.Sc., Assoc.M.Inst.C.E., M.I.Mech.E. A.F. Royal Ae. Soc.)
(pp. 1435–1450)

REVOLUTION INDICATORS.
(pp. 1450–1452)

GUEST & CHRIMES LTD

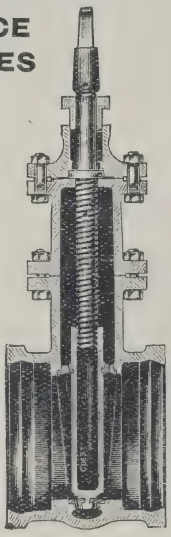
ROTHERHAM

Telegrams: 'Guest, Rotherham.' *Telephone: 205 Rotherham (2 lines)*

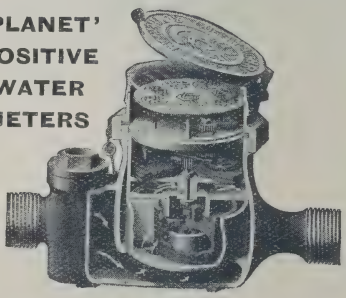
SLUICE VALVES

For Water, Steam, Sewage, Gas and Oil.

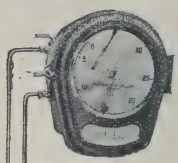
All Sizes from 2-in. upwards.



'PLANET' POSITIVE WATER METERS



Also
'TURBINE,'
'CAPSTAN' FAN, 'SPIRAL'
and
VENTURI METERS
with
PATENT RECORDERS.



Recorders Supplied in Various Types to meet Customers' Requirements

TYPE B2E.
Patent Recorder and Integrator

HIGHEST QUALITY BRASS TAPS & FITTINGS.

WATERWORKS CONSTRUCTIONAL ARTICLES OF EVERY DESCRIPTION.

London Office: **Cecil Chambers, 76 Strand, W.C.2**

'Grams: **Birbinchri, Rand, London.** 'Phone: **Temple Bar 4200**

SECTION XXVII

METERING AND MECHANICAL TESTING APPLIANCES.

METERING OF WATER, AIR, GAS, STEAM, AND COAL.

(Contributed by J. L. Hodgson, B.Sc., Assoc.M.Inst.C.E., M.I.Mech.E.)

The metering of fluids is undertaken to enable running costs to be apportioned, sales to be effected on an equitable basis, to control the mixing of reagents in various chemical processes, or to detect wasteful or inefficient use so that working costs may be reduced and unnecessary additions to plant obviated, etc.

Various efficient and accurate types of water and gas meters have been in use for many years; it is only recently, however, that accurate and reliable air and steam meters have been developed. The first large installation of air meters was made on the Rand in 1911, where now more than 300 million h.p. hours per annum are metered, at an ascertained annual aggregate money saving of 130,000*l.**

The following is a brief summary of the principal classes of fluid and gaseous meters, with notes as to their application, and the names of the leading makers :—

1. Meters Based on Volume or Weight Measurement.

(*a*) Tanks or weighing. [For liquids, condensed steam.]
(*b*) Receivers of known capacity discharging between known pressures and temperatures. [For air and gas.]
(*c*) Piston displacement meters. [For water (Messrs. George Kent, Ltd., Luton; Messrs. Beck & Co. Ltd., Southwark, London; Messrs. Frost, Manchester).]
(*d*) Water-sealed drum meters. [For gas (The Gas Meter Co., London).]
(*e*) Bellows meters. [For gas (The Gas Meter Co. Ltd.).]

2. Meters which Measure the Velocity at a Point or Points in the Cross Section of the Stream.

(*a*) Pitot tube meters. [For water, air and gas (George Kent, Ltd.). For air and gas (Rotary Meter Co. (1905), Ltd.) For steam (The General Electric Co. of Schenectady and British Thomson Houston Co. Ltd. of Rugby).]
(*b*) Professor Morris's hot wire velocity indicator. [For air and gas (Cambridge Instrument Co. Ltd., Cambridge).]
(*c*) Impact meters. [For measuring air velocity in mine tunnels (Messrs. Davis).]
(*d*) Anemometer meters. [For water in open channels (George Kent, Ltd.). For air in conduits and passages (Messrs. Davis).]
(*e*) Floats. [For measuring the velocity of water in open channels.]

3. Meters which Depend on the Drop of Pressure due to an Obstruction Placed in the Stream.

(*a*) Weirs. [For liquids in open channels (Lea Recorder Co. Ltd., 28 Deansgate Manchester, and George Kent, Ltd.).]
(*b*) Venturi tubes. [For water, air, steam and gas (George Kent, Ltd.).]
(*c*) Nozzles. [For gas (Messrs. George Kent, Ltd.). For steam (The General Electric Co. of Schenectady, and Messrs. George Kent, Ltd.).]
(*d*) Circular, cord, or tongue orifices. [For water, air, gas and steam (George Kent, Ltd.).]
(*e*) Variable orifices. [For water, air, gas and steam (George Kent, Ltd.).]

---

* See Hodgson, *Proc. Inst. C.E.*, **Vol.** CCIV, Part 2, and Mavor, *Trans. Inst. Mining Engineers*, **Vol. L**, Part 4.

ROTARY OR TURBINE METERS.

(*a*) Which measure the whole of the flow passing. [For water (Guest & Chrimes Ltd., Rotherham; Messrs. Hayward Tyler, Luton; George Kent, Ltd.; the Self Engineering Co., London). For air and gas (Rotary Meter Co. (1905) Ltd., Victoria Works, Whitefield, Manchester, and George Kent, Ltd.).]

(*b*) Which are placed in a shunt circuit. [For air, steam, and gas (George Kent, Ltd.).]

5. METERS BASED ON THE MOVEMENT OF A SPRING OR WEIGHT CONTROLLED OBSTRUCTION PLACED IN THE PIPE LINE.

(*a*) In which a disc moves axially in a circular chamber with conical or perforated walls. [For water (Palatine Engineering Co. Ltd., Bootle, Liverpool). For gas or air (Rotary Meter Co. (1905) Ltd., Victoria Works, Whitefield, Manchester). For steam (The Drayton Meter Co.).]

(*b*) In which a shaped plug opens up an orifice. [For steam (The St. John Steam Meter Co., New York).]

(*c*) In which a weighted hinged gate is opened by the flow. [For water and air (George Kent, Ltd.).]

6. METERS BASED ON THE DROP OF PRESSURE CAUSED BY FRICTIONAL LOSSES IN A LENGTH OF CHANNEL OR PIPE LINE.

[For measuring irrigation water (George Kent, Ltd.).]

7. METERS BASED ON THE INTRODUCTION OF A MEASURED AMOUNT OF (*a*) HEAT, OR (*b*) CHEMICAL REAGENT, AND THE CONSEQUENT RISE OF TEMPERATURE OF THE FLUID, OR DILUTION OF THE REAGENT AT SOME POINT FURTHER DOWNSTREAM.

(*a*) The Thomas meter. [For air and gas (Messrs. Humphreys & Glasgow, London).]

(*b*) [A method used for measuring the discharge of rivers and canals, recently further developed for the measurement of air and steam flows. Salt solution is usually introduced when measuring water-discharges, and ammonia or carbon dioxide gas when measuring steam or air flows.]

8. METERS BASED ON INCREASE OF WEIGHT DUE TO CHEMICAL ABSORPTION.

Notes.

Meters of classes 1 (*c*), (*d*) and (*e*) can be made accurate to within plus or minus 1 per cent. from zero flow up to the maximum flow which they are designed to measure. The chief source of error of meters of class 1 (*d*) is variation in the water level, and in meters of classes 1 (*b*) and 1 (*e*) wear of the valves.

Meters of classes 2 (*a*), (*b*) and (*c*) are usually only accurate down to one-third or one-sixth of the maximum flow for which they are designed, as their indication depends upon the square of the velocity of the fluid. Few anemometers are accurate when working at less than one-tenth of their maximum rated capacity.

Meters of classes 3 (*a*), (*b*), (*c*) and (*d*) are usually accurate down to one-fifth or one-tenth of the maximum flow. If specially sensitive differential pressure-measuring devices are provided, such meters may be made accurate down to one-twentieth or one-thirtieth of the maximum flow. By using a variable orifice, the range over which accurate measurement is possible with any given indicator or recorder is greatly extended. These meters have the very real advantage of having in the pipe line no moving parts which can be deranged by dirt or erosion. The Venturi tube, with its smooth contours which eliminate eddying, is a valuable device when measuring dirty gas. Meters of this class show many ingenious arrangements for obtaining an accurate measurement of the differential pressure over a wide range and for compensating for variations in the specific gravities of the fluids or gases to be measured.

Meters of class 4 are not usually accurate in practice below one-tenth of the maximum flow, though many makers claim much higher ranges of accuracy. Endeavours have been made to increase the range of accurate measurement by opening up additional jets at low flows, or by introducing additional resistance at the high flows. While these devices frequently give excellent results under test conditions, they rapidly get out of adjustment under working conditions. Water meters of this class can often carry considerable overloads, thus greatly increasing the range over which accurate measurements may be obtained.

Meters of classes 5 (*a*), (*b*) and (*c*) are capable of high accuracy over a large range of flow when measuring clean fluids. They rapidly lose their accuracy, however, when measuring dirty fluids, owing to the small clearances becoming choked.

The accuracy of meters of class 6 is dependent upon an accurate knowledge of the area of the channel and its resistance coefficient. These are apt to change owing to silting up and the growth of weeds.

THREE
CELEBRATED
PAPERS

"THE ENGINEER"
(*Established 1856*)

"THE IRONMONGER"
(*Established 1859*)

"THE CHEMIST & DRUGGIST"
(*Established 1859*)

Proprietors:—

MORGAN BROTHERS (PUBLISHERS) LTD.
28 ESSEX STREET STRAND
LONDON W.C.2

FLUID METERING EQUIPMENT

Waterworks	-	Venturi Meters Weir Recorders Level Recorders Small Positive Meters Filtration Controllers
Power Station	-	Boiler Feed Meters Condensate Meters Steam Meters Combustion Meters
Oil Fields and Refineries		Mechanical Oil Meters Orifice Flow Meters Temperature Controllers Specific Gravity Recorders
Gas Works	-	Gas Meters Steam Meters Water Meters Automatic Controllers

COMPLETE FLUID METERING EQUIPMENT FOR EVERY INDUSTRY

Send for latest Catalogues

GEORGE KENT LTD.

LONDON & LUTON.

F. 1437

In the following pages the main features of some of the best known, or the most noteworthy, meters are described and illustrated.

Water Meters.

The Kent Mark III Positive Water Meter.

This meter is shown diagrammatically in fig. 1. Its sequence of operations is as follows:—

1. The piston A descends, and the disc P throws over the valve C.
2. The piston B descends, and the disc R throws over the valve D.
3. The piston A descends, and the disc Q throws over the valve C.
4. The piston B descends, and the disc S throws over the valve E; and so on.

Stops M limit the motion so that an exact quantity of water is measured off each stroke.

Projections K enter the outlet passage at the end of the stroke and prevent jar by slightly throttling the flow.

The cylinders are lined with brass, and the valves and valve faces are made of gun-metal. The valves are caused to oscillate slightly each time they are thrown over in order to keep the rubbing surfaces polished. For the same reason the pistons are caused to rotate a slight amount at each stroke.

The valves are held up against the port block by rollers L carried by springs which press inwards towards the valve faces.

The meter body is jointed along the line XY, which enables all parts to be readily accessible (actually the inlet and outlet flanges are at right angles to the positions shown in the diagram).

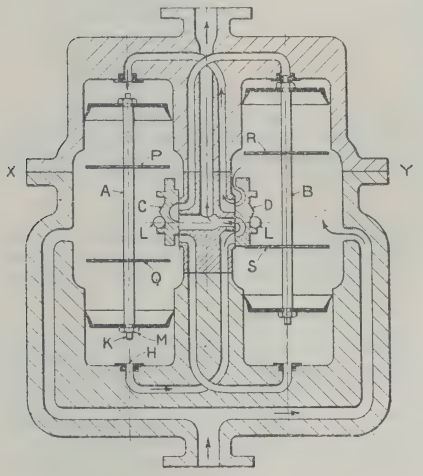


Fig. 1.

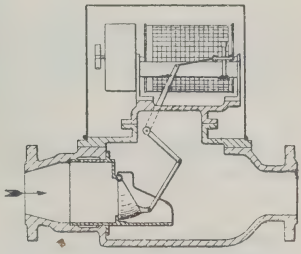


Fig. 2.

The Kent Waste Water Meter.

This meter (fig. 2) consists of a weighted gate pivoted on a horizontal axis and surrounded by a machined shield. The gate opens with increase in the flow, and takes up a definite position at each flow. Its motion is transmitted to the diagram through a stuffing box by means of link mechanism.

It is possible with this meter to measure with accuracy down to very small fractions of the full flow.

Owing to the absence of all sliding parts in this meter, the friction of the mechanism is extremely small.

The bulk of the meter, and the loss of pressure across it, are also reduced to a minimum by its arrangement, which allows of a direct flow between inlet and outlet.

The Beck 'Imperial' Water Meter.

In this meter (figs. 3 and 4) the rotation of the arm A attached to the spindle D, which drives the counter, causes the spherical valve cap V to rock on its seating S, thus opening the lower parts of each of the three cylinders C successively to the inlet and discharge sides of the meter.

The valve cap V is prevented from turning round on its own axis by the projections Q, which engage in recesses left in the valve seat S. The length of the stroke of the pistons is limited by the flange P on the valve cap V, which comes into rolling contact with a machined surface on the valve seat S.

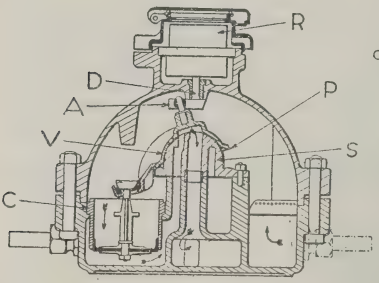


Fig. 3.

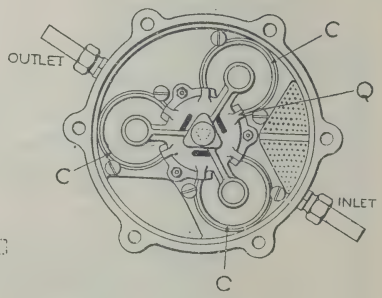


Fig. 4.

It will be seen that the valve cap V moves with a lens-grinding action. This ensures that it and the seating will remain polished and unscratched even when gritty water is being measured.

The valve cap and the pistons are made of hard gun-metal, the valve seat and bushes of vulcanite, and the cylinders of either vulcanite or gun-metal; vulcanite being used for water of exceptional hardness.

The Deacon Water Meter.

In the Deacon Water Meter (fig. 5) a weighted disc moves vertically in a conical shield. Its motion is transmitted to the diagram by means of thin wire passing through a gland. Guide plates are provided to steady the flow at the entrance to the conical shield. See also p. 1420.

The Kent Venturi Water Meter.

This meter (fig. 6) consists of (*a*) a Venturi tube VT through which the water passes; (*b*) a large mercury U-tube provided with floats FU and FT (which are displaced by the differential pressure which exists when water is passing between the upstream and throat pressure holes of the Venturi tube); and (*c*) diagram counter and recording mechanisms, which are actuated by the movements of the floats.

The motion of the floats is transmitted to the outside of the U-tube by means of gland spindles GU and GT, which actuate toothed wheels that engage with the racks RU and RT.

The law of flow through a Venturi tube is given by the equation:—

where,

$$Q = \frac{\Omega\, a_1}{7{\cdot}88}\left[\frac{h\,W}{n^2-1}\right]^{1/2},$$

Q = flow in lbs. per sec.; Ω = coefficient of discharge for the tube (usually about 0·99); a_1 = area of tube at upstream pressure holes in sq. ins.; na_1 = area of tube at throat pressure holes in sq. ins.; h = difference of pressure between upstream and throat pressure holes in ins. of water; W = weight in lbs. of 1 cub. ft. of the fluid passing.

The counter recording mechanism operates as follows:—A cylindrical cam LC, which has a slightly raised portion, is rotated once every ten minutes by clockwork. A pivoted arm A, which carries the roller L, which is kept in contact with the surface of the cam by a spring (not shown), is lowered by the rack to which it is attached when the flow increases. The greater the distance the arm is lowered, the larger is the fraction of each revolution of the cam during which the roller is on the unraised portion of the cam. By means of mechanism (not shown) it is arranged that a toothed wheel which drives the counter is put into gear with the clock train during the period that the roller L is on the unraised portion of the cam. The boundary line between the raised and the unraised portions of the cam is so shaped that when the pivoted arm A carrying

the roller L moves down a distance H from its zero position an amount proportional to $\sqrt{H}$ is added on to the counter reading. In this way, with suitable gear wheels in the counter train, a counter record of the amount of fluid that has passed is obtained.

Valves U and T are provided so that the upstream and throat pressures may be shut off, and the meter checked by opening the equalising valve E.

Any air that accumulates in the pressure pipes may be blown off through the air vents provided.

To reduce the chance of air getting into the pressure pipes, these should be taken off horizontally, and not at the top, as shown for clearness in the diagram.

These meters will measure accurately down to one-fourteenth of the maximum flow (and special meters of this type down to one-thirtieth of the maximum flow).

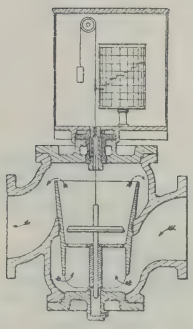


FIG. 5.

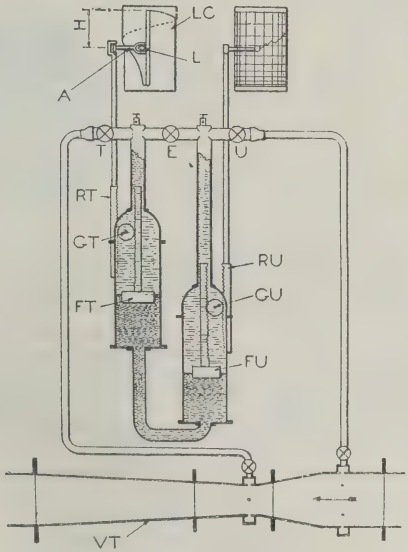


FIG. 6.

The functions of the downstream cone of the Venturi tube is to recover the kinetic energy which exists at the throat, thus enabling the major portion of the pressure difference which has been created between the upstream and throat pressure holes to be recovered.

The friction loss due to the insertion of the water Venturi tube is about 2 lbs. per sq. in. at the maximum flow.

Venturi meters have been used to measure water in all sizes of pipes from about ½ in. to 20 ft. in diameter. They provide the most convenient and accurate method known of measuring the flow of water in large mains. There is a minimum of obstruction to the flow, and there are no moving parts in contact with the flowing water.

Water Weighers.

The 'Avery' Automatic Water Weigher.

The principle of the 'Avery' Automatic Water Weigher (fig. 7) is that of an equal-armed beam A, with the weights contained in the weight box B, suspended at one end and the weigh hopper C at the other.

A quantity of water, determined by the weights in the weight box B, is allowed to enter the weigh hopper C through the feed valve D.

When the correct amount of water is obtained, the supply from the feed valve or valves D is, by means of a combination of levers, automatically cut off.

Immediately after the supply is cut off, the weigh hooper C automatically tips and discharges its load.

THE 'AVERY' AUTOMATIC WATER WEIGHER—*cont.*

As soon as the contents of the weigh hopper C have been discharged, the weigh hopper automatically returns and assumes the weighing position, at the same time opening the feed valve or valves D, and the cycle of operations is repeated.

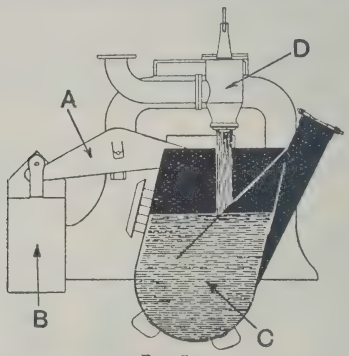


FIG. 7.

On the large-capacity machines there are two valves at D, the larger one feeding the main flow into the weigh hopper C, then closing, with the result that the final dribble of liquid to complete the weighing and ensure accuracy is fed through the supplementary small valve D.

A mechanical counter keeps an exact record of the number of times the machine operates.

Truck and Hopper Weighers.

THE 'AVERY' PATENT AUTOMATIC WEIGHBRIDGE AND TOTALISER.

This machine, fig. 8, automatically weighs trucks while at rest on the weighbridge. It also records on special counters the total weights during any period, and the number of weighings made by the machine.

If all the trucks have the same tare weight the machine can be arranged to record the total net weight of material passed over the scale.

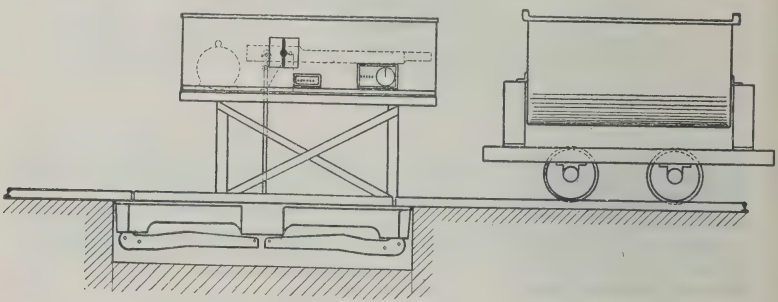


FIG. 8.

With suitable weighing levers, etc., the machine can be arranged for use with overhead ropeways, grabbing hoists for transporters, overhead runways carrying coke skips for gasworks, etc.

A special type of the mechanism is also used with the Avery 'Automatic Hopper Weigher and Totaliser,' which automatically records the weight of material discharged from grabs or skips. The material dealt with includes coal, coke, ore, pitch, etc., and pieces of any size from nuts to the largest lumps can be weighed, as there are no feed valves.

When grabs are used to unload barges or steamships this type of machine is most suitable, as the total intake of coal is then automatically recorded; also the load of each barge or ship can be ascertained. All the machines described are approved by the Board of Trade as legal weighing instruments, and the recorded weights are therefore beyond dispute.

Weir Recorders.

The Lea 'V' Notch Weir Recorder.

The discharge for a 90° V notch with sharp edges bevelled at 45° on the downstream side is given by the equation.

$$Q = 2{\cdot}48H^{2{\cdot}48}$$

where,

Q = discharge in cub. ft. per sec.; H = head, in ft. above the bottom of the notch.

In the Lea Recorder (fig. 9) the movement of the float F in the weir chamber, which is proportional to H, is transmitted to a cylindrical cam C. By means of this cam, a movement of the pen arm P, which is proportional to Q, is obtained.

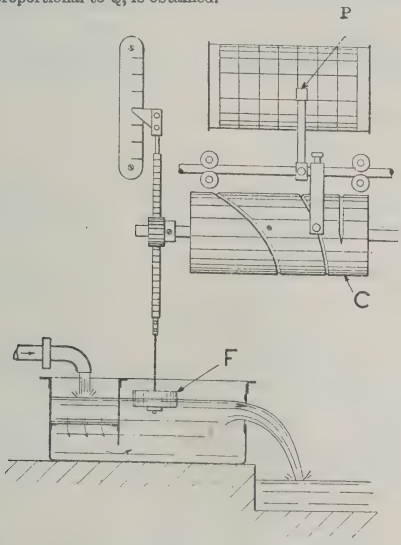


Fig. 9.

These recorders are specially suitable for the measurement of boiler feed-water and condensates, also rivers, streams, reservoir supplies, sewage effluents, etc., or in such cases where it is necessary to measure the water in some form of open channel.

In the Kent Weir Recorder a flat cam is used, and the straight line movements are replaced by movements about fixed pivots.

In the Wright Weir Recorder the weir notch is made of such a shape that the motion of the float is proportional to the quantity passing. In this recorder there is, therefore, no cam between the float and the recorder pen. The approximate shape of the weir notch is shown in fig. 10.

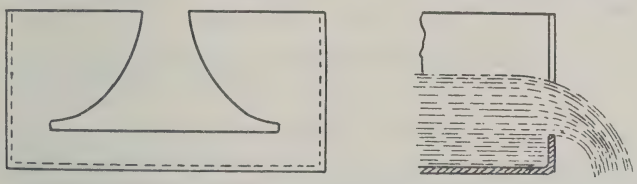


Fig. 10.

Notes on the Use of Orifices, Nozzles, and Venturi Tubes for Test Purposes.

Q = discharge in lbs./secs.;
E = coeff. of discharge;
a_1 = area of main in ins.;
a_2 = area of orifice on throat of nozzle in sq. ins.;
$n = a_1/a_2$;
$N = a_1 \div \sqrt{n^2 - 1}$;
d_1 = dia. of main in ins.;
d_2 = dia. of orifice on throat of nozzle in sq. ins.;
h = differential pressure across orifice or nozzle in ins. of water;
p_1 = abs. pressure in lbs./sq. ins. at upstream pressure hole;
p_2 = abs. press. in lbs./sq. ins. at downstream or throat pressure hole;
T_1 = abs. temp. in F.° at upstream pressure hole;
W_1 = specific weight of fluid at upstream pressure hole in lbs./cu. ft. = $2 \cdot 7P_1/T_1$ for air (values for water, steam, and other fluids can be obtained from various published tables);
μ = viscosity of the fluid in foot, pound, sec. units;
γ = ratio of the specific heats if the fluid is gaseous.

The discharge through an orifice, shaped nozzle, or Venturi tube for gaseous or liquid flows can be calculated from the formula:

$$Q = 0 \cdot 127 \ EN\sqrt{hW_1} \text{ lbs./secs.}$$

The value of the coefficient of discharge, E, is the same for the same values of the viscous flow criterion $Q/d_2\mu$, whatever be the density and viscosity of the fluid used, provided that when the fluid which is being metered is a gas, it does not appreciably change in density as it passes through the orifice or nozzle (*i.e.*, p_2/p_1 should be between 1·0 and 0·98).

For shaped nozzles and Venturi tubes, if the numerical value of the viscous flow criterion $Q/d_2\mu$ is greater than 3,000,

$$E = 0 \cdot 94\text{–}1 \cdot 00;$$

for orifices:

$$E = 0 \cdot 605\text{–}0 \cdot 615,$$

provided that:

(*a*) The numerical value of viscous flow criterion $Q/d_2\mu$ is greater than 1,200;
(*b*) d_2/d_1 is not more than 0·6, and that orifices are used which have:
(*c*) square edges if the orifice plate is thin (less than $\frac{1}{30}$th the diameter of the orifice);
(*d*) square edges bevelled off on the downstream side if the orifice plate is thick;
(*e*) pressure holes not more than one pipe diameter upstream, and not more than half a pipe diameter downstream of the orifice.
(*f*) the orifice or nozzle should be installed with at least 6 diameters upstream and 3 diameters downstream of straight parallel pipe free from bends, tees, branch pipes and control cocks.

When using orifices or nozzles for measuring flows of viscous liquids, curves connecting the differential pressure 'h' with the numerical factor multiplying $\sqrt{h/W_1}$ in the volume discharge equation, can be plotted for various values of the kinematic viscosity μ/W.

If the relation between the temperature and the kinematic viscosity for the liquid has been previously determined, a knowledge of the temperature enables the correct numerical factor for any observed head to be read from the plotted curves.

If the flow is gaseous and pulsating, sufficient capacity or throttling must be introduced between the source of pulsation and the metering point to smooth out the pulsation of flow at the metering point, otherwise the meter will read fast. In the case of liquid flows, an air vessel or an open tank may be used.

If gaseous discharge takes place above the critical pressure ratio (*i.e.* if p_2/p_1 is less than $\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$ when $n = \infty$), the following formulæ hold for shaped nozzles:

$$Q = 0 \cdot 325 \ EN\sqrt{p_1W_1} \text{ for air;}$$

$$= 0 \cdot 316 \ EN\sqrt{p_1W_1} \text{ for steam.}$$

For fuller treatment see Hodgson on 'The Orifice as a Basis of Flow Measurement,' *Proc. I.C.E.*, 1925; 'The Metering of Industrial Fluids,' *Proc. Inst. Mining Eng.*, Nov. 1927.

Useful variations of the formulæ will be found on p. 1308.

Gas Meters.

The 'Drum' Type Station Gas Meter.

In this meter (fig. 11) gas enters through the hollow axis in the four compartments of the drum in a pipe which is put just above the water level, and fills each compartment successively, and by its pressure causes the drum to revolve in the direction of the arrow and to register the volume passed on a set of dials.

Meters of this type are used to measure gas in mains of all sizes, from ¼ in. upwards.

Owing to their costliness, the larger meters of this description are now being superseded by new patterns of meter, such as the Thomas and the Kent-Hodgson Meters.

The main sources of error of this meter are deformation or corrosion of the drum, deposition of naphthalene in the drum, and variations in the water level There are various ingenious devices for preventing the meters from being over-filled, and for maintaining the correct water level. If the meter is overloaded, the water level is depressed, and the meter reads slow.

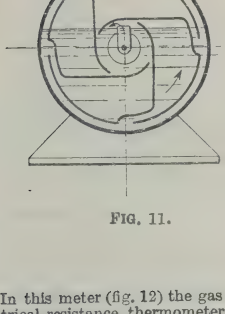


Fig. 11.

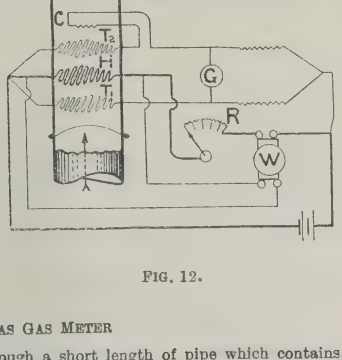


Fig. 12.

The Thomas Gas Meter

In this meter (fig. 12) the gas is passed through a short length of pipe which contains two electrical resistance thermometers T_1 and T_2, which form part of a Wheatstone Bridge. An electrical heater H is placed between the two thermometers. Any increase in the flow of gas causes a lowering of the temperature of the downstream thermometer and a deflection of the needle of the galvanometer G. This is made automatically, by means of a suitable mechanism, to reduce the resistance in the heater circuit, so that the temperature of the gas is always raised the same amount (about 2° F.) by the heater.

The readings on a Wattmeter W, placed in the heater circuit, give the number of cubic feet of gas reduced to standard temperature and pressure.

The Kent-Hodgson Venturi Gas Meter.

In this meter (fig. 13) the 'Venturi Head' $(p_1 - p_2)$ is measured by means of the bell B and an amount proportional to $(p_1 - p_2)$ is added on to the counter reading by means of the cam, feeler, ratchet, and pawl device C.F.R. every time the cam G, which is driven by the wet gas meter W, makes one revolution.

The speed of rotation of this meter is determined by the amount of gas that escapes to atmosphere through a small orifice at O, across which the pressure difference is maintained constant by means of the regulating valve RV.

In this way the counter is made to register the volume of gas passing through the Venturi tube, irrespective of variations in the specific gravity of the gas; the registration of the meter being identical with that of the ordinary Drum type Station Gas Meter.

A diagram, D, is provided which shows the rate of flow of gas at any instant. The readings of this diagram are, however, not corrected for variations in the specific gravity of the gas.

If tar, naphthalene, or other depositable matter settles in the Venturi throat, the meter tends

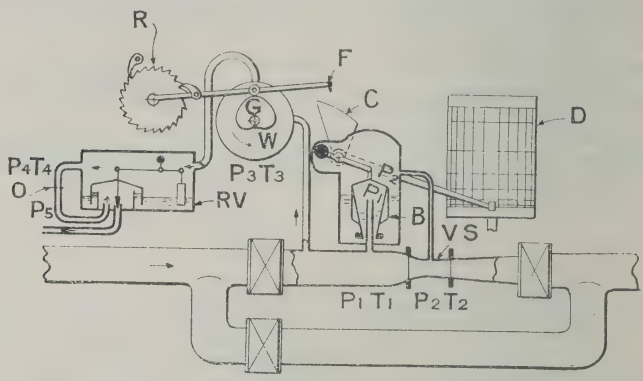


FIG. 13.

to read fast. It is found that when measuring coal gas, such deposits can be entirely prevented by keeping the walls of the throat at a few degrees higher temperature than that of the gas passing through.

The Venturi tube is installed on a by-pass, and the throat section VS is made so that it can easily be swung out of the pipe line for inspection and cleaning.

The loss of pressure at the maximum flow due to the insertion of the gas Venturi tube is arranged not to exceed 1 in. of water gauge.

The range of accurate registration is down to one-tenth or one-fifteenth of the maximum flow.

If it is desired to reduce the meter readings to standard cubic feet of dry gas, corrections must be made for variations in the barometric pressure, atmospheric temperature, and the tension of aqueous vapour by means of tabular numbers, as is the case with the Drum type Station Meter.

The Venturi Gas Meter is also made in a cheaper form in which the integrating mechanism is driven by clockwork, and any variation in the specific gravity, temperature, and moisture content of the gas is corrected for by means of tabular numbers, *i.e.* the meter readings are multiplied by

$$\frac{(p_1 - p'_1)}{p_1} \sqrt{\frac{W_D}{W_S} \cdot \frac{p_1}{T_1} \cdot \frac{T_S}{P_S}},$$

where, W_D = weight of a cub. ft. of dry gas at p_1 and T_1; W_S = weight of a cub. ft. of gas containing water vapour at p_1 and T_1; p'_1 = pressure of water vapour at temperature T_1, P_S and T_S the standard temperature and pressure.

See Hodgson, *Proc. Inst. C.E.*, vol. cciv., pp. 164–165.

THE THORP ROTARY GAS METERS.

In the Thorp Rotary Gas Meter (fig. 14), the counter C is actuated by an anemometer fan F, which rotates about a vertical axis.

The gas, after being carefully guided by a series of nozzles N into a vertical direction, impinges on the underside of the blades, thus helping to take the weight of the fan off the bottom pivot. The rate of revolution of the fan is proportional to the actual volume of gas passing,

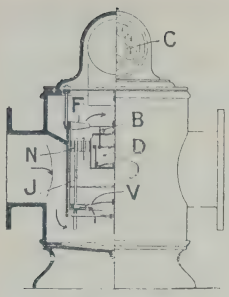


FIG. 14.

except at low flows, when the friction of the mechanism is appreciable. At these flows a weighted valve V (which lifts at the higher flows) causes the whole of the gas to pass through a few jets J, and so to impinge on the fan at an increased velocity. In this way the friction of the mechanism is compensated for, and accurate readings obtained down to one-tenth of the maximum flow.

The dashpot D is provided to steady the motion of the weighted valve. A brake B is attached to this valve, which stops the fan immediately the flow is turned off, as the valve then falls.

This meter registers the actual volume of gas passed, and is subject to the same corrections for variations in the temperature, pressure, and moisture content of the gas as is the ordinary Drum type Station Meter, or the Kent-Hodgson Venturi Gas Meter.

Clogging of the nozzles through which the gas discharges on to the fan causes the meter to read fast. A special form of meter in which the nozzles can be cleaned from the outside, and in which any deposit thrown off from the fan does not accumulate round it, has been devised for measuring dirty gas.

THE 'KENT-HODGSON' ROTARY GAS METER.

This meter (fig. 15) consists of a body casting inserted in the main, or by preference in a by-pass in the main, a turbine T placed in such a position that gas passing through the two nozzles J on the upstream side of the orifice K impinges on its blades, and a counter mechanism actuated by the turbine. The turbine, orifices, and nozzles can be inspected and cleaned through the hand-holes provided.

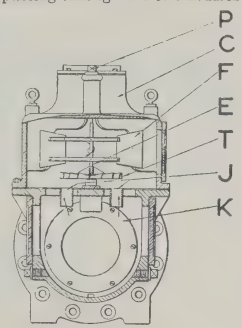


FIG. 15.

The meter carries the measurement of gas on the rotary principle to the utmost refinement. The speed of rotation of the turbine is kept low by means of the damping fan F, so that wear on the bearings is reduced to a minimum. The weight of the turbine is carried on a mercury float, so that the all-important pivot bearing never wears appreciably, and is also protected by being covered with mercury. The upper thrust bearing and the high-speed wheels of the counter train run in an oil bath. The motion of the turbine is transmitted to the counter dial by means of a magnetic drive similar to that shown in fig. 15, thus eliminating all gland friction and the possibility of moisture depositing on the counter glass.

The turbine chambers and the mechanisms they contain are identical, whatever be the size of the main. The only parts of the meter which vary in size are the body casting which contains the orifice, the orifice itself, and the change wheels which are fitted into the counter train. The length of the body casting in practically all sizes is 24 ins., so that the meter is very light to handle and requires only a short length of pipe line for its installation. It is accurate down to $\frac{1}{10}$th of full flow, and will stand considerable temporary overloads without damage.

HOT WIRE ANEMOMETER.

In this instrument a bare wire with a large temperature coefficient is heated electrically and exposed to the gas flow whose velocity it is required to measure. The velocity V of the gas can then be obtained from the relation

$$I^2 = A\theta + B\theta \left[V.W\, K_p \right]^{\frac{1}{2}}$$

where,

θ = temperature elevation of the wire ;
I = current in amps. ;
V = velocity of the gas ;
W = specific weight of the gas ;
K_p = specific heat of the gas at standard pressure ;
A & B = constants.

Referring to the diagram (fig. 16), R_1, R_2, and R_3 are resistances with no temperature coefficient, R_4 is the anemometer wire which is maintained at constant temperature when the gas velocity varies by adjusting the resistance R_5 so that the needle of the galvanometer G remains at zero. The current through R_4 is thus proportional to the reading of the high resistance voltmeter V.

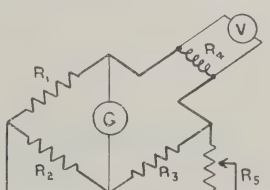


FIG. 16.

The formula given above holds for ordinary gas speeds (9·5 to 30 ft. per second). At speeds below these, free convection currents carry away a considerable proportion of the heat, and the formula requires modification.

The accuracy of the instrument is affected by changes in the temperature of the gas itself, and for this reason it is desirable, when high accuracy is required, to work at a temperature elevation of about 1,000° C. In this case the anemometer wire should be of platinum. For ordinary work a temperature elevation of 200° C. or 300° C. is sufficient, and the anemometer wire may be made of electrolytic nickel.

It will be noticed that the velocity of the gas is approximately proportional to the fourth power of the current. This makes the accurate range of the instrument small, and it also makes it unsuitable for obtaining the mean velocity of fluctuating flows.

In Professor MacGregor Morris's Hot-Wire Anemometer, as made by The Cambridge Instrument Co., Ltd., the four arms of the Wheatstone bridge are made of electrolytic nickel.

The wires which compose the two opposite arms of the bridge are shielded from the air current, and the two remaining wires are exposed to it. The instrument is thus automatically compensated for changes in the air temperature, so that it is not necessary for the temperature of the wires to be more than 100° F. above that of the air current to be measured. A constant potential difference is maintained across the bridge, and the air velocity is measured in terms of the out-of-balance current of the galvanometer.

The instrument is calibrated against known velocities in a wind tunnel.

Steam Meters.

The Kent-Hodgson Steam Flow Meter.

This meter (fig. 17) consists of an orifice O placed in the steam main, two cooling chambers CU and CD (whose function it is to condense any steam that passes into them and to serve as reservoirs of water), and pressure pipes PU and PD which transmit the difference of pressure between the upstream and downstream sides of the orifice, due to the steam flow, to the two sides of the differential pressure diaphragms D and to the inside of the pressure diaphragms P.

Any increase of the flow causes an increase of the difference of pressure between the upstream and downstream sides of the orifice, and consequently causes the diaphragms D to contract and

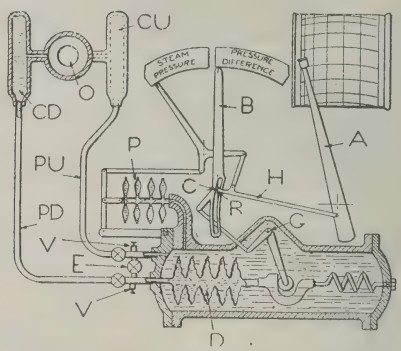


FIG. 17.

the pointer B to move about its pivot C (the motion of the diaphragms D is transmitted to the outside of the diaphragm case through a stuffing box G); while any increase in the pressure of the steam causes the diaphragms P to expand and the arm H to descend.

Air vents V and an equalising valve E are fitted.

The product of the angular movement of the arm B (which is proportional to the differential pressure $(p_1 - p_2)$ and the travel of the roller R in its slot (which is proportional to the absolute pressure of the steam p_1) is transmitted to the pen arm A, which is thus given a motion proportional to $(p_1 - p_2)\, p_1$.

Since the rate of discharge of steam through an orifice is represented approximately by the equation,

$$Q = K\sqrt{(p_1 - p_2)p_1}$$

where,

p_1 = absolute pressure of the steam on the upstream side of the orifice; p_2 = absolute pressure of the steam on the downstream side of the orifice; Q = lbs. of steam per hour; K is a constant depending upon the size and proportions of the orifice used.

The diagram can be divided to show the rate of steam flow at any instant, automatically corrected for variations in the pressure. (The corrections for variations in the superheat or wetness of the steam* are made by means of tabular numbers.)

The meters can be arranged to record by counter also if desired.

With any given orifice plate the meter is accurate down to one-fifth of the maximum flow for which the plate is designed to deal. If it is desired to measure over a greater range of flow than this, a new orifice plate must be inserted. Each meter is designed to work in connection with any orifice plate. Each orifice plate is calibrated with actual fluid before leaving the works, and the accuracy obtained in practice with this meter is within plus or minus 1 per cent.

With slight modifications, the meters can be used for the measurement of compressed air.

The Hodgson Rotary Steam Flow Meter.

This meter, which is being manufactured by *George Kent, Ltd.*, of London and Luton as shown in section by fig. 18. A portion of the steam passing through the instrument is diverted by means of an orifice plate through nozzles A, by means of which it is caused to impinge on a turbine. The speed of this turbine is kept low by means of a damping fan C, which rotates in water, resulting in a minimum of wear on the turbine bearings. The retarding torque at any given speed due to this fan is made so great that any variation in the corresponding torque due to the turbine, caused by variations in the density of the steam in which it rotates, can be neglected. The turbine rotates at normal flows with a speed proportional to $Q\sqrt{W}$, where Q is the weight of steam passing per second, and W is the density of the steam in pounds per cubic foot. From this it will be seen that a variation of 2 per cent. in the density of the steam will involve a correction of only 1 per cent. in the flow reading. Besides reducing wear on the pivots, the use of the damping fan increases the range of the meter, since it increases the relative velocity between the impinging steam and the turbine. The counter E is mounted in a compartment below the main, a magnetic drive D from the turbine being employed so that leakage of steam or water into this compartment is impossible. Radiating fins assist in keeping the counter cool. The maximum flow which the meter will measure can be altered by changing the orifice plate B; and by fitting a blank plate in place of this orifice plate, and putting the meter in a by-pass in a large main into which a suitable orifice plate is inserted directly, the largest mains can be metered at a small cost.

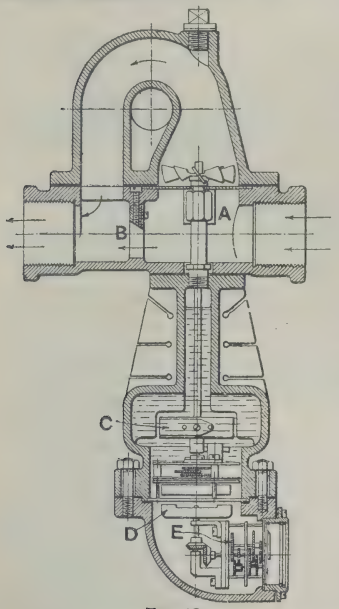


FIG. 18.

The ratio of the revolutions per minute to the rate of flow is constant.

The meters as supplied are accurate down to $\frac{1}{10}$ of the maximum flow, and will stand temporary overloads of 100 per cent. without damage. The meters are also suitable for measuring compressed air.

* About 1 per cent. for every 2 lbs. change in the pressure, or 20° F. change in the superheat, or 2 per cent change in the wetness.

The Kent-Hodgson Spring-Controlled Flexible Diaphragm Instruments.

The diaphragms in these instruments being required to be sensitive to small differences of pressure, have, with the springs which control them, to be protected from any damage due to overload, or to the application of any considerable unbalanced static pressure, to which they may by mischance be subjected; this, in the Hodgson instrument, is effected by means of a cradle on both sides of the diaphragm. Further, in order that the accuracy of the instrument shall be maintained in use, the diaphragm is reinforced with flexible material which will not stretch radially or circumferentially. The cradle is in the form of a portion of a cone, on which a suitably shaped diaphragm can roll and unroll without stretching. From fig. 19 it will be seen that whatever position along its axis the stiff central plate BC may take up, the length AB is always equal to the length AB_1.

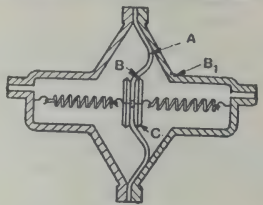


FIG. 19.

In order to obtain a good zero, the controlling springs are under stress when there is no load on the instrument, so that there is no backlash to be taken up when the diaphragm begins to move.

The motion of the centre of the diaphragm is transmitted to the indicating pointer or the recording pen by suitable means (not shown in the figure).

The instruments as constructed show a lag between the up and down readings of only about 1/500th part of the maximum applied pressure. The diaphragm can be removed and replaced without altering the adjustment of the instrument.

The arrangement is more especially suitable for use in connection with the measurement of flow through orifices and Venturi tubes, where small pressure differences at high static pressures have to be measured.

The instruments as supplied are fitted with an 18-in. rate of flow scale, and can be arranged to record by diagram or by diagram and counter.

Coal Meters.

These meters are designed to measure either (*a*) *the total quantity* of coal going to any particular furnace, or (*b*) *the rate* at which the coal is fed. The Avery meter (type *a*) actually *weighs* the coal supplied. The Lea and Kent meters (type *b*) *measure the volume*, i.e. not the weight, of the coal supplied.

The 'Avery' Automatic Coal Weigher.

The principle of the 'Avery' Automatic Coal Weigher (fig. 20) is that of an equal-armed beam A with the weights contained in the weight box B suspended at one end and the weigh hopper C at the other.

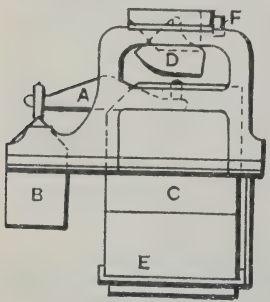


FIG. 20.

A quantity of coal, determined by the weights in the weight box B, is allowed to enter the weigh hopper C through the feed shoot controlled by the valve D.

When the correct amount of coal is obtained, the supply from the feed shoot is, through a combination of levers, automatically cut off by the valve D.

Immediately after the supply is cut off, the door E of the weigh hopper C automatically opens and the load is discharged.

When the load is clear of the weigh hopper door E it closes again and automatically opens the feed valve D, and the cycle of operations is repeated.

A mechanical counter F keeps an exact record of the number of times the machine operates.

Coal or other granular material which is carried by belt conveyors can be weighed to an accuracy of *plus* or *minus* 1 per cent. by supporting a short length of the conveyor on a weigh-bridge and introducing a device which automatically integrates the product of the speed of the conveyor, and the *net* reading of the weigh-bridge.

In the 'Avery' Automatic Conveyor Weigher, the pull on the steelyard arm is balanced by a cylindrical float immersed in mercury, the movement of the arm being thus made proportional to the weight on the weigh-bridge. Integration is effected by means of a feeler and free-wheel device, the steelyard arm being clamped while the feeler is in contact with it.

Lea Coal Meter—Chain Grate Type.

The object of this meter is to measure the amount of coal consumed by a chain grate mechanical stoker.

The coal, as it passes from a hopper underneath a movable fire door, assumes the form of a continuous stream of rectangular cross-section. The volume of coal, therefore, having passed during any given time is the product of the width of the chain grate, the height of coal as it passes under the fire door, and the linear distance travelled by the grate.

Of these three factors two are variable: firstly, the depth of coal passing over the grate (due to the variation in lift of the fire door); secondly, the speed of the grate (or the distance travelled by the grate over a given time). The meter is so constructed as to take both these variables into account in the following manner:—

A specially cut toothed drum, having at one end teeth cut round the whole of its periphery, and a uniformly decreasing number of teeth cut along the whole of its length, is driven by a small driving chain from the grate itself; thus the speed of the drum is proportional to that of the grate, and accordingly proportional to the linear distance travelled by the grate over any given time.

Meshing with this toothed drum is a sliding pinion, which is directly connected to the fire door by means of a steel cable; its position on the toothed drum, and consequently the number of teeth with which it engages, is, therefore, dependent on the amount to which the fire door is raised or lowered. Thus, the rotation of this pinion is dependent on both the grate speed and the fire door lift, and, consequently, this rotation is in a direct proportion to the volume of coal having passed.

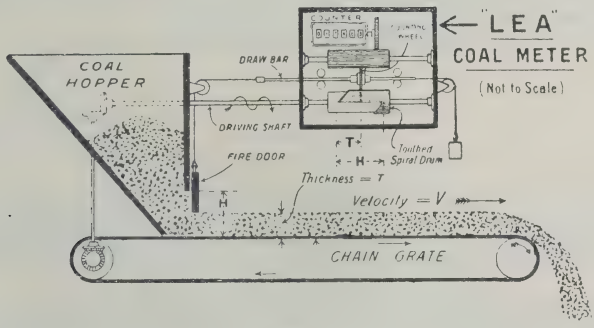


Fig. 21.

The principle of operation of the Lea Coal Meter is as follows:

If W = width of grate in ft.; T = thickness of fire in ft.; V = velocity if grate in ft. per hour; then, cubic ft. per hour = W × T × V = cross-sectional area × V.

The rotation of the pinion is transmitted directly to a row of seven counting dials, the movement of which gives a visible indication of the volume of coal having been fed on to the grate.

It should be pointed out that the meter cannot weigh anything directly; it is solely a measurer of volume, but it eventually determines the nett weight of coal consumed. Lea Recorder Co., Ltd., guarantees the accuracy of its Coal Meters to be within 2·5 per cent. by volume.

It is quite a simple matter to arrive at the actual weight of coal consumed, by means of tests carried out to determine the average weight of a cubic foot of the particular fuel being used. No variation is caused by the size of the particles of coal, as it is a fact that equal volumes of dry graded coal weigh the same, irrespective of the size of the particles, or whether they are in the form of nuts, peas, or slack.

These Coal Meters can be fitted with patent 'Rate of Flow' Indicator, which, when worked in conjunction with the various speeds of the Stoker drive, gives, on specially graduated scales, the actual amount of coal being consumed at any moment in lbs. per hour.

Lea Coal Meter—Lancashire Type.

In this type of meter the volumetric principle of measurement is adhered to, and the meter is operated directly by the rams, or pusher plates, of stokers of the Proctor, Bennis, and other similar types. The meter is so arranged as to take into account variations of both the speed at which the rams are operated and the differences of travel of the rams, so that a true indication of the amount of fuel fed into the furnaces is given.

The Lea Recorder Company guarantee the meter to be correct within 2·5 per cent. by volume.

Lea Cubi-Meter.

The object of this machine, which is a development of the Lea Coal Meter (chain grate type), is to measure bulk supplies of coal, ores, grain, or any other similar granular material, in a continuous manner, in conjunction with elevators or conveyors, on which the material is being handled.

The principle of measurement is volumetric, and is as follows :—

The material to be measured is passed on to an endless belt, in a similar manner to the coal passing on to the grate of a chain grate stoker. The meter itself is similarly driven by a small chain from this belt.

As the material travels over the belt it passes underneath a swinging door, the lift of which varies with the thickness passing under it. Directly connected to this swinging door is a link motion which operates the sliding pinion in the meter. Thus the rotation of the meter dials is proportional to the speed of the belt, and also to the height of the material passing along it, the two factors being combined in the meter giving a true indication of the volume of material having passed through the machine.

These machines are guaranteed to be accurate within 2 per cent. by volume.

Revolution Indicators.

The Jaquet Indicator.

(*John Dickinson, 76 Gray's Inn Road, London, W.C.* 1.)

This indicator, fig. 22, is an accurate standard instrument which in the shortest time gives positive readings of revolution rate per minute or, using the disc wheel fitted in the case, of linear speed per minute. The standard time unit adopted is 6 seconds, and this unit is controlled by the precision watch movement fitted in the indicator. 1½ seconds are allowed for starting and stopping the watch movement, which also automatically switches the counting mechanism in and out at the start and finish of the 6 second reading unit.

The lever with button which projects from the case winds the watch mechanism ; for each reading one pressure on the button suffices, and the lever is designed not only to wind the watch but also to zero the counting mechanism from the previous reading. This does away with any necessity for resetting the counter after taking a test ; the last reading remains on the dial *until the indicator has been placed in position on the machine* under test and a new reading is required ; then the finger is pressed on the knob and allowed to slide over it so that it can fly back to its full extent, thus resetting the counter and winding the watch.

Either a black or red disc is shown on the face of the counter dial, the colour altering with the direction of rotation and corresponding to that of the set of graduations to be used, black for right-hand, red for left-hand rotation.

One revolution of the large pointer = 1,000 revs. (or 100 m. if the measuring disc is attached) of the small pointer = 10,000 revs. (1,000 m.).

The indicator is designed to stand up to 30,000 revs. per min., and all pivots and bearings are suitable for such high-speed rates.

The indicator is claimed to combine the accuracy of a summating counter with the rapid reading rate usually associated only with tachometers of the centrifugal or magnetic types.

Three fittings are supplied with the instrument: B = rubber point, C = hollow or cup centre, G = measuring disc. The triangular steel point of the spindle can also be used.

The measuring disc G is used for *periphery or rate of cut tests* ; this is fitted with a rubber rim 10 cm. circumference, or if readings are required in feet, the diameter is altered to suit. In cases where shaft centres are inaccessible the revolution rate of same can be determined by using

the measuring disc G and converting the peripheral speed reading to the corresponding revolution rate.

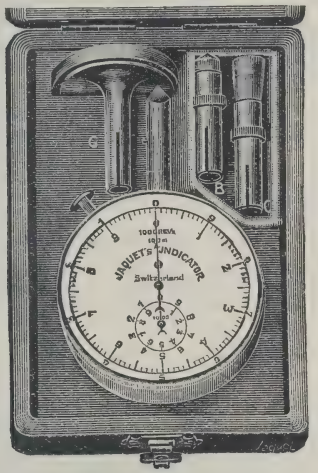


FIG. 22.

An additional model of the foregoing is now made in which 1 rev. of the long pointer = 100 r.p.m.

THE JAQUET HAND TACHOGRAPH.

(*John Dickinson, 76, Gray's Inn Road, W.C.* 1.)

The Jaquet hand tachograph, fig. 23, is used for recording revolution rate, drawing the curve in rectangular co-ordinates on a moving paper strip, marking the time units horizontally and the revolution rate vertically.

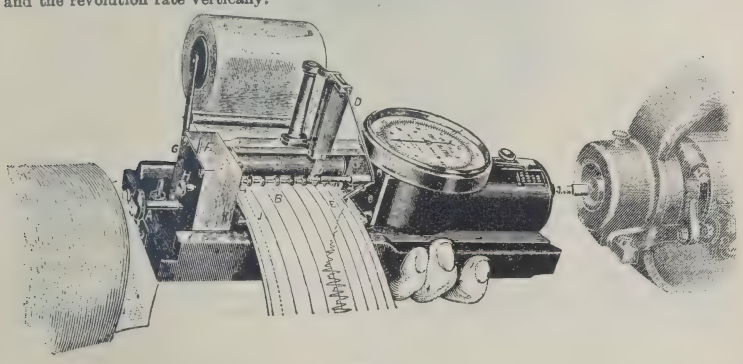


FIG. 23.

It can be used for any desired revolution rate per minute between 30 and 4,000, and allows records of wide speed fluctuations, such as the running-down rate of rotating bodies, and also of such speed rate changes of only 1 to 2 per cent. as occur under governor control.

The mass effect of the recording system is reduced to a minimum, so that records of cycle speed fluctuations are possible.

The record strip is fed forward between two clockwork-driven feed rolls, the clockwork is wound up by the thumb nut A, whilst a trigger at the back (not visible in the fig.) and triggers H and F act respectively to stop or vary the strip feed rate.

Narrow discs are mounted on the top feed roll at positions corresponding to the calibrated revolution rates indicated on the tachometer dial; these discs are in contact with an inking roll C and mark on the record strip parallel lines spaced to agree with the calibration of the instrument.

After a test the r.p.m. value of the inked-in parallel lines can be written in on the record.

The revolution rate during test is taken from the tachometer dial. The thumb nut H winds the watchwork actuating the time recorder J, which marks the record at definite intervals of 1 or $\frac{1}{5}$ second as selected. Usually the apparatus are provided with 1 second registering time markers.

Record of the revolution rate is effected by the pen E, connected through a system of levers D to a spindle projecting from the tachometer and working at right angles to the line of strip advance. Suitable recording ink, which is supplied with the instrument, is put into the pen by a small brush.

An additional fitting may be had, by the use of which the chart spacing may be increased or multiplied, at a small extra charge.

Optical accessories are also available in the way of a flexible drive attachment, whereby readings may be taken from inconvenient positions to allow for easy reading; a neat reducing and multiplying gear-box is also available, at a nominal cost.

The Veeder Magnetic Counters.

(*F. E. Dickinson, St. Andrew's House, Holborn Circus, London, E.C.* 1.)

The 'Veeder Magnetic Counters' differ from the general run of Veeder counting mechanisms in that they operate by the making and breaking of an electric circuit. Their uses are many and varied. They can be placed at a distance from the machine or apparatus the operations of which it is desired to keep account, and they are useful where the articles to be counted are small, or thin, or light in weight, making it difficult to count by mechanical means. No extra contact device need be provided where the counter is to be used in conjunction with other devices already

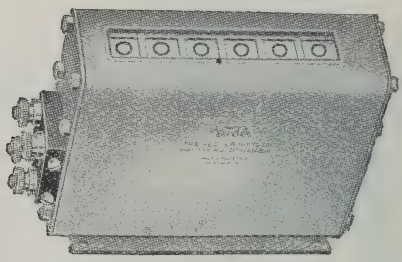


Fig. 24.

electrically operated, and where it is desired to know the number of times these devices are started or stopped. In other cases it is necessary to provide some mechanical device which will close and break the electrical circuit which operates the counter. The counters are made for either direct or alternating current.

Mechanically operated Veeder Counters are used for counting, measuring, and particularly in recording output of many varied products often at different stages of manufacture.

The counter shown by fig. 24 is the larger of the two magnetic devices and is furnished with or without a reset attachment (for the purpose of resetting the counter to zero at any particular point).

SECTION XXVIII

HARBOURS AND BREAKWATERS — DOCKS — QUAY-SIDE APPLIANCES—GRAVING AND FLOATING DOCKS — DREDGING—SUBAQUEOUS ROCK REMOVAL—CHANNEL DEMARCATION — BUOYS — LIGHTSHIPS — LIGHTHOUSES — SUBMARINE SIGNALS—DIVING—FLOATING LANDING STAGES.

(Pp. 1455–1502.)

(Contributed and revised by Brysson Cunningham, D.Sc., B.E., F.R.S.E., M.Inst.C.E.)

SECTION XXVIII

HARBOURS AND BREAKWATERS — DOCKS— QUAY-SIDE APPLIANCES—GRAVING AND FLOATING DOCKS—DREDGING — SUBAQUEOUS ROCK REMOVAL — CHANNEL DEMARCATION — BUOYS — LIGHTSHIPS — LIGHTHOUSES — SUBMARINE SIGNALS—DIVING—FLOATING LANDING STAGES.

Contributed by Brysson Cunningham, D.Sc., B.E., F.R.S.E., M.Inst.C.E.

(Author of 'Dock Engineering,' 'Harbour Engineering,' etc.)

HARBOURS AND BREAKWATERS

Harbours are sheltered areas of water affording safe anchorage for ships generally also provided with quays and facilities for landing passengers and cargo. There are many harbours where the requisite shelter is forthcoming naturally from the physical configuration of the coast line (Queenstown, Rio de Janeiro, etc.). Where protection cannot be obtained in this way it has to be provided by means of breakwaters, as at Portland, Dover, and elsewhere.

Breakwaters are of two types : (*a*) the mound of rubble stone, and (*b*) the upright wall of masonry or concrete. Frequently the types are combined. Sometimes the mound predominates

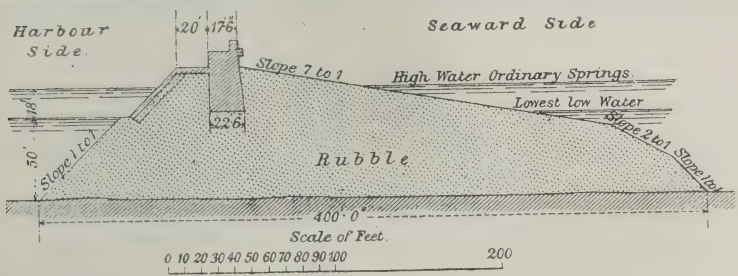


FIG. 1.—Holyhead Breakwater.

and is simply capped by a slight superstructure of regular masonry, as at Algiers and Oran ; in other cases it is reduced to a minimum, becoming a mere foundation layer for a wall of massive proportions, as at Ymuiden and Zeebrugge.

The relative advantages of the mound and the wall are as follows :—

1. As regards *efficiency*, the mound is not so serviceable as the wall. The undulations of the sea are transmitted through the interstices of a mass of rubble, and the harbourage area is kept more or less in a state of agitation.

2. As regards *initial cost*, the mound can be formed by unskilled labour, and where that is plentiful and stone abundant, this type of breakwater will commend itself on account of the facility with which it can be constructed. In the absence of these essential conditions, and provided the depth of water be moderate and the foundation sufficiently firm, the wall, involving a much less quantity of material, will be found preferable. Especially will this be the case where skilled labour is readily procurable at reasonable rates. Even the difficulty of a defective foundation may be overcome by one or other of several expedients without perceptibly altering the relative positions. But where the sea bottom lies at a considerable depth (40 feet or more) the superior economy of the pure wall cannot be maintained.

CONSTRUCTIONAL COST OF BREAKWATERS.*

Harbour.	Type.	Mean Depth of Water.	Cost per lin. ft.	Remarks.
		Ft.	£	
Alderney .	Composite	90	259	Mound terminates 12 ft. to 15 ft. below L.W.
Algiers. .	Mound	50	122	Large random concrete blocks surrounding rubble.
Dover . .	Wall	40	415	Concrete blocks weighing 26 to 40 tons.
Holyhead .	Composite	54	162	Mound very predominant.
Marseilles .	Mound	35	87	
Bizerta .	Composite	58	60	Wall built on caisson system.
Portland .	Mound	44	127	
Naples . .	Composite	93	186	
Valparaiso .	Composite	60 to 150	560	Concrete monoliths on rubble mound.

An interesting comparison of the estimated (pre-war) cost of four different types of breakwater, based on the conditions at Zeebrugge, Belgium, is made by MM. Bonnet and Braeckman in a paper to the Fourteenth International Congress of Navigation. The depth of the sea bed at Zeebrugge is 26 ft. 3 ins. below zero datum, and the tide rises 15 ft. above zero, giving a total depth of 41 ft. 3 ins. of water. The unit costs per lineal metre (3·28 ft.) are given as follows :—

	Francs.
(*a*) Solid wall of the Zeebrugge type. Total height 66 ft., mean thickness 23 ft.	6,760
(*b*) Composite breakwater with mound of unassorted rubble, the slopes being lightly covered by large natural blocks of 10 cwts. to 6 tons ; wall superstructure 33 ft. high	7,600
(*c*) Composite breakwater, mound having core of unassorted rubble, larger material on slopes protected on outer side by artificial blocks of 33 tons deposited haphazard	8,700
(*d*) Composite breakwater with base as before, but with slopes protected by artificial blocks of 31 tons laid symmetrically to inclination	9,000

3. As regards the cost of *maintenance*, this is generally greater in the case of mounds than in the case of walls. A wall, provided it be properly constructed in the first instance, calls for no further attention, save for such rare and occasional damage as results from some storm of

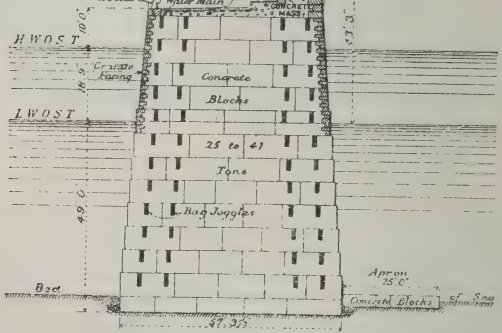


FIG. 2.—Admiralty Pier Extension, Dover.

exceptional severity. The rubble mound, on the other hand, is peculiarly susceptible to wave action, and needs constant replenishment. The cost of this varies very considerably according to situation and exposure. At Holyhead it was for many years as low as 1*s.* 3*d.* per lin. ft. per annum,

* The impossibility of accurately adjusting costs and prices, where they are quoted, to the altered and variable conditions arising out of the war has rendered it advisable to retain them on their old pre-war basis, leaving the necessary adjustments to be made as occasion may require.

but latterly there has been some extensive replenishment at considerable outlay ; at Genoa, the cost is stated to be 7*s.* ; at Naples, 13*s.* ; while at Alderney, during a certain period, it rose as high as 45*s.* This last, however, is quite an abnormal figure, and under ordinary conditions the pre-war annual expenditure on mounds of suitable design hardly exceeded a pound a foot.

Wave Action.

Wave Action.— Wave action on breakwaters is most keenly marked between the levels of high and low water, but it extends to a depth of 30 ft., and there are instances on record of serious disturbances at even greater depths. Thus, at Peterhead, blocks weighing upwards of 41 tons have been displaced at 36½ ft. below L.W.O.S.T. But cases of this kind are exceptional. Generally speaking, at a depth of 5 fathoms the sea is fairly quiescent and a rubble mound may be counted upon to stand therein with side slopes of from 45° to 50°.

Above this level much flatter slopes become necessary : 2, 3, 4, 5, and even 7 to 1. Unfortunately, while the equilibrium of the mound demands such gradual inclinations, they serve also to foster the disruptive action of waves, which, passing up a long seaward incline, have their oscillatory character changed into one of translation. At the summit the impact delivered is so powerful as to produce a pressure which Stevenson records as being six times as great as that against a steep wall. This inimical effect can only be counteracted by the use of huge blocks and monoliths, covering the exposed surface of the mound. The blocks, which in practice rarely weigh less than 25 or 30 tons apiece, and often considerably more, may be deposited either in courses or at random. In the former case they are generally stepped so as to conform to the general inclination. In either arrangement they should be laid as headers, pointing seaward, so as to expose the least extent of surface to wave-action and at the same time gain the utmost

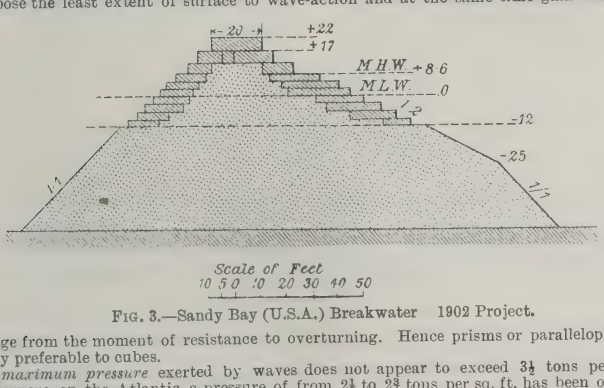


FIG. 3.—Sandy Bay (U.S.A.) Breakwater 1902 Project.

advantage from the moment of resistance to overturning. Hence prisms or parallelopipeds are decidedly preferable to cubes.

The *maximum pressure* exerted by waves does not appear to exceed 3½ tons per sq. ft. At Skerryvore, on the Atlantic, a pressure of from 2½ to 2¾ tons per sq. ft. has been observed ; at Bell Rock, in the North Sea, 1½ ton ; at Dunbar (East Lothian) 3½ tons, and at Buckie (Banffshire) 3 tons.

Composite breakwaters, in which the wall plays any important part at all, should be so designed that the *back draught* or *after-tow* of the waves cannot undermine the superstructure. To be really effective, this means that the wall should be carried down to a depth of at least 30 ft. below low water, and though in the past many such walls have been founded at low water level, or very slightly below it, yet the experience gained has proved that this depth is quite inadequate (the low water zone is particularly dangerous), so that of late years there has been a continual tendency towards increasing the depth of the foundation level. At Bilbao the wall commences at 16½ ft. below low water ; at Madras 27½ ft. ; at Naples 33 ft. ; and at Peterhead 43 ft.

Methods of Construction.

Methods of Construction.—Breakwaters are constructed in several ways, according to situation and type.

For Mound Breakwaters, rubble may be deposited—(*a*) from barges or scows; (*b*) by means of cranes running on temporary staging; and (*c*) from wagons passing over roads laid at or about the level of the top of the mound and extended as the work proceeds.

The first method is best adapted to sheltered situations, and affords an opportunity of depositing rubble uniformly and simultaneously over the whole site, but it is not practicable to carry on operations in this way at or about surface water level nor above it. Moreover, the progress of the work is very dependent on weather conditions, and there may be long periods of enforced inaction. It is to be noted in passing that irregular deposits of rubble lead to current action and scour. The second method (staging) admits of operations being carried on uniformly over the site, though not with quite the same degree of freedom as in the case of barges. But the advantages of a stable working base with infrequent interruptions and higher degree of safety and

protection for workers, are very pronounced. The damage sustained by timber staging in storms and from the attacks of marine borers are drawbacks which detract from the merits of the system, though, generally speaking, the former is proportionately not a serious matter, and any

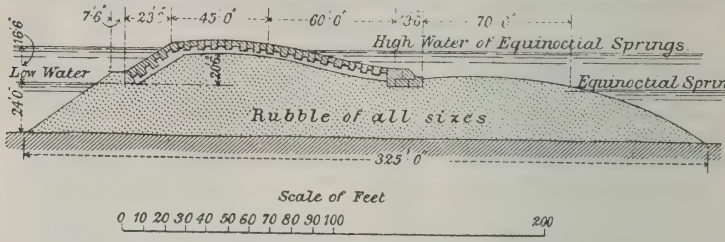


FIG. 4.—Plymouth Breakwater.

evil results of the latter can be to some extent guarded against by careful inspection. The third, or low-level system of construction, by means of tipping from wagons running on tracks laid on the breakwater itself, assists in the consolidation of the work as it proceeds, and saves the cost

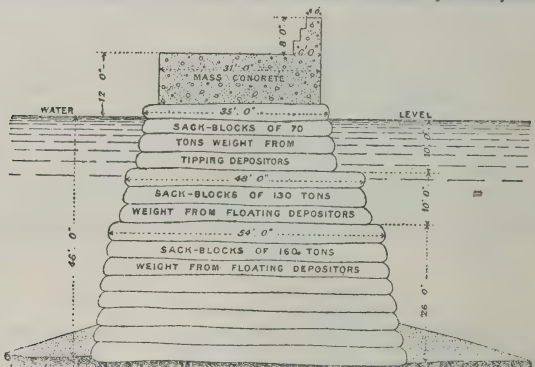


FIG. 5.—La Guayra Breakwater.

of staging, but the area of operation is limited and there is liability to interruption during the prevalence of heavy seas. It is a method more adapted to small embankments, where time also is perhaps not a matter of prime importance.

Wall-Breakwaters.

Walls are constructed either by means of staging, or on the low-level, end-on system: the former being generally more suitable when concrete is deposited in mass, and the latter for blockwork. The blocks, or monoliths, are deposited in place by huge revolving cranes, known as Titans, and overhead travellers known as Goliaths (p. 807). A modern Titan is a powerful cantilever crane capable of dealing with blocks of from 30 to 60 tons weight, at a working radius of as much as 100 ft. A 41-ton steam Titan used on the breakwater at South Shields had a radius of 75 ft., and was carried on 16 central-flanged wheels running on rails set to a 22 ft. gauge. The speeds of the various motions were as follows: Lifting full load, 6 ft. per min.; lifting loads up to 10 tons, 25 ft. per min.; racking full load, 20 ft. per min.; racking loads up to 10 tons, 50 ft. per min.; slewing, one revolution in 4 min. Goliaths are constructed to the same lifting capacity as Titans, and some command a lateral range of travel of 100 ft. or more, but the span is generally limited to about 50 or 60 ft. At Dover, a 42-ton steam Goliath had a range of 130 ft., a clear headway of 25 ft., and a total lift of 120 ft. The speeds of movements were: Lifting, 10 ft. per min.: crab travel, 50 ft. per min.; and main travel, 60 ft. per min.

FIG. 6.—Sloping Bond.

Special Methods of Construction.

Special methods of breakwater construction include the use of bags, or sacks, of concrete, weighing from 50 to 200 tons, for deposition under water by divers, as at La Guayra; large caissons, or tanks, of reinforced concrete, with volumes of 60,000 to 70,000 cu. ft., floated out into position, sunk and filled in solid, as at Bizerta; and blocks set on end, slightly tilted (15° to 20°) out of the vertical, so as to form what is called *sloping bond*, the object being to allow for unequal settlement in the foundation. Small breakwaters in shallow waters are sometimes constructed of cribwork—box-shaped frames of timber in openwork with cross partitioning and ties, weighted when in position with rubble—but the system is not very durable and only suitable for comparatively sheltered situations.

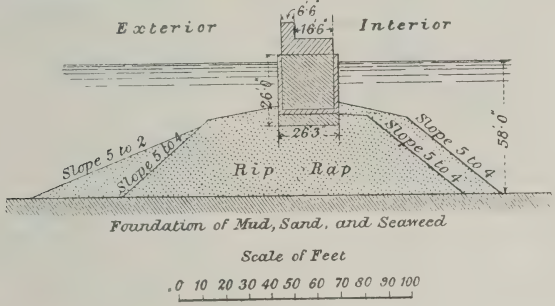


Fig. 7.—Bizerta Breakwater.

DOCKS.

Docks are enclosed repositories for shipping. Where the enclosure is not complete and there is free communication with the sea, the term *Basin*, though not universally employed, is more appropriate. Docks proper are fitted with gates or caissons, either in order to impound the water at a fairly constant level, in which case they are distinguished as *wet docks*, or to exclude the water, in which case they are known as *dry* or *graving docks*. *Floating Docks* are movable structures capable of lifting vessels out of the water and supporting them by their own displacement.

Constructional Cost of Wet Docks.*

(Note: The variety and range of design and equipment render any very effective comparison out of the question.)

	Per acre of Water Surface.		Per acre of Water Surface.
Alexandra Dock, Hull . .	£28,900	Barry Docks	£12,950
Alexandra Dock, Liverpool .	23,300	Prince of Wales Dock, Swansea	18,600
Avonmouth Dock, Bristol .	22,600	Queen's Dock, Glasgow . .	24,450

King George V. Dock, London.

One of the most notable docks constructed in recent years is the King George V. Dock at London, which adds 64 acres to the enclosed water area of the port. The dock has a depth of 38 ft., and the quayage aggregates approximately 10,000 ft. A section of the north, west, and east quay walls is given in fig. 11. An unusual feature on the south side of the dock, shown in fig. 12, is the provision of seven reinforced concrete jetties, each 520 ft. long and 22 ft. wide, which have been constructed at a distance of 32 ft. from the face of the main quay wall, the only shore connection being a timber foot-bridge. The arrangement is due to the fact that a preponderating portion of the goods arriving at the port is transferred to barges for conveyance to their ultimate destination up river. Ships are berthed on the outside of the jetties, and the jetty cranes can discharge goods therefrom either on to the quay for delivery to carts or railway trucks, or for sorting or temporary storage at the sheds behind, or, alternatively, into barges lying in the space between the jetties and the wall.

The entrance lock from the river Thames is 800 ft. long by 100 ft. wide, with a depth of water on the sill of 45 ft. below Trinity high water. The chamber is divided into two compartments, of 550 ft. and 250 ft. respectively, by three pairs of steel lock gates, each leaf of which weighs 309 tons. By the use of a floating caisson, for which provision has been made at the inner end of the lock, the available length can be increased to 910 ft. The cost of the dock with its equipment was about 4,130,000*l.*

* See footnote, page 1456.

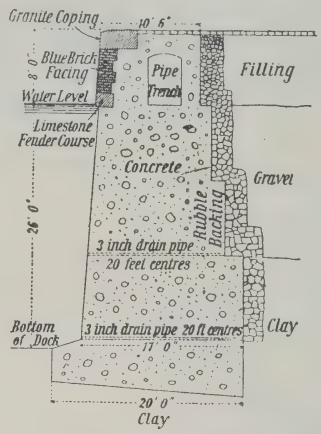


FIG. 8.—Manchester Dock Wall.

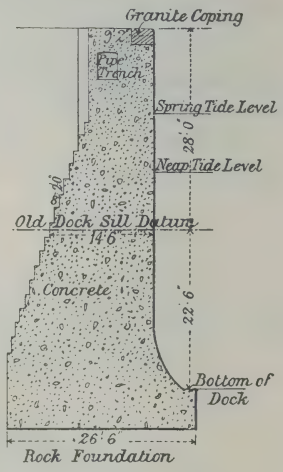


FIG. 10.—Liverpool Dock Wall.

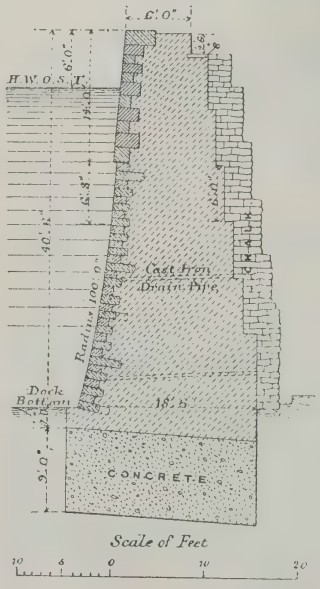


FIG. 9.—Hull Dock Wall.

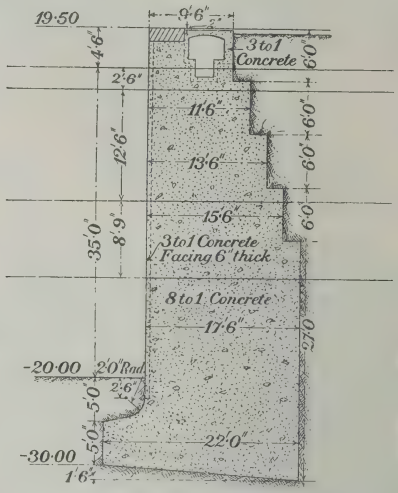


FIG. 11.—King George V. Dock, London: Section of N., E., and W. Quay Walls.

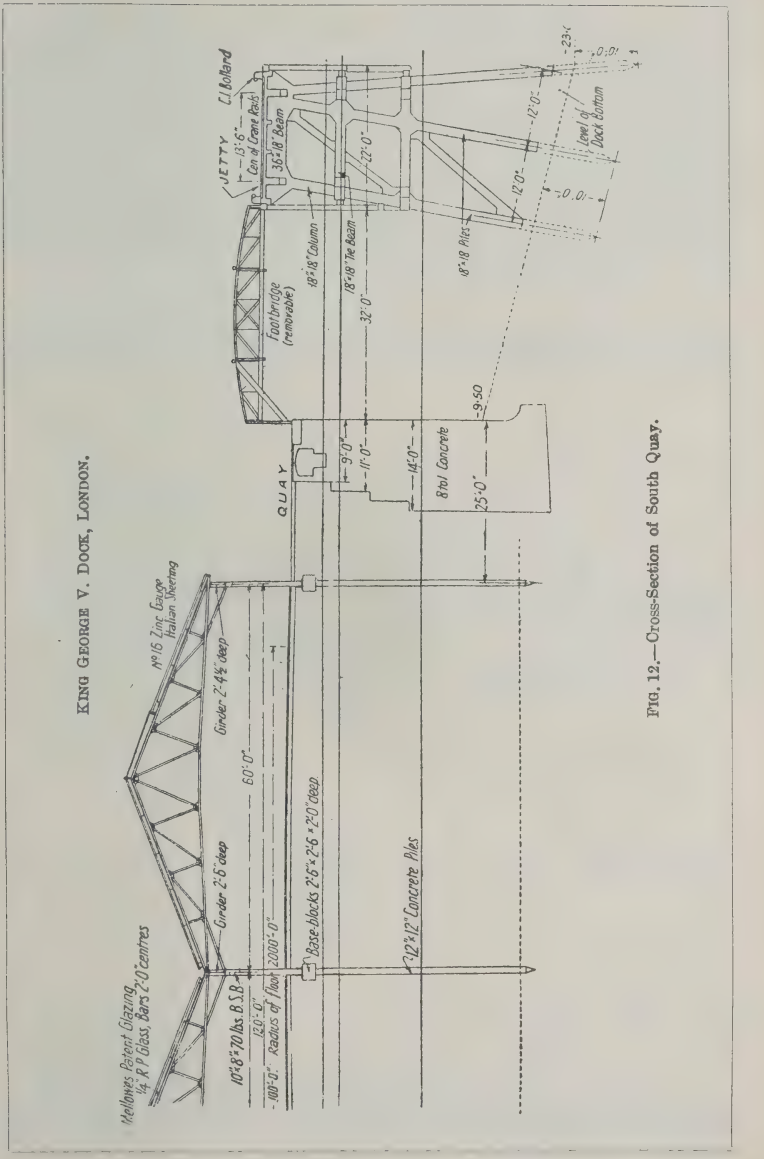


FIG. 12.—Cross-Section of South Quay.

The Gladstone Docks, Liverpool.

Another notable example of recent dock construction is the Gladstone Dock System of the Mersey Docks and Harbour Board, opened to traffic in 1927. It consists of a vestibule, or turning, basin with two branch docks, the whole comprising a water area of 58¼ acres. The vestibule dock is entered from the River Mersey by a lock of outstanding dimensions. It is 1,070 ft. in length and 130 ft. in width, divided into two compartments, with sills 20 ft. below Bay Datum, giving 48 ft. 4 ins. depth of water at H.W.O.S.T., 35 ft. at half tide level, and 22 ft. at L.W.O.S.T. Both the branch docks are 400 ft. wide; one is 1,420 ft., and the other 1,285 ft. long. They are equipped with treble storey sheds of reinforced concrete construction, providing floor areas (including the flat roofs which are designed for light loading) of 32 acres and 23 acres respectively. The system includes a graving dock which is the largest in Europe. The length of the dock from the head to the inner caisson sill is 1,050 ft. 4 ins. The width of entrance is 120 ft. and there is a depth of water over sill of 43 ft. 6 ins. at H.W.O.S.T. This dock has the peculiarity that it can be used as a wet dock, and for that purpose it is equipped with a single storey shed on its north side.

Eastern Arm Extension of Royal Edward Dock, Avonmouth.

On May 23, 1928, H.R.H. the Prince of Wales opened a new arm or extension, 1,750 ft. long and 400 ft. wide, of the Royal Edward Dock at the port of Bristol which, with a water area of about 17 acres, affords on each of its two main frontages three deep-water berths for large vessels,

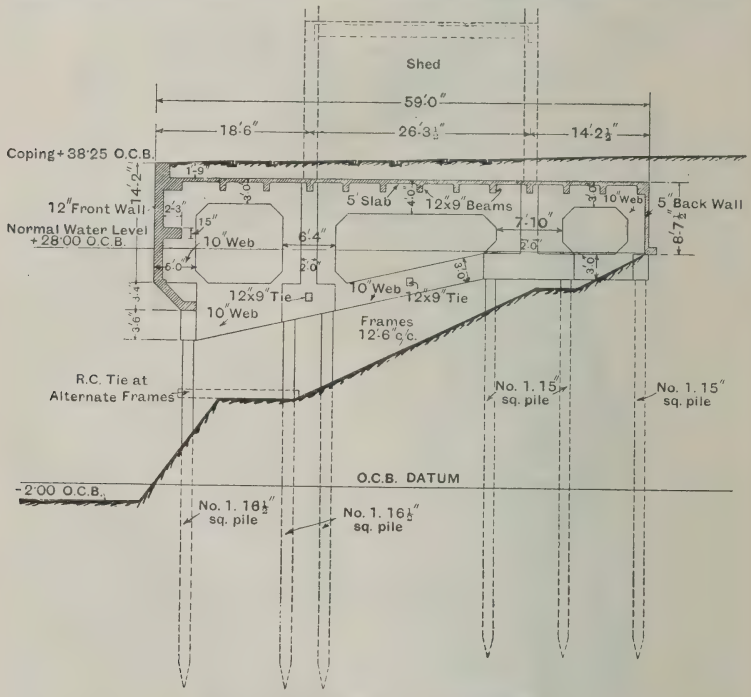


Fig. 13.—Royal Edward Dock, Avonmouth.

with an aggregate quayage length of 3,600 ft. The extension has been carried out on lines which, while generally applicable to wharfing, are of some novelty in dock quay construction. In place of a massive wall of the gravity type, there is a system of openwork framing and decking in reinforced concrete, supported on piles of the same material, as shown on the typical section in fig. 13.

The number of piles driven amounts to over 4,500, and they are of lengths ranging from 55 to 60 ft. The main frames are set at 12 ft. 6 ins. centres, and the front spacing is made up of concrete slab and beam construction. The cost of the quay was between 70*l.* and 75*l.* per ft. run, covering the reinforced concrete work and the excavation down to the 2½ to 1 slope, within the limits of the front and back wall.

Dock Walls.

Dock Walls.—Under present-day conditions it is necessary for quay walls to possess a perpendicular, or almost perpendicular, face. Ships are now built to a square, box-like section amidships, with vertical sides inclining inwards towards the top. Unless, therefore, there is an absence of batter in the walls, vessels will be unable to lie as closely in contact with the quay coping as is desirable, and quay cranes for loading and discharging them will have to be provided with a greater outreach than is absolutely necessary, at an enhanced cost.

Dock walls are constructed of masonry, brickwork and concrete—nowadays generally the latter, deposited in mass with sometimes a brick or stone facing. There is nearly always a coping course of granite. A toe in front of the wall adds to its stability and prevents any undermining of the foundation. The material forming the backing should be carefully selected so as to admit of proper drainage, and weep or vent holes should be provided in the wall at intervals of 50 feet or so, to admit of water escaping through to the front when the level of the dock is lowered in any way.

The composition of *Concrete* for the hearting of dock walls may safely be put at 8 parts of aggregate by volume to 1 part of sound Portland cement. Of the aggregate the sand while not exceeding 3 times the cement will at least suffice to fill the interstices in the gravel. Stone rubble plums can be used to advantage, provided they are completely bedded in concrete and do not come within 6 inches of each other or the face. The facing of the wall, if of concrete, will be composed of equal parts of gravel, sand and cement.

The *essential features of a quay wall* are weight and resistance to slip. The first is secured by using heavy material and providing ample sectional area; the second by giving a suitable bevel to the surface of the foundation so that it slopes downward from the front of the wall to the back, at the same time taking care that the maximum intensity of pressure does not exceed a certain limit, which, in the case of stiff clay, is 3 tons per sq. ft.; in the case of unconfined sand and gravel, is 2 tons; and in the case of soft, uncertain ground, is 10 cwts. per sq. ft.

Stresses in Dock Walls.

Stresses in Dock Walls.—The calculation of the stresses in a dock wall is a very complex problem, involving a considerable number of elements, the exact determination of which is impracticable, so that they have largely to be based on conjecture. Briefly stated, the conditions are as follows: The wall is subjected to the overturning influence of a wedge-shaped mass of earth which represents the excess of material behind the wall lying above the plane of natural slope of the material. The pressure, of course, is intensified by any surcharge upon the surface area of the wedge. The restraining force is the weight of the wall itself, acting vertically downward, assisted to some extent by the earth backing which may rest upon offsets, if any, at the back of the wall.

In fig. 14, let A B D K G be the outline section of a dock wall.

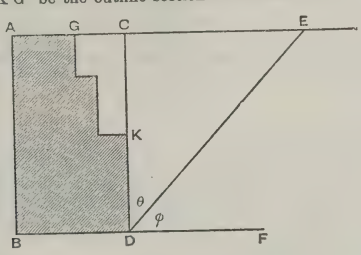


FIG. 14.

Let E D be the plane of natural slope of the ground behind the wall, i.e., the slope which the material would assume if left unsupported. The angle E D F will be designated ϕ and is known as the Angle of Repose.

Let the height of the wall A B be h, and w be the weight per unit volume; and let v be the weight per unit volume of the earth behind the wall.

Then the weight of the wedge of earth behind the wall, tending to overturn it, is

$$\frac{vh^2}{2} \cdot \tan \theta.$$

In order to cover cases where the back of the wall is not vertical and where the surface of the ground is not horizontal, this expression may be written in general terms:

$$\frac{wh^2}{2} \times \text{constant},$$

and the value of the constant, as determined by Professor Rankine in accordance with a theory which neglects any cohesion between the earth particles, is

$$\frac{1-\sin\phi}{1+\sin\phi}.$$

The line of action of the overturning force,

$$\frac{wh^2}{2} \cdot \frac{1-\sin\phi}{1+\sin\phi},$$

is taken as parallel to the surface of the ground and the point of application at one-third the height of the wall measured from the base.

For simple practical calculations for dock walls where the quay surface is almost invariably

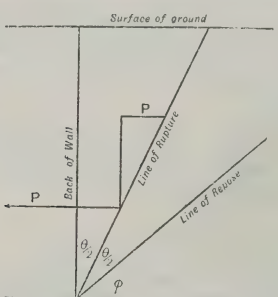


FIG. 15.

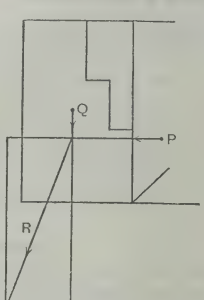


FIG. 16.

horizontal, the angle θ ($=90°-\phi$) is bisected and the value $\frac{\theta}{2}$, fig. 15, taken as representing the angle of the wedge producing maximum pressure, the other moiety being ineffective as regards overturning owing to the support it receives from the earth beneath it. The weight of the triangle $\frac{h^2}{2}\tan\frac{\theta}{2}$ is computed, and the horizontal component, P, taken of the force acting down the plane of rupture, of which $w\frac{h^2}{2}\tan\frac{\theta}{2}$ is the vertical component. This constitutes the overturning force, which may accordingly be written:

$$P=\frac{wh^2}{2}\tan^2\frac{\theta}{2}$$

The point of application of P is again at one-third of the height of the wall.

The restraining force Q is measured by the sectional area of the wall A B D G, fig. 14, multiplied by w, together with the sectional area of the earth G K C resting on the wall, multiplied by v, the two acting vertically downwards through their common centre of gravity.

Calling the overturning force P, and the restraining force Q, fig. 16, and completing the parallelogram of forces, the resultant R will be obtained, representing in magnitude and direction the force acting at the base of the wall.

So long as the line of action of R falls within the base of the wall, there is theoretical stability, but, for practical constructional purposes, it is essential:

(1) That the line of action of R should not fall outside the middle third of the base line, and

(2) That the magnitude of R should not be sufficient to crush the base; in which connection it is to be noted that the intensity of pressure increases with the eccentricity of the resultant (i.e. its distance from the centre) and that when R passes through the limiting point of the middle third of the base, the intensity of pressure on the outer edge is just twice the mean intensity. As a rough practical rule in designing dock walls, the mean width or thickness may be made about one-third of the height. It should never be less than one-fourth. The base width may be about one-half the height, but need never be more than two-thirds.

EXAMPLES OF PROPORTIONS ADOPTED FOR DOCK WALLS.

Port.	Total Height. h	Base Width. b	Sectional Area. A	Mean Width $= \frac{A}{h}$	Ratios. Base. / Height.	Ratios. Mean Width / Height.
	Ft.	Ft.	Sq. Ft.	Ft.		
Manchester	38	20	476	12·5	·53	·33
Middlesbrough	48	21	648	13·5	·44	·28
Immingham	51	31	910	17·8	·60	·34
Hull	52	20	736	14·1	·38	·27
Liverpool	55	22	760	13·9	·40	·25
Southampton	71½	39	1,692	23·6	·54	·33

CONSTRUCTIONAL COST OF QUAY WALLS.*

	Height of Wall. Ft.	Cost per lin. yard. £
Albert Dock, Hull	49	58
Albert Dock, London	37½	70
Alexandra Dock, Hull	52	84
Antwerp River Quays	56½	250
Dublin River Quay	42	120
Princes Dock, Glasgow	57½	120
White Star Dock, Southampton	71½	195

Entrance Locks.

Entrance Locks.—During periods when the water in a dock is impounded at a higher level than that of the external water (in some cases continuously), the dock can only be entered through the medium of a lock. The size of lock chambers is governed by the dimensions of the ships making use of them, and these vary at different ports. The largest vessels in existence at the present time are 900 ft. to 950 ft. long by 95 ft. to 100 ft. broad, and they draw from 35 ft. to 40 ft. of water when loaded to their full capacity. No port can claim to be of the first rank which does not receive vessels from 500 to 600 ft. long, with 60 to 70 ft. beam and drawing 30 ft. of water.

Some examples of modern dock entrance locks are as follows:—

Name.	Port.	Length of Chamber.	Width of Entrance.	Depth of Water over Sill at H.W.
		Ft.	Ft.	Ft.
Gladstone (1927)	Liverpool	1,070	130	48
Tilbury (1929)	London	1,000	110	45½
Kruisschans (1928)	Antwerp	886	115	47

See also some prominent examples of Ship Canal Locks in Section XXIX., page 1528.

Dock Gates.

Dock Gates.—Where the rise and fall of the tide does not exceed about 12 ft. the inducement to provide completely enclosed areas for shipping is not great, but above that limit the necessity for impounding the dock water becomes evident on account of the inconvenience arising from rapid changes of level, frequent alteration in moorings, and chafing against the quayside.

Dock gates are either of steel or wood—in the latter case of greenheart, jarrah, or other hard, heavy, durable timber. Greenheart gates cost more than steel gates—in some cases nearly 50 per cent. more—but their natural durability is very great and the cost of maintenance practically nil. Unfortunately, they are subject to the attacks of sea vermin, and in certain ports considerable trouble has been experienced from this cause. Salt water, especially if contaminated in any way with sewage, is extremely deleterious to ironwork, and the life of a steel gate, under these circumstances, can scarcely be expected to exceed 30 years. Greenheart gates have been found in excellent condition at the end of 50 years.

Stresses in Dock Gates.

Stresses in Dock Gates.—Dock (double-leaf) gates are maintained in equilibrium under the action of four external forces:—(1) the resultant pressure of the water against the back of the gate $= \frac{wh^2}{2}$ per lineal foot of gate, where w (62½ lbs. for fresh, 64 lbs. for salt water) is the weight

* See footnote, page 1456.

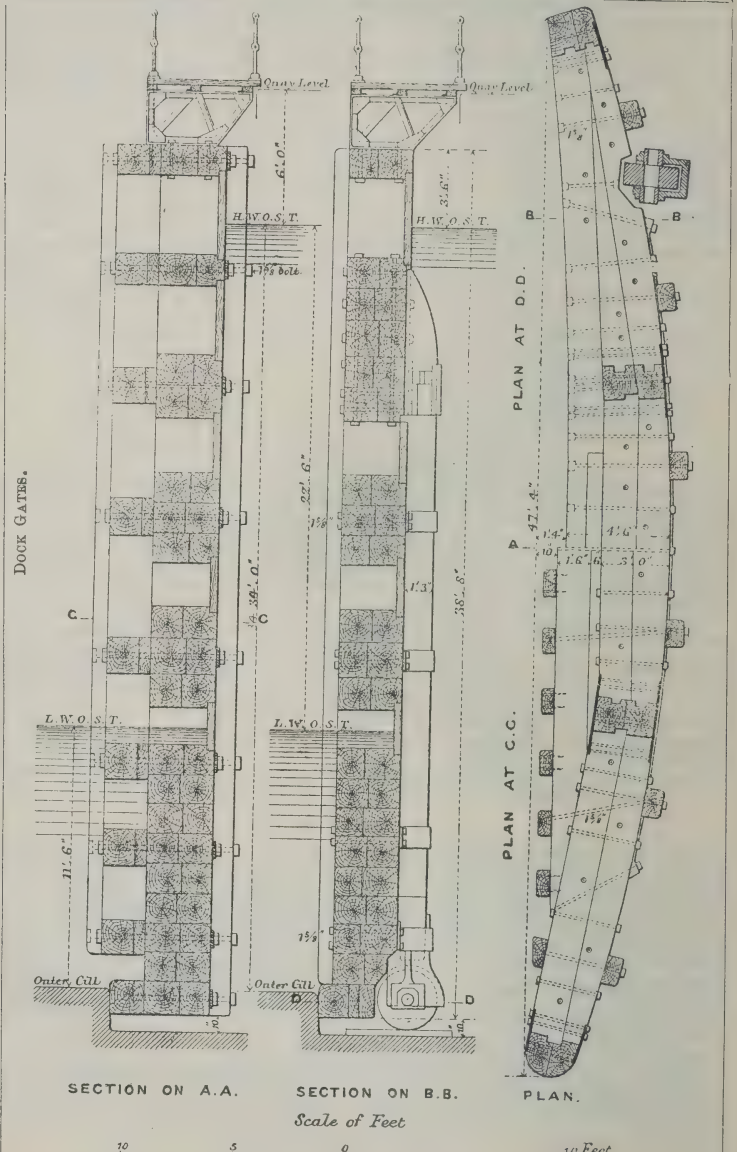


FIG. 17.—Dock Gates. (Timber.)

of a cubic foot of water, and h its height behind the gate. This force acts horizontally at one-third of the height. When there is water on both faces $p = \frac{w}{2}(h^2 - h_1^2)$, and the line of action is determined by the centre of gravity of a quadrilateral representing the difference in pressure. (2) The reaction of the mitre or meeting posts of the gates acting perpendicularly to the meeting faces, and therefore also perpendicular to the centre line of the waterway. (3) The reaction of the hollow quoins passing through the heel-post of the gate and also through the point of intersection of the other two forces. (4) The reaction of the sill, which is generally neglected.

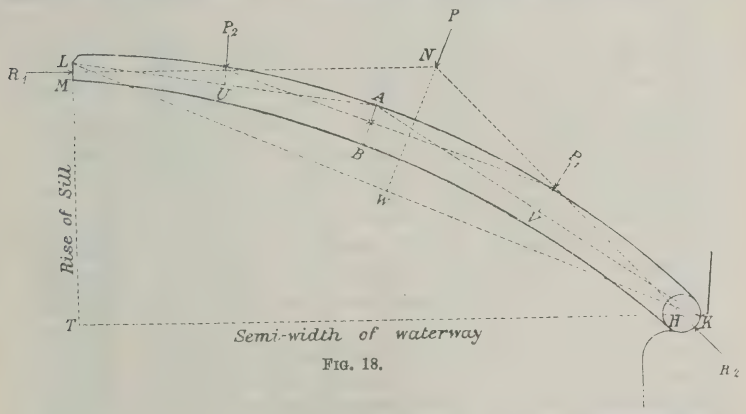


FIG. 18.

Fig. 18 represents the plan of one leaf of a pair of gates. P is the total water pressure upon the back of the leaf, assumed concentrated at its centre. R_1 is the mitre-post reaction of the adjoining leaf, taken as passing through the centre line of the abutting surfaces. R_2 is the reaction of the hollow quoin, assumed to act through the axis of the heelpost. These three forces being in equilibrium, the triangle of forces *ors* (fig. 19) can be drawn, having its sides parallel to

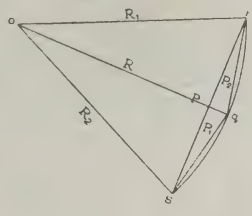


FIG. 19.

the forces P, R_1, and R_2 respectively, and since the magnitude of P is known, the magnitudes of the other two are determined.

If now it be required to find the position and amount of the resultant stress across any section AB (fig. 18) of the gate, the procedure is as follows:—Join the point A to each of the extremities K, L of the water-bearing surface of the leaf; bisect these lines at U and V respectively, and draw perpendiculars to represent the total water pressure on each section. Each section is in equilibrium under the action of these forces: in one case, the water pressure, the heelpost reaction, and the stress across AB; in the second case, the water pressure, the mitre-post reaction, and also the stress acting across AB in the opposite direction. Join the points of intersection of the known forces to obtain the line of action of the resultant pressure at the section AB. Its magnitude may be determined by drawing a parallel line, *qo*, in the force diagram from the point *o* (fig. 19). By taking a series of sections in this way, it will be found that the locus of *q* is sensibly the arc of a circle, and therefore that, except in the case of very flat gates, the resultant pressure is so nearly constant as to be justifiably considered so without serious error.

Rise of Sill.—The 'rise of sill,' MT (fig. 18), varies in different cases from $\frac{1}{4}$ to $\frac{1}{8}$ of the 'span of the gates' or width of waterway. The vicinity of the latter limit is the more common, especially for large cylindrical gates, where a large rise would entail a correspondingly long and deep recess in the side walls.

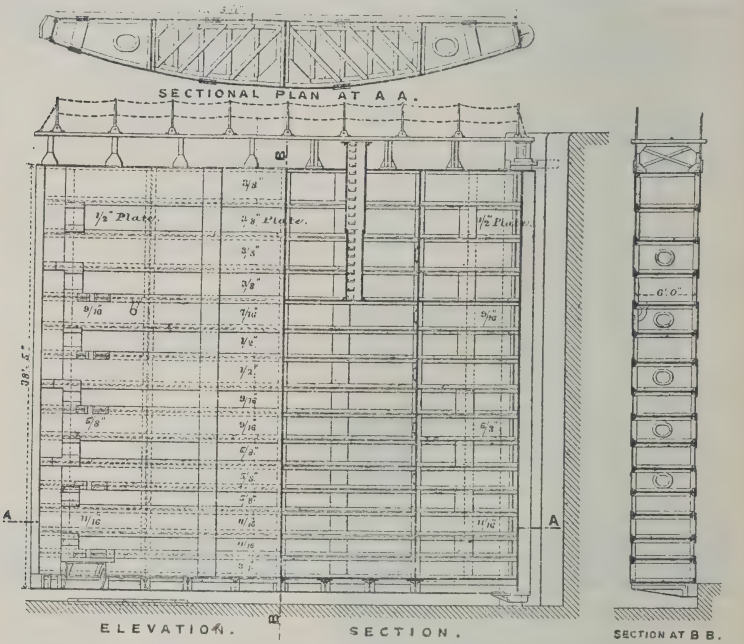


FIG. 20.—Dock Gates. (Steel.)

PARTICULARS OF DOCK GATES.

Port	Material.	Width of Waterway.	Depth of Water at H.W.O.S.T.	Height of Gates.	Weight of one Leaf.	Cost per sq. ft. of Gate.*	Remarks.
		Ft.	Ft.	Ft.	Tons.	*s. d.*	
Liverpool.	Greenheart	90	39½	44½	165	45 3	
London .	Steel	80	36½	38½	185	—	
Do.	Steel	100	45	38	309	—	
Hull .	Greenheart	85	34	38½	176	48 0	
Manchester	Steel	65	40	—	—	33 9	Similar Greenheart gates cost 49*s.* 9*d.* per square foot.
Limerick .	Steel	70	23½	26	45	25 3	
Avonmouth	Greenheart, pitch pine and memel	70	38½	48	102	26 0	

CAISSONS.

Caissons are sometimes preferred to gates on the ground that they do not absorb the length of side walls which is required to house gates when a passage is open. Furthermore, although the first cost of a caisson is undoubtedly in most cases greater than that of a pair of gates, yet

* See footnote, page 1456.

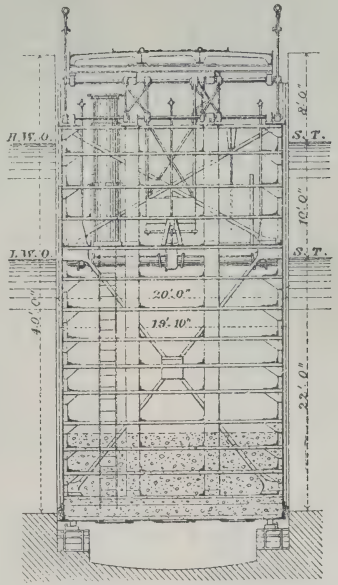


FIG. 21.—Section of Traversing Caisson.

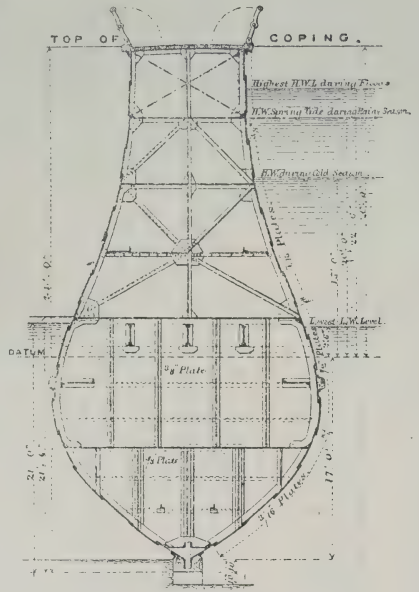


FIG. 22.—Section of Ship Caisson.

if the cost of a swing bridge for vehicular traffic, which is a necessary adjunct in the case of gates, be also taken into consideration, the advantage will be found to lie with the caisson.

Caissons are mainly of two types: traversing and floating. Traversing caissons are rectangular boxes with a rectilinear motion either sliding or rolling. When withdrawn from the passage way, they are housed in a special chamber constructed in the side wall. Floating caissons are usually ship-shaped, having more or less the form of a navigable vessel, though the curvature of the sides varies very much with the depth of water in which they have to float. Floating caissons are not easy to manage in boisterous weather and strong currents.

PARTICULARS OF CAISSONS.

Port.	Type.	Width of Waterway.	Depth of Water on sill at H.W.O.S.T.	Height of Caisson.	Cost per sq. ft. of Waterway.*
		Ft.	Ft.	Ft.	s.
Quebec . . .	Rolling .	62	25½	32	72
Malta . . .	Sliding .	94	36½	40½	85·7
Belfast . . .	Ship . .	80	31	33	35
Barry . . .	Floating .	60¼	26¾	34½	47·2

QUAYSIDE APPLIANCES.

Quay Cranes.

For general work, quay cranes range in lifting capacity from 1 to 3 tons. For hydraulic cranes the usual capacity is 30 to 40 cwts. It very often happens that the parcels from a ship's hold do not exceed 12 to 15 cwts., and, under these circumstances, a hydraulic crane of single and much higher power than this cannot be worked to advantage, as the consumption of pressure water is the same whether the load be infinitesimal or the maximum. Electric cranes are better adapted to variable loads, since the current consumed is directly proportional to the work done. On the other hand, the initial cost of electric cranes is some 30 to 40 per cent. more than that of hydraulic cranes, and the capital charges (interest and depreciation) are factors of considerable importance in estimating the relative economy. Electric cranes are now commonly installed at leading ports with 3 tons maximum capacity.

Crane Types.—Quay cranes are generally movable and include several types.

(*a*) The ordinary crane, with a wheel gauge of 10 to 15 ft. running at quay level.

(*b*) The portal crane, with an arched base spanning one or more lines of railway sidings and admitting of the passage of rolling stock beneath the crane.

(*c*) The semi-portal crane, like an inverted L, with one leg or support running on a rail at quay level, and the back support carried by a rail at a higher level on an adjacent shed structure, the object being the same as in the case of the portal crane.

(*d*) The roof crane, carried entirely on the roof of the shed or warehouse.

Types (*a*) and (*d*) are adapted to cases where there are no sidings, and type (*d*) where the shed is situated close to the edge of the quay.

The *cycle of operations* of a quay crane consists of lifting from a hold, slewing, depositing, and returning unloaded, and should, in ordinary cases, where the range of lift does not exceed 50 or 60 ft., be performed in about one minute. A good average under favourable conditions would be 50 cycles per hour; but spread over any lengthy period. this rate could hardly be maintained, for reasons entirely unconnected with the crane.

Speeds of Quay Cranes.—The ordinary speeds of 30 to 35 cwts. quay crane machinery are as follows:—

Lifting, full load	250 ft. to 300 ft. per min.
Lifting, light	350 ft. to 400 ft. per min.
Lowering, full load	400 ft. per min.
Lowering, light	150 ft. per min.
Slewing	one revolution in 25 or 30 seconds.

Luffing Cranes.—It is an important consideration in modern quayside work that cranes should have considerable outreach. Not only is it necessary for them to plumb the hatchways of vessels moored alongside, but in many cases it is desirable to command a barge or lighter beyond. This involves an outreach of 50 to 60 ft. from the face of the quay for even vessels of moderate beam, and entails a luffing or derricking movement of the jib. In view of the obstruction to radial movement caused by ships' yards and stays, the luffing of the crane jib has of recent years become an important feature of crane design. Formerly, the load was unavoidably raised during the inward travel of the jib with a correspondingly unnecessary expenditure of energy and time. This has now been obviated by the introduction of compensating luffing gear by means of which the load is carried at a uniform level throughout its range of travel.

* See footnote, page 1456.

Compound Cranes.—Quay cranes have generally a radial action, and though the luffing gear provides a rectilinear motion, the movement of goods is mainly along a more or less circular path. The joint effect is to afford considerable scope for the deposit of goods on the quay in contradistinction to cranes of the transporter type, which can only function along a straight line and often only deposit at a single point. Such cranes, however, have an advantage in being able in certain cases to transmit their loads within a transit shed and so under cover. At Hamburg, a combined type of quay crane has been introduced with the usual radial jib on a rotating platform above, while below, within the framework of the pedestal, there are one or two crabs travelling on horizontal tracks at right angles to the quay front. The aggregate power of these cranes is considerable, reaching to as much as 6, and even 9, tons.

Fixed Quay Cranes.—Fixed quay cranes are provided for dealing with special loads of considerable calibre. The powers provided are generally 5, 7, 10, 12, 15, and 25 tons, with appliances at first-class ports ranging up to 150 or 200 tons. The rates of lifting are correspondingly slower; about 150 ft. per min. for 5-ton cranes 30 ft. per min. for 25-ton cranes, and 20 ft. per min. for 50-ton cranes.

(See also 'Wharf Cranes,' p. 810.)

Burtoning.

General cargo is mainly, and in fact almost universally, handled at North American ports by a system known as Burtoning, which is the same in principle as the Union Purchase used at English ports in conjunction with ships' gear. At American ports, the sheds are equipped with cargo masts, generally prolongations of the shed front columns, rising to a height of 80 ft. or so above the quay level and supporting a longitudinal girder, or rail, running the full length of the shed. To this rail, at any selected point, a pulley may be attached, worked with a line, one extremity of which is joined, with a similar line from a pulley at the extremity of a ship's boom, to the hook carrying the load. Two winches are required for operating: one for each line. The ship's winch and line serve to lift the load from the ship's hold to deck level; the shed winch then takes charge, and by means of the shed line draws the load overside to the quay, where it is lowered to the ground.

Floating Cranes.

Floating Cranes.—For the heavier lifts, floating cranes are often preferred. It is a convenience to bring the crane to the ship rather than *vice versâ*. Against this must be set the greater initial and working cost of a floating crane. Floating cranes are provided at most first-class ports of lifting power ranging from 25 to 100 or 150 tons. A floating crane recently built for a Japanese port has a working load of 200 tons at 105 ft. radius, 100 tons at 150 ft. radius, and 30 tons at 160 ft. radius. The heaviest lifts are performed at the rate of 5 ft. per minute. A revolution takes 6 to 8 minutes.

A floating crane of 200 tons capacity is among the plant of the Mersey Docks and Harbour Board. It is an electrically-driven, derricking, and revolving crane, 154 ft. in length, 88 ft. in breadth, and 16 ft. in depth. The hoisting speeds vary from 6 ft. per min. for loads of 200 tons, to 20 ft. per min. for loads of 60 tons. The highest point of the jib at its highest position is 246 ft. above the water-line, with a maximum radius for loads of 60 tons amounting to 183 ft., and for loads of 200 tons, to 110 ft. The vessel is named the *Mammoth*; in conjunction with its outreach it is claimed to possess the largest lifting capacity of any existing floating crane. There are, however, two floating cranes of 250 tons capacity at the Panama Canal.

Even for relatively small power lifts (5 tons or less) floating cranes are employed with advantage at some French ports where the conditions are favourable for their use. The length of the crane pontoons are about 60 ft., a convenient dimension from the point of view of manœuvring; the width increases (to prevent transverse oscillation) from 17 to 31 ft. in proportion as the power of the cranes increases from 1½ to 5 tons and the outreach from 25 to 50 ft. The height of the jib pulley varies from 53 to 65 ft. above water level.

(See also CRANE Section, p. 811.)

Transporters.

Transporters.—Where loads are uniform, can be readily handled, and have to be transmitted some distance, a transporter offers considerable facilities for dealing with them at high speeds, ranging up to 1,200 or 1,500 ft. per minute. Quayside transporters have generally to be furnished with hinged cantilever ends capable of being raised when out of action. The travelling motion is not necessarily rectilinear, curved tracks being employed where necessary.

Coaling Appliances.

Shipment of Coal.—Coal is shipped in a variety of ways, principally in this country by means of the hoist and tip or the coaling crane, but also by means of the belt conveyor, perhaps more especially abroad. The belt conveyor requires a layout on the quay, which is not always practicable or convenient in congested areas such as prevail at English docks. The hoist raises the wagon to the requisite height and tips it at an angle of 45° or 50°, so that the contents pass into a shoot. The crane lifts the wagon bodily, slings it over the ship's hatchway and there empties it. At Cardiff there is an installation on the Lewis-Hunter system by which the

wagon, instead of being lifted, discharges its contents into a hopper placed in a pit on the quay, and the hopper is lifted by the crane and lowered into the hold, where, being fitted with cone valves, it can be emptied within a few inches of the floor. This system is adopted on account of the great liability of the South Wales coal to breakage and comminution. The protection of coal from breakage is of such importance that in nearly all cases where coal is tipped from a considerable height, an anti-breakage crane is provided to lower the first few loads to the bottom of the hold.

Coal wagons in this country are commonly built to contain 10, 12, 15 or 20 tons of coal, and the gross weight ranges from 15 to 30 tons. To this must be added from 6 to 9 tons for the cradle or platform. Consequently the lifting capacity of a coaling appliance will range between 20 and 40 tons according to the class of wagon dealt with. In certain special cases, as at Goole Docks, provision is made to lift a gross load of 50 or 60 tons, of which up to 40 tons is coal. The coal is brought along the Aire and Calder Canal in tanks, which are floated under the hoists.

In America the size of coaling wagons is very much greater and varies from 40 tons upwards to 100 or 120 tons. At South Wales ports attempts are being made to secure the general adoption of larger-sized wagons (20 ton capacity) than has hitherto been the practice.

The rate of coaling by an ordinary crane or hoist is from 300 to 450 tons and may even reach 600 tons per hour, but it is largely governed by the rate of trimming operations in the hold. The lifting and lowering of the wagons is done at speeds of between 100 and 180 ft. per minute and slewing at about one revolution per minute.

The belt conveyor consists either of a continuous rubber belt or of articulated steel plates. Of the two, the rubber belt has shown itself to be more convenient and more durable than the steel plating, the wear on which is considerable. Rubber belts are run at rates up to 600 or 700 ft. per minute, and they discharge the coal in a continuous stream. The belt is fed by means of a hopper recessed in the quay, into which the coal wagons are emptied. Installations in this country are of moderate proportions, the width of the belt running up to about 48 ins., but in the United States very much larger installations have been effected with belts of 60 ins. diameter. These belts travel up inclined ways to the summit of movable towers, whence the coal is passed on to a transverse belt discharging into the ship. Considerable expedition has been gained by the use of the mechanical trimmer, which is a short, rapidly rotating belt at the foot of the vertical shoot leading into the hold. The speed of the trimmer belt ranges up to 2,000 ft. per minute, and the acquired velocity projects the coal forward into all parts of the hold to distances up to 40 or 50 ft.

Ship's Bunkering.—The foregoing appliances are in vogue in connection with shipping coal cargoes. For the more moderate requirements of ships' bunkers, the loading appliance has generally to be brought to the ship's side, so as to proceed simultaneously with cargo discharging or loading operations. A good deal of bunkering is done by hand, with such primitive assistance as is afforded by baskets, ropes, pulley blocks, and winches, but the rate of coaling under such conditions is very slow and relatively costly—perhaps 2 tons per man per hour. There are a number of special floating machines provided at important ports for doing the work more expeditiously. A typical example, and one of the most modern designs, is the *Suisted Elevator*, built for the company of that name by *Spencers (Melksham) Limited*, and at work in the port of London. It is capable of handling 300 tons of coal per hour, but such a rate is manifestly only practicable under favourable circumstances; it could not be maintained with trimming operations going on.

Coal Discharge.—For the importation of coal, as also of ore, the grab is perhaps the appliance most commonly in use in this country, as also on a much more gigantic scale at the Lake Ports of North America. As used at English ports, the capacity of the grab runs up to 6 tons, with a total combined load, including the grab itself, of about 12 tons. At the Lake Ports of North America the standard type is a grab capable of taking a load of 12 tons, and the largest can take 17 tons. There is a 'bucket' for clearing up the holds at the close of operations, which when outspread has a span of 24 ft. between the bucket lips.

Grain Elevators.

Grain Elevators.—Grain is handled either by the bucket elevator and conveyor, or by the pneumatic elevator or suction apparatus. The latter has the advantage that the pipes, being flexible, can easily be taken to the furthermost parts of a ship's interior, and that no restrictions are imposed by variations in freeboard and depth of hold. The bucket elevator, though cheaper to construct and requiring less power, is hampered in working by the necessity of ploughing and trimming a preponderant portion of the grain—often as much as 80 per cent.—to the elevator leg. A special gang of competent men is required for the purpose, and this, of course, increases the cost of handling.

It is impracticable to do trimming with a ship's winch at a higher average rate than about 80 tons per hour. Beyond this limit the pneumatic machine becomes more effective, and pneumatic elevators can be constructed so as to give outputs of between two and three hundred tons per hour. On the other hand, conditions at ports are often such that so high a rate of discharge cannot be maintained, operations being governed by the rate of delivery. In practice something like 100 tons per hour is found to be the maximum consistent with effective working.

It has recently been realised that quite small pneumatic plants can be installed in warehouses, mills, etc., with extremely satisfactory and economical results in working. They vary in capacity from 15 to 50 tons per hour, and are generally driven either by the mill shafting or by electric

current where available. Convenience in handling, absence of dust and avoidance of trimming are strong points in their favour.

An objection urged against the suction apparatus is that the surface friction of the pipes tends to injure some of the more delicate classes of grain, such as malting barley, but this drawback, for all practical purposes, has been eliminated of late years by the easing of bends and the removal of all internal projections in the pipes. In any case it should not be overlooked that the cutting and driving action of the buckets of a bucket elevator cannot be without some similar attritional effect.

Bucket elevators are either housed at the quayside or are mounted on barges, in both cases being fitted with derrick supports and suspension gear to enable them to be projected forward and set into the ship's hatchway. Elevator speeds generally range between 200 and 400 ft. per minute, but may be as high as 600 ft. per minute. Band or belt conveyors (p. 843) run at 300 to 600 ft. per minute, but for light grain, such as oats, the speed should not exceed 500 ft. per minute.

The development of the pneumatic operation of handling grain has been very marked in recent years. The slow-speed horizontal reciprocating steam-driven suction pumps have been largely superseded by quick-acting vertical pumps driven by internal combustion engines which are coupled direct to the pumps. A further development is the introduction of a multi-stage air exhauster, also driven in many cases by an internal combustion engine through speeding-up gear. This type is perhaps coming mostly into favour, especially where electric power is available.

Timber Appliances.

Timber, as received at ports, is generally divided into the categories of hard wood and soft wood, the former comprising logs of greenheart, teak, oak, etc., and the latter consisting chiefly of deals, battens, and boards of pine and spruce. Logs are handled overside by crane and either floated to timber ponds or conveyed on trollies to storage sites. In Liverpool a special arrangement of tandem axles is employed to which the logs are underslung.

Cargoes of deals may consist of from 150,000 to 300,000 pieces of various sizes and marks. This renders the use of apparatus of the conveyor type difficult and unsuitable, although such appliances have been successfully installed in certain cases. Generally speaking, cargoes of this class are best dealt with by steam cranes of moderate capacity (30 cwts. to 2 tons) which, arranged to run at right angles to the quay front, convey their loads to an adjacent storage ground at speeds up to 600 ft. per minute. The lifting speed is about 160 ft. per minute, and the slewing speed about 1½ to 2 revolutions per minute.

There is an appliance in use at American ports as well as at sawmills and lumber yards which has proved very serviceable for handling timber. It is known as the Ross carrier, and is a framed motor-driven carriage, standing about 9 or 10 ft. high, the driver's seat being at the top. The framework is of arched or pierced pedestal type, so that the carriage can run astraddle of the load to be lifted, which is placed on 'bolsters,' or cross-bearers, on the ground. Suspended from the carriage chassis are two adjustable side frames with angle iron clips which can be lowered, brought under the bolsters, and then raised to a height of a foot or so to give the necessary clearance for travelling. The carrier transports its load at a speed of 12 to 15 miles per hour. The load to be carried may be 3 ft. wide, 4 ft. high, and of any ordinary length of deals; the carrier frames being about 12 ft. long, the deals project back and front. The carrier can also be used for logs and timber in baulk within the same limits of width and height. The carrying capacity is 10,000 lbs. The appliance is used in connection with a sorting table, known as the Ross break-down rig or drop table, which receives the parcel of deals from the carrier and drops, thereby releasing the deals from the bolster and placing them on a series of chains which serve to break down the load, so that it can be spread over the table and sorted to grade and size.

GRAVING AND FLOATING DOCKS.

A modern graving dock is an excavated chamber, two sides, one end and the floor of which are lined, either naturally or artificially, with watertight material, generally concrete with brick or granite facings and dressings. The remaining end is the entrance and is provided with a pair of gates or a caisson. After the entry of a ship, the entrance is closed and the water is pumped from within the dock, though, in certain cases, where graving docks are entered direct from a river or open waterway where tidal conditions prevail, the operation may be assisted, and sometimes be altogether effected, by the fall of the tide.

The floating dock is an open-ended, hollow structure of steel or iron, which, in order to receive a vessel, is sunk to the requisite depth, by allowing its air chambers to fill with water. When the vessel has been berthed, this water is pumped out, the dock rises bodily and so lifts the vessel above the water-line.

It is not altogether easy to make an effective comparison of the respective merits of graving and floating docks, on account of the wide divergence in the conditions to which each structure is specially applicable, and, although there are strong and convinced advocates for the general superiority of one or other of the two systems, a brief summary of the most salient considerations will show that it is scarcely justifiable to decide conclusively in favour of either. Floating docks are almost exclusively adopted at Hamburg, Rotterdam and other Continental ports; in Great Britain there are considerably more graving docks than floating docks.

A graving dock requires a good foundation at a reasonable depth, and a disposition of site which frequently involves the appropriation of a considerable area of valuable land.

A floating dock requires a sheltered position with an adequate depth of water, which, if not obtainable naturally, will have to be gained by dredging. The depth of water to be provided for a floating dock must be sufficient to cover the draught of the ship, the depth of the pontoons and the requisite clearances of a foot or two between keel and blocks and between pontoon and bed. Where there is tidal variation, the amount of this must also be added, if the dock is to be continuously accessible. In this country the tidal range may be anything up to 35 ft., and between 15 and 20 ft. is very general. For a modern ship of average size drawing 28 to 30 ft. of water, the pontoons would have to be at least 20 ft. deep. This means that the total depth of the floating dock berth below high water level would be about 70 feet, and there are circumstances under which dredging to so great a depth would be not merely difficult, but almost impracticable.

In sites where the foundation is defective, the depth of excavation required for the floor of a graving dock may easily equal the depth of the floating dock pontoons, and the difficulties and expense in these cases are just as pronounced.

The utility of a graving dock is measured by the linear dimensions of the largest vessel which it can accommodate. The weight of the vessel is relatively a matter of little or no importance.

On the other hand, the utility of a floating dock is gauged by its maximum lifting capacity and is independent of the exact size of the vessel. Floating docks can, and often do, take vessels many feet in excess of their own length and, in the case of one-sided docks, in excess of their own width.

The size of the entrance of a graving dock is influenced by the cost of the gates, or caisson, to be used for closing it, and by the greatest width that can safely withstand the pressure of water. In floating docks these considerations do not enter into account.

Relative Costs of Graving and Floating Docks.*

Initial Cost.—The initial cost of graving docks, including pumping and general equipment, measured in terms of their serviceable dimensions (i.e. internal length by breadth of entrance by depth of water over blocks or sill), has varied from 1*s.* 3*d.* to 3*s.* 6*d.* per cubic foot, and, in a rough-and-ready way may fairly be taken for a modern concrete and granite dock of ordinary size, constructed in this country under normal conditions, on a pre-war basis, at 2*s.* per cubic foot.

The cost of floating docks is usually measured in terms of their lifting power, and varies inversely as the size of the dock, ranging from about 14*l.* per ton for a dock which would take small coasting ships to about 6*l.* per ton for one to take large Atlantic liners.

As regards time of construction, a graving dock of any importance could hardly be completed within less than eighteen months or even a couple of years, whereas a floating dock could be ready in half the time. The floating dock could, moreover, be constructed in any convenient locality and transported to its destination at some additional expense for towage and insurance. Many floating docks for the colonies are constructed in home shipyards.

Durability and Upkeep.—A stone or concrete graving dock is practically indestructible, and the only charge brought against it, in this respect, is that of outlasting its utility and becoming obsolete. This is, no doubt, true in regard to the class of vessel for which it is constructed; but, on the other hand, when the growth of shipping has left the graving dock too small for the larger vessels, it is still utilisable for those of smaller size. With the exception of the expenditure on the gates or caisson, which close the entrance, there is practically no maintenance outlay. It is all probably covered by an allowance of $\frac{1}{2}$ per cent. per annum.

The life of a floating dock, considered as an iron or steel structure constantly afloat, will vary with its situation, some waters being more destructive than others. It may be fairly reckoned at from 40 to 50 years, and there is no reason why, with reasonable care and attention, the longer period should not be exceeded. The annual cost of maintenance estimated from several existing examples lies between 1 and $1\frac{1}{2}$ per cent. of the capital outlay. Any cost of dredging required to keep the berth clear of mud deposits must be added to this allowance.

Working Cost.—The amount of pumping to be done in the case of a graving dock generally greatly exceeds that in the case of a floating dock—it may be as much as 3 or 4 times. In a floating dock only so much work is required to be done as is necessary to raise the vessel and the deck of the pontoon above water level. The graving dock chamber, on the other hand, has always to be completely dewatered, no matter what may be the size of the vessel occupying it, and the smaller the vessel the greater the quantity of water to be pumped.

Small graving docks on the banks of a tidal river, in cases where time is unimportant, may be emptied of water by natural means. These cases are exceptional.

But even in the larger docks the cost of pumping is small in comparison with the capital charges. For one hundred dockings a year they probably do not amount to more than 10 or 12 per cent. of the whole annual expenditure.

Again, the pumps being only intermittently employed, one pumping station may be made to serve two graving docks, whereas each floating dock requires its own installation. Possibly an extra hand or two is also needed to attend to the valves of the floating dock in order to

* See footnote, page 1456.

maintain an even keel during pumping operations, but generally all the valves are worked from a central position.

As regards general adaptability, floating docks have the advantage, already pointed out, of mobility, and they may be transferred from one port to another, whenever the circumstances require. This has already been done on several occasions to fall in with altered trade conditions, or for purposes of war.

A floating dock can be trimmed to take a vessel having a list which conceivably might interfere with its passage through a graving dock entrance with vertical sides. This operation was frequently performed during the war.

The mobility of a floating dock is not free from certain drawbacks. The dock may founder or be wrecked.

All these varying considerations point to the conclusion that there is scope for both graving and floating docks, and that each may be employed to better advantage than the other under some particular set of local conditions.

SOME OF THE LARGEST FLOATING DOCKS IN THE WORLD.

Location.	Lifting Capacity.	Length.	Inside Width.	Depth of Water over Keel Blocks.	Owners.
	Tons.	Ft.	Ft.	Ft.	
Southampton	60,000	960	130½	38	Southern Railway Co.
Singapore	50,000	855	132	40	British Admiralty.
Rotterdam	46,000	695	142	35	Wilton's Eng. & Slipway Co.
Hamburg	46,000	575	123	33	Blohm & Voss.
Malta	65,000	962	140	38	British Admiralty.
Devonport	33,000	680	113	36	Ditto.
Hamburg	38,000	625	123	33	Blohm & Voss.
Portsmouth	32,000	680	113	36	British Admiralty.
Nikolaieff	30,000	656	136	30	Soviet Government.
Brooklyn, U.S.A.	30,000	685	100	31	Morse Dry Dock Company.

THE SOUTHAMPTON FLOATING DOCK.

The following particulars are extracted from a paper read before the Institution of Civil Engineers in January 1925, by Dr. E. H. Salmon (*Min. Proc. Inst. C.E.*, vol. 219):

Weights and Lifting Power.—The total net estimated weight of the dock in its final form was 18,990 tons, made up as follows:

	Tons.
Steelwork in hull, including mooring-booms	16,482
4½ per cent. for rivets	741
Joint bolts	17
Machinery	404
Fittings	736
Timber	407
Cranes, crane-rails, and collecting gear	183
Stores and spare gear	50
Proportion of weight of brow and moorings carried by dock	33
	19,053
Less half-weight of mooring-booms	63
Total	18,990

The above weight does not include any allowance for overweight and extras. From measured displacements of the finished dock its actual weight was found to be 19,330 tons. The displacement of the pontoon at Southampton (density of water = 35·09 cubic feet per ton) is 81,840 tons. At the conclusion of the test made to determine the lifting power of the dock, 1,530 tons of water remained in the pontoon. The true lifting power of the dock, therefore, is 81,840 − 19,330 − 1,530 = 60,980 tons.

Cost.—The original contract price for the dock delivered at Southampton was £375,000. The corresponding price of the steelwork in the hull comes to £17 per ton; the machinery, about £120 per ton; the fittings, about £32 per ton; and the timber, £22 per ton. A number of extra fittings over and above the original contract have been added, somewhat increasing the total cost of the finished structure. The above figures include the dock proper and mooring booms, but not the shore work, nor the preparation of the site.

PARTICULARS OF SINGAPORE FLOATING DOCK.

(*Built* (1928) *by Swan, Hunter, & Wigham Richardson, Ltd., Wallsend-on-Tyne, for the British Admiralty for service at Singapore.*)

Lifting capacity	50,000 tons
Length	855 ft.
Breadth outside	172 ,,
Clear width between fenders	132 ,,
Height of side walls and pontoon	75 ,,
,, walls from pontoon deck	50 ,,
Steel weight, including joint bolts, castings, etc.	21,750 tons
Number of rivets	3,500,000

Pumping capacity: 30,000 tons per hour, viz.:

Three	pumps,	each	expelling	6,000	tons	of water per hour.
Two	,,	,,	,,	4,000	,,	,, ,,
Two	,,	,,	,,	2,000	,,	,, ,,

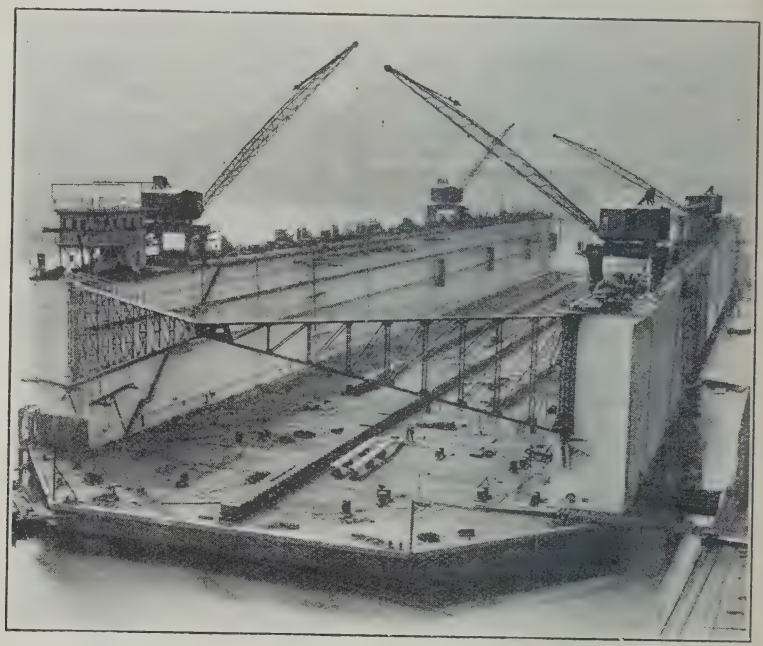


FIG. 23.—Singapore Floating Dock.

The pumps are of centrifugal type and driven by electric power. They can work at full capacity except when there is less than approximately 4 ft. of water in the tanks.

There are also four wash-down and fire-extinguishing pumps, each with a capacity of 75 tons per hour.

Electrical equipment:

(*a*) 1,000 volt, 3-phase, alternating current system for main generation and pumping.
(*b*) 220 volt, 2-wire, direct current system for auxiliary generation, lighting and power.
(*c*) 20 volt, 2-wire, direct current system for valve control in docking operations.
(*d*) Complete telephone system with central exchange.
(*e*) Provision for supplying ships in dock with electrical energy for lighting and power.
(*f*) Provision for supplying submarines in dock with electrical energy for charging large storage batteries, etc.

A view of the dock lying alongside the builder's yard is shown in fig. 23.

Notes on Floating Docks.

(Contributed by Swan, Hunter, & Wigham Richardson, Ltd.)

The development of different types of floating docks in this country is largely due to Messrs. Clark and Standfield, Westminster, who have designed docks of almost every kind, and for service in many parts of the world.

Floating docks may be conveniently divided into two classes : (*a*) those which are not self-docking, or ' box docks,' and (*b*) those which can be self-docked.

(*a*) The *box dock* is double sided and of ⌊ ⌋ section. It has two side walls, and is the only form of modern floating dock which cannot be divided into more than one part for the purpose of self-docking. The two side walls and the bottom pontoon are one permanent structure. Box docks of great size have been built ; for instance, the British Admiralty floating dock of 33,000 tons lifting capacity built at the Wallsend Shipyard and stationed at the River Medway.

(*b*) By a ' self-docking ' type is meant a dock divided into sections any one of which can be lifted on the remainder of the dock for painting or repairs. The following are some of the various types of self-docking docks : the *Bolted-Sectional*, the *Sectional Pontoon* or *Rennie*, the *Off-Shore*, the *Havana*, the *Sectional Box*, the *Multi-Sectional*, and the *Depositing*.

The *Bolted Sectional Dock* combines the advantages of the great longitudinal strength of the ' *Box* ' type with facility for self-docking. This type is usually built in three sections of approximately equal lengths, the two end sections being stepped to form landings when carrying out a self-docking operation.

Fig. 24 shows, firstly, the whole dock assembled ; secondly, the centre section lifted by the two end ones ; and thirdly, one of the end sections lifted by the centre and other end section. Examples of the former type of dock may be seen at Port of Spain (Island of Trinidad), Rotterdam, at Callao in Peru, and at Sténia in the Bosphorus. (See fig. 29, page 1479.)

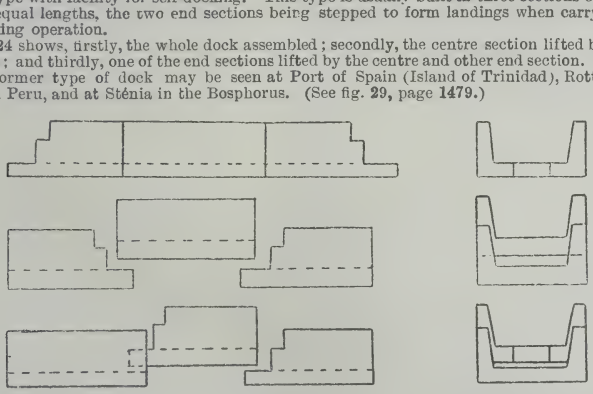


FIG. 24.—Bolted Sectional Floating Dock.

The Sectional Pontoon Dock (also called ' the Rennie Dock,' after its inventor), consists of a number of separate pontoons lying transversely to the length of the dock. Upon these pontoons and bolted to them rest two continuous side walls. The length of each pontoon, i.e. on the keel line, is such that when detached from the rest of the dock it can be passed end-wise between the walls of the latter and so lifted in the same manner as a ship. Each pontoon can be successively dealt with in the same manner. Fig. 25 shows first the whole dock assembled and then

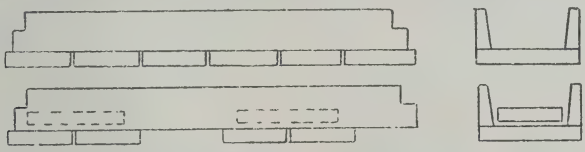


FIG. 25.—Sectional Pontoon Floating Dock.

two of the pontoons lifted by the remainder of the dock. This *Sectional Pontoon* type is very suitable for vessels of moderate dimensions, and owing to its form can easily be constructed in places where docks of other types could not be completed. The separate pontoons not being very large in size can be easily erected and launched, after which the walls can be built upon them. Examples of this type of dock were built at Wallsend-on-Tyne for the Government of Southern Nigeria, and for service at Saratov on the River Volga.

The Off-Shore Dock is an ingenious design patented by Messrs. Clark & Standfield in 1884 ; a well-known example of it was built at Wallsend-on-Tyne for a ship-repairing establishment at

Sunderland, and there is another good example of it also constructed by Messrs. Swan, Hunter, and Wigham Richardson, Ltd., at Penarth, near Cardiff. The off-shore dock (fig. 26), of which the end elevation is 'L' shaped, is very easily handled. The side wall is connected to the shore by booms or girders hinged at each end. When the dock is sunk to receive a vessel the

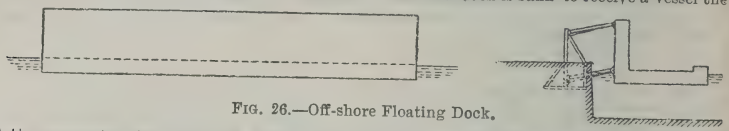


FIG. 26.—Off-shore Floating Dock.

latter can enter the dock from either end or may be warped in sideways. It is then rapidly centred by telescopic side shores and supported underneath by mechanical bilge-shores. This type of dock is divided into two sections, and as it has only one wall one-half of the dock can be lifted, so to speak, in the lap of the other. Fig. 28 on this page illustrates the self-docking operation.

The *Sectional Box Dock* (fig. 27) was originated by Messrs. Clark & Standfield, Westminster. It combines the advantage of ease and simplicity in self-docking possessed by the Sectional Pontoon dock, with the strength of the solid or *Box* dock. The dock is divided, like the ordinary sectional dock, into sections of such a length that any one section can be turned round 90 degrees and docked on the rest of the dock. In the Sectional Box dock, however, the sections are all

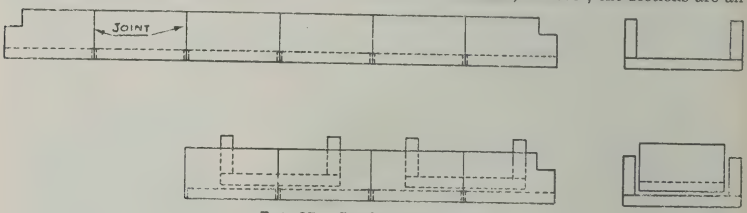


FIG. 27.—Sectional Box Dock.

joined together by a strong, rigid joint of a similar type to that used in the bolted sectional docks running round the whole profile of the dock. The 60,000-ton dock at Southampton, designed by Messrs. Clark & Standfield, and the 50,000-ton dock built by Messrs. Swan, Hunter, & Wigham Richardson for the British Admiralty for Singapore (fig. 23, p. 1476), are notable examples of this type.

The *Multi-Sectional Dock* is a modification of the Sectional Dock type. In small docks it is not necessary to connect the bottom plating right across the width of the pontoon. The joints at the sections only occur, therefore, round the profile of the side wall, the spaces between the pontoons being decked over to form a continuous platform.

The *Depositing Dock* was designed as early as 1877, by Messrs. Clark & Standfield. In section it is 'L' shaped. The single side wall is continuous, but the bottom pontoon is made up of a series of 'fingers' with spaces between them (fig 28). The necessary stability is given to the dock

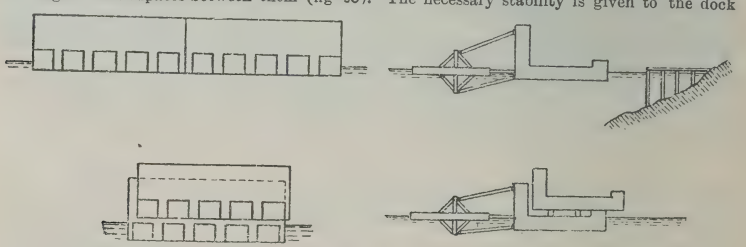
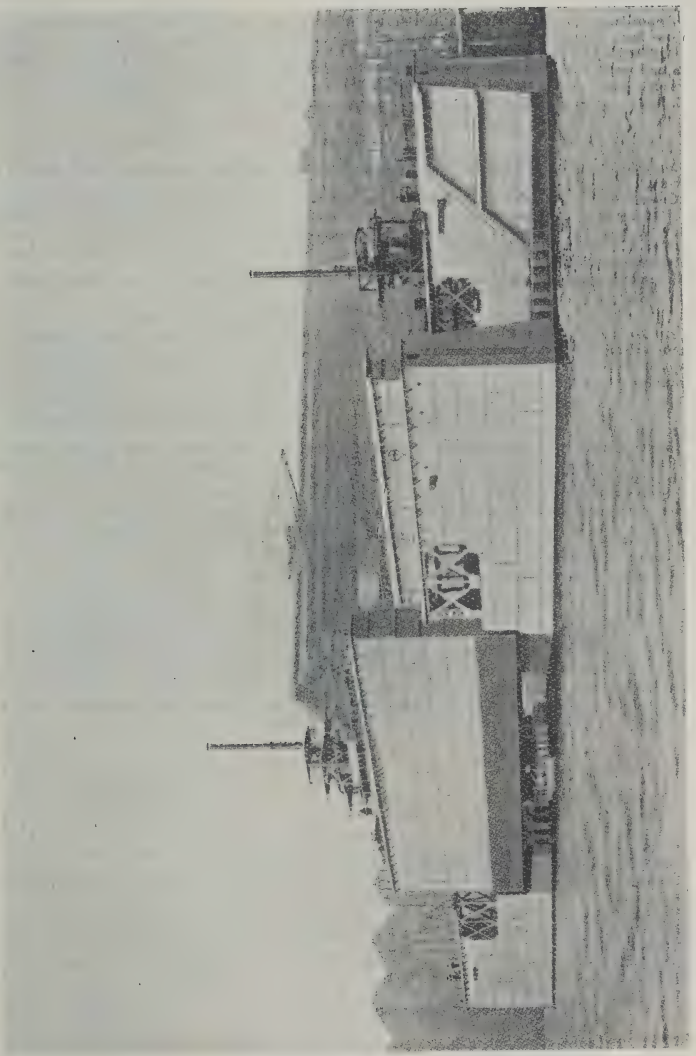


FIG. 28.—Depositing Floating Dock.

by means of parallel booms hinged at each end and connected to a floating outrigger. This type of dock can not only dock vessels, but owing to the peculiar construction of the bottom pontoon is enabled to deposit them on a gridiron formed by groups of piles. There are spaces between each set of piles of such a width as to admit the 'fingers' of the dock. The dock carrying a ship is pushed forward so that the 'fingers' are engaged between the piles and then on sinking the



FLOATING DOCK.

FIG. 29.—Bolted-Sectional Dock.

(Built by *Swan, Hunter, & Wigham Richardson, Ltd.*, Wallsend-on-Tyne, England.)

Self-docking operation; centre section lifted by the two end-sections.

dock the ship is left upon the gridiron. The wall of the dock is divided into two equal lengths, either of which can be docked by the other as in the case of the off-shore type.

Depositing docks at Kobe and Barcelona have been used for the purpose of lifting large, hollow ferro-concrete blocks for harbour construction. These were built upon the depositing grids, lifted off by the dock, and finally floated off so that they could be towed to their final position and sunk there. In the case of the Kobe dock, some of these monoliths were 120 ft. long by 41 ft. high, and weighed as much as 2,200 tons. In this way the difficulty of launching such large masses was avoided and much expense saved.

DREDGING.

Modern dredging appliances may be summarised under two classes, each with several subdivisions.

Class I.—Digging dredgers, comprising (*a*) the ladder, or elevator, dredger, with a continuous band of buckets; (*b*) the dipper dredger, with a single bucket at the extremity of a long arm, or lever; and (*c*) the grab or grapple dredger, with a drop bucket suspended at the end of a chain.

Class II.—Suction or hydraulic dredgers, comprising (*a*) the sand pump; and (*b*) the suction cutter dredger.

Rock cutters are not dredgers, but constitute an auxiliary class which will be treated separately

Ladder Dredgers.

Ladder Dredgers, or Elevator Dredgers (sometimes in Europe called bucket dredgers), are characterised by high power and great serviceability under nearly any conditions. They are adaptable to the removal of every variety of material except the harder kinds of rock. The principle upon which they are constructed is that of an endless chain of buckets travelling in an elliptical path about two pivots, or tumblers, set one at each extremity of an inclined plane. The buckets excavate material at the lower tumbler and discharge it into a shoot while passing over the upper tumbler.

Single Ladder v. *Double Ladder Dredgers.*—Dredgers of the ladder type have either one or two ladders. In single ladder dredgers, the ladder is situated along the centre line of the vessel; in the other case, the ladders are placed at each side.

Single ladder dredgers are most in favour for ordinary operations. In machines of the same power, they have the advantage of few working parts and of less working friction. As seagoing craft, they possess a more convenient outline, and their lesser width enables them to pass through narrower waterways and lock chambers. But they cannot work so closely to a quay wall as side ladder dredgers, and, if any breakdown occurs in the chain of buckets, the whole dredger is thrown out of action, whereas, with the double ladder dredger, this is not necessarily the case.

Capacity of Ladder Dredgers.—The capacity of the buckets of ladder dredgers ranges from 5 to 54 cubic ft., but, generally speaking, about 35 cubic ft. is looked upon as the maximum size consistent with convenient utility. A moderately sized dredger, with buckets of 20 to 25 cubic ft. capacity, travelling at speeds from 15 to 20 ft. per minute, and working in water of depths up to 50 ft., may be expected to raise from 400 to 500 tons of stiff clay per hour; in suitable material, such as ballast and gravel, the rate obtained may be as high as 1,000 tons per hour. The most powerful dredgers can double this performance.

Examples of Ladder Dredgers.—The following particulars relate to the 'Corozal,' built in 1911 for the U.S. Government, for work on the Panama Canal:—

Dimensions.	Ft.
Length on load water-line	239
,, between perpendiculars	259
,, over all	268½
Breadth moulded	45
Depth moulded	19½
Draught fully loaded	15½

Propelling speed: 10 knots.

Dredging capacity: 1,200 cubic yds. of mud and sand per hour.

Buckets: Two sets; one of 54 cubic ft. capacity for soft material and one of 35 cubic ft. for stiff clay.

Maximum dredging depth: 50 ft.

The vessel is a hopper dredger, i.e. she discharges into herself instead of into an independent hopper ranged alongside.

The 'Silurus,' built by Messrs. W. Simons & Co., Ltd., Renfrew, is a twin-screw bucket hopper dredger of the following dimensions:

	Ft.
Length between perpendiculars	260
Breadth moulded	46
Depth moulded	20
Propelling speed	8½ knots.
Dredging capacity	2,200 tons per hour.
Buckets, two sets	one of 42 cu. ft. and one of 22 cu. ft.
Maximum dredging depth	50 ft.
Hopper capacity	1,500 tons.

Utility of Ladder Dredgers.—Although the serviceability of ladder dredgers does not extend to the removal of hard rock, yet they are capable of scraping away the surface of the softer varieties of sandstone and chalk, and, for this purpose, the bucket edges are sometimes fitted with spikes or tines. A ladder dredger works admirably in marl and stiff clay, and no difficulty even is found in dealing with heavy boulders which are frequently encountered in clays of glacial formation. The buckets of a first class dredger will readily lift stones weighing from 30 cwts. to 2 or even 3 tons apiece, bringing them up to the surface and within the range of a deck crane. As illustrative of the character of the material dealt with in many cases, the following particulars may be cited, relating to twelve months' work on the river Tees. A quantity of 417,963 cubic yds. was dredged at a cost of 22,432*l.*, equivalent to about 1*s.* 6¾*d.* per cubic yd. About two-thirds of this was hard, boulder clay and there were removed therefrom 1,651 stones, varying in weight from 5 cwts. to 4 tons apiece, of which 1,523 were raised by the dredgers unaided by divers; and, in addition, there were 19 tree trunks, ranging in length from 10 ft. to 20 ft., 16 tree roots and stumps, and 107 miscellaneous obstructions.

Dipper Dredgers.

Dipper Dredgers.—The dipper dredger is almost purely an American type, and has not yet been found in Europe. It is mainly employed on the great lakes of the Northern Continent of America, and on river beds and channels when the working depth does not generally exceed 20 ft. or so, and where space for manœuvring is somewhat restricted. This is accountable for by the fact that the dredger is not manipulated by chain moorings, as in the case of the ladder dredger, but is maintained in position by vertical spuds, or posts, which pass down through the vessel to the bottom of the channel, and are raised and lowered as occasion may require. The dipper arm itself assists in manœuvring, being allowed to remain on the bottom while the position of the spuds is changed.

Capacity of Dipper Dredgers.—The single bucket, or scoop, at the end of the lever arm has a capacity ranging from 1 to 12 or 15 cubic yds. The bucket is operated in a curved upward sweep by suitable gearing, on the same principle as a steam navvy or land excavator. The speed of a good dipper dredger in sheltered inland waters is from 30 to 40 seconds per dipper load, and on the sea coast from 40 to 50 seconds. For hard material, the bucket, as in the case of the ladder dredger, is fitted with teeth.

Examples of Dipper Dredgers.—The following particulars relate to a very large dredger, the 'Onondaga,' which worked in New York Harbour:—

Length of hull	140 ft.
Breadth „	50 ft.
Depth „	15 ft.
Capacity of dipper bucket	12 cubic yds.
Dredging depth	50 ft.

Size of dipper arm: 80 ft. long by 36 ins. square.
Spuds: Four timbers, 80 ft. long by 5 ft. square.
The dipper arm and spuds are both of Oregon fir.
Engines: Double cylinder, condensing.
Cylinders: 20·5 ins. diameter by 24 ins. stroke.

Another instance of a dipper dredger at Montreal Harbour:—

Length of hull	90 ft.
Width „	36 ft.
Depth „	10 ft.
Capacity of dipper bucket	7 cubic yds.
Dredging depth	40 ft.
Diameter of hoisting rope	2¼ ins.
Maximum pull on dipper	120,000 lbs.

Forward spuds: Single timbers (Douglas fir) 60 ft. by 42 ins. square.
Engines: Double cylinder (16 ins. by 18 ins. stroke) non-condensing.

Two dipper dredgers, the 'Gamboa' and the 'Paraiso,' each with buckets of 15 cub. yds. capacity, were built by the Bucyrus Co. in 1914 for work on the Panama Canal.

These dredgers have a displacement of about 1,500 tons, and the hulls, which are of steel, have an over-all length of 136 ft., a width of 44 ft., and a depth at bow of 15½ ft. and at stern of 13½ ft. Each dredger has two dipper buckets, the larger, of 15 cub. yds. capacity, for soft material, and the smaller, of 10 cub. yds. capacity, for harder material. The buckets, of ¾-in. plate, have lips of manganese steel, 2 ins. thick, which are 50 ins. deep at the centre, reduced to 9½ ins. at the back edges. They have no teeth. The handles are of Oregon fir, 72 ft. long.

The rate of working is said to average 6,000 cub. yds. per day each, in two 8-hour shifts.

A dipper dredger at work in Bombay Harbour has a bucket capacity of 6 cub. yds., dredges to a depth of 40 ft., and is capable of lifting boulders of over 12 tons weight.

Comparison of Dipper Dredger with Ladder Dredger.—Compared with the ladder dredger the dipper dredger is not quite so serviceable a machine for all-round purposes, and, for the more intractable kinds of material, it has been demonstrated that the former is more efficient than the latter. Thus, in excavating a channel in indurated clay in the St. Lawrence River, dipper dredgers of the most powerful design failed and were replaced by a ladder dredger which completed the work in a satisfactory manner.

The Grab or Grapple Dredger.

The Grab or Grapple Dredger has a bucket which consists of two or more curved plates, or jaws capable of opening and closing under the action of suitable gear. The grab is suspended from the jib head of a crane which does the requisite raising and lowering. The working is actually done very largely under the influence of gravity, the bucket being allowed to fall with open jaws under its own weight and to bury itself in the ground, where the jaws are closed and the bucket with its contents is then withdrawn. The principle of the grab is simple, but there are many complex adaptations in practice. Thus, the appliance may be worked by one or by two chains, each of which arrangements admits of several variations, so that there are quite a number of different machines on the market, each possessing special characteristics and advantages.

Comparison of Single and Double Chain Grabs.—The drawback of a double-chain grab is that it cannot be fitted to an ordinary crane; but, on the other hand, it has less intricate and less delicate mechanism, and, consequently, does not run the same risk of getting out of order. With two chains, moreover, there is less likelihood of the bucket being lost through fracture of the hauling gear. There is the further consideration that in many, if not most, single-chain grabs, the full height of lift has to be made before the grab can be opened in order to discharge its contents. Double-chain grabs can discharge at any stage of the lift. This advantage is shared by a few of the single-chain systems. Its importance lies in the fact that the grab might close upon some immovable object below water, and therefore necessitate the sending down of a diver, unless the jaws can be opened mechanically in that position.

Utility of Grab Dredgers.—The grab is not a suitable appliance for extensive operations. It is best adapted to dealing with material in confined and awkward situations inaccessible to other types of dredger. It is thus excellently suited for the removal of mud and silt in the vicinity of dock entrances and passages. Silt is generally of so impalpable a character that any disturbance of the water in which it lies causes it to rise in suspension, and renders it difficult of removal. The grab proves a most effective appliance for dealing with this soft material, entrapping it with the least admixture of water and the least alteration in consistency. Moreover, the grab is an extremely convenient implement for picking up miscellaneous articles which find their way to the bottom of docks; logs of hard wood, rails, coal, bales of hay and wool, barrels of nails and cement, hoop iron, etc.

Capacity of Grab Dredgers.—Grabs are constructed of a capacity up to 10 cubic yards, but for general purposes the most useful size lies between 70 and 120 cubic ft. They work very successfully in water up to 50 or 60 ft. in depth, at a speed of about one dip per minute, raising in this way from 150 to 250 cubic yds. per hour.

Examples of Grab Dredgers.—Several excellent examples of this class of dredger are constantly at work in the Liverpool docks. The grabs are mounted on barges of 1,100 to 1,400 tons hopper capacity. There are four 5-ton or 7-ton cranes of 16 ft. and 19 ft. radius respectively on each barge, each actuating a grab of 70 or 115 cubic ft. capacity. The principal dimensions of the vessels are:—

1,100 ton barges . . . . 200 ft. long by 35 ft. beam by 15½ ft. deep.
1,400 ton barges . . . . 225 ft. long by 38½ ft. beam by 16 ft. deep.

The speed of the barges is about 11 knots.

Bag and Spoon Dredging.—This notice of dredging appliances would be incomplete without some reference to a very rudimentary form of apparatus still practised in restricted areas and where operations are on a small scale. This apparatus consists of a strong leather bag of a capacity of from one-half to one cubic yard attached to a steel rim at the end of a stout spar, or pole, from 20 ft. to 35 ft. long. The bag is kept at the bottom by means of the pole, while it is drawn along the surface by a line attached to the rim. The operation is performed by manual labour, and about half a dozen men are required to man a barge for the purpose. The quantity raised and loaded into the barge will range from 60 to 100 cubic yds. per day. The appliance is really only suitable for dealing with mud in timber ponds and in canals.

Suction or Hydraulic Dredgers.

Suction or Hydraulic Dredgers.—Turning now to the second class of dredging plant, the suction or hydraulic type may be described as consisting essentially of a continuous pipe, or tube, through which material is drawn up by suction from the bottom and discharged either into a hopper or through a long pipe line to a depositing site some distance away. The first application of the principle was to the removal of sand—hence the name Sand Pump—but it has since had a much wider application. As regards sand, the method proved extremely effective from the first, the light granular nature of the substance lending itself admirably to easy incorporatio with the body of water flowing into the pump intake. Such difficulty as has been experience has been merely in regard to securing the deposition of the sand within the hopper, the tendency being for a large proportion to flow out over the sides with the escaping water. Considerable reduction in this unremunerative pumping has been made by giving the effluent stream as long a distance to travel as possible and providing the channel with adjustable coamings.

Sand Pump Dredgers.

Sand Pump Dredgers.—For the purposes of suction dredgers, centrifugal pumps are universally employed, and of these, those with a single inlet have an advantage over pumps with a double inlet, in that there is less concussion and interference with the flow. A double inlet pump, on the other hand, disposes of any question of balancing, which has to be met in a single inlet pump by thrust bearings. The most efficient peripheral speed is stated by the Mississippi River Commission to be about 50 ft. per second. The form of the nozzle of the intake pipe materially influences the percentage of sand raised which may be anything up to 50.

Examples of Sand Pump Dredgers.—The largest sand pump dredger in existence at the present day is the 'Leviathan' of the Mersey Docks and Harbour Board. Her dimensions are 465 ft. 9 ins. long between perpendiculars, by 69 ft. wide by 30 ft. 7 ins. deep moulded. She is fitted with four suction pumps, each of 42 ins. diameter, and is capable of filling herself with 10,000 tons of clean sand in 50 minutes from a maximum depth of 70 ft. The speed of the vessel is 10 knots. On arrival at the depositing ground, she discharges herself in 10 minutes through large cylinders extending from valve openings, 5 ft. 6 ins. diameter, in the bottom of each hopper to deck level. These valves have a lift of 4 ft. and are operated by hydraulic rams. At work on the bar and entrance channel of the Mersey, the Leviathan pumps up over 12 million tons of sand per annum.

The following particulars relate to the 'Coronation,' a vessel in the same service :—

Length between perpendiculars	332 ft.
Breadth moulded	52·7 ft.
Mean draught, loaded	17·2 ft.
Maximum dredging depth	65 ft.
Speed	10 knots.
Loaded capacity	3,500 tons.
Pumping capacity	4,500 tons per hour.
Suction pipes (two)	36 ins. diameter, 81 ft. long.
Propelling engines	2,000 i.h.p.
Pumping engines	800 i.h.p.
Cylinders of propelling engines	19 ins., 30 ins., and 49 ins. diameter by 30 ins. stroke.
Cylinders of pumping engines	12 ins., 20 ins., and 32 ins. diameter by 18 ins. stroke.

Suction Cutter Dredgers.

Suction Cutter Dredgers.—The success attending the introduction of the sand pump led to an investigation of the possibility of dealing with less tractable material by the same means. Water jets for loosening and disintegrating hard and compact strata were added, and some satisfactory results being obtained, the further step of introducing rotary cutters was taken. There was found to be little difficulty in treating in this way earth of a marly or brittle nature, easily resolvable into disconnected particles. Trouble, however, was experienced in dealing with clay and tenacious material, which clogged the cutters and choked the intake. Eventually this was overcome by the evolution of cutter blades of a suitable form and pitch.

Capacity of Suction Cutter Dredgers.—Suction cutter dredgers are now constructed capable of removing over 1,000 cub. yds. of stiff clay per hour. In material usually dealt with by bucket dredgers, the cutter suction and discharging dredger 'Lord Willingdon' at Cochin dredged and discharged, through 4,500 ft. of pipe line, 1,500 cub. yds. per hour over an extended period, and on occasion this output has been considerably exceeded.

Use of the Pipe Line.—The use of the pipe line for discharging purposes is becoming very general, especially in cases where it is practicable to use the dredged material for filling and embankment purposes. These lines are often of great length. A dredger working at Oakland, California, U.S.A., has successfully delivered material through 6,170 lineal ft. of 20-in. diameter pipe. Diameters reach as high as 42 ins. when altogether afloat, and 30 ins. to 36 ins. when partly ashore. In calm water thin sheet tubing will suffice, but in exposed situations long lengths of solidly built sections must be provided. The carrying capacity of a 36-in. pipe with 80 per cent. of solid matter conveyed at a rate of 10 ft. per second would be 7,400 cubic yds. per hour. Sand causes considerable wear and abrasion of the piping and requires a high percentage of water with greater velocity to keep it in suspension. Clay, from its unctuous nature, exercises far less frictional resistance and is sometimes rendered sufficiently fluid by 10 per cent. of water. A velocity of 7 ft. per second for clay and mud, and 10 ft. per second for sand, has been found to give good results.

Compound Dredgers.

Compound Dredgers.—A compound type of dredger has recently been introduced fitted with both the suction, or suction cutter, apparatus and a bucket ladder. This enables the dredger to be applied to a wide variety of material in the most effective manner. The bucket ladder is generally placed in the centre well and the suction apparatus at the sides of the vessel, but in some cases the suction pipe is also centrally situated.

Example of Compound Dredgers.—The dredger 'La Loire,' built for service on the river of that name, is a twin screw combined bucket and suction dredger with cutter gear and pipe line shore delivery. Her dimensions are 173 ft. 10 ins. in length by 32 ft. beam by 14 ft. deep. She is fitted with two centrifugal suction and force pumps, capable of raising material from a depth of 25 ft., and of driving it ashore through floating delivery pipes about 30 ft. long and 24 ins. diameter. The suction pipe is 26 ins. diameter and works, together with the bucket ladder, in a centre well.

The cutter has a diameter of 6 ft. 10 ins. and is driven at 12 revolutions per minute, the engine running at 140 revolutions.

The bucket ladder enables the dredger to work to a depth of 52 ft. 6 ins. The buckets have a capacity of 27½ cubic ft. They are driven at a speed of 16 buckets per minute with the engine running at 170 revolutions.

Both suction and bucket apparatus enable the dredger to excavate her own flotation in advance.

The machinery consists of two complete sets of direct acting, triple expansion surface condensing, reversible propelling engines, each arranged to drive a centrifugal pump as well as the bucket dredging gear. Each set of engines indicates about 450 h.p. when running at 170 revolutions per minute. The cylinders are 13 ins., 21 ins., and 33½ ins. by 18 ins. stroke.

The cutter gear is driven by means of belting from a separate engine of 200 h.p., which also serves for a centrifugal pump delivering spoil from the buckets into the upper part of the shoot.

For the purpose of working the bucket dredging apparatus the suction pipes and cutter gear are dismounted, but in deepening the bed of the river Loire the latter proved so effective that over 13 million cubic yds. of alluvial material have been removed without recourse to the bucket apparatus.

Hopper Service.

When dredged material has to be removed some distance to a depositing ground it is excavated by the dredger into its own hopper, or into hopper barges ranged alongside.

A combined hopper dredger costs less in initial expenditure and subsequent upkeep than a separate dredger and hopper of corresponding capacity, and it occupies less valuable space in a crowded waterway.

On the other hand, it involves the discontinuity of dredging operations and the frequent getting out and taking in of moorings. This latter consideration is of importance more to dredgers of the bucket ladder type.

On the whole, it is found expedient on undertakings of considerable magnitude, and in cases where the depositing ground is several miles distant from the dredging site, to maintain a fleet of hopper barges in attendance on bucket ladder dredgers. Suction dredgers, when they do not discharge through a pipeline, more commonly are compound vessels with self-contained hoppers.

Capacity of Hopper Barges.—Barges are constructed capable of receiving from 500 to 1,000 cubic yds. of dredged material (say 650 to 1,350 tons) and of making journeys to sea and back at speeds of 10 to 11 knots.

Example of Hopper Barge.—A typical barge of 800 tons capacity has a length of about 190 ft., breadth 30 ft. and depth 14 ft. The engines are triple expansion, surface condensing, with cylinders 16½ ins., 26 ins. and 43 ins. diameter by 27 ins. stroke, which at 110 revolutions per minute develop 820 i.h.p. and give a speed of 10 knots, steam being supplied at 180 lbs. per sq. in. from marine multitubular boilers.

Cost of Dredging Plant.*

The following figures may be useful as approximate indications of the cost of dredging plant which must, of course, vary from time to time with fluctuations in the price of material and the engagements of the shipbuilding yards.

Ladder Dredgers.—A 5 cubic ft. bucket, single ladder dredger, capable of raising 150 tons of soft material per hour from a depth of 40 ft., has cost in round figures 7,000*l.*

A 15 cubic ft. bucket, single ladder dredger, capable of raising 400 cubic yds. of sand, gravel and clay per hour from a depth of 45 ft., has cost about 27,000*l.* and was constructed in six months.

A one cubic yd. bucket, single ladder dredger, capable of raising 750 cubic yds. of ballast per hour from a depth of 55 ft., has cost about 43,000*l.*, and was built in nine months.

Dipper Dredgers.—A one to two yard dipper dredger costs from $8,000 to $12,000; say 1,600*l.* to 2,400*l.* The hull is of timber, about 60 ft. long by 25 ft. wide, with a draught of 3 ft.

A 3½ to 4 cubic yd. machine dredging to a depth of 30 to 35 ft., would cost about $80,000 to $90,000, say 16,000*l.* to 18,000*l.*

A 10 cubic yd. machine dredging to a depth of 35 ft. would cost about $130,000 to $140,000, say 26,000*l.* to 28,000*l.*

* See footnote, page 1456.

Grab Dredgers.—A twin-screw hopper dredger, with a hopper capacity of 1,200 tons, and fitted with 5 hydraulic grab cranes each of 5 tons bucket capacity, working to a depth of 47 ft., has cost about 35,000*l.*

Suction Dredgers.—A suction hopper dredger with a hopper capacity of 70,000 cubic ft., capable of raising 3,500 to 4,000 tons of sand per hour from a depth of 70 ft., has cost about 75,000*l.* A 2,000 ton suction hopper dredger would cost about 50,000*l.*

Compound Dredgers.—A sea-going hopper dredger, fitted with both ladder buckets (18 cubic ft.) and suction pumps, and capable of dredging by the former means 400 cubic yds. per hour, and by the latter means 650 cubic yds. per hour, from a depth of 50 ft., has cost about 53,000*l.*

Hopper Barges.—A steam hopper barge of 800 cubic yds. capacity, with a speed of 11 knots, costs from 18,000*l.* to 20,000*l.*

Cost of Dredging Operations.*

Conditions and circumstances vary with each locality, so that it is difficult to give reliable figures for general guidance. Sand pump dredging can be done for as little as 1*d.* per cubic yd. or even less, and grab dredging from 2*d.* to 6*d.*, but a very great deal depends on the distance the spoil has to be carried. In bucket dredging, either by ladder or dipper, the range of cost is considerable, depending in addition, as it does, on the kind of material dealt with. The following are a few collected records from various sources :—

BOMBAY HARBOUR AND DOCKS.

Total Maintenance Dredging during Five Years, 1920–1925.†

Costs exclusive of Interest and Depreciation on Dredging Plant.

Location.	Total Quantity Lifted : Cubic Yards.					Cost per Cubic Yard : Annas.				
	1920–1921	1921–1922	1922–1923	1923–1924	1924–1925	1920–1921	1921–1922	1922–1923	1923–1924	1924–1925
Alexandra Dock	24,210	297,500	20,380	144,800	388,000	9·23	7·78	8·01	5·78	6·62
Prince's Dock	50,500	263,500	68,500	17,700	474,000	7·60	6·81	7·30	4·75	5·50
Victoria Dock	159,500	114,000	48,900	344,000	49,000	6·22	6·18	6·87	5·88	10·60
Basins	206,800	131,400	3,335	—	9,330	7·78	12·26	11·60	—	9·10
Alexandra Dock Channels	650,000	149,600	615,000	591,000	1,170,000	4·40	8·56	5·76	4·97	3·60
P. & V. Docks Channels,	689,800	1,098,000	931,500	857,200	1,098,800	5·37	5·01	5·38	4·54	3·83
P. & V. Channels, S.E.Flank	40,700	27,500	—	182,800	82,600	6·83	3·61	—	4·17	3·66
P. & V. Docks Channels, N. Flank	276,500	72,500	184,800	316,000	400,000	4·98	6·18	4·67	4·61	3·44

Port Natal.—The following particulars (extracted from a paper by Mr. William Brown on 'Recent Progress in Dredging Machinery'; *vide* Min. Proc. Inst. C.E., vol. cciii.) relate to the performances of the four principal dredgers at Port Natal, Durban, within a period of several years ended December 31, 1914. The costs include all charges for working, maintenance, insurance, depreciation and interest on capital. The price of coal consumed by the dredgers was low, and seldom exceeded 8*s.* per ton. The three large vessels (Labrus, Cetus, and Nautilus) each employed fourteen Europeans, eighteen Kaffirs, and eleven Indians, or a total complement of

* See footnote, page 1456.

† Extract from paper on 'Maintenance Dredging in Bombay Harbour and Docks,' by E. L. Everatt, *Min. Proc. Inst. C.E.*

forty-three hands, at an annual cost of about 3,850*l.* per vessel. 'With the possible exception of Liverpool, no port hampered by sandbars has made such progress in comparatively recent years as Durban.'

PARTICULARS AND WORKING COST OF SEA-GOING PUMP DREDGING PLANT AT PORT NATAL.

Name of Dredger.	Year Built.	Tonnage.		Hoppr. Capacity (Tons of Sand).	Pumping Speed (Tons of Sand per hour).		Average Spoil handled per ann. (Tons).	Average number of days worked per ann.	Cost per ton handled. (*d.*)	Remarks.
		Net.	Gross.		Contract.	Record.				
Labrus .	1911	1,215	2,097	2,000	2,000	2,000	719,500	244	6·806	Worked exclusively in stiff clay, being fitted with patent clay cutter. All spoil dredged repumped ashore from ship's hoppers for land reclamation.
Cetus .	1905	1,327	2,414	3,000	3,000	7,430	2,151,000	232	2·227	Largely employed for bar work. Spoil mostly coarse loose sand and fine stone, all dumped at sea, two miles from harbour entrance.
Nautilus	1903	1,050	2,034	2,500	3,000	7,894	1,466,000	247	3·109	Employed chiefly on maintenance and deepening of harbour bed and channels. Material mostly fine sand, with occasional admixture of loose clay and mud, all deposited at sea two miles from harbour entrance.
Snipe .	1903	180	265	200	300	340	358,000	256	3·201	Used for clearing up wharf and quay berthage, cutting and maintaining boat channels, occasional reclamation, etc. Spoil deposited within harbour.

Panama Canal.—During 1910 the sea-going suction dredgers 'Culebra' and 'Caribbean' removed 6,083,902 cubic yds. at a cost of 3·7*d.* per cubic yd. The ladder dredger 'Marmot' removed 1,372,665 cubic yds. at 7·82*d.*, and dredger No. 5 removed 560,622 cubic yds. at 13·65*d.* per cubic yd. The small 5 cubic yd. dipper dredgers 'Cardenas' and 'Mindi' removed 1,013,485 cubic yds. at 27·89*d.* per cubic yd. and the 20-in. pipe-line suction dredgers, Nos. 82 and 85, dealt with 2,472,822 cubic yds. at a cost of 14*d.* per cubic yd. During a period of twelve months ended May, 1916, the dipper dredgers 'Gamboa' and 'Paraiso' each removed over 2,900,000 cubic yds., at costs varying from 2·25*d.* to 5·45*d.* per cubic yd., including operation and maintenance.

Mississippi River.—In 1907 three pipe-line suction-cutter dredgers dredged 1,279,000 cubic yds. at 4·12*d.* per cubic yd. including the cost of idle time. In 1908 the same dredgers accounted for 989,000 cubic yds. at 5¼*d.* per cubic yd.

New York Harbour.—In 1909, the sea-going suction dredgers 'Manhattan,' 'Atlantic,' 'Raritan,' and 'Navesink,' removed 10,786,638 cubic yds. at a total cost of $424,734 (say 85,000*l.*) or at the rate of 1·97*d.* per cubic yd. made up as follows: Dredging, 1·785*d.*; surveys and examinations, ·125*d.*; contingencies and supervision, ·06*d.* The 'Delaware' removed 1,675,562 cubic yds. at a total cost of 2·39*d.* per cubic yd.

River Thames.—The contract cost of removing hard material (ballast and clay) by ladder dredger from the Thames below London Bridge has ranged at various times before the war from 1*s.* 4*d.* to 2*s.* 6*d.* per cubic yd. largely dependent on the distance the material had to be conveyed. Some work has been carried out departmentally, including all establishment and capital charges, at a figure as low as 7½d. per cubic yd. This was for a quantity of about one million cubic yards, and the distance conveyed was about 30 miles.

River Mersey.—At the bar and in the entrance channels of the Mersey nearly 15 million cubic yds. of sand were removed annually at a pre-war cost of ·6*d.* per cubic yd. exclusive of interest and depreciation on the plant.

Siam Irrigation Canals.—The following particulars of the working expenses of a 2½-cu. yd. dipper dredger, employed on the Siamese Irrigation Service, are given by Lobnitz & Co., Ltd.:—

'During the year 1920, working nine hours per day, the total quantity dredged was 312,130 cu. yds., measured in place, or an average of 26,010 cu. yds. per month. The best month in that year was October, with 43,700 cu. yds. at a total cost of 1·95 pence per cu. yd., and the average cost per cu. yd. for the year was 2·87 pence, including labour, fuel, supplies, repairs, and every expense except interest and depreciation. The fuel used was oil, and represented about half the total operating cost. The quantity used was about one ton per day.

'While larger outputs have been made for short periods and under more favourable conditions, this record is notable chiefly for steady running for a long period without appreciable delays or expense on account of the machinery, and this in a tropical country remote from base of supplies. The cost of repairs was 4·3 per cent. of the total cost, and consisted mainly in the replacement of some wire ropes.'

SUBAQUEOUS ROCK REMOVAL.

The removal of hard rock under water is generally effected by one or other of two means: (*a*) by drilling and blasting; and (*b*) by the use of the chisel or rock breaker. In both cases a dredger is subsequently required for the purpose of clearing the débris from the site.

Drilling and Blasting.

Drilling and Blasting.—The first is the older method and remains still a most effective means of dealing with rock of adamantine hardness in huge solid masses, such, for instance, as reefs of granite and other igneous rock, and for the harder varieties of limestone.

The holes are drilled by suitable plant at distances apart dependent upon the nature of the rock and the quantity to be removed. In very extensive operations chambers may be formed for the reception of powerful charges.

Two classes of drill are generally available for boring purposes: (*a*) diamond drills and (*b*) jumper drills. The former are hollow rods, fitted with a head or crown of diamonds, costing from 50*l.* to 60*l.*, and driven at from 200 to 300 revolutions per minute. The latter are bars or shafts of steel, furnished with bits, preferably of circular form, which are worked by percussion at the rate of from 20 to 30 blows per minute.

The blasting material is usually dynamite, or any of the numerous compounds of nitro-glycerine to be found on the market.

The following are brief notes on the methods employed in several instances.

Blyth Harbour.—The rock at Blyth harbour is sandstone of the hardness of basalt in places, and the work was done by six drills lifted by steam power and guided by hand. The distance between the shot-holes was 5 ft. in one direction and 6 ft. 2 ins. in the other. The blasting material, bellite, was lowered in canisters through the drilling tubes and fired by fuses and detonators, the holes being tamped with small gravel. The cost of drilling and blasting came to 3*s.* per cubic yd. of rock *in situ*, adding to which 2*s.* 6*d.*, the cost of dredging, and 8·2*d.* for interest and depreciation on the dredger, the total amounts to 6*s.* 2·2*d.* per cubic yd. The average quantity of rock drilled and blasted per week by one barge was 488 cubic yards. The system was afterwards abandoned in favour of the rock-cutter.

Black Rock Harbour, Buffalo, U.S.A.—Here the rock formation is a very hard limestone containing veins of flint. The blasting material was dynamite and, for a quantity of 14,450 cubic yds., the holes were drilled to an average depth of 9 ft. 9 ins. The cost of drilling and blasting only, including an allowance for interest and depreciation of 2 per cent. per month on 8,000*l.* (the value of the plant), came to 2*s.* per cubic yd. On the other hand, a smaller quantity of 333 cubic yds. with holes drilled only to 3 ft. 6 ins. deep cost 23*s.* 2*d.* per cubic yd.

The drill boat employed on this work was a steel hull, 137 ft. by 33 ft., divided into 20 water-tight compartments. It was equipped with five drill frames located along one of the gunwales each with a travel of 25 ft. on longitudinal tracks. Each frame was fitted with an Ingersoll-Rand drill 2½ ins. in diameter, with 4-in. bits, having a vertical range of 21 ft. without changing the drill bars. Each frame was also provided with an engine for hoisting the drills, two davits for handling drill steels and charging pipes and a pipe which was driven, whenever required, through loose material to the rock to prevent the drill holes from clogging. A 250 h.p. boiler supplied steam to the drills, spud engines and capstans.

Detroit River.—On three sections of the Detroit River, where the blasting material was dynamite and the holes ranged from 5 ft. to 8 ft. in average depth, the total cost is stated to have been 3*s.* 4*d.*, 4*s.* 4*d.*, and 4*s.* 1*d.* per cubic yd., respectively, of useful material, or about half the rates for the whole quantity removed. The holes were spaced 5 ft. apart, and the total number drilled was 94,691 at a cost of about 52,000*l.* It is not apparent that this figure includes any allowance for interest and depreciation.

Fremantle Harbour, Western Australia.—On account of the shallowness of the water, the work of drilling at Fremantle Harbour could be carried on from light wooden stages resting on four-footed trestles. In a couple of days it was possible to erect 20,000 square ft. of this staging, which carried 120 to 160 men drilling with regularity in 20 ft. or more of water. The holes were from 8 ft. to 12 ft. apart, and the charges consisted of 12 to 15 lbs. of dynamite or gelignite. One ton of explosive accounted for 5,600 cubic yds. of rock blasted. The cost of drilling and blasting 1,503,099 cubic yds. was 238,345*l.*, equivalent to 3*s.* 2*d.* per cubic yd.

Rock Breakers or Chisels.

Rock breakers, or chisels, are steel rams weighing from 15 to 20 tons, with pointed ends of hard chrome steel which are detachable for renewal. The appliance is suspended from a frame, or tripod, mounted on a barge, and is worked by a winch. A drop of from 6 ft. to 10 ft. is generally sufficient to produce disintegration. Blows can be repeated with advantage until a depth of 3 ft. is reached, with holes spaced about 3 ft. to 4 ft. apart (about eight blows will give the required penetration), when it becomes desirable to dredge the broken material before proceeding further. A net depth of 2½ ft. over the site will probably thereby be obtained. It is found that the average result in hard rock is 2 cubic ft. per blow and that about 150 blows can be given per hour. This is equivalent in uninterrupted working to about 10 cubic yds. per hour for a single-cutter machine; about 50 per cent. more will be achieved by a double cutter. In moderately hard rock the output will be greater, as about 250 to 300 blows can be given per minute. About 200 cubic yds. may be reckoned a good day's work.

There are several types of cutter, but perhaps the Lobnitz rockbreaker is that which is best known. The following are some instances of its employment. It may be added that some of the latest examples of this type of cutter have been fitted with an under-water sleeve, to afford more accurate guidance for the ram or chisel.

Suez Canal.—On the Suez Canal, where the rock consisted of limestone, calcareous agglomerates and tufa, gypsum and alabaster, 132 blows were delivered per hour of effective working, or 83 blows per hour of whole time, allowing for interruptions from passing ships and stoppages for repairs. The average thickness of rock shattered was 2 ft. 7½ ins., and each cubic yd. of rock required a mean of 5·4 blows. The cost, exclusive of dredging but covering everything else except general charges, varied from 2½*d.* to 7*s.* 6½*d.* per cubic yd.

Blyth Harbour.—At Blyth in hard sandstone of the coal measures, a ram of 15 tons weight, falling through a distance of 8 ft., penetrated the rock to a depth of 3 ft. on an average of 8 or 9 blows. The holes were spaced 4½ ft. apart. The plant comprised two Lobnitz rock-breakers, each consisting of a steel barge carrying sheer legs, from which the rams, 40 ft. to 50 ft. long and 17 ins. to 19 ins. diameter, were suspended. The cost of one rockbreaker was about 6,800*l.* The quantity broken per week by a single machine working night and day was 908 cubic yds., at a cost (including wages, coal, stores, repairs and insurance) of 46*l.* 11*s.* 3*d.*, equivalent to 12·3*d.* per cubic yd. Adding 6½ per cent. for interest and depreciation on plant, the total cost was 14·5*d.* per cubic yd. The cost of subsequently dredging the loose material (including capital charges) was 2*s.* 9·1*d.* per cubic yd., making 3*s.* 11·6*d.* per cubic yd. in all.

Black Rock Harbour, Buffalo, U.S.A.—The cost of working two Lobnitz machines at Black Rock Harbour in hard limestone rock was 2*s.* 9½*d.* per cubic yd. for a quantity of 52,500 cubic yds. This does not cover the cost of dredging.

Panama Canal.—On the Panama Canal, a Lobnitz cutter worked mounted on a barge 100 ft. long by 28 ft. beam by 8 ft. deep. The extreme variation in the tides necessitated the use of 3 rams, weighing 15, 16 and 19½ tons respectively. The holes were spaced 4 ft. apart, and 3 ft. of penetration was made in each case. From records during 1910–11 the work done was an average of 16·2 cubic yds. per hour of actual working time, or 11 cubic yds. per hour of total time, at an average cost of 3*s.* 4*d.* per cubic yd.

Value of the method of Drilling and Blasting as Compared with the use of Rock Breaker.—While drilling and blasting is most commonly adopted for hard rock in large masses, it does not necessarily follow that the cutter always gives better results in soft rock. In some cases it is found that the cutter has a tendency to stick in the hole which it makes, and for that reason the extent of penetration should be carefully limited. On the other hand, the cutter offers less interruption to the passage of ships close to the area of operations, and it can be used in positions where blasting is inadmissible. Moreover, the rock is broken to a more uniform and more convenient size, which facilitates the action of dredging. The cost of the work done by the cutter is generally, though not invariably, less than the cost by the alternative method.

CHANNEL DEMARCATION.

Entrance channels to ports are outlined by buoys and important changes of direction are also indicated by beacons and lightships.

Uniform System of Buoyage.

The general Lighthouse Authorities of the United Kingdom have adopted the following uniform system of buoyage.

Conical Buoys (those with pointed tops) denote the starboard hand, or that side which would be on the right hand of a mariner, either going with the main stream of flood tide, or entering a harbour, river or estuary from seaward.

Can Buoys (those with flat tops) denote the port hand, being the left-hand side of the mariner under the circumstances described.

Spherical Buoys, with domed tops, mark the ends of middle grounds.

Pillar Buoys, having a tall central structure on a broad base, mark special positions.

Colour.—Starboard-hand buoys are always to be painted in one colour only; port-hand buoys are to be painted another characteristic colour, either single or particolour. Spherical buoys are to be distinguished by white horizontal stripes. Wreck buoys are to be coloured green with the word 'wreck' in white letters across.

Types of Buoys.

Buoy Construction.—Buoys are generally hollow structures of steel or iron plating, at least ¼ in. thick. They should be constructed in two watertight sections to prevent sinking in case of penetration of one part.

The size and weight of buoys is as follows:—

		Diam. Ft.	Weight. Tons.
Day Mark Buoys	First class	12	4
	Second class	10	2½
	Third class	8	1¾
	Fourth class	6	¾
	Fifth class	5	⅔
Gas Buoys		10	4

Gas buoys, in which gas is stored under pressure, should be made of welded iron or mild steel, not riveted. Under high pressures the riveted joints tend to leak and are not satisfactory.

Buoy Moorings.—Mooring chains for first-class buoys are about 1¼ ins. diameter, and they are attached to sinkers weighing about 50 cwts. For second-class buoys the chains are usually 1¼ ins. diameter, and the sinkers weigh 40 cwts.

Bell Buoys are provided with a bell from 3 to 7 cwts. in weight struck under the action of the sea by pendant clappers, or by a hammer actuated by carbonic acid gas.

Whistling Buoys emit a sound by means of air imprisoned in a long central tube during the period of rising in a swell and expelled through a whistle during the period of falling. The tube descends into the water to a considerable depth—about 20 ft.

Luminous Buoys.

Luminous Buoys are lighted by means of either oil gas, Blau gas, or acetylene. On account of the high pressure required for compact storage, coal gas is unsuitable for use as a naked flame, the luminosity of the gas being rapidly reduced by pressure. The heating power, however, remains almost intact, and with a mantle satisfactory incandescence is produced, though the brilliancy and luminous intensity are lower than those of the other agencies. The ratio of calorific value is approximately as follows :—Coal gas, 5 ; oil gas, 10 ; Blau gas and acetylene, 14.

Oil gas, as manufactured on the *Pintsch* system, is produced at a cost of about 30*s.* per 1,000 cubic feet, about 75 cubic feet of gas being obtained from one gallon of oil. The average burner of a fixed light consumes about 0·75 cubic feet per hour. The gas is stored under pressure, generally of about 90 to 105 lbs. per sq. in., but reaching up to 150 lbs. A reservoir of 380 cubic feet in a buoy of 10 ft. diameter, charged to 30 lbs. pressure, would hold a sufficient supply of gas for about four months.

Lenses are usually of 200 to 300 millimetres (8 to 12 ins.) diameter. The candle power of lanterns fitted with incandescent burners and lenses of 200 mm. diameter is about 220, and with 300 mm. lenses about 330.

Blau gas is derived from the decomposition at a lower temperature of the raw or residuary products from which ordinary oil gas is extracted, and is richer than the latter in heavy carbides of hydrogen. It is liquefied for storage in bottles. Before use, the gas requires to be expanded in an intermediate receiver. The cost of production of Blau gas is considerably higher than that of oil gas, but the cost of transport is much less, and the increased heat of the flame gives an increase of 30 per cent. over oil gas to the intrinsic brilliancy of the mantle and the intensity of the lens.

Acetylene, a gaseous compound of hydrogen and carbon, is produced by the access of water to calcium carbide. In the case of buoys, the water is generally admitted to the chamber containing the carbide through a hole in the bottom, the generation of gas being regulated by the pressure arising from its formation. The system is wasteful of carbide, and only suitable for positions where the buoys can be periodically cleaned and recharged. The better principle, wherever feasible, is the inverse arrangement of the application of carbide to water.

From 1 lb. of commercial calcium carbide can be produced about 4½ cubic feet of acetylene, at a pre-war cost of about 1½*d.* for material only.

Acetylene gas produced as above is not chemically pure, and gives rise to deposit on the burners, but this difficulty can be overcome by the addition of a suitable purifier between the gas-generating chamber and the burner. Unattended lights working on this system will run for six months, or even a year, without recharging.

Dissolved Acetylene.—The compression of acetylene for storage purposes is generally illegal, on account of the liability of the gas, so stored, to explode. On the other hand, compressed acetylene in what is known as the dissolved form is perfectly safe, and cylinders of dissolved acetylene, prepared in accordance with official regulations, are classed by the Home Office as non-explosive. Acetylene in this form is now most commonly used both for buoys and unattended lights, because of its convenient storage and transportation.

The solvent commonly employed is acetone, which is a liquid hydro-carbon, possessing the property of absorbing about 25 times its own volume of acetylene at atmospheric pressure, and multiples of this to correspond with the number of atmospheres to which compression is carried.

The steel cylinders for the reception of the dissolved acetylene are first completely filled with a substance having a porosity of 80 per cent., which is then caused to absorb as much acetone as would occupy 40 per cent. of the volume of the cylinder. The actual gas capacity of the cylinder is thus 40 per cent. of 25, or 10 volumes per atmosphere of pressure.

Dissolved acetylene cylinders should not be discharged below the atmosphere gauge pressure, because the rate at which the acetone is carried off by the gas greatly increases as the pressure approaches zero. Acetone vapour has no detrimental effect upon the consumption of the gas in the burner. It is generally found necessary to make up the quantity of acetone, after five to ten refillings of the cylinder with acetylene.

Economies in the consumption of dissolved acetylene have been effected by the introduction of mechanism for emitting flashing or occulting lights invented by Mr. Gustav Dalen and others whereby a wide range of intermittent light signals can be produced automatically.

To the same scientist is attributable the *Sun-valve*, or, more correctly, the *Light-valve*, which is a mechanical device sensitive to light and actuated entirely by the alternation of daylight and darkness. It opens the gas supply at nightfall and closes it at daybreak with perfect regularity, resulting in a saving of some 25 to 40 per cent. of the gas, according to climate, as compared with a constantly burning light. The use of the instrument is generally restricted to shore lights, or lights of considerable magnitude.

Mr. Dalen has further produced complete automatic apparatus for mixing in correct and constant proportions the dissolved acetylene and air and for burning the compound in conjunction with incandescent mantles, either as a constant or intermittent light. This apparatus, by reason of its nature, is more applicable to lights of considerable power, superior to those generally fitted to buoys.

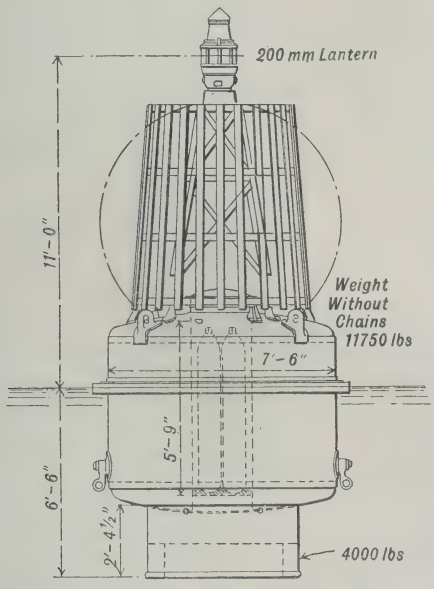


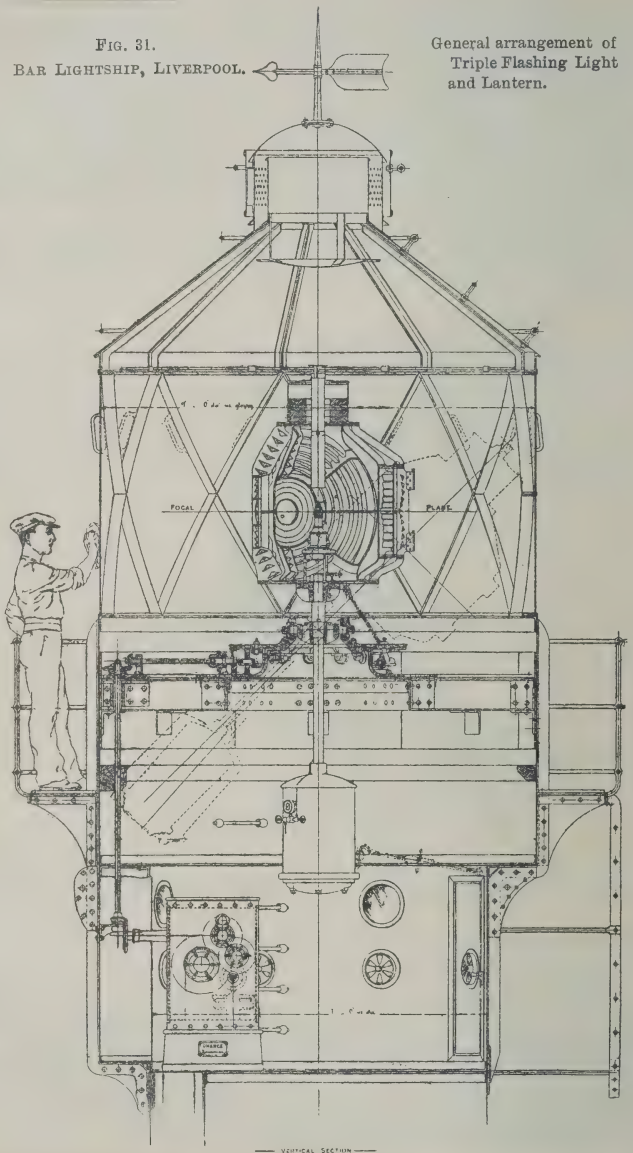
FIG. 30.—Luminous Buoy.

The elevation of a luminous buoy given in fig. 30 above (the Gas Accumulator Co. (United Kingdom), Ltd.) has been selected, as an example of buoy design with an unusually varied range of application. It has been designed primarily for use as a spare or wreck-marking buoy.

The lantern, which can be of any size up to that carrying a 300-mm. lens, is mounted upon a trestle superstructure. In this way it is possible to place any one of the three conventional types of daymark in position, namely, can, cone, or spherical, without in any way interfering with the light apparatus. The lantern carries a special type of flashing mechanism which is so designed that any one of three characters can be exhibited at will by simply turning a valve on the burner stem. The characters are single, double, or triple flashing.

FIG. 31.
BAR LIGHTSHIP, LIVERPOOL.

General arrangement of Triple Flashing Light and Lantern.



Two accumulators are the minimum number with which the buoy may be equipped and retain its calculated stability, but, if necessary, the number can be increased to four, and the unattended service period can be as much as twelve months.

The candle-power depends, of course, upon the size of lens and burner used. With a 300-mm. lantern and 15-litre burner this would amount to 180 s.c.p. The coloration of the light can be varied at will by the insertion of suitably tinted glass shades inside the lens and surrounding the flame.

Further reference to the use of acetylene lighting will be found on page 1497.

Lightships.

Lightships afford a more powerful light than luminous buoys, and the focal plane is at a much higher level—generally about 35 ft. above the water surface. They are, of course, more costly both in first outlay and maintenance, and are usually manned by a crew, though this is not essential in all cases.

A notable example of an attended lightship is the Bar at the entrance to the River Mersey. The illuminant in this vessel is petroleum with an incandescent mantle, and the lens is of small third order (750 mm. diameter). The whole optical apparatus is swung on gymbals, so that the beam emitted is kept practically horizontal. It is revolved by clockwork, and gives three powerful flashes, so that the whole installation resembles a floating lighthouse. Fig. 31.

Small lightships, without constant attendance, are stationed at several places on the coast of this country and of France. A vessel of this type is about 40 ft. long, 12 ft. beam, and 8 ft. deep, fitted with a storeholder containing 2,500 cubic ft. of gas, which is sufficient to maintain a light for 3 months. Many unattended light vessels in all parts of the world are equipped with dissolved acetylene lights and run without recharging for six or twelve months.

Larger vessels, with crews, range from 60 to 150 ft. in length, from 20 to 30 ft. in breadth, and from 10 to 20 ft. in depth, the draught being from 7 to 10 ft.

Lenses vary from 500 mm. to 750 mm. in diameter and the candle-power from 2,000 to 250,000. Catoptric reflectors with a candle-power of 4,000 are used in the older types of light vessels.

LIGHTHOUSES.

CLASSIFICATION OF LIGHTHOUSE APPARATUS.

Order.	Focal distance, mm.	Order.	Focal distance, mm.
Hyper-radial . . .	1,330	Small third order . .	375
Meso-radial . . .	1,125	Fourth order . . .	250
First order . . .	920	Fifth „ . . .	187·5
Second „ . . .	700	Sixth „ . . .	150
Third „ . . .	500		

THE EDDYSTONE LIGHTHOUSE.

The most recent type of foundations and general construction of lighthouses is well exemplified by the new Eddystone Lighthouse. In this structure the tendency of the curvilinear outline of Smeaton's and of other similar sea-towers that have succeeded it, to elevate the centre of force of each wave-stroke on the structure, induced Sir James Douglas to adopt a cylindrical base, from which base, at a level of 2½ feet above high-water spring-tides, the curved shaft of the tower commences. The difference in the rise of heavy seas on the two structures during stormy weather is remarkable. The cylindrical base has the further advantage of affording a convenient landing platform, thus adding considerably to the opportunities of relieving the lighthouse.

The base is 44 feet in diameter by 22 feet in height. The tower is a concave elliptic frustum, the generating curve having a semi-transverse axis of 173 feet, and a semi-conjugate axis of 37 feet. With the exception of the space occupied by the fresh-water tanks, the tower is solid for 25 ft. 6 ins. above high-water spring-tide level. At the top of the solid portion the wall is 8 ft. 6 ins. in thickness, diminishing to 2 ft. 3 ins. in the thinnest part of the service-room. All the stones are dovetailed both horizontally and vertically. The system consists in having a raised dovetailed band 3 inches in height on the top bed and on end joint of each stone. A corresponding dovetailed recess is cut in the bottom bed and end joint of the adjoining stones, with just sufficient clearance for the raised band to enter it freely in setting. From experiments made with blocks put together in this manner with Portland cement, it is found that the work is so homogeneous as to be as nearly as possible as strong as the solid granite.

Each stone of the foundation-courses (figs. 32, 33) is sunk to a depth of not less, at any part, than 1 foot below the surface of the surrounding rock, and is further secured by two Muntz metal bolts, 1½ inch in diameter, passing through the stone, and 9 inches into the rock below, the top and bottom of each bolt being fox-wedged.

The masonry consists of 2,171 stones, containing 62,133 cubic feet of granite, or 4,688 tons.

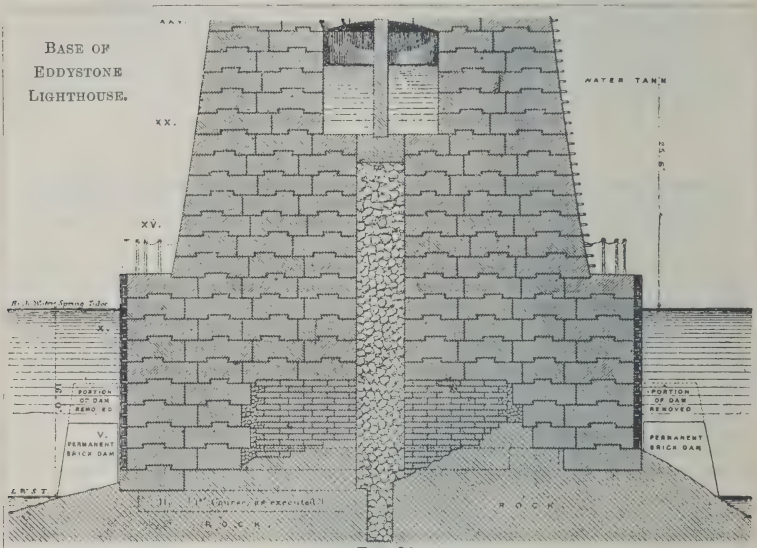

BASE OF EDDYSTONE LIGHTHOUSE.
A A.
XX.
XV.
WATER TANK
PORTION OF DAM REMOVED
PERMANENT BRICK DAM
ROCK

FIG. 32.

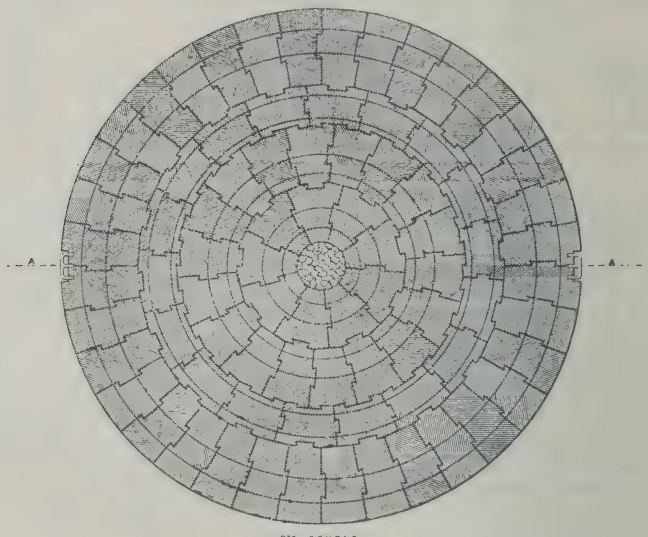

A
A
BIT COURSE

FIG 33.

Cost of Construction of Various Lighthouses.

Name of Structure.	Total Cost.	Cubic Feet.	Cost per Cubic Foot.
	£ s. d.		£ s. d.
Eddystone (Smeaton)	40,000 0 0	13,343	2 19 11½
Bell Rock	55,619 12 1	28,530	1 19 0
Skerryvore	72,200 11 6	58,580	1 4 7¾
Bishop Rock	34,559 18 9	35,209	0 19 7¾
Smalls	50,124 11 8	46,386	1 1 7¼
Hanois	25,296 0 0	24,542	1 0 7¼
Wolf Rock	62,726 0 0	59,070	1 1 3
Dhu Heartach	72,584 9 7	42,050	1 14 6
Longships	43,869 8 11	47,610	0 18 5
Eddystone (New)	59,255 0 0	65,198	0 18 2

Different Descriptions of Lights.

(1) Fixed. (2) Revolving light, gradually increasing at definite equal periods to full power, and then gradually decreasing to eclipse. (3) Fixed light varied by flashes; shows a fixedlight, which at certain periods is varied by white or coloured flashes. (4) Flashing; shows a single flash at short intervals of a few seconds. (5) Group-flashing; shows groups of two or more flashes in quick succession, separated by a period of eclipse between the groups. (6) Occulting, which bursts instantaneously into full power, and after remaining for a period as a fixed light is suddenly eclipsed. (7) Alternating lights of different colours, generally white and red alternately.

Distance visible from Lighthouses.

The following Table shows the distance of the horizon visible from towers of the heights given. In it the necessary correction has been made for refraction :—

H. Heights in Feet.	λ. Lengths in English Miles.	H. Heights in Feet.	λ. Lengths in English Miles.	H. Heights in Feet.	λ. Lengths in English Miles.
5	2·958	80	11·832	250	20·916
20	5·916	90	12·549	300	22·912
30	7·245	100	13·228	350	24·748
40	8·366	120	14·490	400	26·457
50	9·354	140	15·652	600	32·403
60	10·246	150	17·201	800	37·416
70	11·067	200	18·708	1,000	41·833

Optical Apparatus.

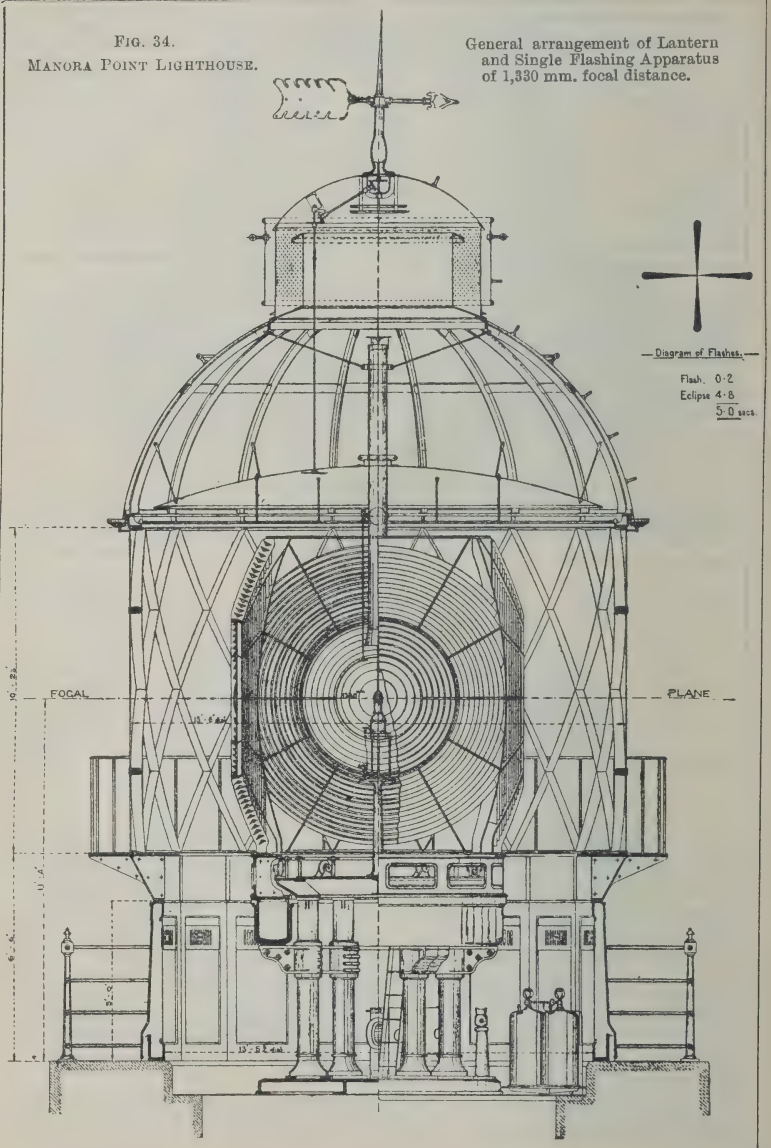
The essential part of a lighthouse is the optical apparatus, which is formed of glass lenses and reflecting prisms in order to concentrate the light from the burner in the required direction. The optical apparatus of a fixed light concentrates the rays in the vertical plane only, and sends out a zone of light which illuminates the horizon over the whole 360° or any less angle which may be required. That of a flashing light concentrates the rays in both the vertical and horizontal planes, and thus throws out a series of beams, the number varying with the character which the light is required to give. The whole apparatus is revolved, and by this means the beams sweep round the horizon and appear as flashes at definite intervals to the mariner.

Large lights are supported on a mercury float whereby friction of movement is reduced to a minimum; ball bearings are satisfactory for lights of the fourth order and upwards. The adoption of the mercury float, and of the group flashing system, has revolutionised lighthouse practice. The former enables the largest type of optical apparatus, the weight of which runs up to 10 tons, such as that illustrated in fig. 34 (p. 1496), to be rotated by the little finger only, and the latter has so greatly increased the distinctive characters which can be given to lights that there is now no danger of confusion between them.

It is usual to arrange the optical system so that the flashes have a minimum duration of 0·2 second; it has been found that a shorter duration is not sufficient for the mariner to detect the flash. Fig. 34 also illustrates the lantern which protects the optical apparatus.

FIG. 34.
MANORA POINT LIGHTHOUSE.

General arrangement of Lantern and Single Flashing Apparatus of 1,330 mm. focal distance.



Sources of Lighthouse Illumination.

For the ordinary necessities of lighthouse illumination, where an electric supply is not available, mineral oil is the most suitable and economical illuminant.

Oil and gas lights, when shown through similar lenses, are equally affected by atmospheric variations. The electric light is absorbed more largely by haze and fog, but in all weathers and at all distances its penetration has proved superior.

The luminary now in general use at lighthouses is the incandescent oil burner, to which paraffin oil is supplied under pressure, and is vaporised in a retort beneath the burner, being afterwards consumed in an incandescent mantle. The sizes of mantle usually employed vary from 35 mm. to 110 mm. in diameter with intensities up to 3,500 candles.

The latest burner of this type is the 'Hood' petroleum vapour burner designed by the late engineer-in-chief to the Trinity House, in which the mantle employed is of the autoform type, with an intrinsic brightness of 325 candles per sq. in. The burner is in four sizes, 35 mm., 50 mm., 75 mm., and 100 mm., with total intensities of 660, 1,150, 2,200, and 3,300 candles, and consumptions of 0·75, 1·25, 2·25, and 3·25 pints of oil per hour, respectively.

The substitution of incandescent oil burners for concentric wick burners has not only greatly augmented the intensity of the lights, but has resulted in a considerable economy in consumption, in addition to obviating excessive labour and expense on landing large quantities of oil at lighthouses, especially at rocks and isolated positions. The following may be cited as an example of the increased intensity and economy which are attained by the use of this luminary. At the Eddystone, which is a biform light having superimposed lenses and burners, the resultant beam with the former concentric six-wick burners was 79,250 candles, the consumption of oil being 2,500 gallons per annum; whilst with incandescent oil burners the resultant beam is nearly 600,000 candles and the consumption 1,200 gallons per annum.

The development in high-powered electric incandescent filament lamps has made possible their use for lighthouse illumination. Lamps up to 10 kw. are now employed and, combined with automatic apparatus, effect economy in maintenance by the reduction of personnel.

For minor lighthouses and portlights (fig. 35, p. 1498) at the entrances to harbours, etc., the illuminant is sometimes a capillary wick lamp having one or more wicks, which can be left without attention for a single night. A distinctive character is given to such lights by means of revolving screens driven by a simple clockwork mechanism which will run for a night without rewinding; the screen occults the light at regular intervals. For the entrance to harbours where range is not a great consideration the light may be coloured red or green.

Many lights are now left unattended for long periods, and the illuminant is generally compressed oil-gas, stored in receivers, or acetylene gas, manufactured in generators on the spot or on the dissolved acetylene principle or electricity.

Ras Serani Lighthouse, Mombasa.

The following description of the modernised lighting equipment has been furnished by the Gas Accumulator Co. (United Kingdom), Ltd:—

The lighthouse was originally equipped with an oil-burning, clock-driven rotating lens installation attended by light keepers.

In 1928 it was coverted by the company, the light source being a 500 watt electric lamp mounted in a catadioptric section of a drum lens having an arc of illumination of 180°. The focal distance of the lens is 300 mm.

Electric current is derived from the town lighting mains, and is at a pressure of 240 volts, the supply being 50 periods, single-phase.

In order to render the apparatus automatic in operation it became necessary to make provision to cover lamp filament failures and also supply current failures. This problem was overcome by designing the mechanism as shown in fig. 36. Two lamps are employed, one being in the focal point of the lens, the other being carried on an arm at 120° to it; a third arm, again at 120°, carries an acetylene burner, the whole being arranged to rotate about a vertical axis. Should the filament of the lamp in service break, the cessation of the flow of filament current is utilised to operate a release gear, whereby the second bulb is brought into the focus and the defective one removed. As the new lamp comes into position, it lights up and thereby stops the action of the changing mechanism.

Should a failure of the electric supply occur due to a breakdown at the power station or for any other reason, the effect is to remove one or both electric lamps from the focus and replace them by the acetylene burner. The renewal of the electric supply would then have the effect of replacing an electric lamp at the focus.

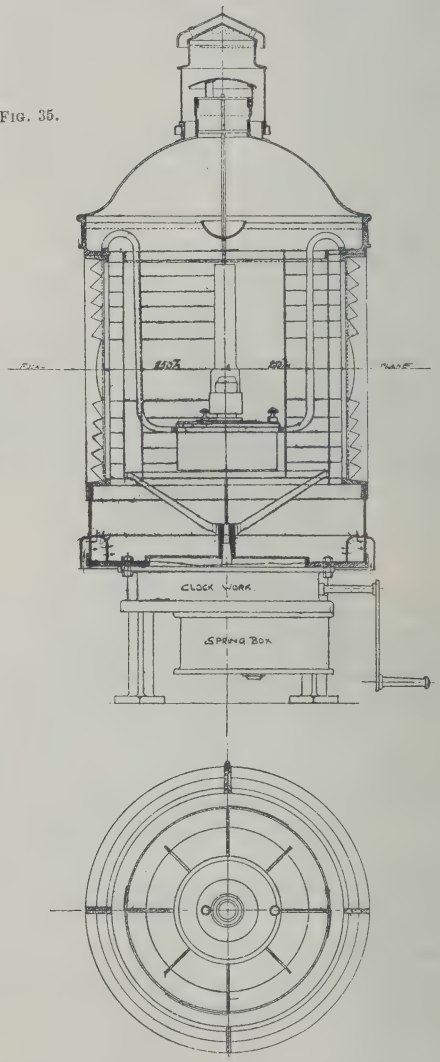
The original character of the old light of 1-in. flash plus 4-in. eclipse has been retained in the new apparatus, both electric and acetylene burners being arranged to emit a flashing light of this character. The lighting and extinguishing of the light is under the control of a Venner clock mechanism.

The candle-power developed with an electric lamp at the focus amounts to 22,000 s.c.p., and with a focal height of 85 ft. above sea-level, the light has a geographical range of 18 nautical miles in clear weather, taking the average height of a ship's bridge as 40 ft.

The dissolved acetylene gas for supplying the standby light is provided by two gas accumulators located in the base of the tower. These supply an open type burner composed of four twin jet burners each of 25-litre capacity. The light is under the care of a member of the port staff, whose duty it is to visit the light for inspection purposes at fortnightly intervals.

FOURTH ORDER OCCULTING PORTLIGHT.

FIG. 35.



RAS SERANI LIGHTHOUSE, MOMBASA.

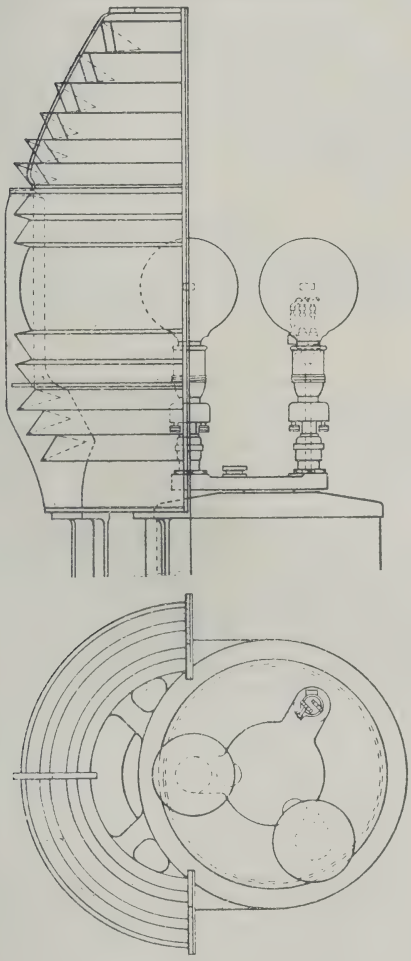


FIG. 36.—Modernised Lighting Equipment for alternative supplies of electricity or acetylene.

Relative Absorptive Powers of Coloured Glass.

With the best red glass the ratio between the powers of the red and white lights is as 2 to 5.

When colour is necessary in fixed dioptric apparatus the coloured beam can be strengthened by condensing prisms, whilst for revolving dioptric apparatus the red and white can be approximately equalised by giving to the red beam a greater breadth and a special arrangement of the lens and prisms.

During thick weather red lights are not so much absorbed by the atmosphere as white.

Glazing.

The glass of lighthouses is usually $\frac{3}{8}$ in. thick, except in peculiarly exposed situations, where it is $\frac{1}{2}$ inch.

The astragals are 1 in. in section, of gun-metal, having a tensile strength of 33,000 lbs. per square inch.

Diagonal astragals do not intercept light in any one azimuth throughout their whole height, and they secure great rigidity and strength.

Ordinary first-order lanterns are 12 ft. in diameter and 10 ft. high in the glazing.

The Bishop Lighthouse has a lantern of 14 ft. diameter and 15 ft. high; Fair Island, 14 ft. diameter, 6 ft. 2 ins. high; Tory Island, 14 ft. 9 ins. diameter, 18 ft. high; Sule Skerry, 16 ft. diameter, 12 ft. 2 ins. high.

Lightning Conductors.

These should be of $\frac{3}{4}$-in. best quality copper rod, carried 18 ins. above highest part of ventilator of lantern, and then terminated in three branches with platinum caps or platinum wire No. 8 B.W.G., screwed and soldered into the rod; the latter should be put into metallic connection with all internal masses of metal. The earth connections should be made by copper plates, 20 ins. by 12 ins., and $\frac{1}{4}$ in. thick. Conductors should be periodically inspected.

Audible Signals.

The types of sound signals in use by the Trinity House are:—(1) aerial and submarine bells; (2) explosive signals; (3) reeds; (4) sirens; (5) diaphones; and (6) electric oscillators.

Of the first of these, **Aerial Bells**, actuated by the motion of the ship or buoy carrying them, are well known, but a new method of operating the striking mechanism has been recently installed at the unattended lighthouse on the Menai Straits and on the Lune Deep unattended gas boat.

The bells in each case weigh 10 cwt. The striking hammer is interior to the bell, and the striking action is performed by the admission of carbonic acid gas under pressure to the underside of a diaphragm which actuates the hammer striking the bell; the gas is then exhausted into the atmosphere and the hammer returns to a vertical position preparatory to another impulse. A battery of 24 cylinders of CO_2 gas is capable of maintaining a bell signal with a character of 1 stroke every 15 seconds for twelve months without re-charging.

Submarine Bells are of several types, one of which is operated by the motion of the floating body to which it is attached, and another by a regulated pressure of air or CO_2 gas, in order to impart a definite character to the signal.

The bells used on Trinity House light-vessels are suspended outboard from a davit, and when lowered are approximately 18 feet below the water line. They are operated by compressed air at a pressure of 15 lbs. per sq. in., which is led from a receiver on board the vessel through a rubber pipe to the striking mechanism actuating the bell hammer.

Explosive Signals consist of the firing of a gun, explosion of acetylene gas, or the detonation of gun-cotton charges from a rising and falling jib on the lantern roof, of which last system there are seven instances in the Trinity House service, possessing a special feature in the safety gear by which premature firing is obviated while the jib is in its lowered position for loading. This is effected by leading the wires from the battery switch on the lantern pedestal to terminals outside the lantern roof, and fixing to the trunnion of the jib spring contact strips which make and break contact between the battery and detonator leads, when the jib is raised and lowered; by this means, after the jib has been moved through a small angle out of the vertical, no firing of the charge is possible until it is again raised, and consequently no danger to the keeper, or damage to the lantern glazing, can accrue.

Explosive signals are now fired by wireless. A recent representative installation on the Clyde consists of a gun which is charged with an admixture of air and acetylene gas and is fitted with wireless receiving apparatus. A synchronous transmitting set is installed on a pier $1\frac{1}{4}$ miles distant. When fog is observed, the transmitting installation is put into operation and the impulses sent out are received on the apparatus attached to the fog signal gun. The synchronising arrangement renders the installations immune from atmospherics and interference from other wireless waves.

The next three important types of sound signals will be dealt with conjointly, as each is operated by compressed air, and the only difference lies in the type of instrument producing the sound.

Reed Signals are, as a rule, sounded at a pressure of 15 lbs. per sq. in. from a vertically fixed conical horn with a bent top. The reed tongues are of steel 3 ins. long by $\frac{1}{8}$ in. wide, and

are tuned to give a note of about 450 vibrations per second, with a consumption of ⅝ cu. ft. of free air per second of blast.

Manual reed signals are sounded at 5 lbs. per sq. in., and the character is produced by a hand-operated valve.

Siren Signals are sounded at a pressure of 25 lbs. per sq. in. from horizontal tapered elliptical trumpets, and from vertical conical horns with bent tops. In all cases, except at Trevose Head Lighthouse, the siren is automatic and produces a note of about 187 vibrations per second, with a consumption of 11 cu. ft. of free air per second of blast.

The siren at Trevose Head is 12 ins. in diameter, with only one port to the trumpet, and is driven by a small Brotherhood air motor. In this instance there is only one trumpet, which is a specially constructed Rayleigh elliptical steel horn.

The modern form of sound signal is the 'Diaphone,' such as that installed by the Trinity House service at Caskets Lighthouse, off Alderney, one of the Channel Islands, Hartland, Skerries, and elsewhere. They are sounded at a pressure of 30 lbs. per sq. in. and have a consumption up to 45 cu. ft. of free air per second of blast.

Electric oscillators consist of a diaphragm vibrated by an alternating electric current, and are used both for aerial and submarine signals.

Audibility.

Under favourable conditions a siren or diaphone may be heard at a distance of 14 miles, but sometimes the range does not exceed 2 miles. A reed horn may have a range of 3 miles.

A compressed-air siren (0·125 metre in diameter) has been heard at a distance of 13 miles, but the range is sometimes as low as 1·75 mile. A small-sized siren (0·042 metre diameter) cannot be heard at a greater distance than 1·85 mile under favourable conditions.

A 4-oz. charge of cotton powder can be heard under favourable conditions at a distance of 20 miles, and under unfavourable conditions at a distance of 2½ miles.

The distance traversed by the sound from submarine bells has reached 15 miles, but 1 to 3 miles is a serviceable distance. The range of a submarine oscillator is about twice this distance.

Wireless Signalling.

A modern aid to navigation is the wireless fog signal, which enables the mariner to obtain bearings when lights are invisible and sound signals inaudible, the system of transmissions being by means of valves for generating continuous or modulated continuous waves within a band of 950 to 1,050 metres. Some light vessels equipped with wireless beacons are also provided with a submarine sound signal, and the combination of the two is utilised for sending out at regular intervals synchronised signals for distance finding as well as for taking bearings.

DIVING.

Forms and Limitations.—Diving is practised in three forms—either naturally, without any extraneous supply of air during submersion, or artificially, under pneumatic pressure (*a*) with a diving dress or (*b*) with a diving bell. The first is the most primitive method and, though still in vogue among natives in Eastern seas, is bound to be intermittent, in spells of short duration, as it causes great physical strain, leading not merely to acute discomfort, but at times to bleeding at the nose, mouth, and ears. When practised with the aid of pneumatic apparatus, the period of submersion may be greatly prolonged. The only practical limitation is in regard to depth. For every 33 ft. below water, the pressure on the lungs is increased by 1 atmosphere (14·7 lbs. per sq. in.) Thus, at 50 ft. below the surface the pressure is 22¼ lbs. in excess of atmospheric pressure; at 100 ft. it is 44½ lbs., and so on. Such pressures, though considerable, may be sustained with impunity, and even a depth of 150 ft. may be reached without ill effects. But greater depths necessitate considerable powers of resistance and are attended with risk. Until recently the maximum depth to which any diver had descended, as recorded by authentic testimony, was 210 ft., at which depth the pressure was rather more than 7 atmospheres absolute. But in August 1924 some trials, made in Walchensee Lake, Bavaria, with a deep-sea diving outfit prepared by a German firm, are reported to have indicated the possibility of executing work at much greater depths than this. One diver went to a depth of 140 metres. The apparatus proved water and pressure tight, and all turning joints could be moved. He could see for half a metre only. Then another diver went 160 metres below the surface of the water, but at that depth there was absolute darkness. He could walk on the bed of the lake without difficulty. Hoisting from this depth to the surface took only 4½ minutes. On the following day, work, such as fastening a shackle to a sunk piece, was done at a depth of 110 metres. At 160 metres, the maximum depth, the pressure amounts to 225 lb. per sq. in. Harbour work does not generally entail diving in water of more than 10 fathoms (60 ft.).

Caisson Disease.—After immersion under pressure the blood becomes more or less saturated with nitrogen in dissolved form, and when the pressure is relaxed the surplus gas is given off—if too rapidly, bubbles of gas form in the circulatory system, resulting in pains ('pins and needles'), asphyxiation, paralysis, and death, according to the intensity. To these symptoms is given the name of *Caisson Disease*, from its association with work carried on in caissons under pneumatic pressure. It is the most formidable antagonist of the diver, prostrating even the most powerful and robust. Fortunately the malady does not manifest itself at all under moderate pressures, viz., those less than 2 atmospheres absolute. For depths exceeding 35 or 40 ft., it is

necessary to use circumspection during the process of decompression, *i.e.* the ascent to the surface should be made slowly, with stoppages at intervals. By this means the evil effects can be neutralised.

The following table indicates the ordinary time limits in deep water, stoppages in ascent, and approximate air supply, in accordance with Prof. Haldane's recommendations and the latest British Admiralty practice :—

TABLE.

Depth.		Pressure, Lbs. per Sq. In.	Time under Water from Surface to Beginning of Ascent.	Stoppages in Minutes at different Depths.			Total Time of Ascent, Minutes.	No. of Cylinders needed.	Revolutions of Pump per min.
Feet.	Fathoms.			30 ft.	20 ft.	10 ft.			
0–36	0–6	0–16	No limit . .	—	—	—	0–1	1	15–20
6–42	6–7	16–18½	Up to 3 hours .	—	—	—	1–1½	1	25–30
			Over 3 hours . .	—	—	5	6	1	25–30
42–48	7–8	18½–21	Up to 1 hour . .	—	—	—	1½	1	30
			1 to 3 hours . .	—	—	5	6½	1	30
			Over 3 hours . .	—	—	10	11½	1	30
48–54	8–9	21–24	Up to ½ hour . .	—	—	—	2	2	20
			½ to 1½ hours . .	—	—	5	7	2	20
			1½ to 3 hours . .	—	—	10	12	2	20
			Over 3 hours . .	—	—	20	22	2	20
54–60	9–10	24–26½	Up to 20 min. .	—	—	—	2	2	25
			20 min. to ¾ hour .	—	—	5	7	2	25
			¾ hour to 1½ hours.	—	—	10	12	2	25
			1½ hours to 3 hours	—	5	15	22	2	25
			Over 3 hours . .	—	10	20	32	2	25
60–66	10–11	26½–29	Up to 15 min. .	—	—	—	2	2	25
			¼ to ½ hour . .	—	—	5	7	2	25
			½ to 1 hour . .	—	3	10	15	2	25
			1 to 2 hours . .	—	5	15	22	2	25
			2 to 3 hours . .	—	10	20	32	2	25
66–72	11–12	29½–32	Up to 15 min. .	—	—	2	4	2	25
			¼ to ½ hour . .	—	3	5	10	2	25
			½ to 1 hour . .	—	5	12	19	2	25
			1 to 2 hours . .	—	10	20	32	2	25
72–78	12–13	32–34½	Up to 20 min. .	—	—	5	7	2	25
			20 to 45 min. . .	—	5	15	22	2	25
			¾ to 1½ hours .	—	10	20	32	2	25
78–84	13–14	34½–37	Up to 20 min. .	—	—	5	7	2	30
			20 to 45 min. . .	—	5	15	22	2	30
			¾ to 1½ hours .	—	10	20	32	2	30
84–90	14–15	37–40	Up to 20 min. .	—	3	5	10	2	30
			20 to 40 min. . .	—	5	15	22	2	30
			40 to 60 min. . .	3	10	15	30	2	30
90–96	15–16	40–42½	Up to 20 min. .	—	3	5	11	2	30
			20 to 35 min. . .	—	5	15	22	2	30
			35 to 55 min. . .	5	10	15	32	2	30

Floating Landing Stages.

The opening of the Tilbury Floating Landing Stage in May 1930 brings into prominence the value of these facilities for passenger traffic at ports where there is an appreciable range of tide.

The Tilbury Landing Stage has a length of 1,142 ft., a width of 80 ft., and is secured to the left bank of the River Thames by means of four hinged steel booms and heavy mooring cables. It comprises a Jarrah timber deck carried on steel girders resting across sixty-three floating pontoons which rise and fall with the tide through a range of about 20 ft. At each end are two massive dolphins of timber pilework.

The Liverpool Landing Stage near the mouth of the River Mersey has been in service for many years. It is 2,534 ft. long by 80 ft. wide and has ten bridges connecting it with the shore, while, in addition, a floating approach bridge 550 ft. long, also carried on pontoons, provides an easy incline for vehicular traffic at all stages of the tide, which during springs attains a range of 30 ft.

Passenger landing stages for overseas traffic are provided with facilities for the handling and examination of baggage, waiting rooms, shelters, and other accommodation according to circumstances.

SECTION XXIX

NAVIGABLE WATERWAYS—VELOCITY OF RIVER FLOW—TREATMENT FOR EROSION—CHANNEL TRAINING WORKS—RIVER DREDGING—COST OF RIVER REGULATION—RIVER CANALISATION—COST OF CANAL CONSTRUCTION—CONTINENTAL CANALS—LOCKS AND MECHANICAL LIFTS—CANAL CURVES—LOSS OF WATER ON CANALS—TRACTION AND HAULAGE—SHIP CANALS.

(Pp. 1505–1528.)

(Contributed and revised by Brysson Cunningham, D.Sc., B.E., F.R.S.E., M.Inst.C.E.)

SECTION XXIX

NAVIGABLE WATERWAYS—VELOCITY OF RIVER FLOW—TREATMENT FOR EROSION—CHANNEL TRAINING WORKS—RIVER DREDGING—COST OF RIVER REGULATION—RIVER CANALISATION—COST OF CANAL CONSTRUCTION—CONTINENTAL CANALS—LOCKS AND MECHANICAL LIFTS—CANAL CURVES—LOSS OF WATER ON CANALS—TRACTION AND HAULAGE—SHIP CANALS.

(Contributed and revised by Brysson Cunningham, D.Sc., B.E., F.R.S.E., M.Inst.C.E.)

NAVIGABLE WATERWAYS.

Classification.

(1) Natural Channels, i.e., those rivers, or portions of rivers, including estuaries, the normal flow of which is not interfered with by artificial works;

(2) Canalised Rivers, frequently called 'Navigations,' the flow of which is more or less under artificial control, and

(3) Canals, proper, entirely artificial in conception and construction.

The distinction between the three classes is fundamental and important. In the case of the natural channel, improvement works are limited to a rectification of the bed, and no attempt is made to control or augment the natural flow of water, which, being derived from the drainage of the basin, is normally sufficient for purposes of navigation, within the limits of the channel capacity. The canalised portions of rivers consist of sections in which the natural flow is conserved and controlled, as in the manner of a canal, but generally without extraneous sources of water supply. Canals, being works of an entirely artificial character, have to be fed with supplies obtained by the interception of streams, or from other external sources, frequently necessitating pumping and reservoir storage.

Internal Waterways in the United Kingdom.

According to the Report of the Royal Canal Commission, issued in 1909, the total mileage of Canals and Navigations in use in the United Kingdom is about 4,670. Of this total, about 3,639 miles lie in England and Wales, 183 in Scotland, and 848 in Ireland. The length of canals in existence in England and Wales is given as 1,927½ miles, and of navigations, 1,312½ miles. There are 812½ miles of open river and estuary.

IMPEDIMENTS TO NAVIGATION OF RIVERS.

In the seaward section the most pronounced disability, where it exists, is the bar. For its removal a very efficient means, producing immediate amelioration, is dredging; but the effect in the majority of cases is merely temporary, and operations must be continuous if the improvement is to be maintained. Where bars are not in evidence, or not very pronounced, a river mouth is often encumbered by shoals, due either to detritus brought down by the river itself or to coastal erosion and drift. In this case also, dredging may be resorted to with immediate advantage.

For permanent benefit it is requisite to supplement this temporary expedient by training works of a scope and magnitude dependent on circumstances. Such works comprise longitudinal training walls, or embankments, and groynes, dykes, or jetties, to be alluded to later.

In the upper sections of rivers the principal difficulties encountered are irregularities in flow, due to floods in rainy seasons and depletion of water in periods of drought, excessive current velocity, instability of channels and the formation of shoals. In low-lying lands, it is often necessary as a protection against inundation to raise the river banks with impervious material, well supported from behind. Insufficiency of water may be met by the introduction of weirs, or the formation of storage reservoirs. The remedy for excessive and irregular currents lies in training works of a suitable character.

Velocity of Flow in Rivers.

The velocity of river flow varies very considerably, as will be seen from the following table. As regards navigation, currents exceeding three to four miles an hour or four and a half to six feet per second make a serious deduction from the speed of propulsion of vessels proceeding upstream, and render control difficult in the case of small craft travelling downstream. See also par. on HORSE TRACTION, page 1525.

River Seine at Meulan .	2·45 ft. per second.	River Rhine at Basle .	6·36 ft. per second.
„ „ at Paris. .	4·00 „ „ „	„ Aar at Berne . .	7·50 „ „ „
„ Thames at London Bridge . .	3·00 „ „ „	„ Durance, below Sisteron . . .	8·65 „ „ „
„ Tiber at Rome .	3·40 „ „ „	„ Maranon, S. America	13·00 „ „ „

Velocity Observations are taken by means of (*a*) Floats, (*b*) Current Meters, and (*c*) Pressure Tubes.

Floats.—For surface velocities an orange is excellent, floating almost wholly immersed. In the winter season, fragments of ice give similarly good results. The topmost layers of water, however, are frequently affected by the wind. To obviate this the float should extend some distance below the surface. A corked bottle, weighted at the bottom, has proved satisfactory for the purpose. Tube floats are also employed, consisting of tin cylinders about 1 in. diameter, weighted so as to maintain perpendicularity, and leave two or three inches projecting above the surface. Double floats are sometimes used, consisting of a small surface float connected by means of a cord to a float of larger bulk below. In Mississippi experiments, the lower floats were kegs 9 ins. high and 6 ins. diameter, or 12 ins. high and 8 ins. diameter, according as depth was less or greater than 5 ft. They were without top or bottom, and ballasted with lead. The connecting cords were $\frac{1}{16}$ in. to $\frac{1}{4}$ in. diameter. The upper float was of light pine, $5\frac{1}{2}$ ins. square by $\frac{1}{4}$ in. thick, or of hollow tin, $5\frac{1}{2}$ ins. by $1\frac{1}{2}$ in.

Current Meters.—These record velocity by means of the revolution of a screw, being calibrated by drawing through still water at various speeds. The price current meter, used by United States Geological Survey, consists of six cups attached to a vertical shaft, which revolves on a conical hardened steel point when immersed in moving water. Meter measurements can be made from a bridge, from a cable and car specially installed for the purpose, from boats or by wading. Current meters are affected by friction, and are not very suitable for low velocities; in fact, in feeble currents they cease to act.

Pressure Tubes.—Pitot tubes, which are L shaped with the lower horizontal limb pointing upstream, record the velocity by the height to which the water in the tube is forced above surface level by the pressure of the current. Darcy's adaptation consists of two tubes in communication, enabling the differential reading to be taken more accurately and eliminating the effect of ripples and oscillations. The apparatus gives good results in small channels with currents of fair velocity, but it is not suitable for large rivers or for very low rates of flow.

Position of Maximum Velocity may be taken generally at or about surface level, but is not infrequently below. The view is generally held that the velocity varies as the abscisse

of a parabola, whose axis is parallel to the surface. The equation for the parabola in the case of the Mississippi was found to be

$$v = -0{\cdot}79222\,d^2 + 3{\cdot}2611$$

where,
v = velocity in feet per second and d is distance from axis in fractional parts of whole depth considered as a unit.

Mean Velocity.—The position of mean velocity has been found by repeated experiments in U.S. Geological Survey to be located at about 0·6 of the total depth. The mean of velocities obtained at 0·2 and 0·8 of the depth gave the mean velocity of the stream very accurately.

In ordinary cases the least, mean and maximum velocities may be taken in the ratio 3 : 4 : 5. In very slow currents they are nearly as 2 : 3 : 4.

Prony's Formula:

$$\frac{\text{Mean velocity}}{\text{Maximum velocity}} = \frac{\text{Maximum velocity} + 7{\cdot}71 \text{ ft. per sec.}}{\text{Maximum velocity} + 10{\cdot}28 \text{ ft. per sec.}}$$

Computed Velocity.—A great number of formulæ have been published connecting the velocity with the hydraulic mean radius (R) = $\frac{\text{area of stream}}{\text{wetted perimeter}}$, and the sine of slope (S) = $\frac{\text{fall in level}}{\text{length of channel}}$. Most of these expressions can be reduced to the form

$$v = c\sqrt{\mathrm{RS}},$$

where c is a coefficient varying with the class and nature of the channel, as well as with the personality of the observer. Some of the coefficients are very complicated, and the values are variable to a considerable degree. Mr. A. A. Barnes ('Hydraulic Flow Reviewed,' 1916) proposes to modify the powers of R and S so as to admit of a coefficient of constant value. His expression is

$$v = 58{\cdot}4\ \mathrm{R}^{\cdot694}\mathrm{S}^{\cdot496}$$

or, approximately,

$$v = 60\ \mathrm{R}^{\cdot7}\ \mathrm{S}^{\cdot5}$$

In nineteen observations with ranges in width of stream from ·26 ft. to 2,598 ft., in depth from ·19 ft. to 69 ft., in hydraulic mean radius from ·0786 ft. to 53·6 ft., and in slope from 1¾ to 241·2 ft. per mile, velocities of from 1 to 7½ ft. per second were computed and found to agree, within 1 per cent., with actual observation.

In the course of investigations made during three years in connection with the regulation of the Whangpoo and Yangtse Rivers, Dr. Herbert Chatley found that in fairly straight and uniformly sectioned reaches when the flow is steady the above exponential form $v = c\mathrm{R}^{\cdot7}\,\mathrm{S}^{\cdot6}$ gave consistent results. But when the flow was unsteady he found it more satisfactory to take the expression

$$v = c\mathrm{R}^{\cdot7}\left(\mathrm{S} - \frac{a}{g}\right)^{\cdot5}$$

where a is the acceleration. In other words, the effective slope was equal to the algebraic difference between the actual slope and an 'acceleration' slope determined by the ratio of the actual mean acceleration of the water to the gravitational acceleration.

Critical Velocities.—The following is a statement founded on the researches of Dubuat and Beardmore of velocities at which various materials in a river bed commence to be eroded:

Soft clay	3 ins. per second.	Rounded pebbles, 1 in. diam. .	2 ft. per second.
Fine sand	6 „ „ „	Rough gravel „ „ .	2½ „ „ „
Coarse sand and pea gravel	9 „ „ „	Pebbles 1½ in. „ .	3½ „ „ „
Gravel as large as French beans	12 „ „ „	Heavy shingle	4 „ „ „

TREATMENT FOR EROSION.

Protection of River Banks.—Water plants form a very efficient natural protection, and they assist in retarding the current—sometimes not an unmixed advantage.

Earthen banks of a soft and erodable character may be artificially protected, where stone is procurable, by rubble pitching, laid at a slope of 1½ or 2 to 1, from 8 to 12 ins. thick at the top, and increasing in thickness downwards at the rate of 3 ins. per yard. The foot of the slope should be adequately secured to some firm abutment, such as piling, otherwise the pitching may be undermined and, in that case, it will subside.

Where stone is not procurable, fascine aprons or mattresses, formed of brushwood, may be utilised. These are bundles of branches and twigs (usually willow) from 9 to 12 ins. in diameter, about 12 ft. long and bound with tarred rope at intervals of 4 ft. or so, laid side by side and tied together so as to produce a surface covering of the desired extent. The 'fascines,' as they are termed, should be secured by stakes or short piles to the bank upon which they are deposited.

A plantation of small willows above water level is useful in counteracting the more deleterious effects of floods.

Mississippi River Bank Protection.—The banks of the Lower Mississippi are protected where necessary from erosion below water surface level by means of brushwood mattresses loaded with stone. The portion of the bank above surface level is protected by a covering of stone only, 10 ins. thick, laid to a suitable slope.

The mattresses were deposited from a single barge, or from a couple of barges, placed end to end, so that the combined length of the barges exceeded the width of the mattress. The ordinary size for mattresses was 800 ft. by 250 ft., but some mattresses were 1,000 ft. long by 300 ft. wide. 'The mattresses are constructed on the mattress-barge by laying parallel to the shore, and, 8 or 10 ft. apart, a series of poles, whose aggregate width equals nearly that of the proposed mattress. These poles are small trees from 3 ins. to 6 ins. diameter at the butt, and 25 ft. to 30 ft. long. Smaller poles are then placed at right angles to the larger series, so as to pass alternately beneath and above the successive larger poles, as in basket-work. These smaller poles, when forced into close contact with each other and with the other series, form a tolerably compact web or mattress. When so many of the small poles have been added that they approach the ends of the larger series, whose poles are parallel to the shore, these larger poles are each continued by having another large pole spliced to it, and so on, for the length of the mattress. The brush in the mattress is fastened together frequently with iron wire and cross poles. The whole mattress is strengthened by wire cables about 20 ft. apart, and by ⅜ in. to ½ in. wire ropes running longitudinally the whole length of the mattress. Transverse cables of wire, about 16 ft. apart, also strengthen the mattress; and transverse wire ropes in addition are sometimes used.'

As each mattress was woven it was allowed to be floated down stream off the barge by the current, and when a sufficient length had been accomplished it was loaded with stone and sunk, the upper end being sunk first. The customary under-water slope did not exceed 1 in 3, and the depth of the outer edge often ranged from 60 ft. to 100 ft. (*From Report by General C. B. Comstock, on Mississippi River Improvement, to Chicago International Congress*, 1893.)

Flexible mattresses of reinforced concrete are being experimented with on the Mississippi.

CHANNEL TRAINING WORKS.

The instability and eccentricity of channels is remedied by training works, comprising (*a*) projecting jetties and groynes, (*b*) longitudinal dykes and embankments, and (*c*) ground sills.

Groynes or Jetties project from the banks to the stream at right angles, or thereabouts (generally within 70° to 90°) to its course, and have the effect of reducing the channel and concentrating the scour of the current within certain bounds. The beneficial effect, however, is largely local, and, in the absence of restraint between the jetties, the stream tends to diffuse itself over a wider area, so that while the bed may be deepened and improved opposite the ends of the groynes it often becomes actually shallower in the intermediate spaces. To be effective, groynes should be spaced apart not more than five times their own length, and this often renders them less economical than a system of longitudinal dyking, which, on the whole, will be found far more satisfactory and is often ultimately adopted. Groynes should, in fact, only be looked upon as of the nature of a temporary expedient. They may be formed of piling, fascine work, or rubble.

Groynes on the River Waal.—These are so arranged as to point upstream with the axis of the groyne on the concave bank forming an angle of 80° with the projected channel, and that on the convex bank an angle of 70°. This system has been found favourable to the deposit of alluvium between the groynes.

The groynes themselves are formed of a core of sand extending to 11 ft. 6 ins. below ordinary low water. The core is covered with mattresses of fascine work, having a lateral slope of 6 to 4, surmounted by a layer of broken brick and rubble stone. The slope towards the river is 1 in 4. The width of the crown of the groynes is 11 ft. 6 ins., and their height along the boundary of the normal river bed 1 ft. 6 ins. above ordinary low water level, corresponding to about 1 ft. 6 ins. below mean water level.

It is stated that sand forms a satisfactory core if banked in large quantities at a time, and that no harmful results ensue from the slipping away of a certain amount during submergence. A few years after formation the groynes receive a protective covering of basalt.

The River Waal has a velocity varying between 2¼ and 3½ ft. per second. The discharge is about 21,000 cubic ft. per second at lowest water level, and 220,000 cubic ft. during highest floods. (*From Report by M. Baucke to Philadelphia Navigation Congress*, 1912.)

Longitudinal Dykes are walls or embankments of timber, stone, concrete, fascines, or other material built up so that the channel is rigidly confined within definite limits throughout the length treated. At the outlet of the Fen Rivers in the Wash, banks of fascine faggots and clay were constructed 22 ft. wide at the base, 17 ft. high, and 13 ft. wide at the top, at a cost of about 1*s.* 8*d.* per cubic yard. On the River Weser, stone and gravel embankments have been found preferable to fascines on the ground that the surface slopes may be rendered less steep than 45°, which is stated to be the limit for fascine work.

River Training Works at the Mouth of the Danube.—Operations for the improvement of the navigable channel at the Sulina mouth of the Danube have been in progress for more than half a century, under the direction of the European Commission of the Danube, an international body, constituted in 1856 by the Treaty of Paris.

There are three main arms, or branches, of the Danube delta, at its influx into the Black Sea. The Sulina arm, with the port of Sulina at its mouth, was chosen for treatment by reason of its greater depth and general use, despite the fact that its discharge was scarcely 10 per cent. of the total.

Rudimentary timber piers, consolidated by concrete mass-work and stone rubble, were constructed in the first instance, thus concentrating the outward current, and by the scour so occasioned, the depth across the bar of the river was increased from 9 ft. to 17 ft. rapidly; and subsequently more gradually to 20½ ft., at which figure it remained, naturally, for a period of 22 years until 1894, when dredging was commenced and a depth of 24 ft. obtained, necessitating, however, continuous maintenance work. In 1915 the depth of 24 ft. was lost and three dredgers proved incapable of re-establishing it. On account of the progressive advance of the delta, the action of natural forces, aided by artificial means, had become insufficient to cope with the deposit of alluvium, which, indeed, threatened to destroy serviceable access to the river for shipping.

In 1922 a prolongation of the piers or jetties was begun, and in 1925 was completed to a point of about 3,300 yds. from the Sulina Lighthouse, which is located at the port entrance. On July 25, 1925, the new channel, 200 ft. wide, was opened to traffic with a minimum depth of 20 ft., which within a month was increased to 23 ft. Deposit, however, continued to take place at the outer extremities of the piers, and a further prolongation is now in progress with the object of reaching deeper water for the pierheads.

Levees constitute a type of longitudinal dyke. They contract the river channel in its higher stages and protect the adjacent country from inundation. As used in American practice, they are generally banks of earth or spoil, well consolidated, with a central trench forming an impervious connection with the natural ground.

Channel Curves.—The natural tendency of river channels is to form a series of deep water pools on the concave sides of bends, with shoals at the intervening points where the current passes over from one side of the river to the other. A judicious system of training will maintain the natural sequence of these deep pools, at the same time suppressing any unduly sharp bends which might be awkward for navigation. Generally speaking, for vessels of 400 to 600 tons moving at a fair rate of speed a radius of 550 yards should be regarded as a minimum—at low speeds, 220 yards will suffice. The minimum for vessels of very small tonnage (say, 200 tons and under) may be as low as 66 to 88 yards.

Ground Sills are serviceable where there is excessive scour of the bed or undue restriction in width of channel. They have the effect of rendering the fall of the stream more uniform. In German practice they are set from 40 to 50 feet apart, and found to be better than continuous pitching or filling of the bed. They are laid across the river so that their highest levels at the bank are about 12 inches below the normal surface, and they slope downwards to the centre of the stream at an angle of about 1 in 40. On the Mississippi the ground sills consist of either rubble mounds or cribs filled with rubble, or, again, of brush mattresses and stone in alternate layers. Whatever form be adopted, the top should be finished off smoothly to prevent injury to passing vessels. Ground sills may be used where it is desired to raise the low water level or lengthen out a slope.

Storage Reservoirs.—For the augmentation of the flow in dry seasons storage reservoirs have been advocated, and in two instances—the Mississippi and the Volga—have been adopted. Circumstances, however, have to be favourable if they are to prove a satisfactory and economical arrangement. The Volga has its source in a lake and swamp region where land is cheap and unfit for agriculture. The dams required were only 17½ feet high, and a storage capacity of 35,000 million cubic feet was obtained. On the Upper Mississippi above St. Paul, five reservoirs were formed out of an extensive lake area of 480 square miles, with a storage capacity of 93,400 million cubic feet. It is stated that the increase in height obtained at St. Paul during the low water period—about 90 days—averages only 14 inches. The system is not regarded as sufficiently satisfactory to justify its extension elsewhere in the United States. Where there is no original headwater accumulation its adoption is out of the question.

RIVER DREDGING.

The improvement of river channels by means of mechanical dredging has received considerable impetus of late years by reason of the development in power and capacity of dredging plant. The various appliances are described in detail in connection with harbour dredging (*vide* p. 1480 *et seq.*). In some cases, as for instance the Volga, with its tributaries the Oka and the Kama, the application of dredging alone, without the assistance of artificial training works, has proved sufficient to ensure the maintenance of a serviceable channel, while in other cases, notably on the Mississippi and the Mersey, engineers employ both systems conjointly.

Dredging Plant on the Volga.—On the Volga, dredgers work from the shallowest depths up to 20 ft. and consist of both bucket dredgers and suction pumps.

One of the former has a hull 131 ft. by 26 ft. with a draught of 3 ft. 3 ins. It discharges about 300 cubic yards per hour by means of shoots, or 370 cubic yards per hour through a floating pipe-line. Two other dredgers have a length of 157 ft., a breadth of 31 ft. 2 ins., and a maximum draught of 4 ft. 11 ins. They are self-propelling with a speed of 4 knots. The pipe-line is 1,740 ft. in length. The buckets have a capacity of ½ cubic yard, and they are reinforced by centrifugal pumps of 5 ft. diameter. The pipe-line has a diameter of 19 ins. A still more recent dredger is 131 ft. long by 28 ft. wide, with a draught of only 2 ft. 4 ins., a dredging depth of 17 ft., and a working capacity of 196 cubic yards per hour in sandy material.

Cost of River Regulation.

Country.	River.	Depth.	Length.	Total Cost.	Average Cost per Mile.	Average Fall per Mile.
		Ft.	Miles.	£	£	Ft.
United States	Upper Mississippi (St. Paul to Missouri River)	4½	632	2,050,000	3,244	0·44
France . .	Rhone (Lyons to Sea) .	4·1	205	2,689,000	13,000	2·5
Germany .	Rhine	4–10 3–4·1	214	2,636,400	12,200	1·14
Ditto .	Weser (near mouth) .	5–10 3·6–4·6	210	463,300	2,210	1·65
Ditto .	Elbe (near mouth) .	6–10 2·6–4·25	252	2,047,900	8,130	0·98
Ditto . .	Oder (near mouth) .	5–8·7	337	1,229,600	3,650	1·29

(*Extracted from Report by Major Harts to Philadelphia Navigation Congress*, 1912.)

PARTICULARS OF DREDGING PLANT USED ON MISSISSIPPI RIVER.

All self-propelling, with two main pumping engines, each of horizontal tandem compound condensing type, and centrifugal pumps with water-jet sand agitators.

Details.	Dredger 'Iota.'	Dredger 'Kappa.'	Dredger 'Henry Flad.'	Dredger 'B. M. Harrod.'
Date when built	1900	1902	1902	1906
Cost, complete	20,296*l.*	27,080*l.*	27,080*l.*	48,000*l.*
Material of hull	Steel	Steel	Steel	Steel and iron*
Dimensions of hull	192′×44′×7′ 4′	192′×44′×7′ 4′	192′×44′×7′ 4′	210′×44′×8½′ 5′
Size of engines	16″×26′	15″×30″	15″×30″	20″×34″
Revolutions per minute	165	140	140	140
Stroke	20″	24″	24″	24″
Pumps:				
Diameter of runner	75″	84″	84″	90′
Peripheral velocity, ft. per sec.	53	51·3	51·3	55
Diameter of suction	Two 24″	Two 24″	Two 24″	Two 27″
Width of suction head	23′	23′	23′	32′
Diameter of discharge	32 11/16″	31⅞″	32″	36″
Maximum working depth	20′	20′	20′	20′
Nominal capacity, sand per hour	1,000 cu. yds.	1,000 cu. yds.	1,000 cu. yds.	2,000 cu. yds.
Sand agitators:				
Kind of pump	Duplex comp. cond.	Duplex comp. cond.	Duplex comp. cond.	Centrifugal
Size	16″×15″	16″×15″	16″×15′	Two 12″
Number and diameter of jets	Twelve, ¾″	Twelve, ¾″	Twelve, ¾″	Eighteen, 2″
Pressure	50 to 80 lbs.	50 to 80 lbs.	50 to 80 lbs.	20 lbs.
Boiler:				
Kind	7 Miss. River	7 Miss. River	7 Miss. River	7 Miss. River
Size	44″×30′	44″×30′	44″×30′	47″×28′

* Iron used for hull plating, the piping, and the chimney stacks, as it was found by experience that the life of iron materially exceeds that of steel for such purposes.

(*Extracted from Report by Mr. J. A. Ockerson to Milan Navigation Congress*, 1905.)

RIVER CANALISATION.

The canalisation of rivers is effected by dividing their beds into a series of reaches separated by dams, or weirs, which are provided with a communicating lock, or locks, arranged generally at one side. Dams are either fixed or movable, and they may also be permeable or impermeable. Fixed, impermeable structures are generally resorted to as being more economical in construction and more convenient in action. Movable dams may be said to be required only in cases of excessive flood flow, or where the conditions admit of free navigation during appreciable periods. Fixed weirs are usually constructed of solid stonework, rubble, piling or concrete provided with the requisite sluices and overflows. Movable weirs are more often of timber or iron framework with mechanism for lowering and raising them, either wholly or in part by means of leaves or panels. They are of various types with distinguishing names, such as Needle, Chanoine, Boulé and Bear-trap.

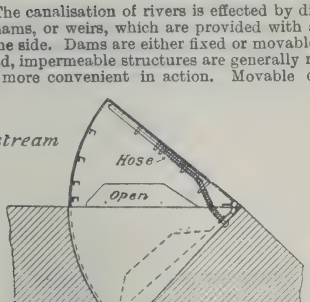


FIG. 1.—Section of Weir near Bremen.

Sector Weir near Bremen.—A notably large sector weir which constitutes a development of the bear-trap type, across the River Weser, near Bremen, is illustrated in cross sectional outline in fig. 1. The weir is in two lengths of 177 ft. each, and holds up the water on an average to a height of about 12 ft.; in winter, the height rises to 18 ft. The structure forms the hollow horizontal sector of a cylinder of 21½ ft. radius turning on a horizontal shaft, the angle of the sector being about 45 degrees. The weight of each sector is 129 tons. A description of the weir with the method of raising and lowering is contained in *Engineering* of July 7, 1916.

TYPES OF UNITED STATES RIVER REGULATION WORKS.

FIG. 2.—Brushwood and Stone Dam, Upper Mississippi.

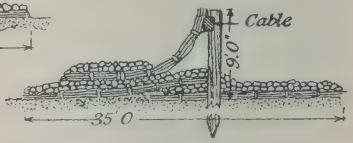


FIG. 3.—Pile and Brushwood Dam, Upper Mississippi.

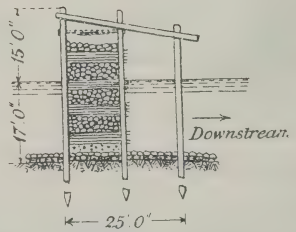


FIG. 4.—Pile, Brushwood, and Stone Dyke, Arkansas River.

FIG. 5.—Pile and Stone Dam, Alleghany River.

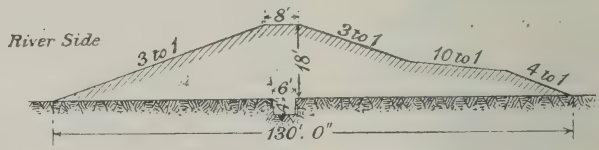


FIG. 6.—Levee on the Lower Mississippi.

COST OF RIVER CANALISATION.

River.	Length Canalised. Miles.	No. of Locks.	Depth. Feet.	Fall in Feet per Mile.	Total Cost.	Cost per Mile.	Operation and Maintenance. Total.	Operation and Maintenance. Per Mile.
					£	£	£	£
FRANCE:								
Saone	232	30	8·2	·85	1,755,000	7,550	12,100	52
Seine (Montereau to Paris)	61	12	—	·85	991,000	16,300	10,870	178
Seine (Paris to Rouen)	145	9	10·5	·85	3,500,000	24,200	18,450	127
Seine (New Works)	140	9	—	—	2,535,000	18,100	—	—
Yonne	67	26	—	2·43	1,118,000	16,700	6,730	100
Marne	113·5	19	7·25	1·16	1,047,000	9,250	7,980	70
Aisne	35·5	7	—	—	195,400	5,470	2,130	60
GERMANY:								
Saar	19·5	6	4·25–7·9	2·15	354,400	18,170	6,410	330
Main	23·5	5	5·9–8·9	1·40	448,200	19,070	8,010	340
Fulda	17·0	7	6·6–11·9	3·30	157,000	9,240	4,385	260
Salle	89·5	15	4·4–8·8	1·27	383,400	4,280	9,080	100
Unstrut	40·5	12	2·5–8·5	1·72	105,600	2,600	2,840	70
Oder	53·2	14	—	1·90	1,211,800	22,780	43,420	815
UNITED STATES:								
Black Warrior	91	7	6	2·76	508,080	5,583	25,207	275
Coosa	25	3	3	2·58	209,687	8,386	1,587	64
Allegheny	25	3	—	2·06	267,574	10,703	9,328	373
Monongahela	131	15	7–8	2·4	1,369,171	10,452	34,673	265
Muskingum	84	10	6	1·55	—	—	10,026	120
Great Kanawha	90	10	6	0·84	844,765	9,394	16,361	187
Kentucky	226	11	6	0·87	580,662	2,578	31,989	141
Green and Barren	190	7	5	0·49	327,682	1,725	15,177	80
Cumberland	131·4	8	6	0·67	525,076	3,996	6,611	50
Illinois	194	2	7	0·05	303,145	1,563	2,568	153

(*Extracted from Report by Major Harts to Philadelphia Navigation Congress*, 1912.)

Lateral Canals.

As an alternative to river canalisation and where the channel exhibits features which do not conveniently lend themselves to treatment, an independent lateral canal may be constructed, generally following the lie of the river, but also making use of any facilities which may offer themselves for straightening and rectifying the course.

The River Rhone, in the chequered history of its amelioration, has been furnished with three lateral canals to overcome the difficulties of navigation at its natural mouth. Each in turn of the earlier two became inadequate for dealing with the traffic which accrued from its construction, and now there is a recently constructed artificial waterway to the port of Marseilles which is described in detail on p. 1516.

COST OF LATERAL CANALS.

Locality.	Length in Miles.	Locks.	Total Cost.	Cost per Mile.	Maintenance and Operation. Total.	Maintenance and Operation. Per Mile.
			£	£	£	£
FRANCE:						
Loire, Digion to Briare	121·7	37	2,071,000	17,000	6,330	52
Garonne	120	53	2,484,000	20,700	7,500	62½
UNITED STATES:						
St. Mary's Falls	1·1	2	1,611,450	1,464,955	20,620	18,740
Des Moines Rapids	8	3	316,600	39,576	7,754	969
Muscle Shoals	18	11	638,345	35,463	10,280	571
Colbert Shoals	8	1	441,588	55,200	—	—
Cascades Canal	¼	2	764,065	1,528,130	2,877	5,751

(*Extracted from Report by Major Harts to Philadelphia Navigation Congress*, 1912.)

BRITISH CANALS.

The following summary of the conditions prevailing in Great Britain is extracted from the Report of the Royal Canal Commission (1909):

'On the whole, the waterways may be roughly divided into two classes, those having locks of more, and those having locks of less, than 14 ft. in width. Corresponding with these two classes of locks are two classes of vessels usually known respectively as barges and narrow boats, the latter being often called "monkey boats," and chiefly in use on the Southern and Midland canals of England. The barge usually measures from 60 to about 72 ft. in length by 14 ft. in width of beam; the narrow boat is about 70 to 72 ft. in length, but only 7 ft. in width. The narrow boat when loaded, and drawing about 3 ft. 3 ins. of water, will carry about 30 tons over many of the canals. The barge will carry about twice that weight, with the same draught. Both the narrow boat and the barge could carry a heavier cargo if the depth of water were increased sufficiently.

'The usual length of lock, both on the wider and the narrower waterways, in the South and the Midlands, is, accordingly, about 75 ft. But on the wider, or barge canals, the locks are usually between 14 and 15 ft. or more in width, and on the narrow canals between 7 ft. and 8 ft. In both classes of lock the depth of water on the sill varies, speaking very generally and with many exceptions, from 4 ft. to 5 ft.'

	Miles.	Chains.
Canals with locks under 14 ft. (narrow)	1,165	5
" " " over 14 ft. (barge canals)	762	29*
Navigations under 14 ft.	478	57
" over 14 ft.	834	20

* Including the Manchester, Gloucester and Berkeley, and Exeter Ship Canals.

DEVELOPMENTS IN CANAL CAPACITY.

In France and South East Belgium the standard boat carries about 300 tons, while in Germany the newest canals are adapted to boats of 600 tons and upwards. In this country opinion as to future policy is divided between the adoption of the 300-ton type and the retention of the smaller class of boat in general use, not exceeding 100 tons. The following are the dimensions recommended for the two classes respectively by Sir J. Wolfe Barry and Partners, who were called in to advise the Royal Commission on the matter.

Dimensions.	100-ton Boat. Canal.	300-ton Boat. Canal.
Length of barge over all	80 ft.	115 ft.
Beam " "	14 "	21½ "
Draught	5 "	6½ "
Area of immersed midship section of barge	69 sq. ft.	138 sq. ft.
Cross-sectional area of canal	350 " "	690 " "
Depth of canal	7 ft.	8½ ft.
Top width of waterway in open country	59½ "	92½ "
" " " " in towns	51½ "	83 "
Width of waterway at bridges	32 "	48 "
Minimum headway of new bridges	10 "	13 "
" " " existing bridges	9 "	13 "
Ruling radius for bends of canal	12 chains	17 chains
Locks: Length, mitre to mitre of gates	253 ft.	266 ft.
" Width between side walls	15 "	22½ "
" Depth on sills	6½ "	8 "

GOVERNING DIMENSIONS OF LARGE BOATS AND CANALS AS DETERMINED BY FRENCH COMMISSION OF PUBLIC WORKS.

Tonnage of Boat.	Dimensions of Boats. Length.	Beam.	Draught.	Width at 6½ Ft. below Surface.	Depth on Axial Line.	Features of Navigable Waterway. Curves. Increased Width.	Radius R.	Locks. Serviceable Length.	Width.	Depth on Sill.
	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.
300 .	126·3	16·4	5·9	32·8	7·9	1,246/R	984	132·8	19·7	8·2
600 .	196·7	23·3	5·9	51·3	7·9	3,018/R	1,919	213·2	29·5	8·2
900 .	246·0	28·2	5·9	65·6	8·2	4,724/R	2,690	262·4	34·4	8·2
1,200 .	295·2	32·8	5·9	77·7	8·4	6,888/R	3,509	311·6	39·4	8·2

A boat, 24 to 25 ft. wide by about 200 ft. long, with 6½ ft. draught, has been recommended by experts as a suitable type for a 600-ton barge on the Canal du Nord.

On the Rhone, barges of the same width and 220 ft. long have been put into service, carrying 650 tons on a draught of 6 ft.

The type of boat on the Rhine is the lighter of 246 ft. by 36 ft., which with the same draught carries about 900 tons, but when immersed to 6½ ft. is capable of taking 1,200 tons, with a displacement coefficient of 0·75.

('*Annales des Ponts et Chaussées*,' *March-April* 1920.)

Capacity of German Inland Waterways.

Inland waterways have reached a very high pitch of development in Germany. Certain inland harbours surpass important seaports in the amount of traffic handled The size of craft accommodated far exceeds anything attempted in this country. The following is a statement of the lengths of waterway available for successive grades of tonnage :—

	Tons.	Kilometres.
For vessels of	3,000 or more	615
,, ,, from	1,200–3,000	1,463
,, ,, ,,	600–1,200	2,064
,, ,, ,,	400– 600	2,607
,, ,, ,,	250– 400	1,705
,, ,, ,,	170– 250	1,565
Smaller vessels	—	4,515

An important project now in contemplation is the Rhine-Neckar-Danube Canal, which will complete the waterway connection right across Europe from the North Sea to the Black Sea. The length of the proposed canal will be about 165 miles, of which 125 miles come upon the Neckar section from Mannheim to Plochingen, and the remaining 40 miles on the section Plochingen-Ulm.

CONTINENTAL CANALS.

Canal du Nord, France, duplicating the St. Quentin Canal, which forms part of the navigable waterway connecting Paris with the Northern parts of France.—The ordinary cross-

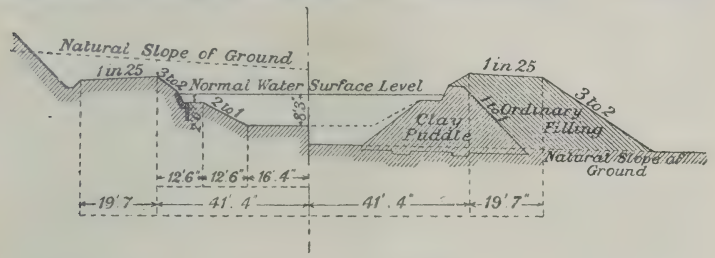


FIG. 7.—CANAL DU NORD.

Section in Excavation. Section in Filling on Impermeable Ground.

section has a minimum width of 33 ft. at the bottom and 39 ft. at a depth of 6½ ft., with a navigable depth of 8 ft. Towpaths, 13 ft. to 20 ft. wide on each bank, one for upstream and one for downstream traffic. Locks provided to accommodate two 300-ton barges in tandem ; dimensions, 279 ft. long by 19½ ft. wide by 8 ft. deep. Lifts, 13 ft. to 22 ft., with minimum reach of 1,100 yards between locks. Tunnels of two types : those for single train of boats are 33 ft. wide at springing of arch, and those for double trains of boats abreast are 52½ ft. wide at springing. Each type has two side platforms, 6½ ft. wide carried on piles. Largest tunnel at Ruyaulcourt, which is 2¾ miles long. Canal designed for mean speed of two miles per hour, with barges mechanically hauled. Fig. 7 gives typical sections.

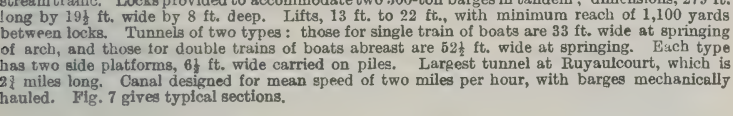


FIG. 8.—Brussels-Charleroi Canal.

The Rhone-Marseilles Canal.—Opened in 1926, the cross-section in the straight lengths gives a width of 82 ft. at a depth of 6½ ft., while the normal depth is 8 ft. 2 ins., and that in the outer portion approaching Marseilles 10 ft. Sills and inverts throughout are laid to this lower level, in order to admit of future deepening. The width of the canal is narrowed to 59 ft. where it passes through the Nerthe Mountain (Rove Tunnel). The canal section on the straight portion has been fixed at five times the immersed section of the present Rhone barges, when loaded to their maximum capacity of 600 tons on a 5 ft. 9 in. draught. These barges have a beam of 26 ft. 3 ins. The dimensions of the locks are 525 ft. internal length, by 52½ ft. in width. The lock at Arles has a lift of 23 ft. at low water. The total cost of the canal is stated to exceed 3,560,000*l.* The section of the Rove Tunnel shown in figs. 9 and 10 cost 52*l.* per ft. run of which the masonry work accounted for 16*l.* per ft. run.

Modern German Canals.—Herr Germelmann, in a Report to the Philadelphia Navigation Congress 1912, remarks that in Germany high economic value is attached only to large efficient canals, suitable for vessels of 213 ft. in length by 26½ ft. in breadth (600 ton barges), or, at least, for vessels of 180½ ft. in length and 26½ ft. in breadth (400-ton barges) with draughts of 5 ft. 9 ins. The cross-sectional area of canals recently constructed is as follows:—Spree-Oder Canal from 705 to 760 sq. ft.; Dortmund-Ems Canal, in cuttings, 640 sq. ft.; in embankment, 810 sq. ft.; Teltow Canal, 652 sq. ft.; Ems-Weser Hanover Canal, in cuttings, 662 sq. ft.; in embankment, 780 sq. ft.; Rhine-Ems Canal, 986 sq. ft.; Berlin-Stettin Canal, about 732 sq. ft.

Trollhätta Canal, Sweden.—This canal, first opened in 1607 and recently enlarged and extended at a cost of about 1,300,000*l.*, has a length of about fifty-two miles, and connects Lake Vänern with the Kattegat, between which there is, at ordinary water level, a vertical difference of 146 ft. The canal is built to accommodate vessels of 13 ft. 1½ in. draught, but with a view to future developments the locks are constructed to pass vessels of 16 ft. 5 ins. draught. The normal bottom width of the canal is 79 ft., but this, at passing places, is increased to 95 ft. and 115 ft. The maximum depth in rock cutting for vessels drawing 13 ft. 1½ in. is 14 ft. 5 ins., and in earth excavation, 18 ft. 3 ins. For the 16 ft. 5 in. vessel these dimensions are increased to 18 ft. and 19 ft. 7 ins. respectively. The lock chambers have a length of 295 ft. and a width of 45 ft., with 18 ft. of water over sills. The greatest change in level at any one lock is 26 ft.

New York State Barge Canal.—This canal, the inception of which dates back to the beginning of last century, has recently been enlarged so as to afford accommodation not only for the largest class of barges at present constructed, but also for those up to 3,000 tons which may be built in the future. A minimum depth of 12 ft. has been provided, with a minimum width of 94 ft. in rock cuttings and of 125 ft. in earth excavation. The locks, 57 in number, have chambers 310 ft. long and 45 ft. wide and are capable of passing vessels with maximum dimensions of 300 ft. by 42 ft. The lock at Little Falls has a range of 40½ ft., exceeding any single lift on the Panama Canal. The syphon lock at Oswego is the first of its type in the United States, and with a range of 25 ft. is probably the largest of its kind in existence. A description of it is as follows (*vide Engineering*, August 2, 1918). The general design of the culverts is not unlike that of an ordinary lock, except that at the upper and lower ends the culverts are curved upwards so as to form necks. These rise a little above the highest water level and are shut off from all communication with the outer air, except through the operating pipes. The flow of water in the syphon is started by means of tanks, one of which is built in each wall near the upper end, and communicates through pipes with the upper and lower levels and with both syphons in the same wall; they are shut off from all other communication with the outer air. The cycle of operations is as follows: The tank is first filled with water, then the intake valve is closed and the outlet valve opened. In consequence there is a body of water suspended by its weight, but tending to escape into the lower pool, and thus producing the necessary vacuum. Therefore, on opening the air valve, the air in the syphon rushes towards the vacuum, and water begins to flow up and over the crest in the neck. When using both syphons, the lock chamber can be filled in from 4½ to 5 mins., and it can be emptied in from 5½ to 6 mins. It has been found that the draught of the vacuum is such that, soon after the flow has started, the direction of the air is reversed and the vacuum is restored in the flask. The operating power is accordingly self-renewing except for a small amount of air leakage.

Cost of Canal Construction and Improvement in Great Britain.

In 1905 Mr. J. A. Saner estimated the constructional cost of new canal, suitable for two lighters of not less than 250 tons carrying capacity, or 200 tons when self-propelled, with locks 230 ft. by 22 ft. having 6½ ft. draught of water over sills, the canal itself being 40 to 45 ft. wide at bottom, 80 to 85 ft. wide at top, and 8 ft. deep in centre, at 30,000*l.* per mile, and the cost of improving existing routes to reach the same standard at 15,000*l.* 'These figures are based on the actual cost of the Weaver Navigation and the Manchester Ship Canal; the former has cost about 42,000*l.* per mile, and the latter about 440,000*l.* per mile, inclusive of docks in each case. The Worcester and Birmingham Canal originally cost 20,000*l.* per mile; and between 1879 and 1891 the French improved 2,500 miles of river, 2,250 miles of canal, and built 875 miles of new canal, for a total of 21,000,000*l.*, or an average of 3,734*l.* per mile; so that the foregoing figures should be ample.' (*Saner on Waterways in Great Britain, Min. Proc. Inst. C.E.*, vol. clxiii.)

RHONE-MARSEILLES CANAL. SECTIONS THROUGH ROVE TUNNEL.

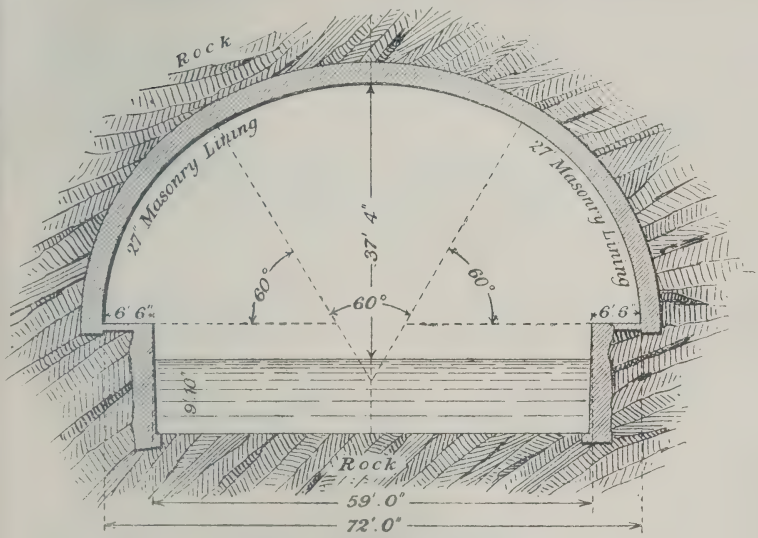


FIG. 9.—In Sections of Low Thrust.

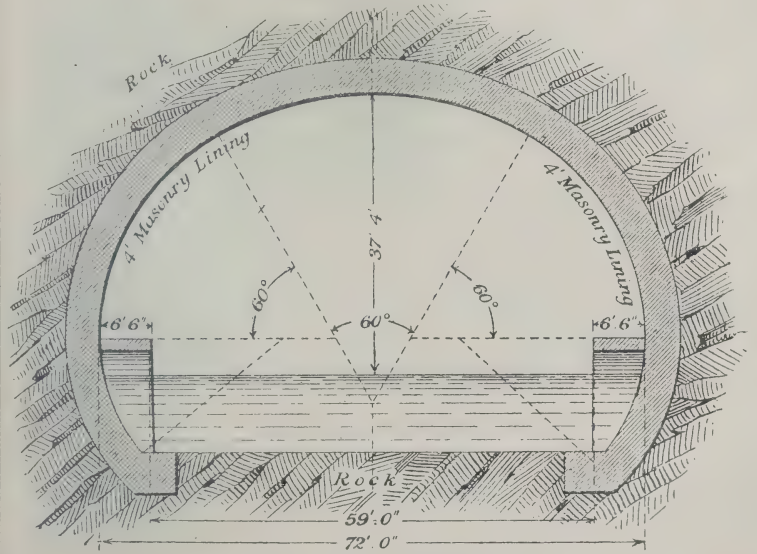


FIG. 10.—In Sections of High Thrust.

EXAMPLES OF MODERN GERMAN CANALS.

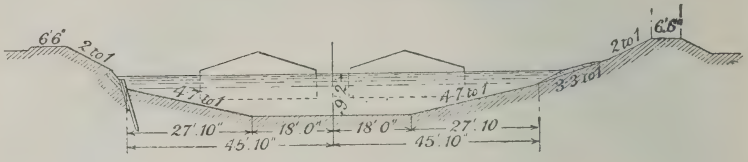


FIG. 11.—Spree-Oder Canal.

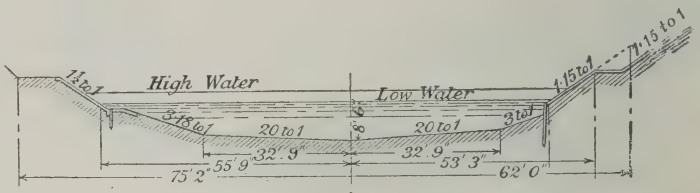


FIG. 12.—Teltow Canal.

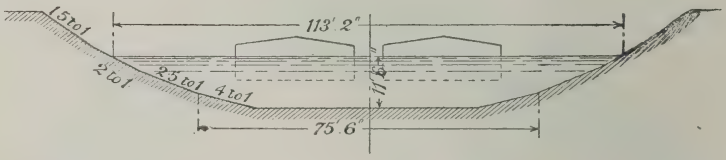


FIG. 13.—Rhine-Ems (Herne) Canal.

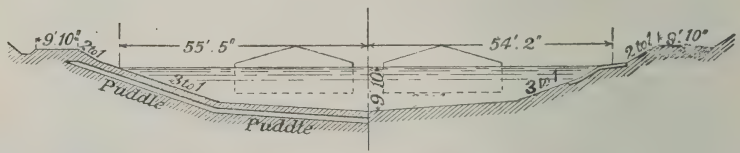


FIG. 14.—Berlin-Stettin Canal.

RELATIVE ADVANTAGES OF LOCKS AND LIFTS ON CANALS.

Changes of level in canals and canalised rivers are effected by (1) locks, and (2) mechanical lifts, the latter being either (*a*) vertical or (*b*) on an inclined plane.

Locks are the simplest and most stable means of attaining the desired object. There is no question of their superiority for all differences of level up to 12 or 15 ft., and circumstances are generally favourable for their adoption for differences up to 25 or 30 ft., and even more. Among

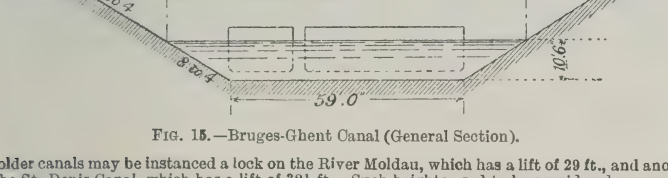


FIG. 15.—Bruges-Ghent Canal (General Section).

the older canals may be instanced a lock on the River Moldau, which has a lift of 29 ft., and another on the St. Denis Canal, which has a lift of 32½ ft. Such heights used to be considered as marking about the limit of the advantageous employment of locks over mechanical appliances. For

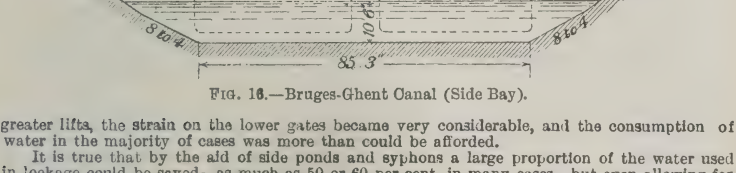


FIG. 16.—Bruges-Ghent Canal (Side Bay).

greater lifts, the strain on the lower gates became very considerable, and the consumption of water in the majority of cases was more than could be afforded.

It is true that by the aid of side ponds and syphons a large proportion of the water used in lockage could be saved—as much as 50 or 60 per cent. in many cases—but even allowing for this the loss was heavy and a mechanical lift was almost universally substituted.

In modern German practice, however, a very striking advance has been made owing to the development in size of vessels, which have now reached to a tonnage of 600 to 1,000 tons. A lock recently constructed at Henrichenburg has a lift of 46 ft. It is provided with five side ponds for the economisation of water, the saving amounting to about 76 per cent. The chamber (312 ft. long by 33 ft. wide with 9 ft. 10 ins. depth of water over sill) is built in concrete on a rock foundation requiring an invert only 19½ ins. thick. Other locks are being constructed which attain lifts of over 60 ft. The former limit therefore can no longer be said to apply, except in so far that, if it be exceeded, due provision will have to be made for the increase in strength of the gates and side walls, and for the economisation of water.

An appropriate test to apply to the serviceability of locks is the gradient or slope of the canal route. Where this is fairly regular and does not exceed 1 in 100 the rise lends itself to convenient subdivision into reaches of a quarter of a mile each, or more, separated by single locks of about 13 ft. lift. Under ordinary circumstances this is found to be a suitable standard for barge traffic, with barges of moderate tonnage, such as are customary in this country.

Where, on the other hand, the level of the ground changes abruptly, locks of higher lift or a series of locks may be employed, but only if an adequate supply of water is forthcoming to make good the loss from lockage. Where water is scarce it will be necessary to fall back upon some system of mechanical transfer.

Mechanical Lifts are generally more expeditious and time saving in action than locks, but they are also as a rule more expensive, and from their nature they are less durable and more liable to derangement.

From a number of examples of direct acting vertical lifts may be mentioned those at Anderton (Cheshire), Les Fontinettes (France), and Peterborough (Canada), *vide* figs. 17–19.

Lift at Anderton.—In this case the difference in level is 50 ft. The lift consists of two wrought-iron troughs, each 75 ft. long by 15½ ft. wide, originally worked hydraulically and counterbalancing one another, so that it was only necessary to reduce the depth of water in the lower trough by 6 ins. in order to set the apparatus in action. Owing to excessive wear and tear which developed in the hydraulic rams and cylinders, these were replaced in 1907–8 by a system of suspension gear passing over pulleys on an overhead gantry with counter weights at

the free ends of the ropes. The total weight so suspended is 1,000 tons, each of the caissons weighing 250 tons, of which 80 tons is iron and the remainder water. The reconstructed lift is worked electrically. The main 30 B.H.P. motor runs at about 750 revs. per min., and the large 6 ft. pulleys make about 1 revolution in 2 minutes.

Lift at Les Fontinettes, near St. Omer, France.—The lift consists of two iron troughs containing water and each capable of taking a boat of 300 tons. Each trough is bolted at its centre to the head of a ram or piston, working in an hydraulic press. The two presses are in intercommunication by means of a pipe and constitute an hydraulic balance, vertical movement being obtained by admitting a surcharge of water to the upper tank. The surcharge is 12 ins. in depth, equal to about 64½ tons weight. The difference in level is 43 ft. The troughs are 129 ft. 7 ins. long by 18 ft. 4½ ins. wide, with a minimum depth of water of about 7 ft. The ends are closed by lifting gates. The rams or pistons are 57 ft. long, 6 ft. 6¾ ins. diameter, and 2 ft. 8 ins. thick. They are formed in sections 9 ft. 2 ins. long, flanged inside, bolted together,

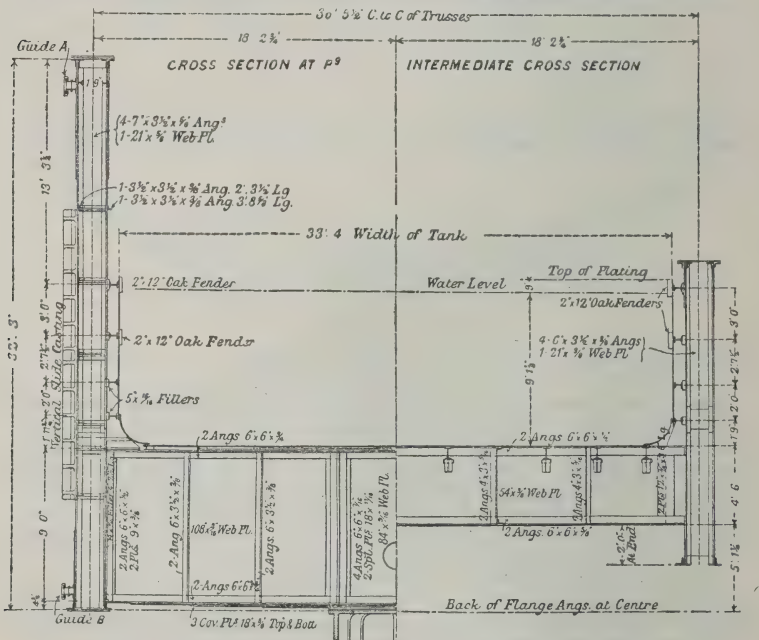


FIG. 17.—Cross-Sections of Canal Lift Lock at Peterborough, Canada.

and made water-tight by a ring of sheet copper inserted between the flanges. The hydraulic presses are 71 ft. high and 6 ft. 10 ins. diameter, resting upon concrete foundations at the bottom of excavated pits.

When one trough is raised to summit level there is a margin of 1½ ins. between its upstream extremity and the downstream end of the canal aqueduct. At the moment of lifting the gates to allow of the passage of a boat in or out, the joint is made good by indiarubber hose running round the end of the aqueduct and protected by springs. This hose is inflated by air at a pressure of 1½ atmospheres. The same connection is made at the lower level.

The machinery consists of two turbines, one of 50 H.P. and the other of 15 H.P.

The actual time of descent and ascent of the troughs is five minutes. An entire operation, including entry and exit of barges, takes twenty minutes.

The total cost of the lift was about 74,800*l.*

Lifts on the Trent Canal, Canada.—There are two large lifts, one at Peterborough and the other at Kirkfield. The former consists of a pair of tanks, or chambers, 137 ft. long by

33 ft. 4 ins. wide, and with 9 ft. depth of water. Each tank is supported by a ram 7 ft. 6 ins. diameter worked hydraulically, so that the descent of one chamber automatically forces up the other. The range of lift is 65 ft. Under ordinary conditions the upper chamber is held in position at a level about 6 or 8 ins. below the limit of stroke, and this surcharge of water is sufficient, when the by-pass is open, to overcome friction and give the necessary movement. The tanks are supported by openwork lattice trusses, and the total load of water and steel is carried to the top of the ram by four plate girders 9 ft. deep. The following is a description of the gates, which are in pairs, one closing the chamber end and the other the canal reach. 'Each gate is made up of three main 15-in. channels, with a small plate and angle girder at the top, and is stiffened vertically by 10-in. I beams. In every space between these vertical I beams there are placed two galvanised iron air chambers about 11 ft. long. These have the effect of making the gates practically buoyant, and this decreases markedly the amount of work which has to be done in raising the gates from a horizontal to a vertical position. The water-tight plating is, in each case, on the outside of the gate, and the hinges are at the bottom. In the case of the lower reach gates and chamber gates, the hinge for the reach gate is 2 ft. 4 ins. below the hinge of the chamber gate, and 1 ft. 10 ins. separates them in a longitudinal direction. Consequently, the chamber gate is approximately 10 ft. deep, while the reach gate is about 12 ft. 6 ins. deep. As it is never necessary to operate any gate singly, the gates are arranged to work in pairs. At

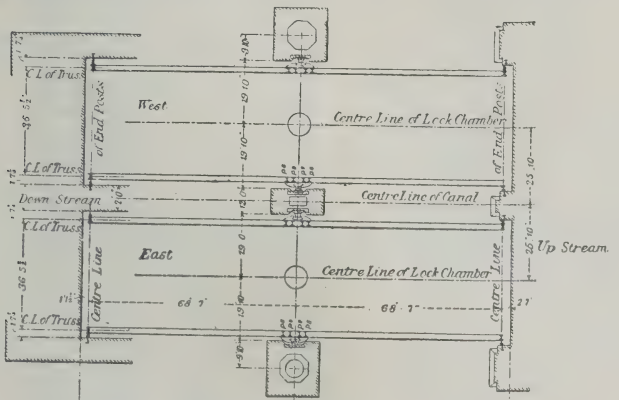


FIG. 18.—General Arrangement of Canal Lift Lock at Peterborough, Canada.

each end of the reach gate there is a pinion meshing into a circular rack set into the dock wall. The centre of the pitch circle of this rack is, of course, coincident with the centre of the hinge of the reach gate. The pinions above referred to are connected by a horizontal shaft running right across the gate, and power is applied to this shaft by means of a sprocket with chain, driven from another shaft, the centre of which is the same as that of the hinge. This shaft is actuated by one of the hydraulic gate engines provided.

'The seal between the reach-gate and the reach is accomplished by a small rubber strip, which lying over the adjacent surfaces is made water-tight by hydrostatic pressure. A similar arrangement is used for making the joints between chamber-gate and chamber water-tight. The seal between the end of the chamber and the end of the reach is accomplished by a collapsible rubber tube, which, when the chamber is away from the reach, lies almost flat against the end of the reach-gate frame. When, however, the chamber is in its proper position, the tube is inflated by compressed air at 90 lbs. per sq. in., and a water-tight joint is obtained.'

The hydraulic cylinders are plain steel castings, built up in flanged sections 5 ft. 3 ins. long. The inside diameter is 7 ft. 8½ ins., and the metal is 3½ ins. thick. The joint packing is of copper, ½ in. wide and $\frac{3}{16}$ in. thick, with $\frac{7}{8}$ in. diameter lead wire in press sections. The rams are of cast iron in flanged sections, also 5 ft. 3 ins. long, 90 ins. external diameter, thus giving a water space of 1½ in. completely round the rams. The jointing is the same as that for the cylinders, except that the lead ring is omitted.

A very full description of this lift appeared in *Engineering* of June 9 and 16, 1916, from which the above particulars were extracted.

Inclined Planes represent a class of mechanical appliances suitable for intermediate conditions, viz., a route gradient too steep for single locks with intermediate reaches, and not

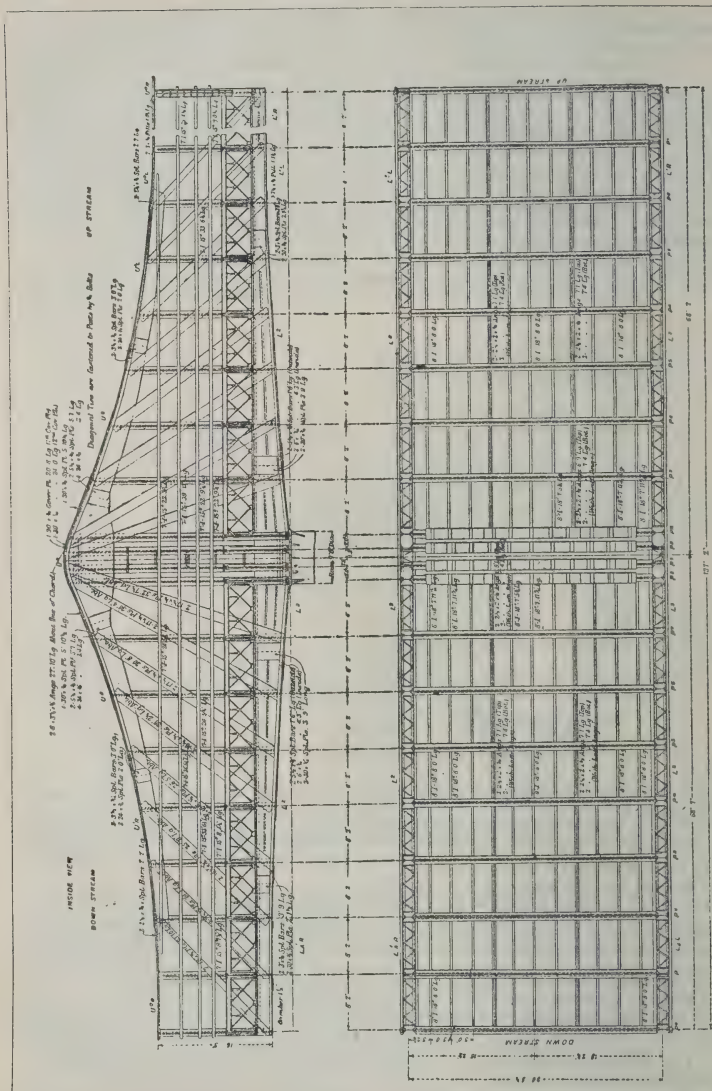


FIG. 19.—Internal View and Plan of Canal Lift Lock at Peterborough, Canada.

DIMENSIONS AND EQUIPMENT OF CERTAIN AMERICAN LOCKS.

	Length in feet.	Width in feet.	Depth at low water in feet.	Lift in feet.	Water Supply.	Valves.	Gates.	Gates operated by	Intermediate Gates.	Double Main Gates.	Means for bringing vessels into locks.
Allegheny River	287 to 298	55 to 56	8	7 to 12	Culverts in walls and under sills	Butterfly	Timber, steel	Hand, spar and cable	No	No	None.
Black Rock Harbour	650	70	22	5	Wall culverts	Stoney	Steel	Electric spar	Yes	No	Electric capstans.
Cascades of Columbia River	450	90	8	24	Wall culverts	Butterfly	Steel arched	Hydraulic cables	No	No	Hydraulic capstan.
Louisville and Portland (Old Lock)	310	80	—	11 to 22	Valves in gates	Butterfly	Timber	Steam chains	No	No	Winding engine.
Louisville and Portland (New Lock)	600	85	11	29	Culvert in walls	Butterfly	Steel	Air chain	No	No	Winding engine.
Monongahela River (New)	360	56	11	4 to 11½	Wall culverts	Butterfly	Steel	Air	No	No	None.
Ohio River	600	110	9½	9	Wall culverts	Butterfly	Steel rolling	Air or steam chain	No	No	None.
St. Mary's River (Weitzel Lock)	515	60 to 80	12½	17 to 20	Floor culverts	Butterfly	Timber	Hydraulic cables	No	No	None.
St. Mary's River (Poe Lock)	800	100	17·7	17 to 20	Floor culverts	Butterfly	Steel arched	Hydraulic cables	No	Yes	Hydraulic capstan.
St. Mary's River (New Lock)	1,350	80	24½	17 to 20	Floor culverts	Butterfly	Steel girder	Electric cables	No	Yes	Electric capstan.
St. Mary's River (Canadian Lock)	900	60	18	17 to 20	Floor culverts	Butterfly	Timber	Electric cables	No	Yes	None.
Panama Canal	1,000	110	40	30 to 60	Wall and floor culverts	Stoney and Cylindrical	Steel girder	Electric strut	Yes	Yes	Electric locomotive.

(*From Report by Col. H. F. Hodges, U.S. Army, to 12th Int. Nav. Cong., Philadelphia*, 1912.)

sufficiently abrupt at any one point for a vertical lift. Generally speaking, gradients of between 1 in 8 and 1 in 10 are most favourable, but they may run up to 1 in 75, or 1 in 80, and, on the other hand, inclined planes have sometimes been adopted where the rise is much steeper, as, for instance, at Foxton on the Grand Junction Canal, where the gradient was 1 in 4. The Foxton incline now out of use, was similar in character to the balanced vertical lift, consisting of two caissons travelling transversely, each maintaining a barge of 70 tons afloat. It is claimed for inclined planes that, in comparison with locks, they save water, and, in comparison with vertical lifts, they save time, since a boat is travelling forward simultaneously with its alteration in level. Moreover, they do not require such deep or expensive foundations as the lift proper. In spite of these advantages, inclined planes have not attained a very marked degree of acceptance among canal engineers.

Canal Curves.—The formula adopted in Belgium for the increased width at curves is

$$S = \frac{A^2 + a^2}{R}.$$

Where S is the increase in bottom width to be added to the normal semi-width on the convex side of the curve, R is the radius of the curve, and A and a are the half lengths of the boats which have to pass one another. The units are all in metres, and the value of S must therefore be multiplied by 3·28 to bring it to feet. The increased width is splayed into the straight length at a distance of 2 $(A+a)$ from the commencement of the curve.

LOSS OF WATER ON CANALS.

Rankine summarises the sources of loss of water on canals as follows :

Waste of Water by leakage of the channel, repairs and evaporation *per day* = area of surface of canal × ¼ of a foot, nearly.

Current from the higher towards the lower reaches, produced by leakage at the lock gates, *per day*, from 10,000 to 20,000 cubic ft. in ordinary cases.

Lockage.—Single locks are more favourable to economy of water than flights of locks. At a single lock, single boats ascending and descending, alternately, cause less expenditure of water than equal numbers of boats in trains. On the other hand, at a flight of locks, boats in trains cause less expenditure of water than equal numbers of boats ascending and descending alternately. For this reason where a long flight of locks is unavoidable, it is usual to make it double, using one flight exclusively for ascending boats and the other exclusively for descending boats.

Loss by Percolation from Erie Canal.
In Cubic Feet per Second per Mile of Canal.

Location.	Material in Bank.	Cub. ft. per Sec. per Mile of Canal.
Near Dunsbach Ferry	Gravel	1·68
Near East of Schenectady	Sandy loam	1·50
Near West of Schenectady	Gravel	2·24
East of Amsterdam	Gravel	3·03
West of Amsterdam	Coarse gravel	12·89
Canajoharie	Gravel	2·00
Fort Plain	Gravel	6·85
East of Utica	Sandy loam	4·45
East of Rome	Sand and gravel	8·48
West of Rome	Clay loam and gravel	1·75

Note.—The Erie Canal has bottom widths of 52½ ft. and 42 ft. and depths of 7 ft. and 9 ft. in various portions.

Canal Bank Protection in Italy.—The sandy banks along the Candiano Canal in the province of Ravenna are strengthened with a row of reinforced concrete piling. The king piles, 10 to 16 ft. long and 8 ins. square, are driven at intervals of 10 ft. They are secured together at the top by a longitudinal runner 8 ins. by 8 ins., behind which is driven a row of sheet piling from 5½ to 6½ ft. long and 2 ins. thick. The length of the piles depends upon the nature of the ground and the depth of water, and, where necessary, further security is obtained by anchorage tie rods and bars to short piles in the rear. At points where it was possible to remove the water from the canal, or where the bottom was occasionally exposed, it was considered desirable and economical to revet the banks with a surfacing of reinforced concrete 2½ ins. thick in which the reinforcement consisted of expanded metal ·4 to 1·6 in. mesh and ·2 to ·32 lbs. per sq. ft. Where the canal bottom is not subject to erosion during floods, this revetment is continued down as far as 18 ins. above the level of the bottom, and the toe is protected by rubble. In cases of erosive action there is a concrete foundation or sill, sometimes supported by piling ; and where the water level cannot be lowered sufficiently for the formation of a sill, the toe of the revetment rests against sheet piling.

TRACTION AND HAULAGE.

Means of Traction.—On ordinary small barge canals horse traction has hitherto remained the principal means of traction, although locomotives have been tried from time to time, generally without success. On the larger canals, principally those on the Continent and abroad, tug haulage is frequently employed and some vessels are self-propelled. Ship canals are navigated by vessels under their own steam, but for passage through the locks of the Panama Canal electric locomotives are provided.

Horse Traction on a straight reach is most effective at speeds between 2½ and 3 miles per hour. Within these limits a steady pull of between 100 and 125 lbs. can be maintained, which is sufficient for loads up to 60 tons. There is the further consideration that at higher speeds the wash against the banks becomes excessive for unprotected surfaces. Where tugs or locomotives are employed some form of bank protection becomes essential.

As regards river navigation Rankine gives the following formula as a rough estimate of the effect of the current of a stream on the load which one horse is able to draw against at a walk.

Load drawn against current = load drawn in still water $\times \left(\frac{3{\cdot}6}{3{\cdot}6 + v}\right)^2$; v being the velocity of the current in feet per second.

Cable Towage.—Lévy system, as adopted on St. Maur Canal between Paris and Joinville.—Cable weighs 2·45 lbs. per lin. ft. Set with permanent tension of five tons. Breaking strength 50 tons. Deflection due to haulage does not exceed 4 ins. Vertical supporting pulleys are 2 ft. 7½ ins. diameter, and have depth of groove of 7¾ ins. Circuit may cover distance of 10 to 11 miles, so that two driving machines may be set 20 to 22 miles apart. For a speed of 1½ miles per hour 2 H.P. is required to draw a barge loaded with 350 tons. For a speed of 2½ miles an hour 4½ H.P. is required. Tractive force $= \frac{Wv^2}{2gl}$, where W = weight of boat in lbs., v = velocity in ft. per sec., g = acceleration due to gravity = 32 ft. per sec. per sec., and l = length of towline in feet. For a boat of 300 tons a velocity of 3 ft. per sec. and length of line of 300 ft., the tractive force = 315 lbs. Estimated cost of plant, about 12*s.* 6*d.* per lin. yd. Cost of working ranges from ½*d.* to ¼*d.* per ton per mile.

(*Report of Dr. Watson to Water Commerce Congress at Chicago*, 1893.)

Electric Traction on an elaborate scale has been installed on the Teltow Canal in Germany since 1905. The tractors run on rails on both banks. The current is continuous at 550 volts, from overhead mains, and each locomotive is driven by two 8 h.p. motors, which are sufficiently powerful to haul two 600-ton barges at a speed of 2½ miles per hour. A similar system is in use on the Douai section of the Canal d'Aire, where the tractors are of 10 h.p. In 1918, some petrol locomotives of 20–25 h.p. were introduced on the Rhine-Rhone Canal in place of horses, but though the experiment was successful the system did not come into general use on the Canal because it would have entailed considerable alteration in the permanent works.

The following advantages among others have been claimed for the use of electric tractors in place of steam tugs:—

1. Assuming that the electric lines are laid on each bank, there will be greater regularity and uniformity of speed in the service than where tugs are used, and there will be less delay in despatching cargoes at short notice and at regular intervals.

2. The banks and bottom of a canal are not so much injured by electric-towing as by tug-towing. In the latter case the peculiar disturbance to the bottom, with consequential injury to the side banks, caused by the motion of the screw, is added to the wave from the moving boats common to each mode of transport.

3. In the case of electric towing there is considerable economy of space and water in locks and lifts, because tugs and steam boats, which, by reason of their machinery, have a smaller cargo-carrying capacity than barges of like dimensions, need not be taken through.

4. Electric power, when once installed along a line of waterway, can be utilised not only for traction, but also for operating locks and other works, and lighting the waterway at night, and can even, perhaps, be utilised for factories establishing themselves along the waterway.

(*Report of Royal Commission on Waterways*, 1909.)

In 1924 there was installed, on a short section of the canal from the Marne to the Rhine, a new system of electric traction known as the Chenean system. The underlying principle is the utilisation of an electric tractor, with proportional contact, travelling on an aerial cable. Owing to the light weight of the tractor (about 12 cwts.) it is possible to space the supports of the aerial cable, in general, at about 20 yards distance. The cable is of steel, 9 in. diameter, weighing 1¾ lbs. per lineal ft. The speed of movement varies from 1 mile per hour in narrow cuttings, tunnels, etc., to 2 miles per hour and over in the open.

SHIP CANALS.

General Considerations.—It is desirable that the ratio of cross-sectional area of the canal to the immersed section of ships, where the traffic is commercial, should not be less than 4 to 1, and it may with advantage be much higher. In the Suez Canal it is about 7 to 1. For vessels of war, the ratio may be slightly less, as, for instance, in the Kiel Canal, where the ratio is about 3½ to 1. The limiting value is, however, dependent on the speed at which vessels are permitted to travel. In the artificial portion of the Suez Canal the speed is limited to between 6 and 7 knots, and about 8 knots represents an approximate upper limit for narrow shallow waterways. In passing through such waters vessels sink to a lower level than in the open sea on account of the displacement. Thus pilots have a rough rule that when a ship is near the bottom her 'squat' or 'sinkage' in feet may be about one-fifth of her speed in statute miles per hour.

For curves, M. Quellenec, Consulting Engineer to the Suez Canal Commission, recommends widening the bottom, so that the longest vessel may preserve in its own direction a free space equal to 1½ times its own length from the slope of the concave side. The formula used on the canal was:

$$W = \sqrt{R^2 + (2a)^2} - (R + b);$$

where,

W = amount of widening in feet;
R = radius of axis in feet;
a = length of ship in feet;
b = ½ bottom width (on tangent) of canal in feet.

Kiel, or North Sea and Baltic Canal.—This canal, which connects the North Sea and the Baltic, was originally constructed between 1887 and 1895 at a cost of about 7,800,000*l.*, including land purchase. It has a length of 61½ miles. The former dimensions have since been very considerably increased—viz. the bottom width from 72 ft. to 144½ ft., the surface width from 226 ft. to 334½ ft., the depth from about 30 ft. to 36 ft., and the cross-sectional area from about 4,500 sq. ft. to 8,868 sq. ft.

At the entrances at each end of the canal there are now twin locks of 1,083 ft. available length, by 147½ ft. clear width and 45 ft. 2 ins. sill depth at mean water level. These locks take the place of the earlier locks, which were 492 ft. long by 82 ft. wide with 32 ft. of water over sill. The new locks are constructed in concrete, and each has three pairs of caissons dividing the chambers into lengths of 725 ft. and 330 ft. The improvement works cost 11,000,000*l.*, and were completed in 1914, one month before the war.

Manchester Ship Canal.—The Manchester Ship Canal was constructed between 1887 and 1893. It has a total length of 36 miles and includes four sets of locks, with an average

MANCHESTER SHIP CANAL: TYPE SECTIONS.

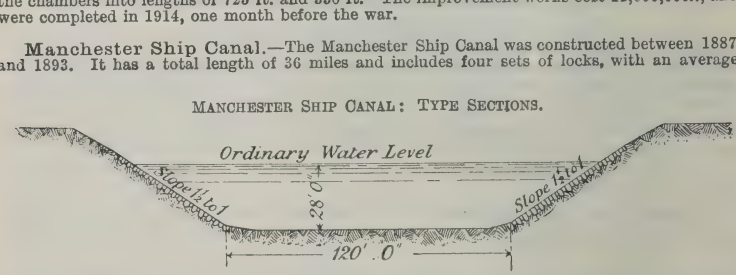


FIG. 20.—Ordinary Section.

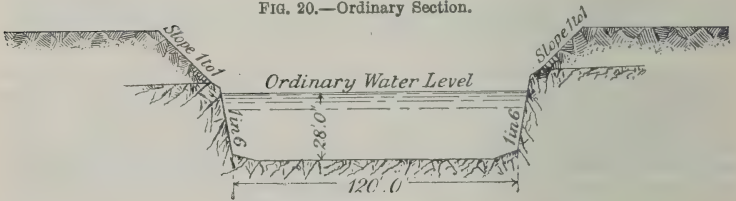


FIG. 21.—Section through Rock.

lift of 15 ft. each, about 160 acres of docks with warehouses and railway sidings, six high level bridges, seven swing bridges, and a swing aqueduct. The cost was 13,500,000*l.*, including land purchase and promotion, the actual constructional cost being about 9,000,000*l.* The capital of the company has since been increased on account of extensions and new works to 19,500,000*l.* The bottom width is 120 ft. The original depth was 26 ft., but this has now been increased to

28 ft. Deepening operations are being continued. A depth of 30 ft. has been extended from Eastham Locks as far as Stanlow Oil Dock, a distance of 4½ miles. The average annual cost of maintenance, which, before the war, was stated to be about 2,000*l.* per mile of canal, in recent years has increased to 6,000*l.* per mile.

Suez Canal.—The length of the Suez Canal is about 100 miles. It was opened in 1869, the cost of construction, apart from promotion, etc., having amounted to 10,680,000*l.* It has been progressively deepened, so that the authorised maximum draught for ships in the canal has been as follows:—

From 1870 to 1890	7 m. 50 = 24 ft. 7 ins.
,, 1890 to 1902	7 m. 80 = 25 ft. 7 ins.
,, 1902 to 1906	8 m. 00 = 26 ft. 3 ins.
,, 1906 to 1908	8 m. 23 = 27 ft. 0 ins.
,, 1908 to 1914	8 m. 53 = 28 ft. 0 ins.
,, 1914 to 1915	8 m. 84 = 29 ft. 0 ins.
,, 1915 to 1920	9 m. 15 = 30 ft. 0 ins.
,, 1921 to 1924	9 m. 45 = 31 ft. 0 ins.
,, 1925 to 1929	9 m. 75 = 32 ft. 0 ins.
,, 1930	10 m. 00 = 33 ft. 0 ins.

The canal is available throughout for vessels drawing 33 ft. of water. The present depth of the canal below the mean level of H.W.O.S.T. is equal at least to 38 ft. or more, over nearly the whole length, and is nowhere less than 36 ft.; dredging works are in progress to increase the depth later on to 41 ft. The width of the canal at a depth of 33 ft. is 147 ft., and 197 ft. in the widened portions. This latter width will be realised over the whole length in a few years' time. The surface width depends on the inclination of the banks, which changes with the nature of the ground through which the canal passes. This width actually varies from 350 ft. to 500 ft. in ordinary parts of the canal, and from 400 ft. to 500 ft. in the widened portions.

Of the total length of 99·86 miles, 90·08 miles are in straight reaches; 3·48 miles in curves of 3,280 to 3,500 yds. radius; 4·97 miles in curves of 2,187 to 3,270 yds. radius; 1·11 miles in curves of 1,105 to 1,930 yds. radius; and ·22 mile with a curve of 321 yds. radius. These curves are in process of being modified so that the minimum radius shall be over 2,200 yds.

SUEZ CANAL.

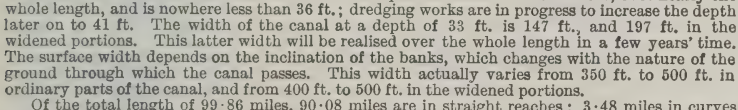


FIG. 22.—Section of Suez Canal, showing progressive deepening, as completed, in course of execution, and contemplated.

There are no locks, the canal being level throughout. At the Suez entrance there is a tidal range, amounting to about 6 ft. in the average at springs. The current flow does not exceed about 3½ ft. per second.

Panama Canal.—The total length of the Panama Canal, measured along its axis, is 50½ miles, but the portion actually within land area between coast lines is only 41½ miles. Of the total length, 14½ miles are at sea level, over 23½ miles in Gatun Lake, nearly 3 miles in locks or alongside approach walls, 1½ miles in Miraflores Lake, and 8 miles in the Gaillard Cut. The

DIAGRAMMATIC SECTIONS OF PANAMA CANAL. (Not to Scale.)

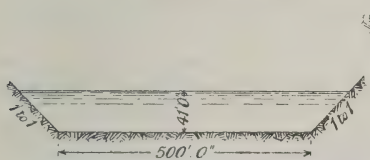


FIG. 23.—From Mindi River to Gatun Locks, and from Miraflores to Panama Bay.

FIG. 24.—Gaillard (formerly Culebra) Cut.

FIG. 25.—From Gatun Locks to Angle North of San Pablo.

summit level is 85 ft. above sea level. The depth lies between 40 and 45 ft. The bottom width varies as follows: For 16 miles the canal is 1,000 ft. wide; for 4 miles, 800 ft.; for 22 miles, 500 ft.; and for 8 miles, 300 ft. wide.

There are six pairs of locks with dimensions of 1,000 ft. in serviceable length and 110 ft. in width; three of these are at Gatun, one at Pedro Miguel, and two at Miraflores. The sills are laid so as to afford a depth of 40 ft. in salt water, and 41 ft. 4 ins. in fresh water. With the exception of the lower lock in the Miraflores flight, the chambers are provided with intermediate gates, dividing them into two sections of 600 ft. and 400 ft. The range of lift of the Pedro Miguel Lock is 30½ ft. All the locks are double, i.e. with parallel chambers, so that vessels may pass in opposite directions. At the Gatun flight the lift is 85 ft. divided among the three locks; the actual operating pressure between consecutive locks is therefore nearly 60 ft.

There are 46 pairs of lock gates, constructed of mild carbon steel with yokes and pintles of vanadium steel. The height of the gates varies from 46 ft. to 82 ft. The largest gate weighs about 700 tons.

The approximate cost of the canal, including purchase money, is 75,000,000*l.*

The following is a statement of the unit cost of excavating the Culebra (now Gaillard) Cut for the year 1912:

	Cost per cu. yd.
Transportation	$0·1331
Drilling and blasting	·1157
Tracks	·0885
Loading by steam shovels	·0681
General expenses and supervision	·0503
Dumps	·0479
Plant, arbitrary	·0395
Drainage, structures and clearing	·0045
Total unit cost	$0·5476

Bruges Ship Canal.—The canal connecting Bruges (Belgium) with the sea was constructed between 1896 and 1907. The works comprised an outer harbour with breakwater at Zeebrugge, and a sea-lock with a chamber 518 ft. long, 65½ ft. wide, the sill being laid at 19 ft. 8 ins. below L.W.O.S.T., giving a depth of 31 ft. 8 ins. at W.H.O.N.T. The canal proper is 6¼ miles long, 72 ft. wide at the bottom, and 229 ft. wide at water surface level. There is a depth of 26 ft. throughout, and the side slopes are about 3 to 1. Two railway and three road bridges cross the canal. At Bruges there are two basins and a lock connecting with the internal barge canal system of Belgium.

Amsterdam, or North Sea Canal.—The distance from Amsterdam to the North Sea on the line of the canal is 15½ miles, and the route traverses two large shallow lakes. As originally constructed, the canal had a depth of 23 ft. and a bottom width of 88 ft. 7 ins. The old entrance locks were three in number, the largest having an available length of 738 ft., a width of 82 ft., and a depth on sill of 33 ft. 6 ins. at ordinary low water. A new lock, now in service at Ymuiden, has a length of 1,313 ft., a breadth of 164 ft., and a depth of 49 ft. The canal itself is designed to be deepened to 49 ft., and to have a bottom width of 328 ft.

New Welland Canal, Canada.—The New Welland Canal was begun in 1913. Work was interrupted from 1917 to 1919 on account of the war. It is at the point of completion and the opening will take place this year (1931). It is estimated that over $115,000,000 will have been spent on the work.

The present Welland Canal locks overcome the 325·5 ft. difference in elevation between Lake Ontario (elevation 247) and Lake Erie (elevation 572·5) in 26 lifts. The locks have a usable length of 270 ft. and clear width of 45 ft., and depth over sills of 12–14 ft. The present canal cost $31,031,226 for construction, and $9,237,961 for maintenance to 1920.

The new canal includes seven locks to overcome the total lift of 325·5 ft., or 46·5 ft. lift per lock. Two additional guard locks are installed, one at each end of the canal, to provide for high and low water differences in the level of the lakes. The locks have a usable length of 820 ft., a clear width of 80 ft., and a depth over sills of 30 ft.

The new Welland Canal is to be excavated to 25 ft. in depth now, and later to 30 ft. When completed it will provide clearances, so that any ship passing through the lock at Sault Ste. Marie can continue to Lake Ontario. Later the Canadian Government expects to provide a waterway of equal dimensions from Lake Ontario down the St. Lawrence River. There is a difference of elevation between Lake Ontario and the river at Montreal of 221 ft.—(*Extract from Report by Edward H. Schulz to the 13th International Congress of Navigation, London*, 1923, *modified by later information.*)

SECTION XXX

STEAM ENGINEERING.

Revised by T. B. Morley, D.Sc., M.I.Mech.E.

PART I

HEAT AND WORK — STEAM — PROPERTIES OF STEAM — FLOW OF STEAM . . . . . pp. 1531–1539

PART II

STEAM GENERATING PLANT . . . . . pp. 1541–1704

PART III

RECIPROCATING STEAM ENGINES . . . . pp. 1705–1765

PART IV

(Contributed by H. L. Guy, A.M.Inst.C.E., M.I.Mech.E. and L. S. Robson.)

STEAM TURBINES . . . . . pp. 1767–1823

PART V

STEAM CONDENSERS—AIR PUMPS—FEED PUMPS—COOLING TOWERS—STEAM ACCUMULATORS . . . . pp. 1825–1858

SECTION XXX

PART I

HEAT AND WORK—STEAM—PROPERTIES OF STEAM—FLOW OF STEAM—POWER PLANT THERMAL EFFICIENCIES.

(Revised by T. B. Morley, D.Sc., M.I.Mech.E.)

1 British Thermal Unit			= 778 ft.-lbs.	2,545 British Thermal Units	= 641 kg. cals.
			= 0·252 kg. calorie.		= 746 watts.
			= 107·7 kg. metres.	1·34 h.p. hours	= 1 k.w. hour
3·968	,,	,, Units	= 1 kg. cal.		= 3,415* B.Th.U.
2,545	,,	,, ,,	= 1 h.p. hour (British).		= 860·6 kg. cals.

STEAM.

When water is converted into steam by the addition of heat no steam is produced until the temperature of evaporation, or boiling point, is reached. This temperature increases with, and is determined by, the pressure. Whilst evaporation is taking place the temperature remains unchanged, and for this reason the heat required during this process is called the 'latent' heat. After the water has all been converted into steam, the addition of further heat raises the temperature and is said to superheat the steam. These general remarks apply to the vapours of all liquids, but the temperatures of evaporation at equal pressures vary widely with different liquids.

If steam contains liquid moisture mixed with it, it is said to be wet. The percentage of actual steam to total steam and moisture is the dryness fraction.

When the steam is not superheated, but yet contains no moisture, it is dry and saturated. The degree of superheat is measured by the rise of temperature above that of saturated steam.

The heat required to superheat steam varies with the pressure and temperature (see Table IV., p. 1535), but is roughly 0·55 B.Th.U. per pound of steam for each degree Fah. of superheat.

Superheating does not alter the pressure of the steam, but increases its volume in proportion (roughly) to the amount of superheat.

The volume of superheated steam may be determined approximately from the equation

$$V = v(1 + \cdot 0015 t_s)$$

where,

v = volume of dry saturated steam at the same pressure; t_s = degrees of superheat Fah.

For the mean specific heat, heat required to superheat, and total heat of superheated steam, see Tables III., IV., V., pp. 1534, 1535.

One cubic foot of water at 70° F. weighs 62·3 lb. At 212° the volume of this weight of water will be 1·04 cub. ft., an increase of 4 per cent. If the water is turned into saturated steam at 100 lb. gauge pressure, its volume will be 242 cub. ft. At 212° the volume of this weight of saturated steam will be 1,670 cub. ft., and at an absolute pressure of 1 in. of mercury it will be 41,000 cub. ft.

Steam Tables.

Prof. Callendar has established equations correlating the principal properties of steam, dry and saturated, superheated and supersaturated. His tables, based on these equations, agree well with the most reliable experimental observations and are also thermodynamically consistent.

In the following Table I., the total heat, volume, and total entropy of steam are abstracted from Callendar's Tables, published by Edward Arnold, London. The remaining data in Table I. and the other following tables have been calculated on the basis of Callendar's figures.

* British Engineering Standards Association 'Glossary of Terms used in Electrical Engineering.' The value 3,415 is based on the British Thermal Unit, which is defined as the quantity of heat required to raise the temperature of 1 lb. of water from 60° F. to 61° F., and is equal to 1·054 joules. See also p. 2204.

TABLE I.—PROPERTIES OF SATURATED STEAM (BASED ON CALLENDAR'S VALUES).

Absolute Pressure. Lbs. per Sq. In.	Temperature. ° F.	Water Heat from 32° F.	Latent Heat of Vapourisation.	Total Heat per Lb. B.Th.U.	Volume, Cub. Ft. per Lb.	Entropy of Water.	Total Entropy of Steam.	Absolute Pressure. Lbs. per Sq. In.
0·5	79·5	47·4	1045	1092	650·5	·092	2·0299	0·5
1	101·7	69·5	1032	1102	333·1	·133	1·9724	1
2	126·1	93·9	1020	1114	173·5	·175	1·9159	2
3	141·5	109·3	1012	1121	118·6	·201	1·8833	3
4	153·0	120·8	1005	1126	90·5	·220	1·8600	4
5	162·3	130·1	1000	1130	73·4	·235	1·8422	5
6	170·1	137·9	995	1133	61·9	·248	1·8277	6
7	176·9	144·9	991	1136	53·6	·258	1·8156	7
8	182·9	150·8	988	1139	47·3	·267	1·8049	8
9	188·3	156·2	985	1141	42·4	·276	1·7956	9
10	193·2	161·1	982	1143	38·4	·282	1·7874	10
11	197·8	165·7	979	1145	35·1	·290	1·7799	11
12	202·0	169·9	977	1147	32·4	·297	1·7731	12
13	205·9	173·9	974	1148	30·0	·303	1·7669	13
14	209·6	177·6	972	1150	28·0	·309	1·7611	14
14·7	212·0	180·0	970	1151	26·8	·312	1·7573	14·7
15	213·0	181·0	970	1151	26·2	·314	1·7557	15
16	216·3	184·3	968	1152	24·7	·319	1·7506	16
17	219·5	187·5	966	1154	23·4	·324	1·7458	17
18	222·4	190·5	964	1155	22·2	·328	1·7414	18
19	225·2	193·3	963	1156	21·1	·332	1·7373	19
20	228·0	196·1	961	1157	20·1	·336	1·7333	20
21	230·6	198·7	959	1158	19·19	·339	1·7294	21
22	233·1	201·3	958	1159	18·37	·343	1·7258	22
23	235·5	203·8	956	1160	17·62	·347	1·7223	23
24	237·8	206·1	955	1161	16·93	·350	1·7189	24
25	240·1	208·4	954	1162	16·29	·353	1·7157	25
26	242·2	210·5	952	1162·5	15·70	·356	1·7126	26
27	244·4	212·8	950	1163	15·17	·359	1·7097	27
28	246·4	214·8	949	1164	14·67	·362	1·7069	28
29	248·4	216·8	948	1165	14·19	·365	1·7042	29
30	250·3	218·7	947	1166	13·74	·367	1·7016	30
35	259·3	228	941	1169	11·90	·381	1·6897	35
40	267·2	236	936	1172	10·50	·392	1·6792	40
45	274·4	243	931	1174	9·40	·402	1·6702	45
50	281·0	250	926	1176	8·52	·411	1·6620	50
55	287·0	256	922	1178	7·79	·420	1·6547	55
60	292·6	262	918	1180	7·18	·427	1·6479	60
65	297·9	268	914	1182	6·66	·434	1·6417	65
70	303·0	273	911	1184	6·22	·441	1·6359	70
75	307·5	278	907	1185	5·82	·447	1·6304	75
80	312·0	282	904	1187	5·49	·453	1·6256	80

TABLE I.—PROPERTIES OF SATURATED STEAM (*continued*)

Absolute Pressure. Lbs. per Sq. In.	Temperature. ° F.	Water Heat from 32° F.	Latent Heat of Vapourisation.	Total Heat per Lb. B.Th.U.	Volume, Cub. Ft. per Lb.	Entropy of Water.	Total Entropy of Steam.	Absolute Pressure. Lbs. per Sq. In.
85	316·2	286	901	1187	5·18	·459	1·6209	85
90	320·2	290	899	1189	4·91	·464	1.6165	90
95	324·1	294	896	1190	4·67	·469	1 6123	95
100	327·7	298	893	1191	4·45	·474	1·6082	100
105	331·3	302	890	1192	4·25	·479	1·6044	105
110	334·7	306	887	1193	4·07	·483	1·6007	110
115	338·0	309	885	1194	3·90	·487	1·5972	115
120	341·1	312	883	1195	3·75	·491	1·5938	120
125	344·2	316	880	1196	3·61	·495	1·5906	125
130	347 2	319	878	1197	3·48	·499	1·5875	130
135	350·1	322	875	1197	3·36	·503	1·5846	135
140	353·0	325	873	1198	3·24	·507	1·5818	140
145	355·7	328	871	1199	3·14	·511	1·5791	145
150	358·4	330	869	1200	3·04	·514	1·5765	150
155	361·0	333	867	1200	2·95	·517	1·5740	155
160	363·5	336	865	1201	2·86	·520	1·5715	160
165	366·0	339	863	1202	2·78	·523	1·5691	165
170	368·4	341	861	1202	2·70	·527	1·5666	170
175	370·7	344	859	1203	2·63	·530	1·5643	175
180	373·1	346	857	1203	2·56	·533	1·5620	180
185	375·3	348	855	1204	2·50	·535	1·5598	185
190	377·6	351	853	1204	2·43	·538	1·5577	190
195	380	353	851	1205	2·38	·541	1·5557	195
200	382	356	849	1205	2·32	·544	1·5538	200
205	384	358	848	1206	2·266	·546	1·5520	205
210	386	360	846	1206	2·216	·549	1·5502	210
215	388	362	845	1207	2·167	·551	1·5483	215
220	390	364	843	1207	2·120	·554	1·5465	220
225	392	366	842	1208	2·076	·556	1·5447	225
230	394	368	840	1208	2·034	·559	1·5429	230
235	396	370	839	1209	1·993	·561	1·5412	235
240	398	372	837	1209	1·954	·564	1·5395	240
245	399	373	836	1209	1·916	·566	1·5379	245
250	401	375	835	1210	1·880	·568	1·5362	250
265	406	382	829	1211	1·780	·574	1·5317	265
315	422	399	815	1214	1·511	·594	1·5179	315
365	436	415	802	1217	1·315	·611	1·5063	365
415	449	429	790	1219	1·165	·627	1·4961	415

It has been established that under certain conditions of expansion, as in turbine nozzles, steam may for a time not be in stable thermal condition, continuing to behave like a vapour, although for equilibrium at the prevailing pressure and temperature a portion of it must become condensed. In other words, condensation is temporarily delayed. Steam in this condition is said to be 'supersaturated,' or, since its temperature is lower than the saturation temperature corresponding to the pressure, to be 'supercooled.' This condition is important in connection with the study of steam turbines.

Atmospheric pressure in this country is taken as 14·69 lbs. per square inch. At this pressure water boils at 212° Fahr., and has a latent heat of 970 British Thermal Units (B.Th.U.) per pound.

Steam at a pressure of 459 lbs. per sq. in. absolute, has a temperature of 459° F.

Gauge pressures are measured above atmospheric and are approximately 15 lbs. per square inch below the absolute pressures.

One inch of mercury column at 32° Fahr. is equal to 0·491 lb. per sq. in. Reduce one-hundredth per cent. for each extra degree Fahr. of mercury column temperature.

TABLE II.—TEMPERATURES OF STEAM AT HIGH VACUA.

Ins. mercury .	29·6	29·5	29·4	29·3	29·2	29·1	29·0	28·9	28·8
Absolute pressure	0·196	0·244	0·293	0·342	0·391	0·440	0·489	0·538	0·587
Temp. Fah. .	52·5	58·5	63·7	68·2	72·1	75·7	78·9	81·8	84·5
Ins. mercury .	28·7	28·6	28·5	28·4	28·3	28·2	28·1	28·0	27·9
Absolute pressure	0·636	0·685	0·733	0·782	0·831	0·880	0·829	0·978	1·027
Temp. Fah. .	87·0	89·3	91·5	93·6	95·6	97·5	99·3	101·0	102·5
Ins, mercury .	27·8	27·7	27·6	27·5	27·4	27·3	27·2	27·1	27·0
Absolute pressure	1·076	1·125	1·174	1·222	1·271	1·320	1·369	1·418	1·467
Temp. Fah. .	104·1	105·6	107·1	108·5	110·0	111·3	112·6	113·8	115·0
Ins. mercury .	26·9	26·8	26·7	26·6	26·5	26·4	26·3	26·2	26·0
Absolute pressure	1·516	1·565	1·614	1·663	1·711	1·760	1·809	1·858	1·956
Temp. Fah. .	116·1	117·2	118·3	119·4	120·4	121·3	122·3	123·3	125·2

NOTES.—Mercury column at 60° F. Referred to a 30-in. barometer. Pressure in lbs. per sq. in. absolute. Height of mercury gauge at a given pressure increases one-hundredth per cent. per deg. F. of mercury temperature above 60. At 60° F. 1 lb. per sq. in. pressure = 2·045 ins. mercury, and 1 in. mercury = 0·489 lbs. per sq. in.

TABLE III.—MEAN SPECIFIC HEATS OF SUPERHEATED STEAM FROM SATURATION TEMPERATURE.

Pressure, Lbs. per Sq. In. Absolute.	Superheat, degrees Fahrenheit.						
	50	100	150	200	250	300	350
20	·503	·500	·497	·495	·494	·492	·491
30	·512	·507	·503	·501	·499	·497	·495
40	·519	·514	·509	·506	·503	·501	·499
50	·525	·519	·515	·511	·508	·505	·502
60	·532	·525	·519	·515	·511	·508	·505
70	·537	·529	·524	·519	·515	·512	·509
80	·542	·534	·528	·523	·519	·515	·512
90	·547	·538	·532	·527	·522	·518	·515
100	·549	·542	·535	·530	·525	·521	·517
120	·560	·551	·543	·536	·531	·526	·522
140	·568	·558	·549	·542	·537	·531	·527
160	·575	·565	·555	·548	·542	·536	·531
180	·580	·570	·561	·553	·546	·540	·535
200	·587	·577	·566	·558	·551	·545	·540
250	·604	·591	·579	·570	·562	·555	·550
300	·618	·603	·591	·581	·572	·563	·556

Table IV. gives the extra heat, above the total heat of saturated steam given in Table I., contained in superheated steam from water at 32° Fahr. The heat added to the water and steam from any higher temperature is less than the value in the table by the number of degrees above 32 from which the heating of the water commences (approximate).

TABLE IV.—HEAT REQUIRED TO SUPERHEAT ONE POUND OF STEAM.

Pressure, Lbs. per Sq. In. Absolute.	Superheat, degrees Fahrenheit.						
	50	100	150	200	250	300	350
20	25·17	50·02	74·62	99·02	123·3	147·5	172·0
30	25·59	50·73	75·54	100·2	124·7	148·9	173·2
40	25·94	51·35	76·43	101·2	125·7	150·2	174·6
50	26·26	51·93	77·20	102·1	126·9	151·4	175·8
60	26·62	52·47	77·92	103·0	127·9	152·5	177·0
70	26·82	52·90	78·55	103·8	128·8	153·5	178·1
80	27·10	53·39	79·20	104·5	129·7	154·5	179·1
90	27·35	53·82	79·76	105·3	130·5	155·4	180·1
100	27·44	54·25	80·33	106·0	131·3	156·2	181·0
120	28·00	55·06	81·43	107·3	132·8	157·9	182·8
140	28·41	55·78	82·42	108·4	134·2	159·4	184·5
160	28·75	56·52	83·31	109·5	135·4	160·8	186·0
180	29·09	57·05	84·17	110·6	136·6	162·1	187·5
200	29·36	57·66	85·0	111·6	137·8	163·5	189·0
250	30·2	59·07	86·9	114·0	140·5	166·4	192·5
300	30·9	60·3	88·6	116·1	142·9	169·0	195·0

Example.—The total heat of steam at 160 lbs. absolute, superheated 150 degrees, from water at 102° F., is approximately as follows. (There is a slight inaccuracy owing to the specific heat of water varying a little, which can be discovered and checked from the third column of Table I.)

Total heat of saturated steam from 32°	=	1201 B.Th.U.
Deduct 102° – 32°	=	70 ,,
Net heat of saturated steam from 102°	=	1131
Add heat of 150° of superheat	=	83
Total heat as required	=	1214

TABLE V.—TOTAL HEAT OF SUPERHEATED STEAM.

Pressure, Lbs. per Sq. In. Absolute.	Superheat, degrees Fahrenheit.						
	50	100	150	200	250	300	350
20	1182·2	1207·1	1231·7	1256·1	1280·4	1304·6	1329·1
30	1191·2	1216·3	1241·1	1265·8	1290·2	1314·5	1338·8
40	1197·6	1223·1	1248·1	1272·9	1297·5	1321·9	1346·3
50	1202·7	1228·4	1253·7	1278·6	1303·4	1327·9	1352·3
60	1207·0	1232·9	1258·3	1283·4	1308·3	1332·9	1357·4
70	1210·5	1236·6	1262·2	1287·5	1312·5	1337·2	1361·8
80	1213·6	1240·0	1265·8	1291·0	1316·2	1341·1	1365·7
90	1216·4	1242·9	1268·8	1294·3	1319·5	1344·4	1369·1
100	1218·7	1245·5	1271·6	1297·2	1322·6	1347·5	1372·5
120	1223·1	1250·1	1276·5	1302·3	1327·8	1352·9	1377·9
140	1226·6	1254·0	1280·7	1306·6	1332·4	1357·6	1382·7
160	1229·7	1257·4	1284·3	1310·5	1336·4	1361·8	1387·0
180	1232·4	1260·4	1287·5	1314·0	1340·0	1365·5	1390·8
200	1234·8	1263·1	1290·4	1317·0	1343·2	1368·9	1394·4
250	1239·9	1268·8	1296·6	1323·7	1350·2	1376·1	1402·2
300	1244·0	1273·4	1301·7	1329·2	1356·0	1382·2	1408·1

Steam at Very High Pressures.

Increasing attention is now being directed to the use of steam generated at very high pressures and temperatures, and researches are being carried out to gain more accurate information in reference to the same.

PROPERTIES OF STEAM AT HIGH PRESSURES.

The following tables are extracted from a report on Professor Callendar's researches, published in 1920 by the Institution of Mechanical Engineers (see *The Engineer*, May 3, 1929):—

TABLE VI.—PROPERTIES OF SUPERHEATED STEAM IN FOOT-POUND FAHRENHEIT UNITS AT PRESSURE FROM 400 LB. TO 4,000 LB.

Temperature t on the Fahrenheit scale. Total heat H in B.Th.U. per pound. Volume V in cubic feet per pound. Entropy ϕ in B.Th.U. per pound per deg. Fah.

NOTE.—The first line under each pressure gives the values of H, V and ϕ for dry steam at the saturation temperature t_s.

P = 400 lbs. per sq. in.				P = 450 lbs. per sq. in.				P = 500 lbs. per sq. in.				P = 600 lbs. per sq. in.			
t	H	V	ϕ	t	H	V	ϕ	t	H	V	ϕ	t	H	V	ϕ
443·7	1211·8	1·1712	1·4933	455·4	1212·1	1·0419	1·4827	466·0	1212·3	0·9380	1·4732	485·4	1212·0	0·7805	1·4559
450	1216·0	1·1841	1·4983	500	1243·7	1·1296	1·5166	500	1237·4	1·0007	1·4998	500	1223·9	0·8055	1·4684
500	1249·8	1·2898	1·5345	550	1276·1	1·2192	1·5495	550	1271·2	1·0847	1·5341	550	1260·8	0·8822	1·5059
550	1281·0	1·3870	1·5664	600	1306·5	1·3033	1·5789	600	1302·4	1·1627	1·5643	600	1294·1	0·9513	1·5381
600	1310·4	1·4788	1·5947	650	1335·4	1·3836	1·6056	650	1332·1	1·2365	1·5916	650	1325·2	1·0159	1·5667
650	1338·8	1·5670	1·6209	700	1363·4	1·4609	1·6302	700	1360·5	1·3076	1·6167	700	1354·7	1·0773	1·5928
700	1366·3	1·6528	1·6451	750	1390·6	1·5361	1·6532	750	1388·1	1·3762	1·6400	750	1383·2	1·1362	1·6168
750	1393·0	1·7364	1·6677	800	1417·2	1·6100	1·6748	800	1415·1	1·4435	1·6619	800	1410·8	1·1937	1·6392
800	1419·4	1·8182	1·6890	850	1443·4	1·6825	1·6951	850	1441·5	1·5092	1·6824	850	1437·8	1·2498	1·6602

P = 700 lbs. per sq. in.				P = 800 lbs. per sq. in.				P = 900 lbs. per sq. in.				P = 1,000 lbs. per sq. in.			
502·1	1211·1	0·6672	1·4407	517·5	1209·1	0·5804	1·4269	531·3	1206·3	0·5120	1·4142	544·1	1203·4	0·4570	1·4021
550	1249·8	0·7361	1·4802	550	1237·7	0·6251	1·4558	550	1224·5	0·5371	1·4321	550	1209·9	0·4649	1·4084
600	1285·4	0·7997	1·5147	600	1276·3	0·6851	1·4931	600	1266·5	0·5953	1·4728	600	1256·1	0·5226	1·4532
650	1318·1	0·8578	1·5448	650	1310·8	0·7389	1·5249	650	1303·1	0·6459	1·5065	650	1295·1	0·5712	1·4892
700	1348·8	0·9126	1·5719	700	1342·7	0·7887	1·5531	700	1336·5	0·6921	1·5359	700	1330·0	0·6148	1·5200
700	1378·1	0·9647	1·5966	750	1373·0	0·8358	1·5786	750	1367·7	0·7354	1·5623	750	1362·4	0·6551	1·5473
800	1406·4	1·0149	1·6196	800	1402·0	0·8811	1·6022	800	1397·6	0·7767	1·5865	800	1393·0	0·6933	1·5721
850	1434·0	1·0640	1·6410	850	1430·2	0·9249	1·6240	850	1426·3	0·8165	1·6089	850	1422·4	0·7299	1·5950
900	1461·0	1·1121	1·6611	900	1457·6	0·9677	1·6446	900	1454·2	0·8552	1·6298	900	1450·8	0·7653	1·6163

P = 1,200 lbs. per sq. in.				P = 1,400 lbs. per sq. in.				P = 1,600 lbs. per sq. in.				P = 1,800 lbs. per sq. in.			
t	H	V	ϕ	t	H	V	ϕ	t	H	V	ϕ	t	H	V	ϕ
566·9	1196·1	0·3727	1·3798	587·1	1186·9	0·3111	1·3587	605·1	1176·0	0·2635	1·3386	621·5	1163·8	0·2255	1·3186
600	1232·7	0·4108	1·4148	600	1204·2	0·3264	1·3751	650	1235·6	0·3099	1·3933	650	1208·6	0·2565	1·3595
650	1277·9	0·4578	1·4565	650	1258·3	0·3746	1·4251	700	1285·6	0·3488	1·4374	700	1267·4	0·2972	1·4114
700	1316·5	0·4980	1·4909	700	1301·8	0·4133	1·4635	750	1327·1	0·3812	1·4724	750	1313·6	0·3292	1·4520
750	1351·3	0·5341	1·5201	750	1339·6	0·4470	1·4954	800	1364·0	0·4099	1·5023	800	1353·3	0·3567	1·4826
800	1383·8	0·5678	1·5463	800	1374·1	0·4777	1·5233	850	1397·9	0·4363	1·5287	850	1389·1	0·3815	1·5105
850	1414·5	0·5997	1·5702	850	1406·3	0·5063	1·5484	900	1429·8	0·4611	1·5525	900	1422·4	0·4046	1·5354
900	1444·0	0·6304	1·5923	900	1436·9	0·5336	1·5713	950	1460·0	0·4847	1·5745	950	1453·6	0·4262	1·5581
950	1472·3	0·6598	1·6129	950	1466·2	0·5597	1·5925	1000	1489·4	0·5076	1·5949	1000	1483·8	0·4471	1·5791
P = 2,000 lbs. per sq. in.				P = 2,400 lbs. per sq. in.				P = 2,800 lbs. per sq. in.				P = 3,200 lbs. per sq. in.			
636·5	1149·8	0·1936	1·2985	663·1	1115·1	0·1433	1·2561	685·4	1071·0	0·1029	1·2078	703·5	1006·2	0·0645	1·1463
650	1175·4	0·2099	1·3218	700	1193·8	0·1842	1·3253	700	1114·9	0·1224	1·2476	750	1158·7	0·1240	1·2769
700	1246·6	0·2543	1·3847	750	1263·9	0·2206	1·3849	750	1219·9	0·1691	1·3363	800	1250·4	0·1597	1·3510
750	1298·8	0·2869	1·4288	800	1316·8	0·2480	1·4274	800	1287·0	0·1991	1·3906	850	1312·8	0·1840	1·3997
800	1342·0	0·3137	1·4638	850	1360·3	0·2706	1·4613	850	1338·1	0·2218	1·4305	900	1362·2	0·2032	1·4366
850	1380·0	0·3374	1·4934	900	1398·8	0·2906	1·4900	900	1381·4	0·2411	1·4628	950	1404·0	0·2195	1·4670
900	1415·0	0·3592	1·5194	950	1433·7	0·3087	1·5154	950	1419·3	0·2580	1·4904	1000	1441·9	0·2343	1·4933
950	1447·1	0·3793	1·5429	1000	1466·7	0·3259	1·5383	1000	1454·6	0·2737	1·5149				
1000	1478·4	0·3988	1·5645												
P = 3,600 lbs. per sq. in.				P = 3,600 lbs. per sq. in.				P = 4,000 lbs. per sq. in.				P = 4,000 lbs. per sq. in.			
750	1067·1	0·0785	1·1953	900	1340·6	0·1729	1·4125	800	1140·7	0·0931	1·2487	900	1316·1	0·1477	1·3850
800	1203·5	0·1254	1·3054	950	1387·3	0·1890	1·4445	850	1246·4	0·1260	1·3316	950	1369·0	0·1642	1·4223
850	1282·8	0·1528	1·3673	1000	1428·3	0·2033	1·4730					1000	1413·8	0·1782	1·4535

TABLE VII.—PROPERTIES OF SATURATED STEAM IN FOOT-POUND FAHRENHEIT UNITS ON A PRESSURE BASIS.

Press., lb./sq. in.	Temp., ° F.	Volume, cu. ft./lb.	Entropy, B.Th.U. per lb. per ° F.		Total and latent heat, B.Th.U. per lb.		
p	t	V	$\phi\omega$	ϕs	h	L	H
400	443·8	1·1712	0·6212	1·4933	423·7	788·1	1211·8
450	455·4	1·0419	0·6357	1·4827	437·0	775·1	1212·1
500	466·1	0·9380	0·6487	1·4732	449·1	763·2	1212·3
600	485·4	0·7805	0·6722	1·4559	471·6	740·4	1212·0
700	502·1	0·6672	0·6929	1·4407	491·6	719·5	1211·1
800	517·5	0·5804	0·7117	1·4269	510·1	699·0	1209·1
900	531·3	0·5120	0·7288	1·4142	527·4	678·9	1206·3
1000	544·1	0·4570	0·7443	1·4021	543·6	659·8	1203·4
1200	566·9	0·3727	0·7734	1·3798	573·4	622·7	1196·1
1400	587·1	0·3111	0·7992	1·3587	601·3	585·6	1186·9
1600	605·1	0·2635	0·8230	1·3386	627·3	548·7	1176·0
1800	621·5	0·2255	0·8454	1·3186	652·4	511·4	1163·8
2000	636·5	0·1936	0·8669	1·2985	676·8	473·0	1149·8
2400	663·1	0·1433	0·9098	1·2561	726·3	388·8	1115·1
2800	685·4	0·1029	0·9556	1·2078	780·4	290·6	1071·0
3200	704·4	0·0645	1·0231	1·1463	861·7	144·5	1006·2

At about 3,650 lbs. per sq. in. absolute, steam is in the 'critical state.' This state has been defined as that in which the surface of separation between liquid and vapour disappears, the specific volume of liquid and vapour are the same, and the latent heat is zero.

Recent research shows that these conditions are not precisely fulfilled simultaneously, so that a new definition of the critical state is necessary.

The generation of steam at the critical pressure (followed by throttling to a lower pressure for use in a turbine), so that the conversion from liquid to vapour takes place without ebullition, is a characteristic feature of the recently developed 'Benson' high-pressure boiler.

Equivalent Evaporation.

The factor of equivalent evaporation 'from and at 212°' is found by dividing the actual total heat of steam as calculated or found from foregoing tables, by 970, which is the latent heat of steam, or heat of evaporation, at atmospheric pressure.

Example.—If the actual total heat is 1067, the factor is 1·1. This means that if the actual evaporation of a boiler is 8·0 lbs. of water per lb. of coal, the 'equivalent evaporation from and at 212°' is $8·0 \times 1·1 = 8·8$ lbs. This basis is widely used because it renders the performance of boilers accurately comparable on a uniform basis.

TO CALCULATE THE AMOUNT OF STEAM REQUIRED TO HEAT A GIVEN QUANTITY OF WATER.

G = gallons of water; T and t = temperatures, Fah., after and before heating; $10 \times G \times (T - t) =$ B.Th.U. added to the water; H = total heat per lb. of steam (column 5, Table I., p. 1532); $H - T + 32$ = heat given to water by each lb. of steam; W = lbs. of steam required; then,

$$W = \frac{10 \times G \times (T - t)}{H - T + 32}$$

Note.—One cubic foot of water contains 6·24 gallons. The volume of each lb. of steam is given in Table I.

Flow of Steam from an Orifice or Nozzle.

A jet of steam issuing from a vessel under pressure first contracts in section, and then expands if the drop in pressure is more than about 42 per cent. of the absolute pressure in the vessel. The absolute pressure at the narrowest section, which corresponds to the neck of a turbine nozzle, varies a little, but averages about 58 per cent. of the initial pressure for saturated steam, and about 55 or 56 per cent. for superheated steam. The velocity of the steam at this narrowest section is nearly constant except at very low pressures.

For typical values of velocity and discharges, see Part IV., Steam Turbines, p. 1767.

When the drop in pressure is less than 42 per cent. of the absolute initial pressure, the full velocity and contraction of section do not occur, and the above data do not hold (see Part IV., Steam Turbines, p. 1767).

Or the quantity discharged per minute may be approximately found by Napier's formula

$$W = 6\,a\,p \div 7,$$

where,

W = weight in pounds; a = area in square inches; p = absolute pressure.

The theoretical flow requires to be multiplied by $k = 0{\cdot}93$ for a short pipe, or $0{\cdot}63$ for a thin opening, as in a plate or a safety valve.

Where the steam flows into a pressure more than two-thirds the pressure in the boiler, then

$$W = 1{\cdot}74\,a\,k\,\sqrt{p_2\,(p_1 - p_2)}, \text{ approximately,}$$

where a, k are as above, and p_1, p_2 are initial and final pressures.

Power Plant Thermal Efficiencies.

At a discussion on Power Station Design at I.M.E.A. Convention, June 1920, the view was expressed by Selvey that 17,500 B.Th.U. per unit generated was then in sight, or 19·5 per cent. over-all efficiency; and Mr. Ferranti was of the opinion that 28 per cent. would be attained with steam plant, eventually.

It is claimed that during October and November 1926 the station thermal efficiency at Columbia Power Station, U.S.A., attained 27 per cent. In this station pulverised coal is used, there are economisers and air pre-heaters, boiler pressure is 600 lbs. per sq. inch, and the steam is superheated to 730° F. The power units are two 45,000 kw. turbines, with four Babcock and Wilcox boilers for each. Steam is bled from the turbines at three stages for feed heating. Steam leaving the H.P. turbines at 80 lbs. per sq. in. is reheated to 700° F. before it passes to the L.P. turbines.

The boiler plant efficiency is over 89 per cent. and the load factor is exceptionally high, from 60 to over 80 per cent.

(*The Engineer*, Sept. 23, 1927.)

Fuel Consumption and Thermal Efficiency of Steam Electric Power Stations.*

The reports on the output and fuel consumption of steam electric power stations in Great Britain, published by the Electricity Commissioners, classify the stations into thirteen groups according to output. For four of these groups, representing the largest, the smallest, and two intermediate sizes of station, the following figures are abstracted from the report for the year ending March 31, 1930:—

Output of Station. Units in Year.	Over 200,000,000.	Between 10,000,000 and 25,000,000.	Between 500,000 and 1,000,000.	Under 50,000.
Number of stations in group . .	11	59	14	10
Total units generated by steam plant .	3,299,385,928	920,965,299	8,829,513	141,853
Total fuel consumed (coal, coke or oil) . . . . . tons	2,498,450	1,024,666	32,263	4,824
Average fuel consumption per unit generated (converted to basis of coal) . . . . . lbs.	1·69	2·47	8·13	—

		1929 Report.	1930 Report.
Total output of all steam stations .	units	10,377,107,062	11,436,600,000
Total coal	tons	9,516,419	10,096,210
Over-all average fuel consumption.	lbs. per unit	2·05	1·97

The lowest fuel consumption recorded for any of the power stations was 1·28 lbs. per unit, corresponding to a thermal efficiency of 22·83 per cent. This was obtained at the North Tees station of the Newcastle Electric Supply Co.

* *Generation of Electricity in Great Britain, Year ending March* 31, 1930. Electricity Commission, H.M. Stationery Office, Adastral House, Kingsway, London, W.C. 2. Price 1*s.* 6*d.* net.

PREVENT OIL LEAKAGE

LANGITE is a highly resilient jointing material which contains no rubber, and resists oils, spirits, benzol and petrol. It is supplied already cut to any shape and in any thickness, plain or reinforced, to any desired special cross-section; or in sheet, strip or block form for users to cut from. Owing to its extraordinary resilience, Langite makes a sound joint without demanding heavy mechanical pressure between the joint faces, and may be used upon uneven or unmachined joints faces.

Langite joint washers are used upon marine engines, commercial and private motor vehicles, on electric control gear and transformers, upon aircraft, in pipe-joints, for oil-retaining rings on running shafts, upon coachwork as a resilient mounting for glass; and for a wide variety of other purposes.

THE CORK MANUFACTURING
COMPANY, LIMITED.
South Chingford, London, E. 4.
Telephone: Silverthorn 1536

LANGITE
JOINTING MATERIAL
RESISTS OILS, SPIRITS, PETROL, BENZOL

F. 1541

SECTION XXX

PART II

STEAM GENERATING PLANT.

BOILER DESIGN—TRANSMISSION OF HEAT IN BOILERS — BOILER EFFICIENCY — BOILER SHELLS — STANDARD CONDITIONS FOR THE DESIGN AND CONSTRUCTION OF MARINE BOILERS — BOILER TUBES — TYPES OF BOILERS — BOILER SETTING — CHIMNEYS AND FLUES — SUPERHEATERS — SCALE AND CORROSION — FEED-WATER SOFTENING — FEED HEATERS — ECONOMISERS — AIR HEATERS — INSPECTION AND LEGISLATION — BOILER FITTINGS — DAMPERS — INJECTORS — SAFETY VALVES — GAUGES — REGULATORS — CLEANERS — STEAM PIPES — STEAM TRAPS.

BOILER DESIGN

General Considerations.

Simplicity and accessibility for cleaning, favour the tank type of boiler, of which the Lancashire is the best known, The ratios of heating surface to weight and space occupied are, however, necessarily low, and for large units or intensive generation of steam, the water-tube boiler has advantages. In order that these advantages may be fully developed, certain problems must be recognised and dealt with, namely—

(*a*) Boiler deposits must be reduced to a minimum, and these constrained in suitable mud drums and other parts.

(*b*) Water and steam spaces must be accessible for cleaning and inspection with a minimum of time and labour for joint breaking and making.

(*c*) Combustion of coal (or oil or gas) in large bulk must be arranged for.

(*d*) As much as possible of the heating surface should be exposed to direct radiation from the furnace and combustion chamber. The remaining heating surface should be large in relation to length of path of gases.

(*e*) The boiler must be capable of dealing with overloads.

The first two, (*a*) and (*b*), used to receive more attention than the other points, but (*c*) and (*d*) are now recognised to be of very great importance.

The efficient combustion of fuel requires that the particles of combustible, whether solid, liquid, or gaseous (including the products of the volatilisation of coal or oil), should be brought into intimate contact with air in the furnace.

In general this requires a large combustion chamber and a long path for the flame before it reaches the tubes or plates of the boiler, so that ample space and time for combustion are ensured.

Large water-tube boilers were at one time unduly limited in combustion space, and in recent years combustion chambers and flame paths have been much increased. A tendency is now showing itself to avoid further increase, and even to reduce combustion space as compared with recent designs, and to secure efficient combustion by turbulence or agitation of the gases in the furnace, so enabling the flame to be shortened.

Ample heating surface exposed to the combustion chamber and high combustion temperature increase the heat transmitted by direct radiation, the importance of which is increasingly recognised.

As regards that part of the heating surface which receives heat by contact with the gases, the area required depends largely on the speed of the gas flow, high speeds resulting in high transmission rates. Restricted gas passages lead to high gas speeds, and require more draught (and hence usually the consumption of power for driving fans).

The tank type of boiler, such as the Lancashire or the cylindrical marine, has the advantage that the large amount of hot water contained enables it to respond quickly to sudden increases in demand for steam.

The Lancashire type is relatively free from breakdowns, the wear and tear are low, so that maintenance is cheap, it is easy to operate with semi-skilled labour, and is better suited than water-tube boilers to deal with impure feed water.

For burning low-grade fuels the water-tube boiler has advantages on account of its large brick-lined combustion chamber and adaptability for fitting with mechanical stokers and ash handling plant. The water-tube boiler can be constructed for very high pressures.

Boiler Output.

The steam generated per hour depends on the amount of fuel that can be properly burned on the grate per hour, and on the extent and efficiency of the heating surface.

For a particular type of boiler, fuel, and draught, the steam-raising capacity is thus proportional to the grate area, though it is obvious that by control of the stoking, and particularly of the draught, the rate of combustion, and hence also the steam generated per hour, may be varied very widely.

Air for Combustion of Coal.

If C = carbon, H = hydrogen, and O = oxygen in 1 lb. coal by weight, the air 'theoretically' required to burn completely 1 lb. of coal = $\left\{ C \times \frac{8}{3} + \left(H - \frac{O}{8} \right) \times 8 \right\} \times \frac{100}{23}$ lbs.

This assumes that all the carbon is burned to CO_2, and neglects combustible constituents other than those named.

Rich coals require more air than poor coals, but over a considerable range of qualities of coals used in boilers, the 'theoretical' air required may be taken with fair accuracy as

$$\frac{\text{higher calorific value}}{10{,}000} \times 7{\cdot}2 \text{ lbs. per lb. coal,}$$

the calorific value being in B.Th.U. per lb.

Quantity of Flue Gases.

In order to specify suitable fans for induced and forced draught, it is necessary to know the weight of the gases and the weight of air actually used per lb. of fuel burned. From this the volume to be dealt with by the fans at the appropriate temperatures can be ascertained.

The air used influences the percentage of CO_2 in the flue gases.

For typical coals, the relation between air used and Co_2 may be expressed by the following approximate formula :—

$$\text{Lbs. air per lb. coal} = \frac{132 \times \text{calorific value of coal}}{CO_2 \text{ percentage} \times 10{,}000}.$$

The weight of flue gases per lb. coal may be taken as (lbs. air + 1).

Excess air : The percentage of excess air is approximately $\left(\frac{18{\cdot}9}{CO_2 \text{ per cent.}} - 1 \right) \times 100.$

Actually the 'constant' 18·9 should vary for coals in which the proportion of hydrogen to carbon is different. The formula gives therefore merely an approximate figure for a typical coal, also it does not apply if there is any CO in the flue gases.

Rate of Combustion and Draught.

With natural or chimney draught the air pressure at the grate depends on the height of the chimney, the temperature of the gases in the chimney, and the lay out of the flues. A typical figure is 0·65 in. of water per 100 ft. of chimney height, above the grate. The effective draught at the grate for a Lancashire boiler battery with economiser will be about one-third of this figure, and rather less for a water-tube boiler installation with economiser.

The coal burnt per hour per square foot of grate will be about :—

20 lbs. average steam coal; 5-10 lbs. anthracite very thinly spread.

For other chimneys and draughts the rate of combustion increases not quite in proportion to height.

The draught at the base of the chimney is due to the difference of the weight of the column of hot gases in the chimney and the weight of a similar column of external air.

If t_1 = temperature of external air; t_2 = mean temperature of gases in chimney ; h = height of chimney in feet ; d = draught in inches of water, measured at foot of chimney ; density of gases (of typical composition) at 32° F. and atmospheric pressure = ·084 lb. per cu. ft. ; density of air = ·0807 lb. per cu. ft., then,

Mean density of gases at chimney temperature $= \cdot 084 \times \frac{493}{t_1 + 460} = \frac{41 \cdot 4}{t_2 + 460}$;

Density of external air $= \cdot 0807 \times \frac{493}{t_1 + 460} = \frac{39 \cdot 8}{t_1 + 460}$;

Pressure difference at base of chimney in lbs. per sq. foot $= h\left(\frac{39 \cdot 8}{t_1 + 460} - \frac{41 \cdot 4}{t_2 + 460}\right)$;

or

$$d \text{ in inches of water gauge} = h\left(\frac{7 \cdot 64}{t_1 + 460} - \frac{7 \cdot 95}{t_2 + 460}\right).$$

This is the ' theoretical ' draught. The actual available draught is less than the theoretical by that required to produce the velocity of flow of the gases and by that required to overcome friction in the flues, economiser, and chimney. The loss due to velocity (in inches w.g.)

$$= d_v = 0 \cdot 10 \times \text{density of gases} \times \text{pressure} \times \frac{V^2}{t_2 + 460}$$

$$= \text{for typical gases } 0 \cdot 123 \frac{V^2}{t_2 + 460}$$

where

V = velocity of gases in chimney.

The friction loss (in inches w.g.) in the chimney

$$= d_f = 0 \cdot 40 \times f \times \text{density} \times \text{pressure} \times \frac{V^2}{t_2 + 460} \times \frac{L}{D}$$

where

f = friction coefficient = 0·016 say ;
L = length of gas path in feet ;
D = diameter of chimney in feet ;

$$\therefore \text{ friction loss } d_f = 0 \cdot 0079 \frac{V^2}{t_2 + 460} \frac{L}{D}$$

It is convenient to relate the losses in the chimney to the weight of gas flowing rather than to the velocity.

If W = rate of flow of gas in lbs. per sec., the velocity loss d_v becomes

$$d_v = 0 \cdot 000117 \frac{W^2 (t_2 + 460)}{D^4}$$

and the friction loss

$$d_f = 0 \cdot 0000075 \frac{W^2 (t_2 + 460) L}{D^5}$$

Example.—A boiler battery consists of six boilers with a grate area of 38 sq. ft. each. What will be the draught at the foot of a chimney 150 ft. high and 7 ft. diameter, when the rate of combustion is 25 lbs. per hour per sq. ft. of grate area and the mean chimney temperature is 600° F. ?

The coal burned per hour is 5,700 lbs.

The flue gases will amount to about 19 lbs. per lb. coal, or 108,000 lbs. per hour = 30 lbs. per sec.

The 'theoretical draught' (with external air at 60° F.)

$$= 150 \left(\frac{7 \cdot 64}{520} - \frac{7 \cdot 95}{1060} \right) = 1 \cdot 08 \text{ inches w.g.}$$

$$\text{Loss due to velocity} = 0 \cdot 000117 \times \frac{30^2 \times 1060}{7^4}$$

$$= 0 \cdot 046 \text{ inches w.g.}$$

$$\text{Loss due to friction} = 0 \cdot 0000075 \times \frac{30^2 \times 1060 \times 150}{7^5}$$

$$= 0 \cdot 064 \text{ inches w.g.}$$

∴ available draught at foot of chimney

$$= 1 \cdot 08 - (0 \cdot 046 + 0 \cdot 064)$$
$$= 0 \cdot 97 \text{ inch w.g.}$$

From this, the friction losses at flues, economiser (if any) and dampers must be deducted in order to find the draught at the boiler furnaces. Note, in the above example, if there were an economiser, the chimney temperature would be lower than 600° F., and hence the 'theoretical draught' itself would be less. The economiser thus reduces draught both by reducing chimney temperature and by its additional frictional resistance.

Effect of Altitude on Draught.

At high altitudes, the barometric height is less than at sea-level, and hence the density of the atmosphere is less. Hence also the 'theoretical draught' of a given chimney is reduced.

In order to obtain the same draught the height of the chimney must be increased.

The approximate ratio of the height required to the height required at sea-level is given in the following table:

Altitude above sea-level (ft.)	500	1,000	1,500	2,000	3,000	4,000
Ratio of heights	1·02	1·04	1·06	1·08	1·12	1·16

The diameter of the chimney at altitudes should also be increased by about 1½ per cent. for each 1,000 ft. of altitude.

Draught and Rate of Combustion.

Even if the draught at the grate is accurately known, this alone does not fix the rate of combustion, which is influenced by the kind of fuel and thickness of fire; the following tables, due to W. S. Hutton, must therefore be regarded as approximate only.

Rate of Combustion for Different Draught Pressures.

Height of Chimney above Grate, in Feet.	Total Draught Pressure, in Inches of Water.	Rate of Combustion, in Pounds per Square Foot of Grate per Hour.	Height of Chimney above Grate, in Feet.	Total Draught Pressure, in Inches of Water.	Rate of Combustion, in Pounds per Square Foot of Grate per Hour.
25	·182	10	130	·948	30
50	·364	16	140	1·029	34
60	·437	17	150	1·095	40
70	·512	18	180	1·313	50
80	·583	19	200	1·459	60
90	·657	20	225	1·641	70
100	·729	22	250	1·825	80
110	·802	24	300	2·189	90
120	·875	27	400	2·553	112

TOTAL DRAUGHT REQUIRED FOR EFFICIENT COMBUSTION OF DIFFERENT KINDS OF FUEL.

Fuel.	Total Draught, in Inches of Water.	Fuel.	Total Draught, in Inches of Water.
Straw	0·20	Slack, very small	0·7 to 1·1
Wood	0·30	Coal-dust	0·8 to 1·1
Sawdust	0·35	Semi-anthracite coal	0·9 to 1·2
Peat, light	0·40	Mixture of breeze and slack	1·0 to 1·3
Peat, heavy	0·50	Anthracite, round	1·2 to 1·4
Sawdust, mixed with small coal	0·60	Mixture of breeze and coal-dust	1·2 to 1·5
Steam coal, round	0·4 to 0·7	Anthracite slack	1·3 to 1·8
Slack, ordinary	0·6 to 0·9		

In hand-fired plants, using coal for fuel, the rate of combustion should not be permitted to fall below 15 lbs. per sq. ft. of grate surface per hour. With stokers the rate of combustion may be as high as the design of the plant will permit. When firing with fine steam-size coal, the rate should not be less than 10 lbs. per sq. ft. of grate area per hour.

A typical working figure for the rate of combustion in stationary boilers with good natural or moderate assisted draught is 25 lbs. coal per hour per sq. ft. of grate area. For locomotives the rate is about 60 to 70, but may rise to 200.

USE OF DRAUGHT GAUGES ON STEAM BOILERS.

(Chas. F. Wade, A.M.I.Mech.E., A.M.I.E.E.)

The weight of oxygen or of air required to produce a given number of B.Th.U., using almost any fuel, is approximately constant, so that a measure of the air supplied to a boiler furnace is a good guide to the amount of fuel being consumed. If, therefore, the boiler is fitted with a steam flow indicator or meter as a measure of the heat transmission taking place, there will be a constant relation between air and steam flow under all conditions of steaming rate for a given efficiency, so that by careful initial tests of the waste gases under best conditions of operation this relation can be established once for all. Air flow may be gauged either by the difference in reading of two separate simple draught gauges connected sufficiently far apart in the path of the gas travel to obtain a large enough difference in air pressure or, preferably, a differential draught gauge may be connected across the two points; so long as the boiler and baffling, etc., remain in good condition the reading of such a gauge will provide a measure of the gas and air flow. The draught scale may with advantage have a second scale added giving the rate of steam flow corresponding to the most efficient conditions for the amount of pressure difference indicated and as established by the preliminary tests of the waste gas composition. Deterioration in the physical state of the boiler and its settings will be shown by inability to maintain the correct relation of steam and air flow, and the cause of the irregularity may be deduced from readings of the temperature of the waste gases.

A single draught gauge connected to one point of the gas circuit affords no indication of efficiency of combustion by itself, as a similar reading may be obtained under very widely differing conditions of fire thickness, initial draught, physical state, relation of forced to induced draught, etc.; thus mere working to a given reading on a simple draught gauge will inevitably lead to serious inefficiency from time to time.

Plain U tube gauges are not sufficiently sensitive to obtain close readings, and in addition cannot be arranged to give bold and distinct indications. Inclined tube gauges are much to be preferred, as each inch of height of liquid may be extended over a much longer scale. Open-scale, dial-type instruments are best of all, as they can be arranged to give a reading which can be observed clearly at a considerable distance, which cannot be done with any existing form of liquid column gauge.

Transmission of Heat in Boilers.

The transfer of the heat developed by the combustion of the fuel in the furnace to the water in the boiler takes place in two ways: (1) by radiation from the incandescent fuel and the hot gases; and (2) by convection and conduction—i.e. conduction through the materials in the

path of the heat flow and by the contact of the gases and water with the metal separating them.

The transmission of heat to that part of the heating surface which is directly exposed to the fire is practically entirely by radiation, and that to the remainder of the heating surface practically entirely by convection.

Transmission by Radiation.

The heat transmitted by direct radiation in most boilers is strikingly large, so that the fire-box surface is much the most powerful evaporating surface.

The amount of heat radiated from a fire of temperature T_1 (° F. absolute) to the fire-box surface at T_2 is

$$1{,}600 \times \frac{T_1^4 - T_2^4}{1{,}000^4} \text{ B.Th.U. per hour per sq. ft.}$$

or practically, since T_2^4 is always negligible in comparison with T_1^4,

$$\text{Radiant heat} = 1{,}600 \left(\frac{T_1}{1{,}000} \right)^4.$$

Whether the radiating surface to be taken into account in calculating the total heat radiated is the grate area, or a larger amount, is debatable. The formula indicates, however, the importance of the direct heating surface, and also the importance of high combustion temperature for high rates of transmission.

Transmission through Plates or Tubes.

The transmission of heat from a fluid to another fluid on the other side of a metal tube depends on a large number of factors, including, as well as the area of tube surface and the temperatures, the velocities of the fluids, the diameter of the tube, and the nature of the fluids and of the tube surfaces. The change of temperature of the fluid passing through the tube is, of course, itself dependent on the factors named, so that the factors are not all independent. Differences of conductivity and thickness of the metals of the tube are negligible, the resistance to the passage of heat through the metal being a very small portion of the total resistance from fluid to fluid.

The coefficient of heat transmission is the number of heat units transmitted per unit time per unit area per degree of mean temperature difference (see p. 1947, and for determination of mean temperature difference see p. 1948).

For the conditions prevailing in the tubes of fire tube boilers, such as boilers of the locomotive type, waste-heat boilers, and air heaters, that is, where the current of hot gas—and for an air heater, the current of air being heated—is along the tubes, the coefficient of heat transmission can be related approximately to $\frac{w}{a}$, where w is the rate of gas flow in lbs. per sec., and a the cross section of the gas path in sq. ft. The results of investigations on different boilers vary widely, but for values of $\frac{w}{a}$ between 1 and 4, mean values of the transmission coefficient h are given by the equation $h = 0{\cdot}00077 \frac{w}{a} + 0{\cdot}00085$, where h = B.Th.U. transmitted per second per sq. ft. of heating surface per degree F. of temperature difference *between the gas and the metal*.

If, as in a boiler, one side of the tube metal is in contact with water, the temperature difference between metal and water is negligible compared with that between gas and metal, and in this case the temperature difference between gas and water may be used instead of that between gas and metal.

When both the substances separated by the tube wall are gaseous, the temperature differences on *each* side of the tube must be considered, or else the overall coefficient of transmission calculated.

The procedure that may be adopted is illustrated by the following examples:

Example 1.—A tubular boiler, in which the gases pass through the tubes at the rate $\frac{w}{a} = 2$, temperature of water (and of steam) = 350° F., gas temperatures 1,000° F. at inlet to 450° F. at outlet.

Mean temperature difference (M.T.D.) between gas and water

$$= \frac{(1{,}000 - 350) - (450 - 350)}{\log_e \frac{1{,}000 - 350}{450 - 350}} = 294.$$

This may also be taken as the M.T.D. between the gas and the tube metal.

Coefficient $h = 0{\cdot}00077 \times 2 + 0{\cdot}00085$

$= 0{\cdot}00239$, say $0{\cdot}0024$ B.Th.U. per sec. per sq. ft. per ° F.

$\therefore$ heat transmission rate $= 0{\cdot}0024 \times 294 = 0{\cdot}706$ B.Th.U. per sec. per sq. ft. (the heating surface being measured on the *gas* side of the tube).

If the tube is 2 ins. internal diameter, $a = 3{\cdot}14$ sq. ins. $= 0{\cdot}0218$ sq. ft.

$$\therefore \quad w = \frac{w}{a} \times a = 2 \times 0{\cdot}0218 = 0{\cdot}0436 \text{ lbs. per sec.}$$

Taking $0{\cdot}25$ as the mean specific heat of the gases, the heat given up in each tube

$$= 0{\cdot}0436 \times 0{\cdot}25 \times (1{,}000 - 450) = 6 \text{ B.Th.U. per sec.}$$

$$\therefore \text{ tube surface required (internal)} = \frac{6}{0{\cdot}706} = 8{\cdot}5 \text{ sq. ft.}$$

$$\therefore \quad \text{tube length} = \frac{\text{surface}}{\text{perimeter}} = \frac{8{\cdot}5 \times 12}{6{\cdot}28} = 16{\cdot}2 \text{ ft.}$$

Evaporation (from and at 212° F.) per tube

$$= \frac{6 \times 3{,}600 \text{ (B.Th.U. per hour)}}{970 \text{ (B.Th.U. per lb. steam)}} = 22{\cdot}2 \text{ lbs. steam per hour.}$$

$\therefore$ for 2,000 lbs. steam per hour

Number of tubes required = 90

Total tube heating surface = 765 sq. ft.

Evaporation per sq. ft. of heating surface = $2{\cdot}62$ lbs. per hour (from and at 212° F.).

Example 2.—A tubular air heater, with tubes 2 ins. internal and $2\frac{1}{4}$ ins. external diameter; gas flowing inside tubes, air flowing in counter current direction outside tubes: $\frac{w}{a}$ for gas = 2; $\frac{w}{a}$ for air = $1{\cdot}86$; gas temperatures 700° F. and 450° F. at inlet and outlet; air temperatures 60° F. and 334° F.

hg = coefficient of heat transmission on gas side = $0{\cdot}0024$.

ha = do. on air side = $0{\cdot}00228$.

Mean temperature difference between gas and air

$$= \frac{(700 - 334) - (450 - 60)}{\log_e \frac{700 - 334}{450 - 60}} = 378.$$

It would be incorrect to take either hg or ha multiplied by 378 as the heat flow per sq. ft. per sec., since hg and ha refer respectively to the transmission between gas and *metal*, and *metal* and air only.

It can be shown that the mean temperature difference (M.T.D.) between gas and metal

$$= \text{M.T.D. between gas and air} \times \frac{1}{1 + \frac{hg \times Ag}{ha \times Aa}}$$

where Ag, Aa are the metal surfaces exposed to gas and air respectively.

In the example, $\frac{Ag}{Aa} = \frac{\text{internal diameter of tube}}{\text{external diameter of tube}} = \frac{2}{2 \cdot 25}$

$\therefore$ M.T.D. gas to metal $= 378 \dfrac{1}{1 + \dfrac{0 \cdot 0024 \times 2}{0 \cdot 00228 \times 2 \cdot 25}} = 195 \cdot 3$

Similarly, M.T.D. metal to air $= 378 \dfrac{1}{1 + \dfrac{0 \cdot 00228 \times 2 \cdot 25}{0 \cdot 0024 \times 2}} = 182 \cdot 7$

$\therefore$ heat transmitted per sec. $= 0 \cdot 0024 \times 195 \cdot 3$ B.Th.U. per sq. ft. of surface exposed to gas $= 0 \cdot 469$ B.Th.U. per sq. ft.

Also heat transmitted per sec. $= 0 \cdot 00228 \times 182 \cdot 7 = 0 \cdot 416$ B.Th.U. per sq. ft. of surface exposed to air.

These heat transmission rates must be inversely proportional to the areas on the gas and air sides, *i.e.* $\frac{0 \cdot 469}{0 \cdot 416}$ should be $\frac{2 \cdot 25}{2 \cdot 00}$, which is fulfilled very closely.

The length of tube required may be calculated as follows:

Rate of flow of gas in tube $= \frac{w}{a} \times$ area of tube

$$= 2 \times \frac{3 \cdot 1416}{144} = 0 \cdot 0436 \text{ lbs. per sec.}$$

Specific heat of gas $= 0 \cdot 25$.

$\therefore$ heat given up by gas in tube

$$= 0 \cdot 0436 \times 0 \cdot 25 \times (700 - 450) \text{ B.Th.U. per sec.}$$
$$= 2 \cdot 725 \text{ B.Th.U. per sec.}$$

$\therefore$ heating surface required (on gas side) $= \frac{2 \cdot 725}{0 \cdot 469} = 5 \cdot 81$ sq. ft.

Internal perimeter of tube $= 3 \cdot 1416 \times 2 = 6 \cdot 283$ in. $= 0 \cdot 5236$ ft.

$\therefore$ tube length $= \frac{5 \cdot 81}{0 \cdot 5236}$ ft. $= 11 \cdot 1$ ft.

The temperature differences are the chief factors in determining the length of tube (for usual values of $\frac{w}{a}$).

The number of tubes is determined from the total gas flow compared with the flow per tube.

Thus if the gas flow is 4·4 lbs. per sec., the number of tubes for the above example would be $4 \cdot 4 \div 0 \cdot 0436 =$ say 100.

Example 3.—An alternative method of finding the heat transmission rates in the previous example is as follows:

If the overall coefficient of heat transmission from gas to air (neglecting the relatively small resistance of the metal) = H,

then $\frac{1}{H}$ = resistance to heat flow

$$= \frac{1}{ha} + \frac{1}{hg} = \frac{1}{0 \cdot 0024} + \frac{1}{0 \cdot 00228}$$

therefore $H = 0 \cdot 00117$.

The overall mean temperature difference is 378°.

$\therefore$ rate of heat transmission

$$= 0 \cdot 00117 \times 378 \text{ B.Th.U. per sec. per sq. ft. of mean area}$$
$$= 0 \cdot 443 \text{ B.Th.U. per sec. per sq. ft. of mean area.}$$

The ratio of the areas on the gas and air sides is 2 : 2·25, or the areas are respectively 6 per cent. below and above the mean area.

The transmission rates are thus $0 \cdot 443 (1 - 0 \cdot 06) = 0 \cdot 416$ and $0 \cdot 443 (1 + 0 \cdot 06) = 0 \cdot 469$ B.Th.U. per sec. per sq. ft. of the respective areas, agreeing with the former calculation.

General Remarks on Tubular Heaters.

For a given range of temperatures, and a given value of $\frac{w}{a}$, the required tube length is proportional to tube diameter.

Hence if the length for a particular diameter has been calculated, the lengths for other diameters can be very readily found.

The tube length is *not* proportional to the drop of temperature of the heating medium; the extra tube length for each additional ten degrees say of temperature reduction at the outlet end becomes successively greater. Thus, if in a waste heat boiler, with water at 350° F., and initial gas temperature 1,000° F., the tube length is 12·2 ft. for outlet temperature 450° F., it will be 13·7 ft. for 430° F., and 15·5 ft. for 410° F. (the rate $w \div a$ and the diameter remaining the same).

The loss of pressure in the tubes, in inches of water gauge, may be taken as approximately

$$\left\{0{\cdot}0000216\,\frac{l}{d}\,T_m + 0{\cdot}000075\left(\frac{T_1}{2} + T_2\right)\right\}\left(\frac{w}{a}\right)^2,$$

where

l = tube length in ft.;

d = internal diameter of tube in ins.;

T_m = mean gas temperature, ° F. absolute $= \frac{T_1 + T_2}{2}$;

T_1 = inlet gas temperature, ° F. absolute;

T_2 = outlet gas temperature, ° F. absolute;

$\frac{w}{a}$ = lbs. of gas per sec. ÷ area of section of tube in sq. ft.

In this formula, the pressure of the gas is assumed to be atmospheric within a few ins. water gauge.

Fry's Empirical Formula.

In the case of boiler tubes with flue gases passing through them, the change of temperature of the gases, and thus also the heat transferred, can be calculated from the following equation (Lawford H. Fry, *Engineering*, Aug. 27, 1920):—

$$\text{lolog.}\,\frac{T_1}{t} - \text{lolog.}\,\frac{T_2}{t} = Mx \quad . \quad . \quad . \quad (1),$$

in which T_1, T_2 are the mean flue gas temperatures, ° F. absolute, at two sections of the tube x feet apart, t the mean temperature (absolute) of the tube wall, and 'lolog.' means 'logarithm of the logarithm.'

M is given by $\log. M = B - m \log.\,\frac{W}{p}$. (2) $\quad \log.(B + 1{\cdot}3) = \bar{1}{\cdot}71 - 0{\cdot}54 \log. d$ . . (3)

$$\log. m = \bar{1}{\cdot}36 + 0{\cdot}37 \log. d \quad . \quad . \quad . \quad (4)$$

where,

d = internal diameter of tube in inches; p = perimeter of tube ($= d \times 3{\cdot}1416$); W = rate of flow of gas in lbs. per hour.

Having determined the fall of temperature of the gases from the above, the heat given up —*i.e.* the heat transferred—is equal to

$$W \times (T_1 - T_2) \times \text{mean specific heat of gases.}$$

The following table gives the values of M for a number of tube diameters and rates of flow. These values may be used directly in equation (1).

VALUES OF M.

d inches.	W in lbs. per hour.				
	100	200	400	800	1,600
0·5	0·133	0·117	0·104	0·092	0·081
1·0	0·074	0·063	0·054	0·046	0·039
2·0	0·050	0·040	0·033	0·027	0·022
4·0	0·040	0·030	0·023	0·018	0·014
8·0	0·037	0·026	0·019	0·013	0·009

EXAMPLES.

(1) To find fall of temperature in flue tube 2 ins. diameter and 12 ft. long, with rate of gas flow of 400 lbs. per hour, boiler pressure 180 lbs. per sq. in., and temperature of gases 2,000° F. at entrance of tubes.

The temperature of the metal of the tube is nearly that of the water in the boiler—i.e. nearly that of steam at 180 lbs. per sq. in.; thus $t = 380°$ F., about. (Slight differences in t have very little influence as a rule.)

From the table, for 2-in. tube, and W = 400 lbs. per hour, M = 0·033

$$\therefore \text{lolog.}\frac{2,000+460}{380+460} - \text{lolog.}\frac{T_2}{380+460} = 0{\cdot}033 \times 12$$

$$\therefore \text{lolog.}\frac{T_2}{840} = \bar{1}{\cdot}273\,;$$

$$\therefore \quad \log.\frac{T_2}{840} = 0{\cdot}1875.$$

$$\therefore T_2 = 1{,}293° \text{ F. abs., or } 833° \text{ F.}$$

(2) To find the heat transferred.

The mean specific heat of flue gases at the temperature in question may be taken as 0·31. Then the heat transferred in B.Th.U. per hour

$$= 400 \times (2{,}000 - 833) \times 0{\cdot}31 = 145{,}000.$$

The tube surface is 6·29 sq. ft., and heat transferred per sq. ft. = 23,100 B.Th.U. per hour.

(3) To find length of tube required to reduce the gas temperature to 700° F., rate of flow and diameter being as before:

$$\text{lolog.}\frac{2,000+460}{380+460} - \text{lolog.}\frac{700+460}{380+460} = 0{\cdot}033 \times \text{length.}$$

From which, length = 15·8 ft.

GRAPHIC CALCULATION OF HEAT TRANSMISSION THROUGH BOILER TUBES.

Fig. 1 (p. 1551) replaces equations (2), (3) and (4), and gives directly, without calculation, the value of M for any given values of W and d, or if required the value of any one of the three quantities W, M, d when the other two are known. In fig. 1 any straight line will cut the three scales graduated in terms of W, M, d at graduations which satisfy equations (2) (3) (4).

Fig. 2 enables equation (1) to be solved graphically. The values of $\frac{T_1}{t}$ and $\frac{T_2}{t}$ are marked on the two end scales and values of Mx on the middle scale. Any straight line cuts these scales at graduations which satisfy equation (1).

Example 1 :—To find the length of flue necessary for a temperature drop from 2,000 deg. F. to 720 deg. F. in a half-inch flue, with a wall temperature of 380 deg. F. and where 100 lb. of gas are passing per hour.

In fig. 1 draw a straight line joining the points W = 100 and d = 0·5. This line cuts the M scale at the required value of M = 0·132.

Since

$$\frac{T_1}{t} = \frac{2000+460}{380+460} = 2{\cdot}93, \text{ and } \frac{T_2}{t} = \frac{720+460}{380+460} = 1{\cdot}41,$$

therefore a straight line drawn in fig. 2 joining these graduations on the $\frac{T_1}{t}$ and $\frac{T_2}{t}$ scales, will cut the Mx scale, at the required value of Mx, which is seen to be 0·50. Therefore, since M = 0·132, $x = \frac{0{\cdot}50}{0{\cdot}132} = 3{\cdot}78$ ft., *i.e.*, a flue 3·78 ft. in length will be required for the given temperature drop.

Example 2 :—To find what drop in temperature will take place in a half-inch flue, 5 ft. in length, when 1,600 lb. of gas are passing per hour and the temperature of flue gas at entry is 2,000 deg. F., the flue wall temperature being 420 deg. F.

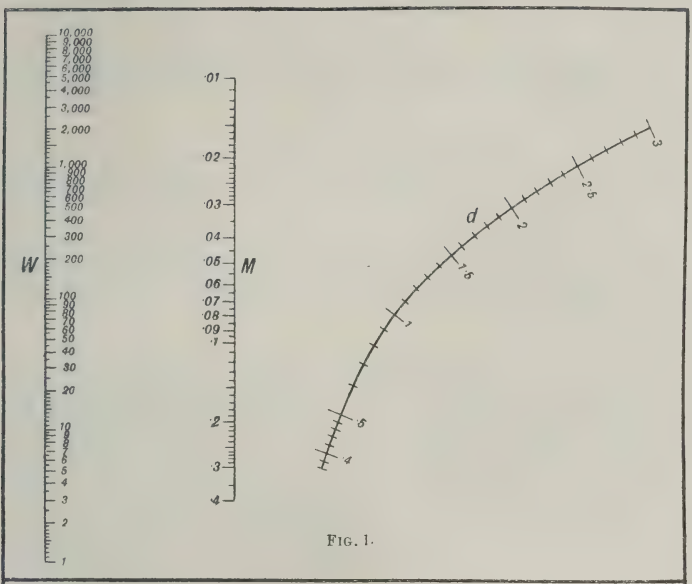


FIG. 1.

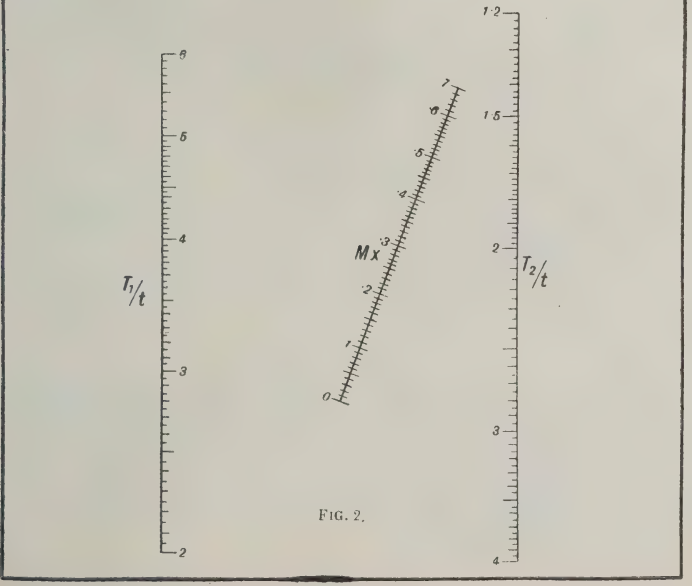


FIG. 2.

From fig. 1, by joining $W = 1{,}600$ and $d = 0{\cdot}5$, we get $M = 0{\cdot}081$, therefore,

$$Mx = 0{\cdot}405.$$

But $\frac{T_1}{t} = \frac{2{,}000 + 460}{420 + 401} = 2{\cdot}80$, therefore, joining $\frac{T_1}{t} = 2{\cdot}80$ and $Mx = 0{\cdot}405$, in fig. 2, we get

$$\frac{T_2}{t} = 1{\cdot}50,$$

therefore,

$$T_2 = 1{\cdot}50 \times 881 = 1{,}322° \text{ C.} = 861° \text{ F.}$$

Actually, lines need not be drawn on the diagrams, as it is sufficient to lay a rule on the scales through the required points.

(A. J. V. Umanski, M.Sc., *Engineering*, June 10, 1921.)

Very high rates of heat transmission have been attained in water-tube boilers when heavily forced. Spyer (Brit. Assoc., 1922) states that rates of working giving an average heat transmission of 24,000 B.Th.U. per hour per sq. ft. of heating surface have been attained on test without causing damage. With a single horizontally inclined tube a rate of 140,000 has been obtained experimentally.

Heat Transfer in Cross Flow.

Experiments have been recently made (Munich Institute of Technology, *Forschungsarbeiten*, No. 269, 1925) on the heat transfer from hot air flowing in a direction normal to the axis of tubes containing water. Tubes 4 6 to 28 mm. diameter were tested with air temperatures 53° F. to 572° F., and air velocities 6 5 to 42 3 ft. per sec., and formulæ applicable to other conditions deduced by the principle of similarity.

Very high temperature differences between the front and rear sides of the tubes. From the tests on single tubes the formula for heat transference, reduced to British units, is—

$$h = 0{\cdot}104 \frac{k}{d} \left(\frac{v\,d\,\rho}{\mu}\right)^{0{\cdot}56},$$

where

h = heat transferred in B.Th.U. per hour per sq. ft. of surface area of tube per ° F. difference between mean air and mean tube surface temperatures;

k = coefficient of heat conduction of air in B.Th.U. per hour per ft. per ° F. (*i.e.*, k = heat conducted in B.Th.U. per hour through 1 sq. ft. of cross-section when the temperature gradient in the direction of flow is 1° F. per ft.);

v = velocity of air in ft. per sec.;

d = diameter of tube in ins.;

ρ = density in lbs. per cub. ft. $\div g$ (32·2 ft. per sec.^{2});

μ = viscosity of air in lb. sec. ft. units.

Average values of v, k, ρ, and μ are to be used. The transfer of heat is increased by—

(*a*) increase of velocity;

(*b*) staggered arrangement of tubes;

(*c*) roughness of tube surface and turbulence in air stream;

(*d*) reduction in tube diameter.

Heating Surface.

The amount of heating surface required for efficient operation is much influenced by its arrangement, the proportion of it receiving heat by direct radiation, the velocity of flow of the gases through the flues or tubes, the circulation of the water, etc. While the ratio of heating surface to grate area thus varies widely, typical approximate values are as follows:—

Lancashire boiler	22–26
Water-tube	50–60
Water-tube, marine	35
Locomotive	60–70
Scotch boiler, marine	35–40
American tubular	50–80

The economiser surface should be about equal to the boiler surface for a Lancashire boiler and less, down to 50 per cent., for a water-tube boiler with ample heating surface.

For superheater surfaces see p. 1613.

As evidence of the effect of high gas speeds, cases are on record in which half the tubes in a tubular boiler have been plugged and yet the same total evaporation per hour obtained.

Recent Progress in Evaporation Rates.

Improvements in design which have raised the proportion of heat transferred by direct radiation have led to great increases in the rate of evaporation. This is illustrated by the following table relating to boilers installed at different dates at Bradford :—

Year.	Lbs. water evaporated per hour per sq. ft. heating surface.
1900	3·0
1902–11	3·25
1916	3·45
1926	6·0
1926 (on order)	12·0

Still higher rates have been achieved ; at Hell Gate Power Station, New York, boilers equipped with Murray fin-tube furnace walls have evaporated 14·7 lbs. per sq. ft. It is reported that boilers are now being built for 69 lbs. per sq. ft., and that a still higher rate is claimed for a boiler at work in an industrial plant at Manchester.

(T. Roles, quoted in *Engineering*, June 3, 1927.)

Geo. Orrok, referring to 'High Pressure Steam Boilers,' says that the earlier rating of 7 lbs. per hour (from and at 212° F.) per sq. ft. of heating surface has now been greatly increased. Rates of 15 to 20 lbs. at good efficiency have been secured, and 25 to 30 lbs. is in sight.

In convection type superheaters 6,000 B.Th.U. per hour per sq. ft., the former maximum reported rate of heat transmission is increased to 8,000 to 9,000 for interdeck superheaters. Steam velocities must not fall below 40 to 50 ft. per sec., or the temperature of the inner and outer tube surfaces will equalise and tend towards the steam temperature. Hence, flooding is necessary during warming up.

In radiant heat superheaters the transmission rate may be 6,000 to 7,000 B.Th.U. per hour per sq. ft., though up to 30,000 is reported.

('*Mechanical Engineering (American)*,' *June* 1928.)

Efficiency and Evaporation Rate.

The efficiency of a boiler depends not only upon good design, but is very much dependent upon good maintenance, cleanliness, prevention of air leakages in flues, and skilful stoking.

The actual evaporation in lbs. of steam per lb. of coal is a common, but somewhat unscientific, standard of efficiency. It obviously varies with the heat value of the coal, the steam pressure, and the superheat. When stated as the equivalent evaporation 'from and at 212° F.' it is a better basis, but is still dependent upon the quality of the coal. Thus, best Welsh coal will evaporate 25 per cent. more than an inferior slack with the same boiler efficiency. To convert actual evaporations into 'from and at 212' multiply by the B.Th.U. given to each lb. of water in heating, evaporating and superheating it, and divide by 970. Thus starting with water at 102° and superheating to 150° at 160 lbs. absolute, the B.Th.U. is 1214, so that the equivalent evaporation factor is 1·255.

If the calorific value of the coal is 11,800 B.Th.U. per lb., then an efficiency of 60 per cent. corresponds to an evaporation 'from and at 212°' of $\frac{11{,}800 \times \cdot 6}{970} = 7 \cdot 25$ lbs. water per lb. coal.

The evaporation per sq. ft. of heating surface depends on the fuel-combustion rate and the ratio of heating surface to grate area as well as on the efficiency.

Thus, in a Lancashire boiler burning 25 lbs. coal per hour per sq. ft. of grate, and having heating surface 25 times the grate area, if 7 lbs. water are evaporated per lb. coal, then the evaporation per sq. ft. of heating surface is 7 lbs. Similarly for a water-tube boiler 5 lbs., and for a locomotive 10 to 12 or more lbs. steam per sq. ft. heating surface are fair average figures.

By increasing the draught and thus increasing the rate of combustion, the output of the boiler can be much increased, but it should be noted that at high combustion rates the efficiency falls off.

For example, the series of trials of a Babcock and Wilcox boiler recorded on page 1598 show a maximum efficiency when burning 25 lbs. coal per sq. ft. of grate per hour. For rates of combustion between 30 and 70 these results correspond roughly to

$$w = \frac{78}{C} + 8{\cdot}3,$$

where C is the rate of combustion, and w the equivalent evaporation per lb. coal.

Typical figures for boilers in good condition, but without specially elaborate technical supervision, are :—

	Per cent.		Per cent.
Small cylindrical boiler, no economiser	50–55	Large water-tube boiler with economiser	70
Large cylindrical boiler with economiser	60–70	Do. highest class	70–80
Do. well fired and maintained	75	Locomotive boiler, moderately fired	65–70
Small water-tube boiler with economiser	50–60	Marine boiler, well fired	60–70

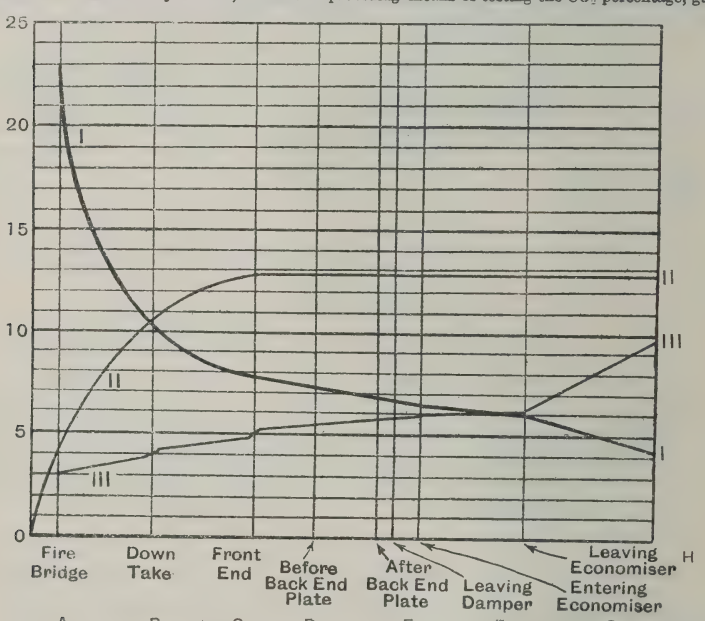
In the modern large power stations with large water-tube boilers and highly skilled technical supervision, efficiencies of 80 to 92 per cent. have been obtained.

That the efficiency of many boiler plants is much lower than the results which can be attained by greater care and skill in maintenance and supervision, is evidenced by the reports of Brownlie (p. 1555).

Maintenance of Boiler Efficiency.

(Chas. F. Wade, A.M.I.Mech.E., A.M.I.E.E.)

The arrangement advocated, and which has been adopted in several important modern power-stations with satisfactory results, consists in providing means of testing the CO_2 percentage, gas



Curve I = $\frac{\text{Temp. F.°}}{100}$; Curve II = CO_2% ; Curve III = Draught ins. W.G. × 10.

FIG. 3.

temperature, and draught pressure at regular intervals at a series of points situated at each consecutive stage in the gas travel through the boiler. Standard conditions being established with the newly overhauled boiler in as perfect a state of air-tightness, soundness of baffles, etc., with,

say, full rate of steaming, then readings are obtained of the three values mentioned, and are plotted to a suitable scale as three curves on a chart, as shewn in fig. 3. The curves shown are ideal ones, and those obtained in practice will deviate more or less, even with a boiler in the best possible condition, both as to physical state and combustion conditions. The significance of the variations which may occur in the various curves for individual defects in state or operation are given in the following table, while combinations of two or more defects will produce modifications of the curves to correspond. The chart may be taken as an equivalent to the indicator diagram of a steam-engine, and is used to check the operation of the boiler and its state in an analogous manner.

INTERPRETATION OF WADE'S BOILER CONTROL CHARTS.

Fault.	I. Temperature Curve.	II. CO_2 Curve	III. Draught Curve.	Remarks.
Forcing boilers beyond economical capacity	High throughout	Low at A „ „ B Normal at D	*Induced and natural draught*—high throughout *Forced draught*—low throughout	
Over-heavy firing beyond air capacity	A low B and D possibly high	High throughout CO present Smoke	High throughout	
Thin fires and bare patches	Low throughout	Low throughout Excess O_2	Low throughout	
Leaky seating blocks or buffles	A and B normal D high	A and B normal D low	A normal or high B onwards low	
Short-circuiting at side walls of downtake or at superheater cross-box wall	A, B and D normal E higher than D	A, B and D normal E rather low	A high. Fall of draught less than normal	
Leakage through back wall into flue	A normal B to E low F high	CO_2 not materially affected; possibly low	Draught not appreciably affected	
Air leakage into flues	A normal B normal or low * D low	A normal B normal or low *	A and B normal* C slightly low	*Marked variations in all curves at point of leakage.
Air leakage around dampers.	E normal F low	E normal F low	E and F (difference too small to measure)	
Dirty plates internally.	High throughout, with reduced fall A to D	Normal throughout	Normal throughout	
Dirty flues and gas passages.	Low throughout	Low throughout CO present	Low A to C High from D onwards	

Efficiency of Steam-Boiler Plant.

(*Contributed by David Brownlie, B.Sc. Hons. (Lond.), F.C.S., A.M.I.Mech.E.*)

(1)

The following are the true average figures for the performance of 250 steam-boiler plants in 27 different industries, comprising 1,000 boilers with an annual coal bill of 2,166,000 tons. Each test, comprising a day test and a check test of one week, was carried out with the greatest care

on the plant as actually working from week to week all the year round. Nearly 2,000 boiler plants were inspected and it was found that all were being worked on the same general lines—that is to say, these average figures apply to the burning of 75–100 million tons of coal burnt per annum in Great Britain for steam generation :—

AVERAGE WORKING FIGURES OF 250 TYPICAL BOILER PLANTS.

TESTS.

Average of 250 Typical Boiler Plants.

(A) *Working Day's Test.*

Type of boiler	'Lancashire'	Water evaporated per sq. ft. of grate area per hour	137·4 lbs.
Number of boilers	4	Water evaporated per lb. of fuel	6·56 lbs.
Grate area	152·6 sq. ft.	Equivalent evaporation from and at 212° F., per lb. of fuel	7·46 lbs.
Duration of test	9·43 hours	Equivalent evaporation from and at 212° F., per 1,000,000 British thermal units	631·0 lbs.
Amount of fuel used	30,131·72 lbs.	Temperature of feed-water before economisers	116° F.
Analysis of fuel used :		Temperature of feed-water after economisers	193° F.
British thermal units	11,822	Percentage of fuel bill saved by economisers	7·1 per cent.
Ash	11·5 per cent.		
Fuel burnt per boiler per hour	798·8 lbs.		
Fuel burnt per sq. ft. of grate area per hour	20·9 lbs.		
Water evaporated	197,776·0 lbs.		
Water evaporated per boiler per hour	5,243 lbs.		

(B) *Long Check Test* (one week of 7 days).

Duration	167·5 hours	Water evaporated per lb. of fuel	6·42 lbs.
Amount of fuel used	128·25 tons		
Water evaporated	184,435·0 galls.		

DETAILS.

Draught in side flue	0·40 in. W.G.	Temperature of superheated steam	346·5° F.
Draught in main flue at exit of economisers or chimney base	0·80 in. W.G.	Steam or power used as auxiliary to the production of steam	2·4 per cent.
Temperature of flue gases before economisers	598° F.	Thermal efficiency :	
Temperature of flue gases after economisers	478° F.	(A) Net working efficiency complete, after deducting 2·4 per cent. steam or power used as auxiliary to the production of steam	60·09 per cent.
Percentage of CO_2 in flue gases from side flue of boilers by means of combustion recorder	7·6 per cent.	(B) Boilers only	56·71 ,,
Steam pressure :		(C) Economisers only	4·35 ,,
Pounds per sq. in. (gauge)	89	(D) Superheaters only	0·51 ,,
Pounds per sq. in. (absolute)	104		
Temperature of saturation of steam	330·5° F.		

The installations from which the above average is derived comprise all sizes of plants with annual coal bills of from 750 tons to 70,000 tons ; whilst the average working pressure varied from 12 lbs. to 200 lbs. per sq. in. The fuels used were of every possible description, from the highest qualities of washed nuts, beans, peas, and fine slack, and large and small screened coal, down to inferior fuels of all qualities, including refuse coal mixed with shale, pit sweepings, washer settlings, etc., as well as coke and coke breeze.

The 250 plants had a total of 1,000 boilers, comprising 935 'Lancashire' boilers, 36 modern 'tubular' boilers, 17 'egg-ended' boilers, 6 'Cornish' boilers, 1 'marine' boiler, 2 'vertical' boilers, and 3 'patent' boilers. The plants varied in size from one small 'vertical' boiler to 35 'Lancashire' boilers, 30 ft. by 8 ft.

The total grate area of the 250 boilers was 38,158·6 sq. ft., corresponding to 152·6 sq. ft. per plant, or 38·1 sq. ft. per boiler, which, on the assumption that all the boilers were 'Lancashire,' would give an average grate-area length of approximately 6 ft.

The average figure shows 20·9 lbs. of fuel per square foot of grate area per hour, referring to the average amount burnt per hour per working day, and not to the amount for any particular hour. This figure is considerably below what is possible on a boiler plant run on modern lines which would give something like 25 lbs. to 30 lbs. per square foot grate area per hour with the average quality of fuel. The boiler plants of the country are working under very easy conditions

in this respect, and much more fuel could be burnt per boiler. Figures for fuel consumption depend very largely on the quality of the fuel. Thus a good quality washed coal of, say, 13,000 British thermal units and 7 per cent. to 8 per cent. ash could easily be burnt economically at the rate of 35 lbs. per square foot per hour, whereas purely refuse coals would only give about 15 lbs. Coke of good quality can be burnt at the rate of 22 lbs. per square foot grate area per hour and coke breeze about 15 lbs.

The average for the 250 plants shows 11,822 British thermal units and 11·5 per cent. ash, which includes almost every description of coal and coke varying in quality from 7,000 British thermal units to 14,000 British thermal units.

As regards coke, a good average sample of coke shows about 12,000 British thermal units to 12,500 British thermal units with 12 per cent. ash, whilst a good sample of coke breeze shows 9,000 British thermal units to 10,000 British thermal units and 15 per cent. to 20 per cent. ash. The great trouble with coke and coke breeze is the amount of moisture contained in them. Good samples should only contain about 5 per cent. and certainly not over 8 per cent. Many samples, however, contain regularly 20 per cent., which is a most serious matter from the point of view of economy.

In a well-managed plant with average quality of coal a 'Lancashire' boiler, 30 ft. by 8 ft., can evaporate easily 850 gallons per boiler per hour, an increase of 62 per cent. of steam output as compared with average figures.

(2)

The following are the figures for two separate industries, namely, collieries, and chemical works:—

(1) *Collieries.*—Representing average figures for the complete investigation of 100 boiler plants, 570 boilers, annual coal bill 1,250,000 tons.

(2) *Chemical Works.*—Representing average figures for the complete investigation of 60 boiler plants, 236 boilers; annual coal bill, 620,000 tons.

Essential Figures of the Tests.	100 Colliery Plants.	60 Chemical Works Plants.
1. Duration of test	9·68 hrs.	9·20 hrs.
2. Amount of coal used	39,815·0 lbs.	38,025 lbs.
3. Analysis of coal. B.Th.U. per lb.	10,500	11,350
4. " " Ash	15·5 per cent.	12·0 per cent.
5. Coal burnt per boiler per hour	721·5 lbs.	1,059·8 lbs.
6. Coal burnt per sq. ft. grate area per hour	18·9 lbs.	26·6 lbs.
7. Amount of water evaporated	216,900 lbs.	23,358 lbs.
8. Water evaporated per boiler per hour	3,931 lbs.	6,510 lbs.
9. Water evaporated per lb. coal	5·44 lbs.	6·10 lbs.
10. Water from and at 212° F. evaporated per lb. coal	6·07 lbs.	7·00 lbs.
11. Temperature of feed-water. Before economisers	137° F.	103° F.
12. " " After economisers	154° F.	215° F.
13. Percentage of coal bill saved by economisers	1·6 per cent.	10·1 per cent.
14. Draught in side flue	0·60 in. W.G.	0·38 in. W.G.
15. Draught in chimney base (or equivalent)	0·95 in. W.G.	0·70 in. W.G.
16. Temperature of flue gases. Before economisers	690° F.	650° F.
17. " " After economisers	650° F.	500° F.
18. Percentage of CO_2 in flue gases	7·5 per cent.	8·0 per cent.
19. Steam pressure. Gauge lbs. per sq. in.	86 lbs.	102 lbs.
20. Temperature of saturated steam	328·4° F.	339·2° F.
21. Temperature of superheated steam	350·0° F.	384·2° F.
22. Steam or power used auxiliary to the production of steam	1·8 per cent.	6·1 per cent.
Efficiency.		
23. (A) Net working efficiency of plant after deducting steam or power used auxiliary to steam production	55·52 per cent.	57·9 per cent.
24. (B) Due to boiler only	55·02 "	54·2 "
25. (C) Due to economiser only	0·90 "	0·1 "
26. (D) Due to superheater only	0·62 "	1·3 "

BOILER

Reference No.		1	2	3
1. Source of data		Inst. Mech. Eng., 1904	Insurance Co. test	Colville Magazine, 1923
2. Duration of test		4 hours	8 hours	—
PARTICULARS OF BOILER.				
3. Type		Lancashire	Lancashire	Lancashire
4. Size		8′ 0″ × 30′ 0″	8′ 0″ × 30′ 0″	8′ 0″ × 30′ 0″
5. Heating surface of boiler	sq. feet	920	954	1,000
6. Heating surface of superheater	sq. feet	217	312	—
7. Heating surface of economiser	sq. feet	2,240 (for 3 boilers)	None	2,560 (for 4 boilers)
8. Grate area	sq. feet	38	34·8	—
9. Method of stoking		Hand	Hand	Terbeek burners
10. Nature of draught		Natural 140 ft. chimney	Natural	—
DRAUGHT (IN INCHES), W.G.				
11. Over fires		0·2	0·45	—
12. Over leaving boiler		—	0·85	—
13. Over leaving economiser		0·52	—	—
14. Pressure of air supply		—	—	—
FUEL.				
15. Description		North Wales coal	Anthracite and slack mixed	Blast-furnace gas
16. Calorific value, B.Th.U. per lb. (higher)		14,040 (dried)	14,038 (dried)	120 (per cub. foot)
17. Moisture	per cent.	4·6	1·35	—
18. Ash	per cent.	4·97	—	—
FUEL CONSUMED.				
19. Weight burnt per hour, lbs. (per boiler)		705	399·9	About 100,000 cub. ft.
20. Weight burnt per hour, per sq. ft. of grate		18·52	11·49	—
21. Ash and clinker removed, per cent. of fuel fired		8·3	13·6	—
STEAM.				
22. Pressure, lbs. per sq. in., boiler		—	—	—
23. Pressure, lbs. per sq. in., superheater		158·9	150	120
24. Temperature leaving superheater	° F.	481·6	547	470
WATER EVAPORATED.				
25. Water temperature entering economiser	° F.	61	—	90
26. Water temperature entering boiler	° F.	245	110	220
27. Feed water, lbs. per hour (per boiler)		5,481	3,035	6,500
28. Water evaporated, lbs. per hour per sq. ft. H.S. from and at 212°		5·24	4·03	—
29. Water evaporated, actual per lb. fuel, lbs.		7·78	7·59	—
30. Water evaporated equivalent from and at 212° F. per lb. fuel		8·45	9·62	—

TEST RESULTS.

4	5	6	7	8
Engineering 1/8/24	The Engineer 5/3/26	Engineering 2/1/25	Amer. Soc. Mech. Eng., Jan. 1925	Amer. Soc. Mech. Eng., Oct. 1926
—	—	5 hrs. 50 mins	—	—
Yarrow Water-tube	Yarrow Water-tube	Stirling Water-tube	Stirling Duplex	Springfield Water-tube
—	—	—	—	—
3,255	7,016	8,282	30,600	12,560
410	2,005	1,120	4,520	4,000
1,550	—	4,100	22,080	10,692
88·7	—	182	—	378
Underfeed chain grate	—	Bennis mech.	Lopulco system	Taylor mechanical stokers
Balanced	—	Forced	—	Forced and induced
—	—	0·22	0·18 0·20 0·58	0·02 0·02 0·03
—	—	0·74	0·06 0·22 1·54	— — —
—	—	—	0·06 0·82 5·15	0·21 1·48 3·41
—	—	—	6·4 6·5 9·9	3·0 3·2 5·1
Washed pearls	Oil	Washed slack	Pulverised coal	Slack
11,000	17,913	12,080 (as fired)	12,590 12,545 12,717	13,840 13,940 13,938
11·5	—	11·21	22·6 2·6 2·0	5·87 5·05 5·48
12·5	—	6·14	10·7 12·2 10·7	6·01 6·41 5·66
1,638 2,390 2,886	—	4,284	5,244 13,561 27,445	5,512 10,430 18,200
18·0 27·0 32·5	—	23·54	— — —	— — —
—	—	6·71	— — —	6·64 7·86 8·15
— — —	—	—	253 254 270	277 281 291
194 201 220	258	208	251 253 265	277 278 281
543 586 622	638	575	571 630 681	560 627 663
108 91 87	—	127	125 157 153	149 140 142
202 219 213	102	211	205 242 265	245 264 268
12,400 17,050 20,300	41,252	34,983	47,582 131,438 254,357	59,957 105,940 161,580
4·9 6·9 8·2	—	4·81	— — —	4·76 9·2 12·85
7·57 7·13 7·0	11·9	8·17	9·07 9·69 9·27	10·87 10·15 8·88
9·55 9·26 9·29	15·49	10·56	— — —	— — —

BOILER TEST

Reference No.	1	2	3
GASES.			
31. Temperature leaving boiler . . . °F.	635	594	600
32. Temperature leaving economiser . . °F.	299	—	500
33. Temperature of boiler-house . . .	—	—	—
34. Vol. analysis of gases leaving economiser, if economiser is used, CO_2, per cent.	6·57	9·5	14·2
35. „ „ CO, per cent.	Nil	Nil	—
36. „ „ Oxygen, per cent.	12·30	9·5	—
EFFICIENCY.			
37. Efficiency of boiler alone (based on higher cal. value where stated) . per cent.	58·0	62·0	—
38. Efficiency of boiler and superheater per cent.	61·0	67·1	—
39. Efficiency of boiler, superheater, and economiser . . . . per cent.	71·6	—	58·2
HEAT ACCOUNT.			
40. Heat absorbed in boiler . . per cent.	58·0	62·0	—
41. Heat absorbed in superheater . per cent.	3·0	5·1	—
42. Heat absorbed in economiser . per cent.	10·6	—	—
43. Heat lost as unburnt fuel in ash . per cent.	3·23	4·7	—
44. Heat lost by incomplete combustion per cent.	—	—	—
45. Heat lost in dry products of combustion . . . . . per cent.	4·05	10·4	—
46. Heat lost in excess air . . . per cent.	8·3	8·9	—
47. Heat lost by moisture in fuel . per cent.	0·47	2·8	—
48. Heat lost due to steam formed by combustion of hydrogen . . per cent.	2·8		—
49. Heat balance, radiation, error, etc. . per cent.	9·55	6·1	—

Notes or Additional Data in connection with some of above tests.

No. 4.—The efficiency stated is evidently based on the *lower* calorific value of the coal.

No. 5 —The oil-fired Yarrow boiler was afterwards provided with an air heater, and about 5 per cent. additional efficiency was obtained, at an evaporation of 40,000 lbs. per hour. Air entering heater at 1110° F., leaving at 240° F. Gas entering at 556° F., leaving at 435° F. Furnace temperature, 2530° F. Suction before heater, $\frac{5}{16}$ in., after $\frac{7}{8}$ in. Air pressure before, 0, after $1\frac{1}{2}$ in. The combustion chamber volume was 2,790 cub. ft.

No. 7.—Boilers at Lake Shore Power Station, Cleveland, U.S.A. Each boiler has a water screen with 835 sq. ft. heating surface. Sixteen fuel burners. The tests quoted are from a series of twenty-four published in the journal of the Amer. Soc. Mec. Eng. (*Mechanical Engineering*), Jan. 1925, which show the efficiency and distribution of losses at a wide range of loads, and at starting and banking conditions.

No. 8.—Boilers at Hell Gate Power Station, New York, U.S.A. The tests shown are selected from a large series showing the performance at different rates of evaporation. The boiler-heating surface includes 460 sq. ft. of water-cooled side walls.

From the above table the losses of heat in a boiler plant may be assigned to their respective causes.

CONSTRUCTION OF BOILERS

Boiler Shells.

Mild steel is now almost invariably used for boiler shells. As a rule, in order to provide a margin for wear and tear, plates below $\frac{3}{8}$ in. thick should never be used.

The bending of plates should be done by rolls, and not by hammering. Rivet holes should be drilled and the rough edge removed by a countersinking tool.

RESULTS—*continued.*

4			5	6	7			8		
—	—	—	—	579	409	509	646	458	576	659
295	354	397	—	281	169	229	318	186	268	300
76	81	77	—	72	—	—	—	77	63	76
15·2 mean			13·8	10·8	13·0	14·7	14·0	13·2	14·0	14·9
—	—	—	—	—	Nil	Nil	Nil	0·1	0·0	0·3
—	—	—	—	—	—	—	—	5·8	5·2	3·7
—	—	—	—	—	75·8	79·0	73·0	—	—	—
—	—	—	83·8	—	81·1	86·4	81·5	84·6	79·5	70·5
89·4	87·0	86·2	—	84·8	86·9	92·9	89·8	92·2	88·6	78·6
—	—	—	—	69·00	75·8	79·0	73·0	77·1	70·6	61·1
—	—	—	—	6·95	5·3	7·4	8·5	7·5	8·9	9·4
—	—	—	—	8·85	5·8	6·5	8·3	7·6	9·1	8·1
—	—	—	—	0·71	—	—	—	0·2	1·1	2·6
—	—	—	—	—	—	—	—	0·4	—	1·1
—	—	—	—	6·77	1·8	3·6	5·8	2·8	5·0	4·9
—	—	—	—					0·6	0·5	0·6
—	—	—	—	4·77	3·6	3·8	3·6			
—	—	—	—					3·1	3·2	3·9
—	—	—	—	2·95	7·7	−0·3	0·8	0·7	1·6	8·3

Steel for Boilers.

The steel is usually required to be made by the open-hearth acid process. The tensile strength is about 26 to 30 tons per sq. in. for shell plates, and 24 to 28 tons per sq. in. for plates for furnaces, flue tubes, and endplates.

Strength of Boiler Shells.

Let

P = working pressure in lbs. per sq. in.;
t = thickness of plate in inches;
D = diameter (internal) of boiler in inches;
E = efficiency of longitudinal joint—i.e. proportionate strength of joint to plate;
F = factor of safety;
f = tensile strength of plate in lbs. per sq. in.;

then,

$$P = \frac{2Eft}{DF}, \text{ or } t = \frac{PDF}{2Ef}.$$

The factor of safety is usually between 4 and 6. Frequently rules are used in which the safe working stress is substituted for $\frac{f}{F}$ in the above.

For permissible values of f, F, or $\frac{f}{F}$, the rules of the Boiler Survey or Insurance Society concerned should be consulted.

For 'Standard Conditions for Marine Boilers,' see page 1563.

The proportionate strength of joint to plate—i.e. the efficiency of the joint—depends upon the type of joint, and also on the pitch and diameter of the rivets, and thickness of plate.

Typical percentage figures for single riveted lap joints are 40 to 58; for double riveted lap 55 to 70; for double butt (a butt strap each side of the plate and double riveted) 72 to 77; for treble riveted butt, each alternate rivet in outer rows being omitted, 82 to 85.

For the method of calculating the efficiency of riveted joints, see page 1567.

There is a strong tendency now in favour of butt joints for the longitudinal seams, by which the circular form of the shell is maintained, and there is less liability to grooving. The butt strap also protects the joint from corrosion.

All longitudinal seams break joint, as it is termed—that is, the seam in one ring of plates comes opposite to the solid plate in the next ring.

The efficiency of a riveted joint depends on the combination of dimensions as well as on the type of the joint, and there is no fixed relation between efficiencies of the different types of joints. However, for strength and sizes of material used in present-day boiler construction, the relative efficiencies obtainable with different types of joints are approximately as follows: Single-riveted lap joints, 57 per cent.; double-riveted lap joints, 74 per cent.; butt and double-strap joint, double-riveted, 82 per cent.; butt and double-strap joint, triple-riveted, 87 per cent.; and butt and double-strap joints, quadruple-riveted, 94 per cent.

(*'Power,'* Feb. 22, 1927.)

Working Strength of Boiler Flues.

(*Modern Dimensions.*)

P = working pressure, in lbs. per sq. in.
D = diameter, in inches.
L = length, in ft., between seams or hoops.
t = thickness of plate, in inches.

$$P = \frac{806300t^{2\cdot19}}{D \times L} \div \sqrt{\frac{300}{L}} \quad (Harrison).$$

$$P = \frac{50t^2}{D\sqrt{L}} - \frac{D}{L} \quad (M.\ Longridge).$$

(t being in 32nds of an inch.)

$$P = \frac{6149t^2}{\sqrt{DL}} \quad (W.\ H.\ Booth).$$

$$P = \frac{48000t^{2\cdot2}}{D\sqrt{L}} \quad (W.\ H.\ Booth).$$

TABLE OF 2·19 POWER FROM ¼ TO ⅝ INCH.
(*For Harrison's and Booth's formulæ.*)

Details.	In.	In.	In.	In.	In.
Thickness of plate . .	·25	·312	·375	·437	·500
Power, 2·19 . . .	·04802	·07829	·116715	·16358	·21915

Every case ought to be finally examined to see that the actual pressure on each square inch of section of plate must not exceed the safe crushing resistance of the material, or $P = 30000t \div Df$. This rule may need applying to test the cases of thick furnaces. The value of f is 3 to 4. The strength of flues can be indefinitely increased by the employment of flanged seams, or corrugations.

See also page 1567.

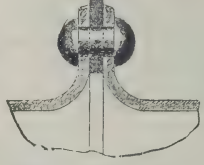


FIG. 4.—Flanged Seam.

The first form of joint in flue and furnace work was ordinary riveting, both transversely and longitudinally, but as weakness developed, the T iron hoop was tried; this, however, though strong, was found to be too rigid, and to lead to leakage in the seams. The flange out of the solid plate (fig. 4) was then designed by Daniel Adamson, and rapidly made its way to the front. The longitudinal joints are now welded, so that when the flange is adopted, no rivets whatever are exposed to the fire in the furnaces.

Pressures Required for Riveting and Flanging.

The pressure required for riveting is 75 to 125 tons per sq. in. of rivet area, preferably the lower value. Excessive pressure causes local deformation of the plate. The riveting pressure should be maintained for a short period, so that the rivet may cool down a little, otherwise the spring of the plate may stretch the rivet while it is still hot.

The pressure required for flanging boiler plates may be found from the formula:

$$F = \frac{2\cdot5(L \times t^2)}{L + 10},$$

where,

F = press load in tons; L = length run of flange in feet; t = thickness of plate in sixteenths of an inch. (*Mechanical World Year Book.*)

Welded Seams in Boilers.

The efficiency of welding depends essentially upon the skill and care of the workman who makes the weld; and in view of the uncertainty of the efficiency in practice as compared with tests, it is not advisable to use welds in parts of boilers subjected to tension, though parts in compression, such as furnaces and flue tubes, may be welded.

For particulars of oxy-acetylene and electric welding, see Section XL., Part II.

Many successful boiler repairs have been effected by welding, particularly in the making up of parts that have been wasted away, and the filling up of grooving.

Welded parts should be hydraulically tested to double the working pressure, the material at and around the weld being hammered while under test.

BOILER RULES AND REGULATIONS.

'Standard Conditions for the Design and Construction of Marine Boilers' have been recommended by a committee of representatives of the Board of Trade, British Corporation, Lloyd's Register, and Bureau Veritas. An abstract of the rules drawn up is given in the following pages. For boilers constructed to the requirements of a particular surveying body or boiler insurance company, the rules of the body concerned should be consulted.

Standard Conditions for the Design and Construction of Marine Boilers.*

(Abstract.)

MATERIALS OF CONSTRUCTION.

(Based on British Standards.)

Process of Manufacture.—Structural steel for marine boilers shall be made by the open-hearth process, acid or basic.

Freedom from Defects.—The finished material shall be free from cracks, surface flaws, and lamination. It shall also have a workmanlike finish, and must not have been hammer-dressed.†

Tensile Tests: Plates.—The tensile breaking strength of steel plates for shells and girders, determined from Standard test pieces, shall be between the limits of 28 tons and 35 tons per square inch, but a range of not more than 4 tons per square inch shall be permitted in any one case. For plates intended for flanging or welding, and for combustion chambers and furnaces, the tensile breaking strength shall be between the limits of 26 tons and 30 tons per square inch. The elongation, measured on a Standard test piece having a gauge length of 8 ins., shall be not less than 20 per cent. for material of $\frac{3}{4}$ in. in thickness and upwards, required to have a tensile breaking strength between the limits of 28 tons and 35 tons per square inch; and not less than 23 per cent. for material of $\frac{3}{4}$ in. in thickness and upwards, required to have a tensile breaking strength between the limits of 26 tons and 30 tons per square inch.

Stay Bars.—The tensile breaking strength of longitudinal stays shall be between the limits of 28 tons and 35 tons per square inch, with an elongation of not less than 20 per cent. measured on the Standard Test Piece B, but a range of not more than 4 tons per square inch shall be permitted in any one case. For steel bars for combustion chamber stays the tensile breaking strength shall be between the limits of 26 tons and 30 tons per square inch, with an elongation of not less than 23 per cent. measured on the Standard Test Piece B.

Where stay bars are tested on a gauge length of 4 times the diameter (Test Piece F), the elongation shall be 24 per cent. and 28 per cent. respectively.

The tensile breaking strength of angle and tee bars shall be between the limits of 28 and 32 tons per square inch, with an elongation of not less than 20 per cent. measured on the Standard Test Piece A.

For material under $\frac{3}{4}$ in. in thickness the elongation may be 3 per cent., but not more than 3 per cent., below the above-named elongations.

Wherever practicable the rolled surfaces shall be retained on two opposite sides of the test piece.

Rivet Bars.—The tensile breaking strength of rivet bars shall be between the limits of 26 tons and 30 tons per square inch of section, with an elongation of not less than 25 per cent. measured on the Standard Test Piece B or 30 per cent. measured on the Standard Test Piece F. The bars may be tested the full size as rolled.

Bend Tests: Cold Bends.—Test pieces shall be sheared lengthwise or crosswise from plates or bars, and shall not be less than $1\frac{1}{2}$ ins. wide, but for small bars the whole section may be used. For rivet bars bend tests are not required.

---

* Published by His Majesty's Stationery Office.

† The Board of Trade have more detailed instructions which they propose to retain.

Temper Bends.—The test pieces shall be similar to those used for cold bend tests. For temper bend tests the samples shall be heated to a blood-red colour and quenched in water at a temperature not exceeding 80° F. The colour shall be judged indoors in the shade.

In all cold bend tests, and in temper bend tests on samples 0·5 in. in thickness and above, the rough edge or arris caused by shearing may be removed by filing or grinding, and samples 1 in. in thickness and above may have the edges machined, but the test pieces shall receive no other preparation. The test pieces shall not be annealed unless the material from which they are cut is similarly annealed, in which case the test pieces shall be similarly and simultaneously treated with the material before testing.

For both cold and temper bends the test piece shall withstand, without fracture, being doubled over until the internal radius is equal to 1½ times the thickness of the test piece, and the sides are parallel.

For small sectional material these bend tests may be made from the flattened bar.

Bend tests may be made either by pressure or by blows.

Tests for Manufactured Rivets.—Rivets selected by the Inspector from the bulk shall withstand the following tests:—

(*a*) The rivet shanks are to be bent cold, and hammered until the two parts of the shank touch, without fracture on the outside of the bend.

(*b*) The rivet heads are to be flattened, while hot, in the usual manner, without cracking at the edges. The heads are to be flattened until their diameter is 2½ times the diameter of the shank.

FORMS OF BRITISH STANDARD TENSILE TEST PIECES.

For Plates and other Structural Material.

TEST PIECE A.

X.—For thicknesses over ¾ths in.: Maximum width allowed = 1½ ins. Y.—For thicknesses ⅜ths in. to ¾ths in.: Maximum width allowed = 2 ins. Z.—For thicknesses under ⅜ths in.: Maximum width allowed = 2½ ins.

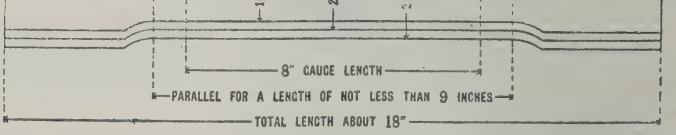


FIG. 5.

NOTE.—It will be observed that the widths given above, being maxima, do not exclude the use of the usual 1¼ ins. by 8 ins. test pieces.

TEST PIECE B.

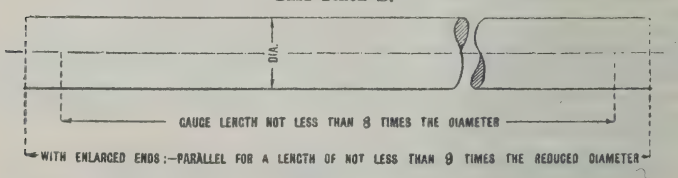


FIG. 6.

TEST PIECE F.

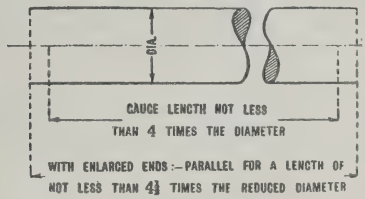


FIG. 7.

SPECIAL IRON FOR SCREW STAYS FOR COMBUSTION CHAMBERS.

In order that iron screw stays may be approved of the same size as would be required for mild steel, the iron must withstand the following tests:—

Tensile Tests.—The tensile breaking strength shall not be less than 21½ tons per square inch, with an elongation of not less than 25 per cent. measured on the Standard Test Piece B, or of 30 per cent. measured on the Standard Test Piece F.

Bend Tests.—Test pieces either of the bar as rolled, or turned down to 1 in. diameter, shall stand bending cold until the sides are parallel and the space between the two sides is not greater than the diameter of the test piece.

Diameters and Thickness of Plain and Stay Tubes.—The following table may be worked to:—

Outside Diameter in Inches.	Standard Thicknesses in L.S.G.				Suitable for Working Pressures of Lbs. per Square Inch.			
Inches.	A	B	C	D	A	B	C	D
2	—	11	10	9	—	155	215	300
2¼	11	10	9	8	140	190	260	315
2½	11	10	9	8	125	175	230	300
2¾	11	10	9	8	110	160	215	275
3	10	9	8	7	140	190	250	300
3¼	10	9	8	7	130	180	230	280
3½	10	9	8	7	120	165	215	260

The minimum thickness of stay tubes measured under the threads shall be ¼ in. for marginal stay tubes, and $\frac{5}{16}$ in. for other stay tubes.

Thickened Ends.—If stay tubes are required to have their thickness increased at the screwed ends, so that the thickness at the bottom of thread is practically the same as in the body of the tube, the thickening is to be attained by upsetting and not by any welding process, and the tubes are to be annealed after the upsetting.

Screwed Threads to be Continuous.—Stay tubes must be screwed at both ends with a continuous thread, which should not be finer than 10 threads per inch. It is desirable, however, that they should be screwed to the standard 9 threads per inch, therefore, after 30th June, 1921, this should be the rule.

MAIN BOILERS, CYLINDRICAL TYPE.

GENERAL CONDITIONS AS TO DESIGN, WORKMANSHIP, HYDRAULIC TEST, ETC.

In the design and construction of marine boilers the following conditions should be observed:—

Riveting of End Seams.—The riveting of the seams joining the end plates to the cylindrical shell shall be not less than 42 per cent. of that of the solid plate. Where the shell plates exceed ⅝ in. in thickness the seams connecting the shell plates to the end plates are to be double riveted

Other Circumferential Seams.—The circumferential seam at or near the middle of the length of single-ended boilers should have a strength of joint not less than 60 per cent. of the solid plates. The inner circumferential seams of double-ended boilers should have a strength of joint not less than 62 per cent. of the solid plate. In any case there shall be three rows of rivets when single-ended boilers have shell plates over 1⅜ ins. in thickness, and when double-ended boilers have shell plates over 1⅜ ins. in thickness. Where the shell plates exceed ½ in. in thickness the intermediate circumferential seams of double-ended boilers are to be at least double riveted.

Welding and other Treatment of Plates.—No steel plates subject to a direct tensional stress are to be welded except where the weld is covered by a butt strap or straps. For small steam domes, by special permission where the welding is done by hammer and the plates do not exceed ½ in. in thickness, the straps may be omitted. The strength shall, in such cases, be assumed to be 50 per cent. of that of the solid plate. All steel plates which are welded, dished, flanged, or locally heated, are to be afterwards efficiently annealed.

Butt Straps.—Butt straps must be cut from plates, and not from rolled strip.

Rivet Holes.—All rivet holes must be drilled 'fair,' and as far as possible they should be drilled in place. After drilling the plates the burrs should be removed and the faying surfaces of the plates cleaned, and the sharp outer edges of holes removed also.

Stays not to be Welded. Annealing of Fire-worked Stay Bars.—No steel stays are to be welded. If plus threads are desired, the ends of the stay bars may be upset or the bars may be drawn down in the central portions from bars originally of the size of the ends. In either of these two cases the bars must be subsequently annealed throughout.

In double-ended boilers the through longitudinal stays must be supported at or near the middle of their length.

Threads of Screw and other Stays.—All longitudinal stays and also screw stays should have threads in accordance with the British Standard Specification, and true to pitch, viz. :—

After the 30th June, 1921, all screw stays 1½ ins. in diameter and upwards should have 9 threads per inch, and all stays 2 ins. in diameter and above passing through plates and secured by nuts on each side of the plate should have not more than 6 threads per inch.

Compensation Rings to Manholes.—Where the cylindrical shell is cut for a manhole, compensation must be provided and must be such that the strength in way of the hole is not less than that required for the longitudinal joint.

Hydraulic Tests of New Boilers.—In all new boilers working at pressures up to 100 lbs. per square inch the hydraulic test must be twice the working pressure. For boilers working at pressures greater than 100 lbs. per square inch the hydraulic test pressure must be 1½ times the working pressure plus 50 lbs. per square inch.

RULES FOR DETERMINING THE WORKING PRESSURE TO BE ALLOWED ON VARIOUS PARTS OF CYLINDRICAL MARINE BOILERS.

Cylindrical Shells.

Formula for Working Pressure of the Shell.—For the cylindrical shells of steel marine boilers the maximum working pressure (which is designated by W.P. and is in lbs. per square inch) to be allowed shall be calculated from the following formula :—

Where the thickness of the shell plates does not exceed 1¼ ins.,

$$\text{W.P.} = \frac{(t - 2) \times \text{S} \times \text{J}}{\text{C} \times \text{D}}$$

Where the thickness of the shell plates exceeds 1¼ ins. and double butt straps are fitted,

$$\text{W.P.} = \frac{t \times \text{S} \times \text{J}}{2{\cdot}85 \times \text{D}}$$

where,

t is the thickness of the shell plates in 32nds of an inch,

S is the minimum tensile strength of the steel shell plates in tons per square inch,

J is the percentage of strength of the longitudinal seams calculated by the methods described on page 1567.

C is a coefficient, which is 2·75 when the longitudinal seams are made with double butt straps; 2·83 when the longitudinal seams are made with lap joints and are treble riveted; 2·9 when they are made with lap joints and are double riveted; and 3·3 when they are made with lap joints and are single riveted.

D is the inside diameter of the outer strake of plating of the cylindrical shell measured in inches.

N.B.—The factor of safety must be in no case less than 4.

Methods of Calculating the Strength of Riveted Joints.—The percentage of strength of a riveted joint (J) is found from the following formulæ (I), (II), (III): (I) and (II) are applicable to any type of joint; (III) is applicable only to that type of joint in which the number of rivets in the inner rows is double that in the outer row. The lowest value given by the application of these formulæ is to be taken as the percentage of strength of the joint.

(I) Percentage of strength of plate at joint as compared with solid plate $= \dfrac{100\,(p-d)}{p}$

(II) Percentage of strength of rivets as compared with the solid plate $= \dfrac{100\,(S_2 \times a \times n \times C)}{S_1 \times p \times T}$

(III) Percentage of combined strength of the plate at the inner row of rivet holes and of the rivets in the outer row

$$= \frac{100\,(p-2d)}{p} + \frac{100\,(S_2 \times a \times C)}{S_1 \times p \times T},$$

where,

p = pitch of rivets at outer rows in inches;
d = diameter of rivet holes in inches;
a = sectional area of one rivet in square inches;
n = number of rivets which are fitted in the pitch p;
T = thickness of plate in inches;
C = 1·0 for rivets in single shear as in lap joints;
C = 1·875 for rivets in double shear as in double butt-strapped joints;
S_1 = minimum tensile strength of plates in tons per square inch;
S_2 = shearing strength of rivets, which is taken generally to be 23 tons per square inch, and may be 85 per cent. of the minimum tensile strength of the rivet bars.

Thickness of Butt Straps.—Where the longitudinal seams are fitted with double butt-strapped joints, the outer butt strap should have at least 0·625 of the strength of the plate, and should be of sufficient thickness to permit of efficient caulking of its outer edges. The inner butt strap should be $\frac{4}{8}$ in. thicker than this.

In cases where the number of rivets in the inner rows is double the number in the outer row, this will require the thickness of the outer strap to be

$$t = \frac{5 \times (p-d)}{8 \times (p-2d)} \times T,$$

and that of the inner strap to be at least

$$t_n = \frac{5 \times (p-d)}{8 \times (p-2d)} \times T + \tfrac{4}{8} \text{ in.}$$

Distances between rows of Rivets and between Rivets and Plate Edges.—In all cases the clear space between a rivet hole and the edge of a plate should not be less than the diameter of the rivet hole—i.e. the centre of the rivet hole should be at least $1\frac{1}{2}$ diameters distant from the edge of the plate.

In joints, whether lapped or fitted with butt straps, in which there are more than one row of rivets and in which there is an equal number of rivets in each row, the distance between the rows of rivets should be not less than $0{\cdot}33p + 0{\cdot}67d$, with zigzag riveting, or $2d$ with chain riveting.

In joints in which the number of rivets in the outer rows is one-half of the number in each of the inner rows, and in which the inner rows are chain riveted, the distance between outer rows and the next rows should be not less than $0{\cdot}33p + 0{\cdot}67d$ or $2d$, whichever is the greater, and the distance between the rows in which there are the full number of rivets should be not less than $2d$.

In joints in which the number of rivets in the outer rows is one-half of the number in each of the inner rows, and in which the inner rows are zigzag, the distance between the outer rows and the next rows should be not less than $0{\cdot}2p + 1{\cdot}15d$, and the distance between the rows in which there are the full number of rivets should be not less than $0{\cdot}165p + 0{\cdot}67d$.

In the above p is the pitch of the rivets in the outer rows.

Maximum Pitch of Rivets in Longitudinal Joints.—The maximum pitch of the rivets in the longitudinal joints of boiler shells is to be—

$$\text{Maximum pitch in inches} = C \times T + 1\tfrac{5}{8} \text{ ins.}$$

where T is the thickness of the plate in inches and C is a coefficient as given in the following table:—

Number of Rivets per Pitch.	Coefficients for Lap Joints.	Coefficients for Double Butt-strapped Joints.
1	1·31	1·75
2	2·62	3·50
3	3·47	4·63
4	4·14	5·52
5	—	6·00

FURNACES.

Plain Furnaces.—The working pressure to be allowed on plain furnaces or furnaces strengthened by Adamson or other joints, and on the cylindrical bottoms of combustion chambers, is to be determined by the following formulæ, the least pressure obtained by either formula being taken:—

$$\text{W.P.} = \frac{C\,(t-1)^2}{(L+24) \times D}, \text{ or W.P.} = \frac{C_1}{D} \times [10\,(t-1) - L].$$

where D is the external diameter of the furnace or combustion chamber bottom in inches;

t is the thickness of the furnace plate in 32nds of an inch;

L is the length of the furnace or of combustion chamber bottom or the length between points of substantial support, in inches, measured from the centres of rivet rows or from the commencement of flange curvature, whichever is applicable;

C = 1,450 where the longitudinal seams are welded, and 1,300 where they are riveted;

C_1 = 50 where the longitudinal seams are welded, and 45 where they are riveted.

W.P. = working pressure in lbs. per square inch.

Corrugated Furnaces.—The working pressure to be allowed on corrugated furnaces is to be determined by the following formula:—

$$\text{W.P.} = \frac{C\,(t-1)}{D},$$

where D is the external diameter measured at the bottom of the corrugations in inches;

t is the thickness of the furnace plate in 32nds of an inch, measured at the bottom of the corrugation or camber;

C is a coefficient which is 480 for the Fox, Morison, Deighton, Purves, and other similar furnaces, 510 for the Leeds Forge Bulb Suspension furnace.

N.B.—No furnace, plain or corrugated, should exceed $\frac{13}{16}$ in. in thickness.

FLAT PLATES SUPPORTED BY STAYS SECURED IN VARIOUS WAYS.

Flat Plates supported by Screwed Stays.—The working pressure to be allowed on flat plates supported by stays is to be calculated by the following formula:—

$$\text{W.P.} = \frac{(t-1)^2 \times C}{a^2 + b^2},$$

but if steel of less tensile strength than 26 tons per square inch is used the working pressure allowed shall be correspondingly reduced.

In this formula,

W.P. is the working pressure in lbs. per square inch;

t is the thickness of the flat plate in 32nds of an inch;

a is the distance apart of the rows of stays in inches;

b is the pitch of the stays in the rows in inches;

C is a coefficient which varies with the method of fixing the stays as follows:—

Where the plates are exposed to flame and the stays are screwed into the plate and the ends are riveted over, C = 50.

Where the plates are not exposed to flame and the stays are screwed into the plate and their ends are riveted over, C = 57.

In these cases the thickness of the plate must be at least half the diameter of the stay required by the rule.

Where stay tubes are screwed into tube plates and expanded, C = 52. If they are fitted with nuts, C = 72.

Where the plates are exposed to flame and the stays are screwed into the plate and fitted with nuts on the outside, C = 75. Where the plates are not exposed to flame, C = 86.

Where the stays pass through the plates not exposed to flame and are fitted with nuts inside and outside, C = 96.

Where plates are stiffened by flanging, the inner radius of which is not greater than 2½ times the thickness of the plate, for the support thus given C = 110 when the plates are not exposed to flame, and C = 96 where they are exposed to flame. The pitch is to be reckoned from the commencement of the curvature.

GIRDERS.

Girders Supporting Combustion Chamber Tops.—For girders supporting the tops of combustion chambers the following formula is to be used :—

$$\text{W.P.} = \frac{\text{C} \times d^2 \times t}{(\text{L}-\text{P}) \times \text{D} \times \text{L}} \times \frac{\text{S}}{28},$$

where,

d is the depth of the girder at centre in inches ;

t is the thickness of the girder at centre, when this is a forging, or the sum of the thicknesses of the plates when the girder is made of two plates, measured in 32nds inch ;

L is the length of the girder in inches, measured between the tube plate and back chamber plate inside, or between tube plates in chambers common to two opposite furnaces ;

P is the pitch of stays supported by the girder, in inches ;

D is the distance apart of the girders, centre to centre, in inches ;

S is the minimum tensile strength of the steel plates forming the girder, in tons per square inch. In the case of forged girders S is to be taken as 24 for iron and 28 for steel ;

C is a coefficient as follows :—

$$\text{C} = \frac{n}{n+1} \times 495,$$

when the number of stays in each girder is odd, and

$$= \frac{n+1}{n+2} \times 495,$$

when the number of stays in each girder is even, n being the number of stays to each girder.

STAYS.

Iron or Steel Screw Stays to Combustion Chambers.—For screw stays with threads not coarser than 9 threads per inch, made of steel or of special wrought iron tested to the requirements, the following formula is to be used :—

$$\text{W.P.} = \frac{(d - 0{\cdot}267)^2 \times 8{,}250}{a},$$

where,

d is the diameter of the stay over the thread in inches ;

a is the area in square inches supported by one stay.

But in no case must the stress exceed 9,000 lbs. per square inch of section.

Steel Longitudinal Stays.—For steel longitudinal stays with threads not coarser than 6 threads per inch, the working pressure is to be calculated from the following formula :—

$$\text{W.P.} = \frac{(d - 0{\cdot}340)^2 \times 9{,}500}{a} \times \frac{S}{28},$$

where d is the diameter of the stay over the thread in inches ;

a is the area in square inches supported by one stay ;

S is the minimum tensile strength of the steel in tons per square inch.

But in no case must the stress exceed 11,000 lbs. per square inch of section, when steel of a minimum tensile strength of 28 tons per square inch is used.

In cases where longitudinal stays are made with enlarged ends, and the body of the stay is smaller in diameter than at the bottom of the thread, and in cases where coarser threads than 6 per inch are used, the working pressure is to be calculated from the following formula :—

$$\text{W.P.} = \frac{(d_1 - 0{\cdot}125)^2 \times 9{,}500}{a} + \frac{S}{28},$$

where d is the diameter of the stay at the bottom of the thread or at the smallest part of the body.

Stay Tubes.—On stay tubes, whether of wrought iron or of lap-welded steel, a working stress of 7,500 lbs. per square inch of the net sectional area at the bottom of the thread is permitted.

Dished Ends (Convex Outside).—For ends of steam chests, etc., dished to partial spherical form, the following formula is to be used :—

$$\text{W.P.} = \frac{15 \times S\,(t - 1)}{R},$$

where

W.P. is the working presures in lbs. per square inch ;

t the thickness in 32nds of an inch ;

R the inner radius of curvature of the end in inches, which shall not exceed the diameter of shell ;

S the minimum tensile strength of plates in tons per square inch.

Safety Valves.—At least two safety valves must be fitted to each boiler. They must be arranged so that the springs and valves are cased in, that the valves cannot be overloaded when steam is up, that they can be lifted by easing gear, and turned round on their seats by hand, and in case of fracture of springs they cannot lift out of their seats.

The easing gear must be arranged to lift all the safety valves on a boiler together, and must be workable from some accessible place free from steam danger.

All the safety valves of each boiler may be fitted in one chest, which must be separate from any other valve chest and which must be connected direct to the boiler by a strong and stiff neck, the passage through which should have a cross-sectional area at least equal to one-half the aggregate area of the safety valves in the chest.

Each safety valve box shall have a means of draining it ; the pipe shall lead to the bilge or tank clear of the boiler.

Size of Safety Valves.—The minimum aggregate area of the safety valves of the ordinary type in each boiler, whether coal-fired or oil-fired, and whether working under natural, forced, or induced draught, shall be found by the following formula :—

Aggregate area of safety valves in square inches

$$= \text{Total heating surface of boilers in square feet} \times \left(\frac{K}{p + 15}\right),$$

where p is the working pressure in lbs. per square inch.

For coal-fired boilers K is 1·25, for oil-fired boilers and boilers with closed stokehold forced draught K = 1·5.

Accumulation Test of Safety Valves.—All safety valves must be set to the required pressure under steam. During a test of 15 minutes with the stop valves closed, and under full firing conditions, the accumulation of pressure must not exceed 10 per cent. of the loaded pressure. During this test no more feed-water should be supplied than is necessary to maintain a safe working water level.

Superheater Safety Valves and Drains.—Where a superheater is fitted which can be shut off from the boiler it must have a separate safety valve fitted with easing gear, etc. The valve as regards construction must comply with the regulations for ordinary safety valves, but the easing gear may be fitted to be workable from the stokehold only.

The superheater must be fitted with a drain cock or valve to free it from water when necessary.

BOILER TUBES.

Approximate Weight of Iron Boiler Tubes.

PER FOOT RUN.

External Diameter.	11 S.W.G.	10 S.W.G.	9 S.W.G.	8 S.W.G.	¼-inch	$\frac{5}{16}$-inch.
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
$2\frac{1}{2}$	2·89	3·18	3·55	3·92	5·89	7·16
$2\frac{3}{4}$	3·20	3·51	3·93	4·34	6·54	7·97
3	3·50	3·85	4·30	4·75	7·20	8·79
$3\frac{1}{4}$	3·80	4·18	4·68	5·17	7·85	9·61
$3\frac{1}{2}$	4·11	4·52	5·06	5·59	8·50	10·4
$3\frac{3}{4}$	4·41	4·85	5·43	6·01	9·16	11·2
4	4·72	5·19	5·81	6·43	9·81	12·0
$4\frac{1}{4}$	5·02	5·52	6·19	6·85	10·4	12·8
$4\frac{1}{2}$	5·32	5·86	6·57	7·27	11·1	13·7
$4\frac{3}{4}$	—	6·19	6·94	7·69	11·7	14·5
5	—	6·53	7·32	8·11	12·4	15·3
$5\frac{1}{4}$	—	6·86	7·70	8·52	13·0	16·1
$5\frac{1}{2}$	—	—	8·07	8·94	13·7	16·9
$5\frac{3}{4}$	—	—	8·45	9·36	14·4	17·8
6	—	—	8·83	9·78	15·0	18·6

Surface of Boiler Tubes.

PER FOOT RUN.

External Diameter.	Surface.	External Diameter.	Surface.	External Diameter.	Surface.
Ins.	Sq. Ft.	Ins.	Sq. Ft.	Ins.	Sq. Ft.
$\frac{3}{4}$	·1964	$1\frac{3}{4}$	·4582	$3\frac{1}{4}$	·8509
$\frac{7}{8}$	·2291	$1\frac{7}{8}$	·4909	$3\frac{1}{2}$	·9163
1	·2618	2	·5236	$3\frac{3}{4}$	·9818
$1\frac{1}{8}$	·2945	$2\frac{1}{8}$	·5563	4	1·047
$1\frac{1}{4}$	·3273	$2\frac{1}{4}$	·5891	$4\frac{1}{2}$	1·178
$1\frac{3}{8}$	·3600	$2\frac{1}{3}$	·6545	5	1·307
$1\frac{1}{3}$	·3927	$2\frac{3}{4}$	·7200	$5\frac{1}{2}$	1·440
$1\frac{5}{8}$	·4254	3	·7584	6	1·571

BRITISH STANDARD SPECIFICATION FOR CHARCOAL IRON LAPWELDED BOILER TUBES.*

(No. 43—1927.) (*Abstract.*)

1. The tubes shall be lapwelded and shall be made from genuine charcoal iron of the best quality. 2. The tubes shall be sound, clean, smooth and well finished, free from surface defects, rust and longitudinal grooving, both internally and externally, and the ends shall be clean and square. 3. The tubes shall be carefully annealed at both ends. 6. The tubes or strips cut from the tubes shall show a tensile strength between the limits of 19 and 24 tons per square inch inclusive, with a contraction of area of the metal of not less than 45 per cent. 7. The tubes

* By permission of the British Engineering Standards Association.

shall stand bulging both hot and cold with a parallel drift (fig. 8) without showing either crack or flaw, until the diameter of the bulged end exceeds the original diameter of the tube by not less than 15 per cent. when tested hot and by not less than 10 per cent. when tested cold. 8. A piece

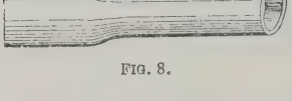


FIG. 8.

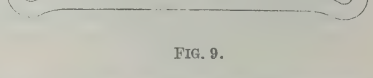


FIG. 9.

of tube 2 ins. long shall be placed on end and shall stand hammering down until it is reduced to 1⅛ in. high, without showing crack or flaw. 9. The tubes shall, when cold, stand flattening (fig. 9) without showing crack or flaw until the interior surfaces of the tubes meet. 10. Each tube shall be tested by an internal hydraulic pressure of at least 750 lbs. per sq. in., and any tubes failing to stand this test shall be rejected.

BRITISH STANDARD SPECIFICATION FOR COLD DRAWN WELDLESS STEEL TUBES FOR LOCOMOTIVE BOILERS.*

(No. 53—1927.) (*Abstract*).

Quality of Material.

1. The tubes shall be cold drawn and weldless, and shall be manufactured from steel of the best quality made by the Open-Hearth process. They shall not show on analysis more than 0·035 per cent. of sulphur or more than 0·030 per cent. of phosphorus. The manufacturer shall supply an analysis when required to do so.

Freedom from Defects.

2. The tubes shall be perfectly sound, clean, smooth, and well finished, free from surface defects, rust and longitudinal grooving, both internally and externally, and the ends shall be clean and square.

Annealing.

3. The tubes shall be carefully annealed throughout their length after the operation of drawing and the ends shall be carefully annealed after the process of enlarging or decreasing the diameter.

Permissible Variation in Weight.

4. The actual weight of each tube shall not differ from the calculated weight by more than 5 per cent. above or 2½ per cent. below the weight calculated from the specified dimensions, allowing 489·6 lbs. per cubic foot of metal in the tubes.

Tensile Test.

6. The tubes or strips cut from the tubes shall, without further annealing, show a tensile strength of not less than 20 tons per sq. in. nor more than 26 tons per sq. in. with an elongation of not less than 28 per cent. in 8 ins. for tubes, and not less than 22 per cent. in 8 ins. for strips cut from the Tubes. If strips are cut for testing from tubes over 2¼ ins. diameter they may be annealed before testing.

Crushing down Test.

8. A piece of tube 2 ins. long shall be placed on end and shall, when cold, stand hammering down until it is reduced to 1⅛ ins. high, without showing either crack or flaw.

Flattening Test.

9. The tubes shall, when cold, stand flattening (fig. 8), without showing either crack or flaw until the interior surfaces of the tubes meet or, when the tubes are over 11 S.W.G. in thickness, until the interior surfaces are brought to a distance apart equal to the thickness of the wall of the tube.

Hydraulic Test.

10. Every tube shall be tested by an internal hydraulic pressure of at least 1,000 lbs. per sq. in., and any tubes failing to stand this test shall be rejected.

* By permission of the British Engineering Standards Association.

TYPES OF BOILERS.

LANCASHIRE AND CORNISH BOILERS.

The distinction between these two general types is that the Cornish has one and the Lancashire two internal flues.

These boilers were originally made with iron plates, and had flat ends (often in two or three pieces), the flat surfaces being strengthened with rod and gusset stays.

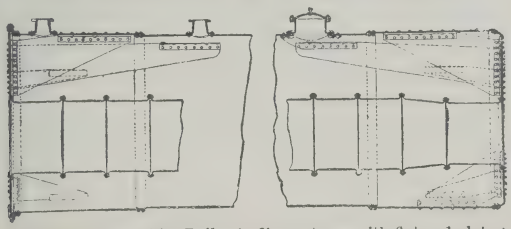


FIG. 10.—Lancashire Boiler (ordinary type, with flat end plates).

When steel took the place of iron, and boiler makers were able to obtain larger plates, the end plates of the largest boilers were then made in one piece, and the shells constructed of seven or nine belts, each in one plate. Fig. 10 shows a section of one of these Lancashire boilers as it is still being made by some firms.

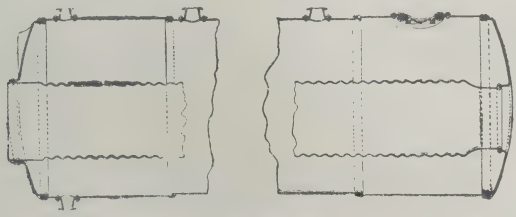


FIG. 11.—John Thompson Type, with dish end plates and corrugated flue tubes.

All internally-fired boilers are subjected to straining due to unequal expansion and contraction, the flue tubes and furnace tubes becoming hotter than the shell while the boiler is at work.

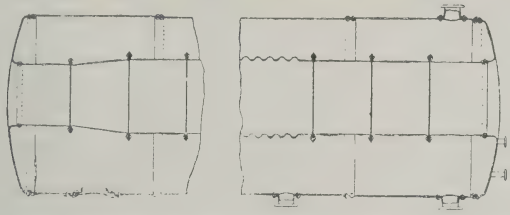


FIG. 12.—John Thompson Type, with flue tubes corrugated for part of their length.

Makers of boilers having flat end plates provide for expansion by the flexibility of the end plates.

Some makers avoid the staying required in the case of flat end plates by using dish end

plates (fig. 11), and provide for the relief of expansion stresses by making the flue tubes corrugated, either for their whole length (fig. 11), or for part of their length (fig. 12).

A length of corrugated flue tube is used by makers of flat end boilers also, for high pressures.

THE GALLOWAY BOILER.

The general design and details of the Galloway Boiler (*Galloways, Ltd., Manchester*) are shown by fig. 13. The boiler has two furnaces at the front, joining behind the bridges into one flue of special shape, containing the well-known Galloway cone tubes, and side pockets; otherwise, and in general appearance, it is somewhat similar to the two-flued or Lancashire boiler.

The advantages claimed for the Galloway boiler are:—

Increased heating surface—about 10 per cent. additional.

Improved circulation, giving more uniform temperature and reduced expansion strains. This circulation, also, largely prevents the deposit of solid matter upon the furnaces and adds to the durability of the boiler.

Improved heat transmission, owing to good circulation and to sinuous path of flue gases past the cone tubes.

Increased output and higher efficiency.

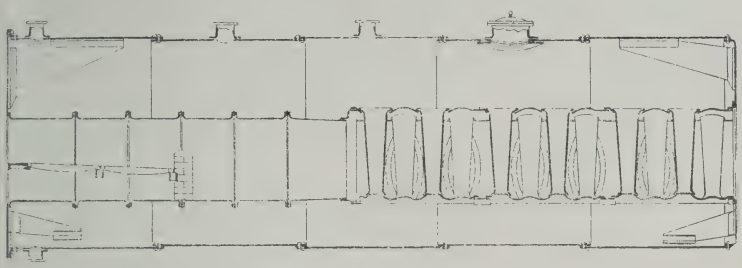
DETAILS OF STANDARD 'GALLOWAY' BOILERS

Size. Length × Diam.	No. of Cone Tubes.	Heating Surface.	Normal Grate Area.	Outside Diam. of Furnace.	Steam Evaporated per Hour, Lbs.			Approximate Net Weight.	
					Coal burnt per Sq. Ft. Grate per Hour.			Working Pressure.	
					20 lbs.	25 lbs.	30 lbs.	120 lbs.	160 lbs.
Ft. Ft.In.		Sq. Ft.	Sq. Ft.	Ft. In.	Lbs.	Lbs.	Lbs.	Tons.	Tons.
14×5 6	6	300	14·5	2 1	2,500	3,100	3,700	7	—
16×5 6	8	350	16	2 1	2,700	3,400	4,100	7·5	—
18×5 6	8	395	17·2	2 1	2,900	3,600	4,400	8	—
16×6 0	8	395	19·5	2 4	3,300	4,100	5,000	8	—
18×6 0	11	445	21	2 4	3,500	4,400	5,300	8·75	—
20×6 0	11	495	22·5	2 4	3,800	4,800	5,700	9·5	—
22×6 0	14	555	22·5	2 4	3,800	4,800	5,700	10·3	—
20×6 6	15	570	23·2	2 7	3,900	4,900	5,900	10·5	12·5
22×6 6	15	605	25	2 7	4,200	5,300	6,400	11	13·25
24×6 6	18	675	27·5	2 7	4,600	5,800	7,000	11·8	14·5
26×6 6	18	725	30	2 7	5,100	6,400	7,600	13	15
24×7 0	18	735	29·7	2 9	5,100	6,300	7,600	14·5	17
26×7 0	20	795	32	2 9	5,400	6,800	8,200	15·5	18
28×7 0	20	850	32	2 9	5,400	6,800	8,200	16·5	19
30×7 0	25	925	32	2 9	5,400	6,800	8,200	17·5	20
28×7 6	20	925	35	3 0	5,900	7,400	8,900	17·75	22
30×7 6	25	1,000	35	3 0	5,900	7,400	8,900	19	23·5
28×8 0	25	1,070	38	3 3	6,500	8,100	9,700	20	25
30×8 0	25	1,120	38	3 3	6,500	8,100	9,700	21	26
30×8 6	25	1,220	40	3 5	6,800	8,500	10,200	23·5	28·3
30×9 0	25	1,310	43	3 8	7,300	9,100	11,000	25·5	31·5

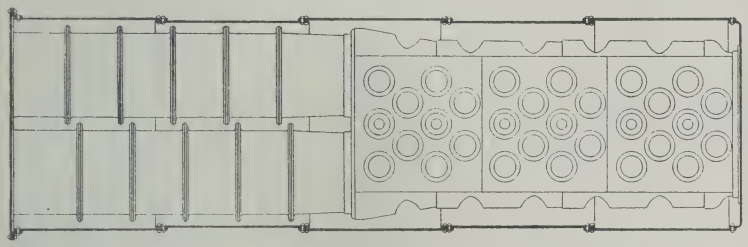
BASIS.—8·5 lbs. of water evaporated per lb. of coal. Boiler supplied with hot feed.

On page 1576 are results of tests made by the British Engine, Boiler, and Electrical Insurance Co., Ltd., on two adjacent boilers working into the same chimney, one being a Lancashire boiler and the other a Galloway patent boiler, but otherwise of similar size and seated in similar manner.

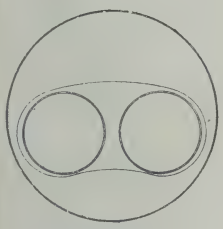
THE GALLOWAY BOILER.



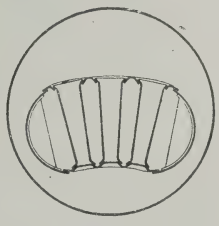
Longitudinal Sectional Elevation.



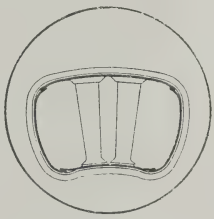
Longitudinal Sectional Plan.



Cross Section behind Fire Grates.



Cross Section through Galloway Flue.



Cross Section at Back End.

FIG. 13.

TESTS OF LANCASHIRE AND GALLOWAY BOILERS.

Details.	Lancashire Boiler.	Galloway Boiler.
Diameter of boiler	8 ft.	8 ft.
Length of boiler	30 ft.	30 ft.
Total heating surface	985 sq. ft.	1,164 sq. ft.
Fuel fired per hour	1,421 lbs.	1,424 lbs.
Combustible burnt per hour	1,348 lbs.	1,317 lbs.
Water evaporated per hour	10,565 lbs.	11,485 lbs.
Dried fuel fired per sq. ft. grate per hour	31 lbs.	31 lbs.
Water evaporated per lb. combustible from and at 212° F.	9·39 lbs.	10·39 lbs.

THE JOHN THOMPSON LANCASHIRE BOILER.

(*John Thompson (Wolverhampton), Ltd., Wolverhampton.*)

This is a boiler (fig. 14, p. 1578) of the dish-end type (figs. 11, 12). The end plates are dished to a convex form and increased in thickness to give the same factor of safety (as obtained with the stayed surface), thereby dispensing with all gusset and longitudinal rod stays. The inside, being free from stays, can be easily cleaned, particularly over the fire-boxes, where the steam is so largely generated.

The table on page 1577 gives some particulars of these boilers, which have been made for working pressures up to 230 lbs. per sq. in.

MULTITUBULAR BOILERS, LOCOMOTIVE TYPE.

THE ROBEY BOILER.

Figs. 15, 16 show a boiler, locomotive type, by Robey & Co. Ltd., of Lincoln. It is rated at 8 h.p. nominal, with enlarged grate suitable for burning inferior fuel, having a grate area of 6·43 sq. ft., and a total heating surface of 164·8 sq. ft. Its working pressure is 150 lbs. per sq. in. The tubes are of steel solid drawn.

The figure shows Messrs. Robey & Co.'s patent roof fire-box girders, which, being supported by the arch plate of external fire-box, take all compressive strains off the fire-box front plate and tube plate.

DIMENSIONS OF 8 H.P. LOCOMOTIVE BOILER.

Length of external fire-box	3 feet 5 inches	Height of internal fire-box	3 feet 4 inches
Length of boiler barrel	6 „ 1 „	Height of external fire-box	4 „ 4½ „
Length of smoke-box	1 „ 11 „	Diameter of boiler barrel	2 „ 9¼ „
Width of external fire-box	2 „ 10 „	Number of tubes	32
Width of internal fire-box	2 „ 3⅜ „	Diameter of tubes	2½ „
Length of internal fire-box	2 „ 10 1/16 „		

THICKNESS OF PLATES.

Boiler barrel	0 ft. ⅜ in.	Tube plates	0 ft. ⅝ in.
External fire-box	0 „ 7/16 „	Total length of boiler	11 „ 5 „
Internal fire-box	0 „ 7/16 „		
Heating surface in tubes	132·5 sq. ft.	Working pressure	150 lbs. per sq. in.
Heating surface in fire-box	32·3 „	Hydraulic test	250 „ „
Total heating surface	164·8 „	Steam test	150 „ „
Grate area	6·43 „		

Figs. 17, 18 (p. 1580) show a Great Northern Railway locomotive boiler for a 6-ft. 6-in. 4-wheel coupled bogie engine.

TABLE SHOWING DIMENSIONS OF JOHN THOMPSON LANCASHIRE BOILERS.

Diameter of Boiler.	Length of Boiler.	Working Pressure, in Lbs. per Sq. In.	Heating Surface, in Sq. Ft.	Grate Area, in Sq. Ft.	Approximate Evaporation, in Lbs. of Water per Hour.	I.H.P., based upon 20 lbs. of Steam per I.H.P. per Hour.	Diameter of Flues.	Thickness of Shell, in Inches.	Thickness of Flues, in Inches.	Thickness of Dish End, in Inches.	Thickness of Flat End, in Inches.
Ft. Ins.	Feet.						Ft. Ins.				
6 6	24	100	640	28	5,000	250	2 6	15/32	3/8	3/4	1/2
6 6	24	120	640	28	5,000	250	2 6	13/32	3/8	3/4	9/16
6 6	24	160	640	28	5,000	250	2 6	15/32	13/32	13/16	29/32
7 3	24	100	680	30	5,500	275	2 9	1/2	3/8	3/4	17/32
7 3	24	120	680	30	5,500	275	2 9	9/16	13/32	3/4	9/16
7 3	24	160	680	30	5,500	275	2 9	21/32	15/32	13/16	21/32
7 3	30	100	875	30	6,000	300	2 9	1/2	3/8	3/4	17/32
7 3	30	120	875	30	6,000	300	2 9	9/16	13/32	3/4	9/16
7 3	30	160	875	30	6,000	300	2 9	21/32	15/32	13/16	21/32
8 3	30	100	1,050	42	7,250	365	3 3	9/16	7/16	13/16	9/16
8 3	30	120	1,050	42	7,250	365	3 3	5/8	7/16	15/16	21/32
8 3	30	160	1,050	42	7,250	365	3 3	23/32	1/2	1	3/4
8 6	30	100	1,100	45	9,000	450	3 5	19/32	7/16	13/16	9/16
8 6	30	120	1,100	45	9,000	450	3 5	21/32	15/32	15/16	21/32
8 6	30	160	1,100	45	9,000	450	3 5	3/4	17/32	1	3/4
9 3	30	100	1,175	49	9,500	475	3 9	5/8	7/16	7/8	9/16
9 3	30	120	1,175	49	9,500	475	3 9	11/16	15/32	15/16	21/32
9 3	30	160	1,175	49	9,500	475	3 9	13/16	17/32	1 1/16	25/32

The above evaporations assume the use of mechanical stokers.

MODERN TYPE OF 'JOHN THOMPSON LANCASHIRE BOILER.' DISHED ENDS—SHELL IN FIVE RINGS—CORRUGATED FLUES THROUGHOUT.
(*John Thompson (Wolverhampton) Ltd., Wolverhampton.*)

Working Pressure 200 lbs. per Square Inch.

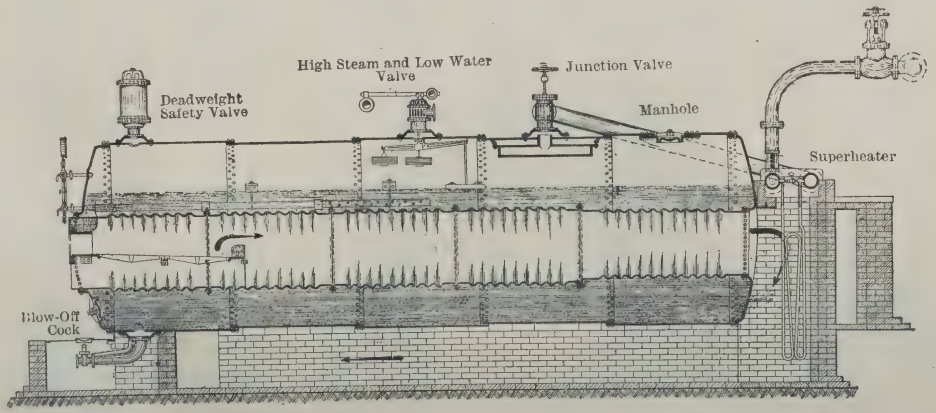


FIG. 14.

ROBEY MULTITUBULAR BOILER LOCOMOTIVE TYPE.

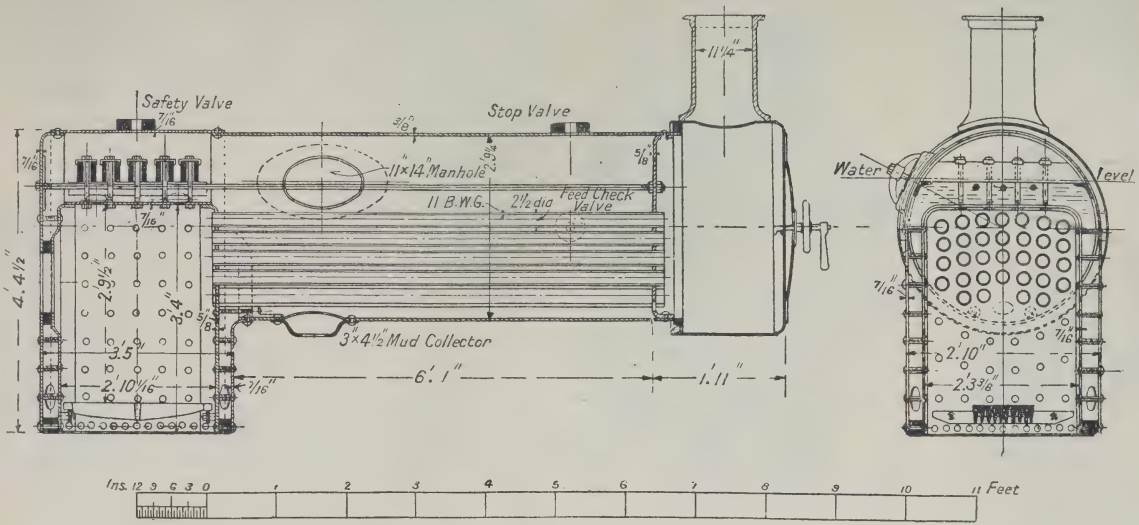
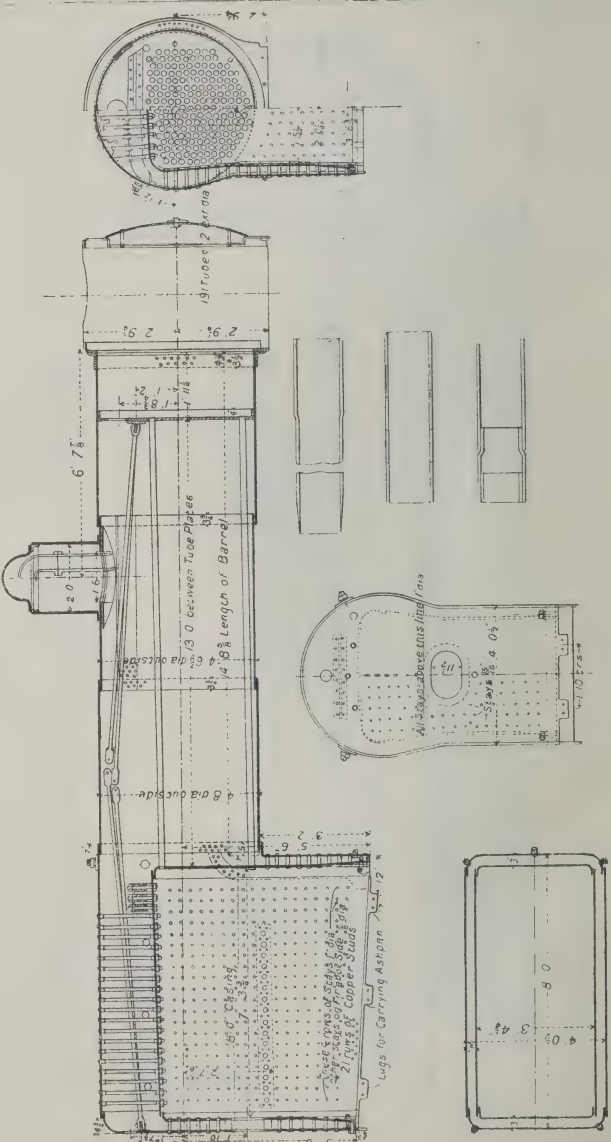


FIG. 15.

FIG. 16.

LOCOMOTIVE BOILER FOR 6 FT. 6 IN. 4-WHEEL COUPLED BOGIE ENGINE, GREAT NORTHERN RAILWAY.


19 Tubes 2 Dia
2 9½
2 9½
6' 7⅝
Dia outside
Length of Barrel
Tube Plates
Dia outside
Dia outside
Lugs for Carrying Ashpan
Copper Studs
Copper Studs
Copper Studs
Dia
Dia
Dia

The Paxman Economic Boiler.

This boiler (fig. 19) consists of a cylindrical shell, with one or more internal flues, and a number of small return tubes running from end to end. A combustion chamber of brick receives the products of combustion at the back of the boiler; they then pass through the small tubes to the smokebox in front, and finally return to the back, outside the boiler shell, through the brick flues in the setting, to the chimney.

Whilst the temperature of the waste gases which leave the boiler on their way to the chimney ranges from 600° to 700°F. with many boilers of the Cornish, Lancashire, or similar types, it is only from 450° to 475°F. in the case of the Paxman 'Economic'; with the latter boiler, on account of its large heating surface, there is about 30 per cent. less waste of heat, the temperature named being just sufficient to create a natural draught, indicating that the utmost possible degree of heat has been abstracted from the fuel burnt, and rendering an Economiser unnecessary.

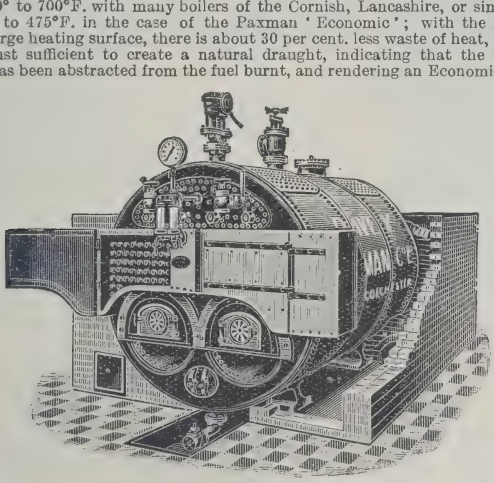


Fig. 19.

An independent test of this boiler by Sir Alexander W. B. Kennedy showed an evaporation of 12·2 lbs. of water from and at 212°F. per lb. of coal.

The operation of sweeping the tubes (preferably daily) can be effected by the stoker, using an ordinary wire tube brush, during any interval of thirty minutes; or, if there is no such interval, a steam cleaning device can be supplied, enabling the operation to be completed in two minutes.

A Paxman 'Economic' boiler being only about half the length of a Lancashire boiler of corresponding capacity, a very considerable saving is effected in the boiler setting and buildings; and in case of shipment the freightage cost may be only one-third of that for a Lancashire boiler.

Principal Dimensions of Standard Sizes of Paxman Economic Boilers with Single or Double Flues.

No.	Length	Diam.	Flues. No.	Flues. Diam.	Internal Heating Surface.	Evaporation per Hour.*	No.	Length	Diam.	Flues. No.	Flues. Diam.	Internal Heating Surface.	Evaporation per Hour.*
	Ft.Ins.	Ft.Ins.		Ft.Ins.	Sq. Ft.	Lbs.		Ft.Ins.	Ft.Ins.		Ft.Ins.	Sq. Ft.	Lbs.
8	6 5	4 9	1	2 3	140	840	30	11 0	6 3	1	3 2	478	2,860
10	7 0	5 0	1	2 4	174	1,044	35	12 6	6 6	1	3 2	565	3,400
12	8 0	5 0	1	2 6	234	1,400	40	12 6	7 0	2	2 3	665	3,990
14	8 6	5 0	1	2 6	248	1,480	45	12 6	7 6	2	2 4	772	4,530
16	9 0	5 3	1	2 6	276	1,650	50	14 0	7 6	2	2 6	890	5,220
20	9 6	5 6	1	2 9	324	1,940	60	14 0	8 0	2	2 8	1,180	6,960
25	9 6	6 0	1	3 0	393	2,350							

* The figures given for the evaporative capacity of each boiler are with feed at 212°F., when burning good coal, and with good draught and stoking. It is of course possible to obtain a much greater evaporation when the draught is ample, and the user is prepared to sacrifice economy to increase the evaporation.

The Cochran Boiler.

(Cochran & Co., Annan, Ltd., Annan, Scotland.)

The boilers (fig. 20, p. 1583) are used very largely both ashore and afloat; the advantage which they present over other kinds for land use and export consists, principally, in great economy of space, in not requiring any brickwork setting, in having a large amount of highly effective heating surface, in obtaining plenty of steam, and in being very economical in fuel, besides being perfectly accessible for cleaning and repairs outside and inside.

For winch boilers this type combines the advantage of the cross-tube boiler and the horizontal boiler without their disadvantage, and has been found the most suitable for large passenger, mail, and cargo steamers running to all parts of the world.

The boilers are made in a range of standard sizes, as per following Table, which gives full particulars of their standard dimensions.

Evaporation of Cochran Boilers.

Particulars of Boiler.					Heavy Steaming.						Easy Steaming.				
					Coal.		Evaporation.				Coal.		Evaporation.		
Diam. of Boiler.	Height of Boiler.	Grate Area.	Heating Surface.	Ratio H.S. / G.A.	Lbs. per hour.	Lbs. per sq. ft. of Grate per hour.	Per lb. Coal.	Per sq. ft. H.S. per hour.	Total per hour by Formula.	Evaporation in lbs. (maximum) nearest round figures.	Lbs. per hour.	Lbs. per sq. ft. of Grate per hour.	Per lb. Coal.	Per sq. ft. H.S. per hour.	Total per hour (calculated).
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Ft. In.	Ft. In.														
3 0	6 9	4·75	60	12·6	72	15·1	5·0	6·0	360	360	43	9·5	6·2	4·5	270
3 3	7 6	5·75	80	14·0	98	17·0	5·1	6·2	500	500	58	10·0	6·4	4·6	375
3 9	8 6	7·50	100	13·3	144	19·3	5·0	7·2	720	700	86	11·5	6·2	5·4	540
4 0	9 0	8·50	120	14·1	170	20·0	5·1	7·2	868	850	102	12·0	6·3	5·4	651
4 3	9 6	9·25	140	15·1	188	20·4	5·2	7·1	996	1,000	112	12·0	6·6	5·3	747
4 6	10 0	9·75	160	16·4	202	20·8	5·4	6·8	1,098	1,100	121	12·4	6·8	5·1	823
4 9	10 3	11·75	200	17·0	253	21·6	5·5	7·0	1,401	1,400	151	12·5	6·9	5·2	1,050
5 0	11 3	12·50	220	17·6	273	21·8	5·6	6·9	1,531	1,520	163	13·0	7·0	5·2	1,148
5 3	11 9	14·00	250	17·8	312	22·4	5·6	7·0	1,760	1,760	187	13·3	7·0	5·2	1,320
5 6	12 3	16·75	300	17·9	383	22·7	5·6	7·2	2,160	2,160	229	13·6	7·0	5·4	1,620
5 9	13 0	18·75	350	18·7	435	23·3	5·7	7·1	2,504	2,500	261	13·9	7·1	5·3	1,878
6 0	12 6	18·75	350	18·7	435	23·3	5·7	7·1	2,504	2,500	261	13·9	7·1	5·3	1,878
6 0	13 6	18·75	350	18·7	435	23·3	5·7	7 1	2,504	2,500	261	13·9	7·1	5·3	1,878
6 0	14 0	18·75	400	21·4	435	23·3	6·1	6·6	2,667	2,600	261	13·9	7·6	5·0	2,002
6 6	13 6	22·50	450	20·0	533	23·7	5·9	7·0	3,144	3,100	319	14·1	7·3	5·2	2,358
6 6	14 0	22·50	450	20·0	533	23·7	5·9	7·0	3,144	3,100	319	14·1	7·3	5·2	2,358
6 6	14 6	22·50	500	22·2	533	23·7	6·2	6·6	3,304	3,300	319	14·1	7·7	4·9	2,478
7 0	14 0	26·75	500	18·7	643	24·0	5·7	7·3	3,665	3,600	385	14·3	7·1	5·4	2,748
7 0	15 0	26·75	600	22·4	643	24·0	6·3	6·7	4,021	4,000	385	14·3	7·8	5·0	3,015
7 6	16 3	31·50	750	23·8	767	24·3	6·5	6·6	4,939	4,900	460	14·6	8·0	5·0	3,705
8 0	16 6	37·00	850	23·0	906	24·5	6·3	6·7	5,760	5,700	543	14·6	7·9	5·0	4,320
8 6	17 0	41·00	1,000	24·4	1,004	24·5	6·5	6·6	6,610	6,600	602	14·6	8·2	4·9	4,957

Messrs. Cochran's formula for evaporation is :—

$$E = 26\,(G - 2)\left(5 + \frac{\frac{H}{G} - 13}{7{\cdot}5}\right)$$

where,

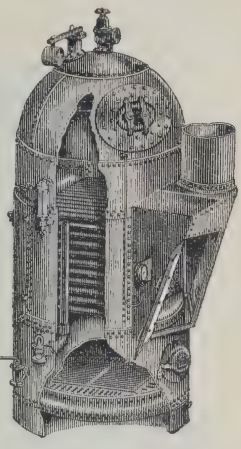
E = Evaporation in lbs. per hour; G = Grate area in sq. ft.; H = Heating surface in sq. ft. The first expression of the formula represents the coal burnt per hour; the second expression represents the water evaporoted per lb. of coal.

The evaporations given by this formula can be obtained under favourable conditions, but the makers' practice is to recommend that about 75 per cent. of his maximum evaporation should be relied upon for continuous working.

COCHRAN BOILERS

Steam and Hot-Water

for Every Industrial Purpose & to burn any fuel, or for Waste-Heat Utilization.



Steam Units Stocked in 23 Standard Sizes & Hot-Water Units in 11 Standard Sizes for prompt delivery.

SPECIAL COMPOSITE UNITS for simultaneous DIESEL EXHAUST GAS & OIL FIRING or DIESEL EXHAUST GAS UTILIZATION ONLY.

▪ ▪ ▪

COCHRAN HOPWOOD TYPE BOILERS for all types of STEAM DRIVEN CRANES.

COCHRAN & CO., ANNAN, LTD.,

Head Office & Works :—ANNAN, SCOTLAND.

Telephone : Annan 111/112. Telegrams : Multitube, Annan.

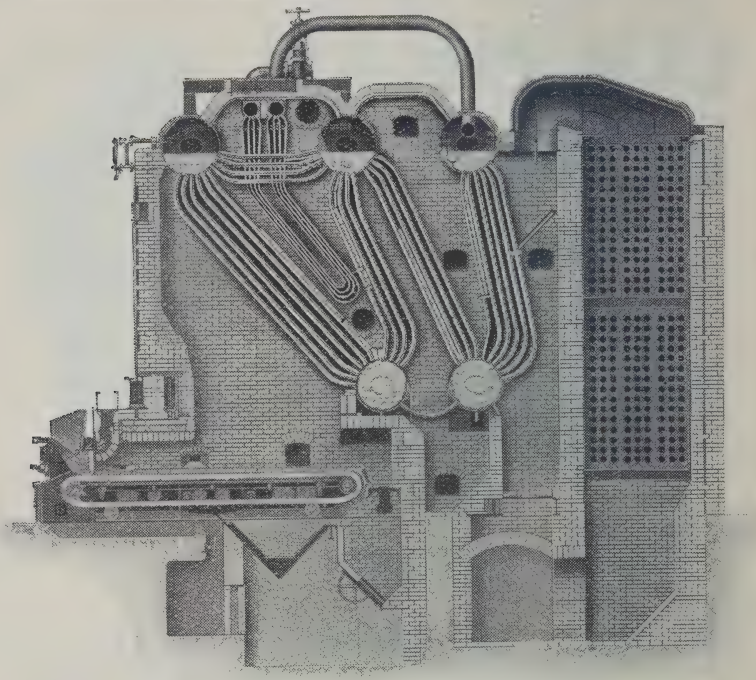
London

137 VICTORIA STREET, S.W.1.
Telephone : Victoria 9341 (3 lines).
Telegrams : Multitube, Sowest.

Newcastle

9 MARKET STREET.
Telephone : Newcastle 21976.
Telegrams : Multitube, Newcastle/Tyne.

F. 1582



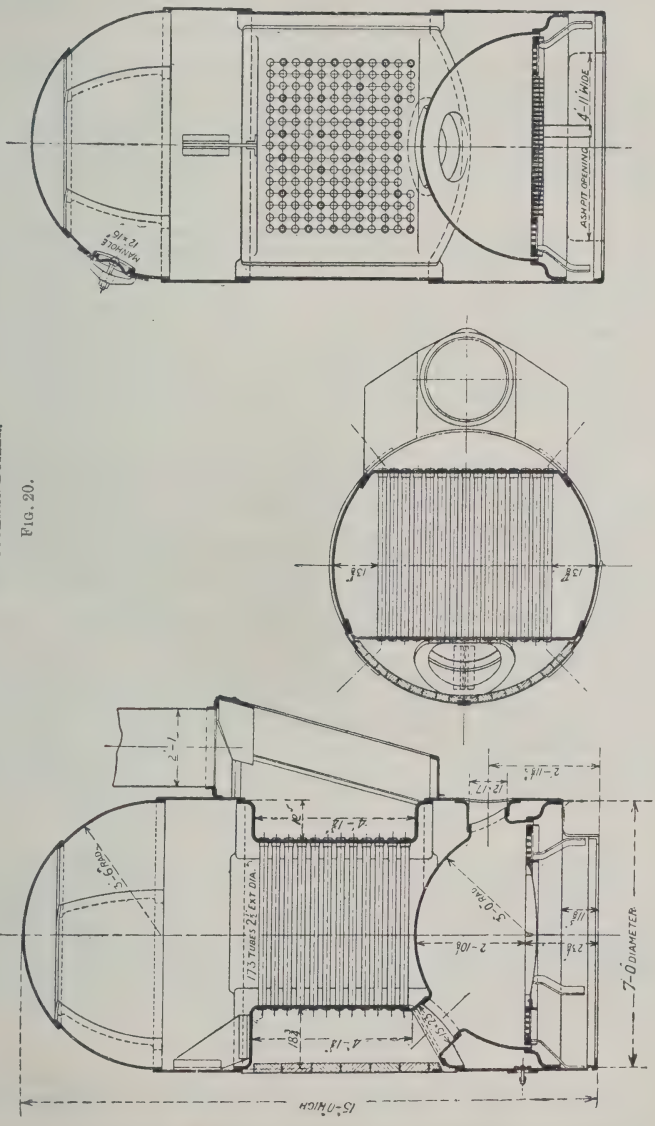
The Ideal Steam-Raising Plant

The sectional illustration shows a 50,000 lb. unit as supplied to the Royal Borough of Kingston-upon-Thames.

THE STIRLING BOILER COMPANY, LTD.
32 & 33 Farringdon Street, London, E.C.4

STIRLING

F. 1583



THE COCHRAN BOILER.

FIG. 20.

In the Evaporation table (p. 1582) it will be noticed that there are two sets of figures, showing coal consumption and evaporation. Under 'Heavy Steaming' are given the evaporations as per formula, and under 'Easy Steaming' the evaporations are calculated at 75 per cent. of this amount.

An adaption of the standard Cochran boiler recently introduced to meet the requirements of hot-water service and heating schemes has met with a considerable measure of success. For particulars of this boiler, see p. 2502.

In addition to the range of standard sizes detailed in the above table, the 'Cochranette' Boiler, 2 ft. 6 in. dia. × 5 ft. 6 in. high × 25 sq. ft. heating surface, is built to stock in the one size only, for 100 lbs. per sq. in. working pressure, and additional large sizes of standard boilers can now be offered 9 ft., 9 ft. 6 in., and 10 ft. diameters to give outputs up to 10,000 lbs. per hour. Also Cochran boilers are now constructed for the utilisation of Diesel engine exhaust gases only, and composite types for simultaneous use of exhaust gas and oil firing.

THE WOLLASTON PRODUCER BOILER.

The Wollaston producer boiler embodies a radical novelty in boiler design—viz. the gasification of the fuel in a producer, together with the arrangement of this producer in relation to the boiler, so that the gas generated is burned in the furnace of the boiler, and also so that losses of heat from the producer, and from the gas on its way to the burners, by external radiation are reduced to negligible amounts.

One design of the producer boiler is shown in fig. 21 (p. 1585). It is mounted directly above the gas-producer, the body of which is surrounded by a water-jacket through which the feed-water passes on its way to the boiler. Air for combustion is supplied by a fan; part of the air passes as 'primary air' into the deep bed of fuel in the producer, the remainder passes as 'secondary air' through passages in a refractory dome above the producer, and then through a number of holes into the gas rising from the producer through a central hole in the dome. The combustion of the gas is similar to that in a Bunsen-burner. The air streams are both regulated by valves and a high degree of combustion is attained, the gases leaving the boiler containing 17 to 18 per cent. of CO_2.

Heat which would otherwise be lost is absorbed by the feed-water and the secondary air, and the external casing is lagged. Ashes are removed through a water seal, as is usual in producers, and holes are provided through which tools can be inserted for poking the fuel. So far, this combination of producer and boiler appears to be limited to units not exceeding in capacity an evaporation of 20,000 lbs. per hour.

Test results showing an efficiency of 80·8 per cent. net—*i.e.* after deducting for all auxiliaries—have been obtained, the fuel being ordinary gasworks coke of about 10,000 B.Th.U. per lb. calorific value. The system is particularly adapted for using poor fuels such as coke breeze, with high ash and moisture content. A fuel which cakes greatly is not suitable, but there is practically no limitation as regards grading, ash moisture, and dust.

While the efficiency is high, even more emphasis is laid upon the economic advantage due to the ability to burn otherwise impossible fuels. No skilled stoking is required, and banking losses are exceedingly low.

The Wollaston Producer Furnace.—This is designed for the application of similar principles of combustion to the Lancashire boiler. It consists of a producer, with fan, primary and secondary air circuits, feed heater, and burner, all arranged in a unit suitable for placing in front of the Lancashire boiler furnace, with the burner projecting slightly into the furnace, the front of which is closed by a plate, with a hole cut in it for the burner to pass. The boiler furnace has, of course, no grate, and is fitted with a cylinder of refractory material of nearly the furnace diameter, which is pierced with a large number of small holes. A water-sealed ash pit is provided below the producer. While radiation is minimised by water and air jacketing as far as possible, the losses in this furnace are necessarily greater than in the self-contained producer boiler.

WATER-TUBE BOILERS FOR LAND SERVICE.*

THE NICLAUSSE BOILER.

This boiler consists of a compact nest of large, straight, short solid drawn tubes, closed at each end and mounted at the front end in solid drawn pressed steel headers, upon the top of which is mounted a capacious steam and water drum. There is thus only a single row of headers. The heating surface of the outer tube is rendered highly efficient by the provision of an inner tube to carry the water to its back end, whereby the circulation must always be in only one, and that the correct, direction. The tubes are closed by a tapered screwed pressed steel cap at the back end, and at the front end by a tapered screwed pressed steel plug, to which is attached the inner tube by a swivelling connection, whereby it can be readily withdrawn. Both plugs and caps can be easily removed for inspection and cleaning. It is generally sufficient to remove the front plug and withdraw the inner tube once a year for washing out by means of a hose applied to the outer end of the inner tube *in situ*.

THE VICKERS WATER TUBE BOILERS.

(*Vickers Boiler Co., Ltd.*)

These boilers (figs. 22 and 23) comprise the 'Bent' Spearing 'Header' type and Tube type. The Vickers 'Bent Tube' type boilers comprise both three-drum and five-drum patterns,

* See Section XXXVI, p. 2353, for Marine Service.

THE WOLLASTON SYSTEM OF BOILER FIRING.

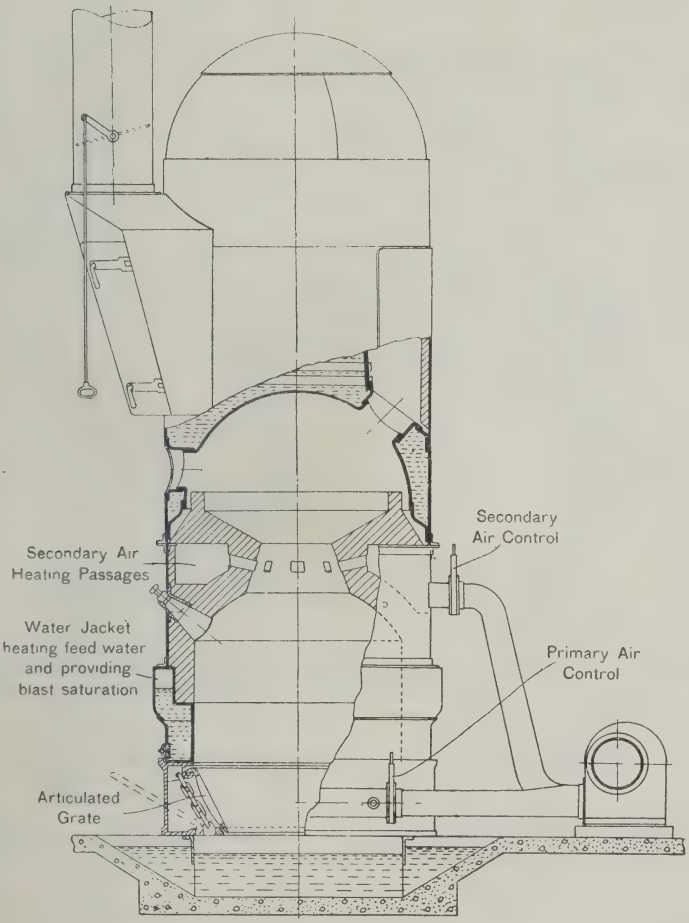


FIG. 21.

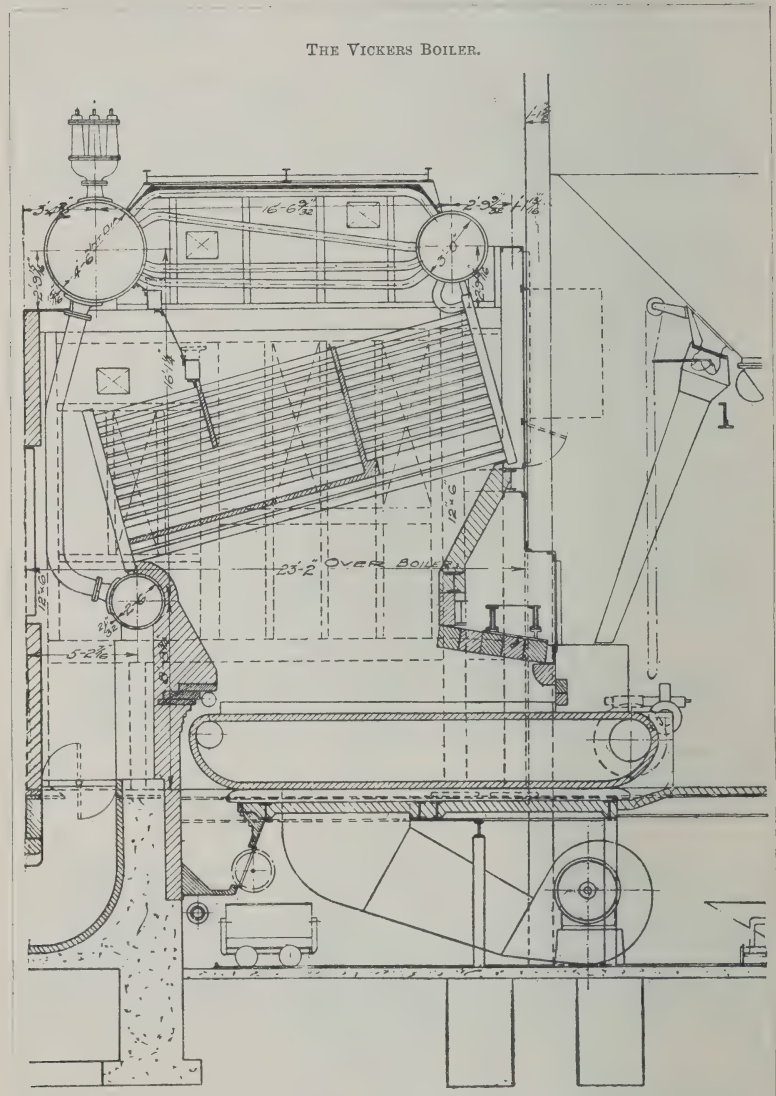


FIG. 22.—Three-drum cross-headed type boiler.

THE VICKERS BOILER.

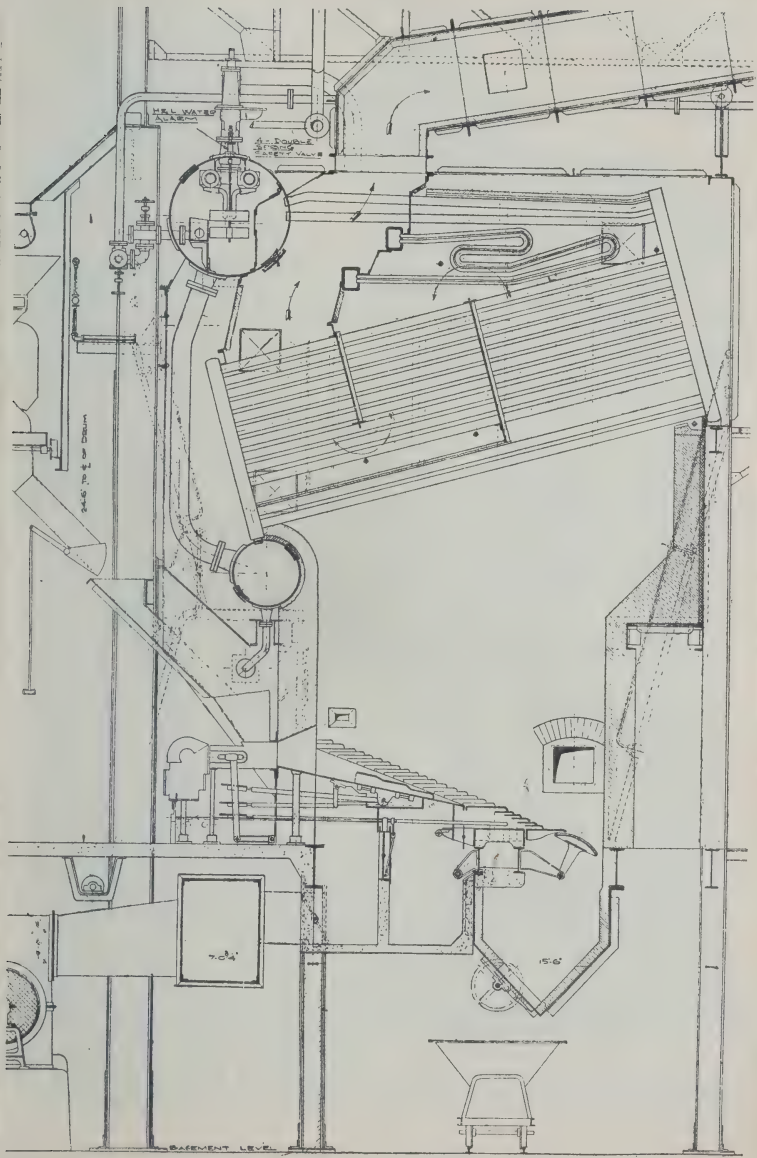


FIG. 23.—Two-drum cross-headed type boiler.

The Spearing 'Header' type boiler may comprise three drums, viz. the main drum at the rear with mud drum and circulating drum at the front, or alternatively two drums only, viz. main steam and water drum and mud drum. Boilers of this type may be arranged either with the two drums at the front or the rear. A further design of the header type boiler is a patent double-circulation boiler which is specially designed for giving a very high rate of evaporation per sq. ft. of heating surface.

All 'Header' type boilers embody the special features associated with the Spearing type of boiler, viz. a mud drum usually of 2 ft. 6 in. internal diameter, downcomer pipes of large area connecting the main steam and water drum with the mud drum, straight solid drawn weldless steel headers of large area with staggered tubes, and also connecting nipples of large area.

These special points in design have been embodied with a view to ensuring positive circulation under all operating conditions, and also to ensure the precipitation of all solids in the mud drum for blowing off. In practice it has been found that the downcomers are of the utmost practical value from the point of view of effective circulation, and in ensuring an uninterrupted supply of feed water in a proper cycle from the main steam and water drum through the mud drum to the bottom rows of tubes first. This positive circulation also makes this type of boiler an extremely efficient and satisfactory steam generator under overload conditions.

The value of the large mud drum in conjunction with the downcomer circulating system, and the precipitation of suspended solids in the water in the mud drum has enabled this boiler to be efficiently used with extremely unsatisfactory feed water without any detrimental effect upon the boiler tubes, and it is very rarely indeed that any deposit whatever is found in these tubes.

Figs. 22 and 23 show: (1) The Vickers three-drum cross-header type boiler; (2) the Vickers two-drum cross-header type boiler.

Boiler units of the header type have been made up to an evaporative capacity of 100,000 lbs. per hour, and of the bent-tube type up to an evaporative capacity of 60,000 lbs. per hour.

The Clarke-Chapman Water-Tube Boiler.

(Woodeson's Patent.)

(Clarke, Chapman & Co., Ltd., Gateshead-on-Tyne.)

The 'Woodeson' Patent Water-tube Boiler, as shown by fig. 24, consists of three sections, each comprising steam drum, water drum, and a number of groups of tubes expanded into same; these tubes are nearly vertical and perfectly straight, and manholes are arranged on steel drums immediately over each group through which any tube can be withdrawn.

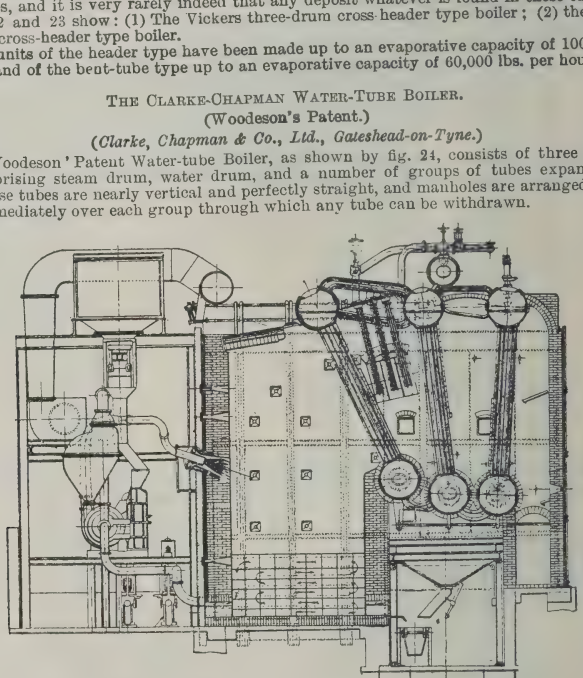


FIG. 24.—The Clarke-Chapman 'Woodeson's' Patent Water-Tube Boiler fired with powdered fuel by 'Resolutor' Pulveriser.

A spacious steam dome is arranged over the steam drums to ensure a supply of dry steam and to maintain a steady water level. The bottom drums are absolutely hanging so that expansion and contraction can take place without effecting any damage to boiler.

The boiler as shown is also fitted with a Clarke-Chapman Patent Superheater between the front and centre sections.

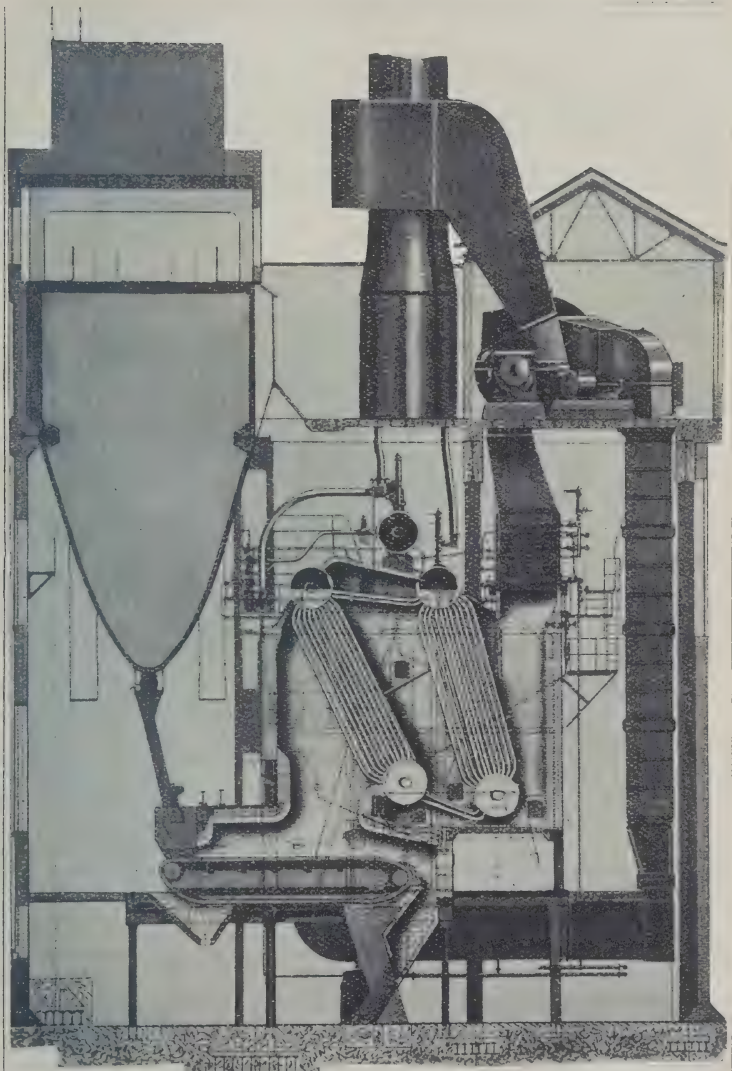


FIG. 25.—Arrangement of one of five Clarke-Chapman Bent Tube Watertube Boilers, fitted with Superheater, Forced Draught Stoker, Steel Tube Economiser, Air Heater, Induced Draught Plant and Grit Collectors.

In the design of the 'Woodeson' boiler, a special feature is the ease of access for purposes of examination or repairs, the time which has to be expended on same being practically reduced to a minimum; thus in a boiler of, say, 6,100 square feet of heating surface, containing 456 tubes, the only work to be done to thoroughly examine the insides of this large number of tubes for their total length is the removal of the manhole door on one end of each bottom drum and one manhole door off each of the three steam drums: thus access can be gained to every tube, and these being perfectly straight they can be thoroughly examined for the whole of their length. The removing and replacing of the manhole doors can be done by an ordinary fireman, no skilled mechanic or complicated tools being required. There is an entire absence of screwed joints in the boiler.

Each tube being nearly vertical and perfectly straight and entering the steam drum direct without intervention of any headers, ensures a perfect steam release, and so adds to the efficiency of the boiler, as there are no sharp turns to retard flow of the steam globules to the steam space. Also, deposits are not retained in the tubes, but are collected in the large mud-drums: thus over-heating and destruction of tubes are reduced to a minimum. The Woodeson Patent Water-Tube Boiler is also eminently suitable for firing with Powdered Fuel as shown in fig. 24.

See also p. 2400.

The Stirling Water-Tube Boiler.

(*The Stirling Boiler Company, Ltd.*, 32–33, *Farringdon Street, London, E.C.* 4.)

This boiler consists of two or more upper drums, containing water and steam, connected by banks of tubes to one or more lower water drums, the upper drums being further connected by several rows of steam and water circulating tubes.

The type of Stirling boiler shown in fig. 26 is termed the 'Tri-drum' boiler. It is the latest type of this generator, and is essentially a high duty boiler, with a rapid circulation, quick steam generation and a large evaporative capacity, the products of combustion being brought directly into intimate contact with the outer surfaces of the tubes, and directed by a system of baffles to secure the maximum heat interchange between these gases and the water.

The tubes are of solid drawn steel, and are 3¼ in. external diameter. As will be seen from the figure, the Stirling tubes are curved at the ends, permitting them to enter the drums radially, thereby allowing not only for a maximum number of tubes to be connected to the drum, but also for a natural expansion and contraction in each individual tube, thus ensuring tightness and at the same time flexibility.

The tubes are expanded into the tube plates, the projecting tube ends within the drums being either belled or beaded over; thus each tube has a discharge outlet of its full area for the escape of the steam as generated, so obviating steam pocketing in the tubes, with its attendant risks of blistering, etc. The Stirling boiler tubes are slightly inclined to the vertical to enable the front bank tubes to absorb much radiant heat and to receive from the gases the maximum heat by convection.

The drums are made of best quality open-hearth steel and undergo hydraulic test pressures before leaving the makers' works.

All portions of the boiler are circular in form, eliminating any tendency to deformation of any part owing to internal pressure, and certain types of this boiler are already operated at a working pressure of 1,400 lbs. per square inch.

The boiler is entirely independent of the brickwork, the upper steam and water drums being either carried on a steel framework or suspended by slings from overhead beams, while the lower drums are supported by the tubes from the upper drums, leaving the whole boiler freedom of movement with changes of temperature.

The water circulation, practically throughout the whole course of its progress through the boiler, is travelling in a direction opposite to that taken by the gases, thus securing throughout the whole surface of the boiler a very high efficient heat transmission rate. The feed water is introduced into the rear drum and becomes highly heated before reaching the front bank, scale forming matter, if present, being thrown down in the rearmost banks of tubes and preserving cleanliness in the main front bank generators.

By means of patent diaphragms in the rear drums, incoming feed water has to traverse two passes in the rear bank of tubes before joining the main body of water already in circulation, giving a gradual rise in the temperature of the water throughout the boiler, with the corresponding gradual approach to the greatest temperature of the gases, *i.e.* the combustion chamber.

A patent separating baffle is also fitted into the upper rear drum to intercept the mixture of steam and water issuing from the ends of the water cross tubes. This mixture is separated into its components within the confined space of the baffle, the water flowing quietly into the water space of the drum, while the steam is discharged above the water level. By directing the steam port downwards there is a tendency to throw down into the water space the last trace of residual moisture left in the steam. The presence of this baffle produces a state of quiescence in the drum (a necessary condition for dry steam), an unusually high dryness factor for the steam (essential for a high degree of superheat), and the cleanliness of the superheater, valves and entire steam circuit.

The overall sizes of the Stirling boiler of a given heating surface can be arranged to suit varying conditions, the design of boiler being adaptable equally well for boiler houses with low roof and ample width as for those having little width and ample height.

A marine type of this boiler is also supplied, which may be used for landwork when the space available is very limited.

Stirling boilers are designed for all evaporations between 4,000 lbs. and 500,000 lbs. of water per hour, and working pressures between 100 lbs. and 1,400 lbs. per square inch.

The boiler is easily cleaned, this being effected on the tubes on the external sides by a system of steam soot blowers, or by hand-operated steam lances; the internal sides (access to which is made by the simple operation of removing three manhole doors) are cleaned by means of water-driven turbine cleaners. This water, supplied through a flexible hose, rotates small cutters, varying in form according to the nature of the scale to be removed. The exhaust water from the turbine washes the refuse into the lower water drum, from which it is easily drained off through the blow down pipes.

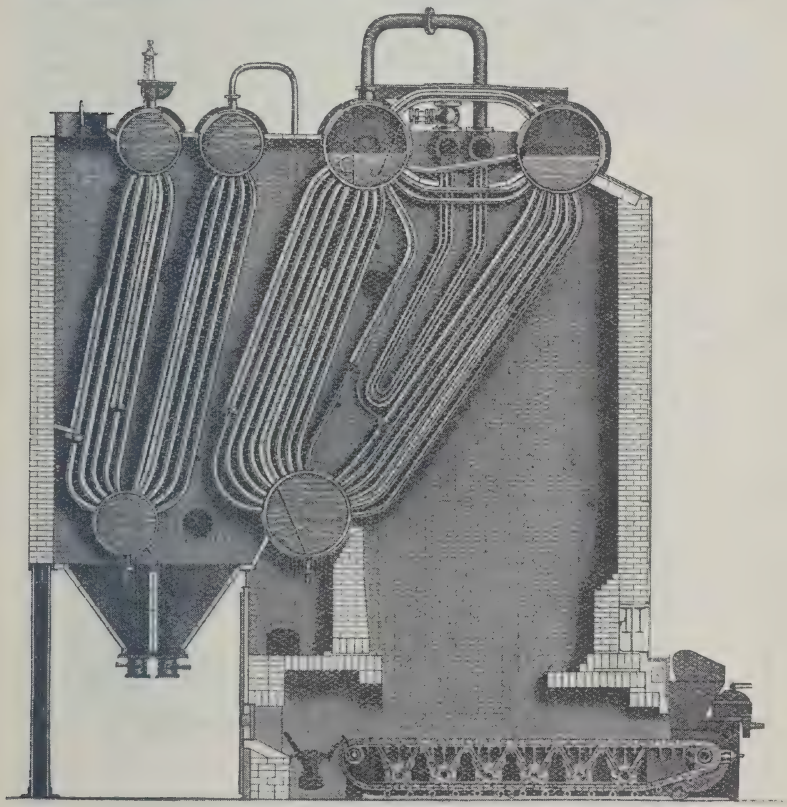


FIG. 26.—The Stirling Boiler.

All other parts of the boiler are easily accessible by the conveniently placed access and sooting doors. The rare occurrence of a renewal of Stirling tubes is provided for by any tube being able to be removed without sacrificing the neighbouring good tubes.

Taking a great number of tests extending over a period of many years, the efficiency of the Stirling boiler averages from 80 to 85 per cent.

This boiler will give the highest satisfactory results with all classes of fuel and methods of firing, whether pulverised fuel, mechanical stokers, or oil burners, while it is largely in use for the utilisation of waste heat, whether in the form of blast-furnace gas, waste heat from coke ovens, or heating and puddling furnaces. It is a boiler much used in power stations, paper works, gas works, etc.; it is also largely employed on sugar estates abroad, fired with bagasse in special furnaces.

The Simon-Carves Boiler.

(Simon-Carves, Ltd., Cheadle Heath, Stockport.)

The Simon-Carves Patent Boiler is of the multiple-drum type comprising at least two steam drums slung from a steel framework, one water drum suspended by the generating tubes, and one mud drum.

None of the drums is divided by partitions or chambers, but the feed water passes straight into the main stream from the feed flash trough B which extends the full length of the rear drum. The main stream descends through the whole of the tubes in both sections of the rear bank P and P1, crosses the water drum WD, ascends the whole of the front bank of tubes N to the front stream drum FD, and completes the circuit to the rear steam drum RD through the water cross tubes O. A portion of the water in circulation enters the outer system through the several external down-comer pipes C connected to the lower portion of the rear steam drum and the mud drum D, which is naturally the lowest point in the whole boiler.

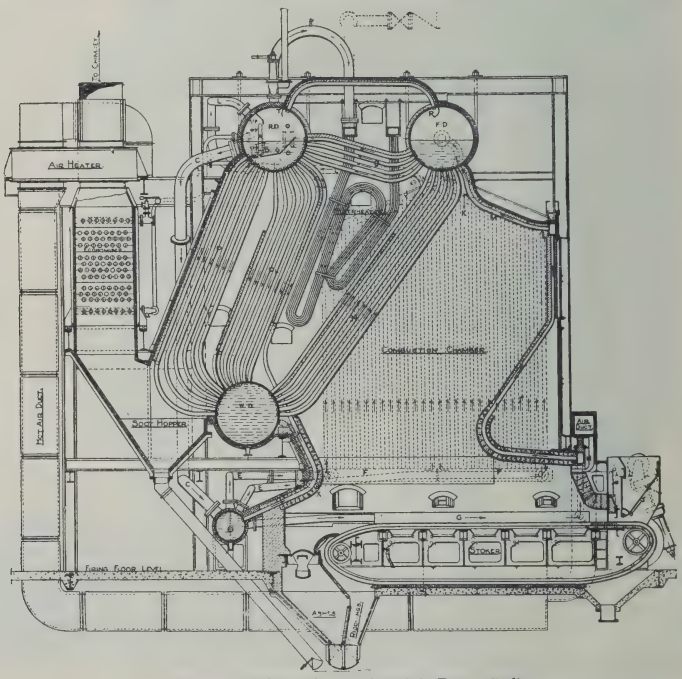
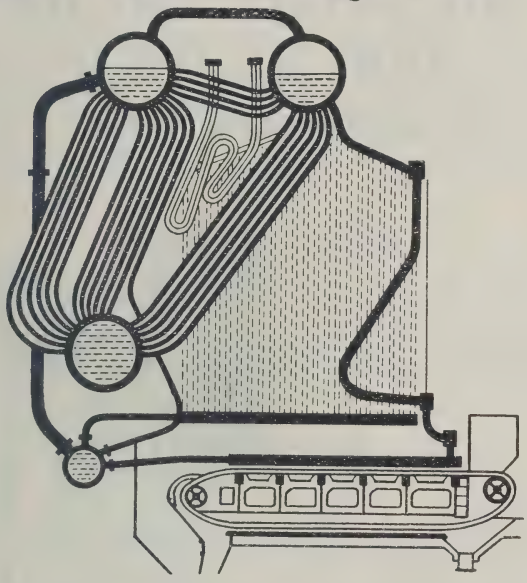


Fig. 27.—The Simon-Carves Multiple-Drum Boiler.

From the mud drum part of the water rises through the tubes forming the rear water-cooled arch X to the rear steam drum. A further portion is fed by external pipes E to the bottom headers F of the water-cooled walls of the combustion chamber, rises through the vertical tubes in these to the top headers, which are connected to the steam drums by external pipes. The remainder passes through the anti-slag stoker side wall tubes G to the bottom headers of the main stoker arch, through the tubes in same, which also form the front wall of the combustion chamber,

SIMON-CARVES MULTIPLE-DRUM BOILERS AND BOILER EQUIPMENT



We undertake the complete installation of steam-raising plants, including boilers, boiler firing equipment, buildings and all auxiliaries. We install the Simon-Carves "Central" or "Unit" systems of pulverised fuel firing or any type of mechanical stoker, gas or oil burning equipment.

SIMON-CARVES LTD.
CHEADLE HEATH, STOCKPORT

The Engineer.

Established 1856. Registered as a Newspaper

PROVIDES THE LATEST AND THE MOST RELIABLE INFORMATION FOR ENGINEERS

IN ALL BRANCHES OF THE PROFESSION.

The last issue in each month contains a Supplement, **The Metallurgist,** *a journal devoted to the science of metals, ferrous and non-ferrous.*

Subscriptions

		Per annum
INLAND, YEARLY		£3 5 0
CANADIAN SUBSCRIPTIONS.	Thin paper edition ...	£2 18 6
	Thick paper edition ...	£3 3 0
FOREIGN SUBSCRIPTIONS.	Thin paper edition ...	£3 3 0
	Thick paper edition ...	£3 7 6

'THE ENGINEER' DIRECTORY AND BUYER'S GUIDE

SENT FREE ON APPLICATION.

to the top headers. These are connected to the front steam drum by a series of tubes which form the combustion chamber roof.

All the tubes in the boiler are as long as practicable, and the gases are made to flow parallel to their axes as far as possible. They are arranged in rows which lie in planes normal to the drum axes with adequate spaces to permit replacement of tubes.

This arrangement aims at external cleanliness and high efficiency, with reasonable allotment of work to the 'tail' of the circuit.

The tubes in the water walls of the combustion chamber are covered, on the inside exposed to the heat of the combustion chamber, with patent refractory tiles of special shape which fit close round the tubes, fill the space between and are interlocked and keyed to the tubes so that they cannot become loose or fall away. The backs of the tubes are clear and may be inspected by removal of the steel plates forming the casing and the insulating slabs or mattresses.

Fig. 27 shows the standard arrangement, but the 'W' type with a central combustion chamber is used for very large units.

The 'John Thompson' Beta Type Water-Tube Boiler.

(*John Thompson Water Tube Boilers, Ltd., Ettingshall, Wolverhampton.*)

This boiler (fig. 28) consists of either two, four or six drums, one, two, or three at the top, for steam and water, and a corresponding number at the bottom. Each series of drums is brought into communication one with the other by a number of short circulating tubes, whilst large bent tubes, set nearly vertically, connect the top drums with those at the bottom. By this means the water, which is fed into the boiler, from the back of the steam and water drums, passes down the back row of tubes, each of which has a discharge outlet equal to its full area, and thus affords a free, rapid, and continuous circulation, which in its turn prevents the deposit of scale on the heating surface of the tubes.

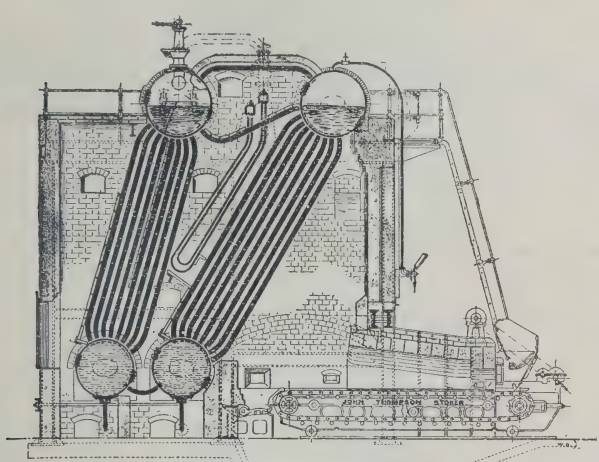


Fig. 28.

The six-drum type of boiler is known as the Integral Economiser Boiler, the back or third bank of tubes being the Economiser.

The tubes are so arranged, as to spacing, that any one tube may be removed and replaced without in any way disturbing any others. They are all passed in and out of the boiler by way of the fire-door opening.

Any scale that may be deposited in the boiler falls into the mud-drums at the bottom.

The drums shown in fig. 29 are of mild steel and longitudinally divided into halves, the portion which receives the tubes being hydraulically formed into a series of impressions, enabling the tubes to enter the tube plate at right angles.

The following points are special features:

1. *Tubes.*—The tubes are so arranged, as to spacing, that any one tube may be removed and replaced without in any way disturbing any others.

The tubes are all passed in and out of the boiler by way of the fire-door opening, thus saving the necessity for extra head room.

Each tube can be examined throughout its whole length by holding a lamp at one end.

2. *Drums.*—The steam and water drums are of a large and ample capacity, so essential for the obtaining of dry steam, and thus giving large thermal storage.

Any scale that may be deposited in the boiler falls into the mud-drums at the bottom.

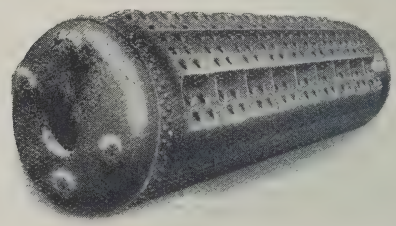


FIG. 29.

3. *Boiler.*—The boiler is suspended from steel girders which rest on steel columns based in cast iron; it is thus entirely independent of the brickwork, ensuring freedom for expansion in every part, and enabling the brickwork to be removed and replaced, if necessary, without in any way disturbing the boiler or its connections.

The largest size of these boilers can be cleaned and restarted in full work within the period of three days, there being only one manhole joint in each drum to be made.

PARTICULARS OF 16 'JOHN THOMPSON' WATER-TUBE BOILERS, VERTICAL TYPE, GIVING WEIGHTS.

(For working pressures up to 250 lbs. per sq. in.)

Heating Surface.	Tubes Wide.	Vertical Centres.	Stokers.	Space occupied over Brickwork.		Evaporation.
				Width over Side Walls.	Length over Back Walls.	
Sq. ft.		Ft. Ins.	Ft. Ft. Ins.	Ft. Ins.	Ft. Ins.	Lbs.
9,275	40	26 6	2–16 × 9 0	23 0	24 6	48,400
8,340	36	23 10	2–16 × 8 0	21 0	24 6	42,000
7,780	34	23 7	2–15 × 7 6	20 0	24 0	38,500
7,350	32	23 7	2–15 × 7 0	19 0	24 0	35,500
6,770	30	23 4	2–15 × 6 6	18 0	24 0	32,500
6,330	28	23 4	2–15 × 6 0	17 0	24 0	30,000
5,870	26	23 4	2–15 × 5 6	16 0	24 0	27,400
5,400	24	23 4	2–15 × 5 0	15 0	24 0	24,800
4,950	22	23 4	1–13 × 10 0	14 0	24 0	22,200
4,430	20	23 1	1–13 × 9 0	13 0	24 0	19,600
3,925	18	22 10	1–13 × 7 6	12 0	23 6	16,800
3,480	16	22 10	1–13 × 6 6	11 0	23 6	14,800
2,855	14	21 10	1–13 × 5 6	10 0	23 0	11,500
1,980	12	20 1	1–10 × 4 6	9 0	20 9	7,500
1,430	10	16 6	1–9 × 3 6	8 0	20 0	5,100
1,100	8	15 9	1–8 × 3 0	7 0	19 6	3,800

'DANKS-NETHERTON' VERTICAL BOILER.

(Multitubular type.)

(*H. & T. Danks* (*Netherton*), *Ltd.*)

Design (refer to illustration).—The fire-box of this boiler is connected by a short neck with a superimposed, centrally placed combustion chamber, and together they form the plate heating surface. Tubular heating surface is provided by the fire tubes radiating from the centre of the boiler, and located between the combustion chamber shell and the boiler shell. These tubes are arranged in staggered tiers.

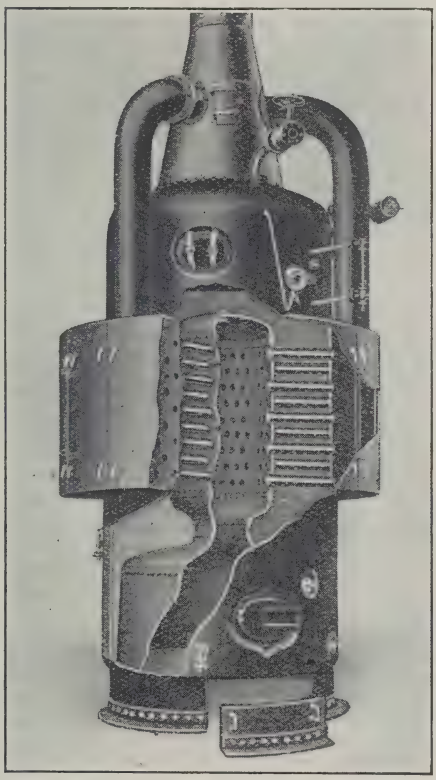


FIG. 30.

Vertical passes for access purposes divide the tubes into three equal nests. The smokebox completely surrounds the shell of the boiler and is divided into three sections by means of baffles corresponding to the tube nests, each section having a separate uptake to the chimney. Hinged doors are provided to give easy access to the tubes.

The boilers are built for evaporations up to 8,250 lbs. per hour maximum.

THE BABCOCK & WILCOX PATENT WATER-TUBE BOILER.

STANDARD LAND TYPE.

The structure of this boiler (figs. 31, 32), composed entirely of wrought steel, may be practically considered as having three distinct parts, namely:

A series of inclined water tubes placed over the furnace, in which the water, being divided into small volumes, is quickly raised to a high temperature and rises, at the front end, through vertical connecting boxes or headers (into which the tubes are expanded).

A horizontal steam and water drum, up to which the water rises from the headers, and where the steam separates from the water, the remaining body of water returning through the vertical tubes and headers at the back end into the inclined water-tubes, where it is again subjected to the action of the fire, and again passes into the steam and water drum; thus, a continuous and rapid circulation is kept up, and a uniform temperature maintained throughout the boiler.

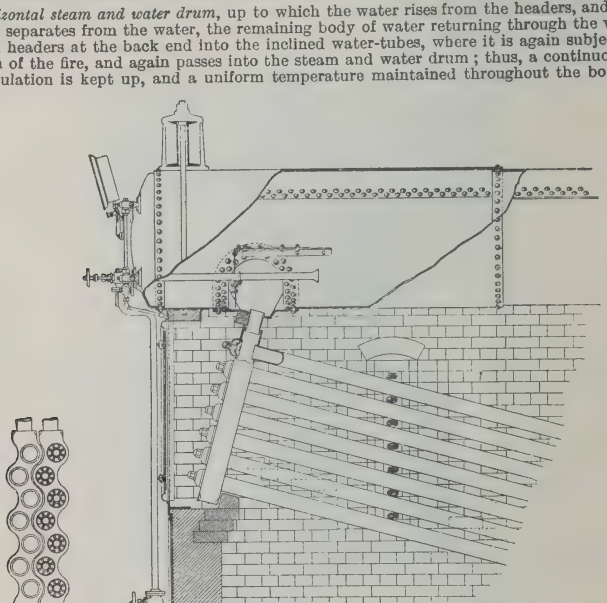


FIG. 31.—Vertical Section showing details.

A *mud-collector* is attached to the lowest point of the inclined water-tubes, into which the matter held in suspension in the water is, to a large extent, precipitated by its specific gravity.

The inclined water-tubes are of steel and the headers at each end into which they are expanded are sinuous (or staggered) and of mild steel, each complete section consisting of one zigzag row of tubes, expanded into a rear and front header. The headers are provided with handholes placed opposite the ends of each tube to permit of cleaning, and, in case of need, the removal of a tube, each handhole being provided with a cap fastened with a wrought steel bolt and clamp and a cap nut, the handhole covers being placed metal to metal. The top ends of each section are connected to the steam and water drum, not directly, but are first expanded into specially designed cross-boxes, which are riveted to and extend from the underside of the drum at each end. The steam and water drum is made of the best selected mild steel double riveted for ordinary purposes; the mud drum is made of mild steel, and is provided with ample facilities for cleaning.

The entire boiler, except the furnace, is suspended by wrought iron slings, from iron girders resting on wrought iron columns, to allow the boiler to expand or contract without straining the brickwork. The boiler and furnace are enclosed in masonry lined with firebrick, the furnace being arranged below the tubes, and firebrick baffles compel the hot gases to pass upwards, then downwards, then upwards again before escaping to the chimney. These hot gases strike the tubes at right angles, thus intercepting the radiant heat from the fuel as effectively as possible; the tubes being zigzagged, the gases are thoroughly broken up as they rise, and, passing into the

THE BABCOCK & WILCOX PATENT WATER-TUBE BOILER.

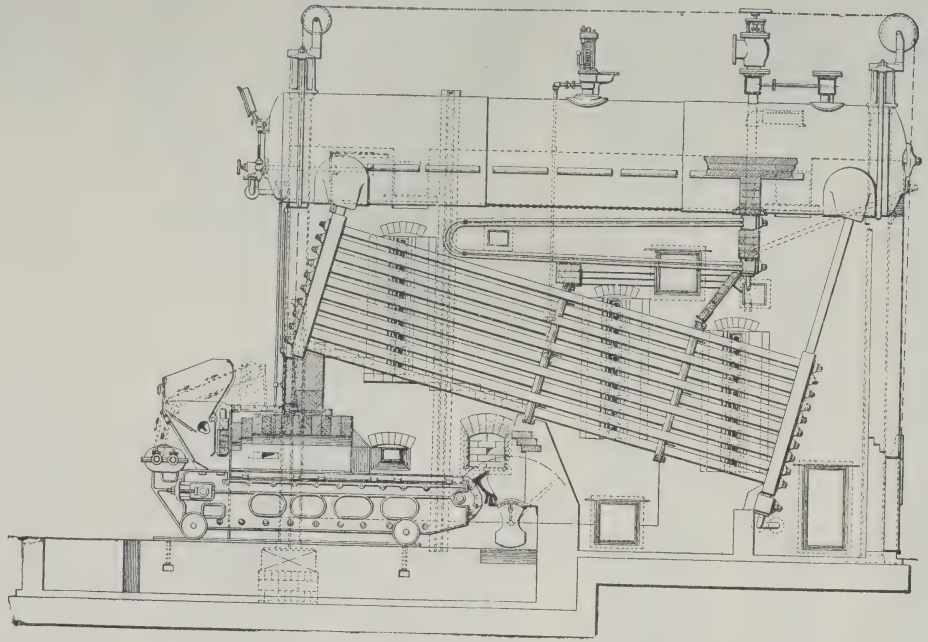


FIG. 32.—Boiler fitted with Superheater and Stoker.

combustion chamber under the drum, they expand and combine before again passing the second and third time through the tubes and on to the flue; thus complete combustion, rapid steaming, and consequent economy are secured.

The damper for regulating the draught and flow of the gases is placed in the back chamber, whilst doors for cleaning the tubes and removing soot are placed on one side of the brickwork.

A comparison of space is afforded by a works, where six Lancashire boilers, evaporating 48,000 lbs. of water per hour, are placed beside four Babcock & Wilcox boilers of the above type, evaporating 40,000 lbs. of water per hour under the same conditions. The ground space occupied by each range of boilers is in the ratio of 10 for Lancashires to 4½ for Babcock & Wilcox boilers.

CROSS-TYPE BOILER.

The general construction of this form of boiler is the same, but the drum is placed transversely, instead of longitudinally, with the tubes, and the whole boiler is constructed in pieces of comparatively light weight suitable for transport in countries where ordinary means of carriage are not available.

PORTABLE BOILER.

This form is similar to the last described, but is still further sectionalised in the drum portion, which is arranged to be riveted up on site. The boiler is also provided with an iron casing lined with firebrick, and is thus self-contained and independent of the brickwork. This type is also used for countries where transport and facilities in the matter of construction are not available.

In addition to the above types, the boilers are arranged with special furnaces for burning sawdust, lignite, rice husks, bagasse, coke breeze, refuse, coal, gas, and oil fuels.

MARINE TYPE BOILER.

In general construction this boiler (see page 2398) is similar to the foregoing, but with the steam and water drum placed transversely and brought to the front. The principal difference between this boiler and those already described is the largely increased grate area, enabling the boiler to give a greatly increased evaporation per square foot of heating surface compared with the other types.

The 'O T' marine type of boiler, of large evaporative capacity, is fitted with a superheater, chain-grate stoker, and superposed economiser, for evaporating up to 250,000 lbs. of water per hour.

Batteries of these boilers are now installed working at pressures up to 1,200 lbs. per square inch.

ACTUAL PERFORMANCES WITH BABCOCK AND WILCOX MARINE BOILERS.

(*Date of Trial, June* 1910.)

Heating surface in sq. ft.	2,571					
Firebar surface in sq. ft.	57·89					
Kind of fuel used	Pocahontas hand-picked.					
Duration of trial—hours	24·06	24·5	24	24	6	3
Amount of draught in stokehole, inches of water	—	—	1·41	1·95	1·78	3·0
Average observed gauge pressure, lbs. per sq. in.	202	201·4	202·8	201·6	200·6	200·4
Average observed temperature of water fed to boiler	211·3	211·4	207·8	204·1	200·9	194·4
Pounds coal consumed per sq. ft. grate per lb.	15·22	24·81	35·48	41·53	43·81	70·24
Pounds water evaporated per sq. ft. heating surface from and at 212° F.	3·88	6·43	9·53	10·52	10·52	14·76
Pounds water evaporated per sq. ft. firebar surface from and at 212° F.	172	286	401	468	467	656
Pounds of water evaporated per lb. of coal per hour (actual)	10·76	10·95	10·71	10·64	10·09	8·77
Pounds of water evaporated from and at 212° F.	11·32	11·52	11·3	11·26	10·67	9·37
Number of boilers	1	1	1	1	1	1

THE HOWDEN H.P. BOILER.

(*James Howden & Co., Ltd.*, 195 *Scotland St., Glasgow.*)

A few of the leading features of this boiler (fig. 33) are as follows :—

1. Reduced weight of boiler and of the water in comparison with Marine Cylindrical Boiler of similar output, approximately 40 per cent.

2. Reduced diameter of shell and thickness of plate.

3. Simplicity of construction.

4. No flat combustion chamber surfaces requiring staying.

5. Shipment of boiler could be made in sections, involving lighter weights to be handled, and also reduction in the spaces left open for the shipping of boilers.

6. In this design the most valuable features of the ordinary cylindrical boiler have been retained.

7. There is ample steam space and steam liberating surface, which will avoid any tendency to priming.

8. A minimum amount of brickwork, and as brickwork and casings are not exposed to high furnace temperatures, the losses by radiation are reduced and the temperature of the stokehold would not be any higher than in the case of a cylindrical boiler.

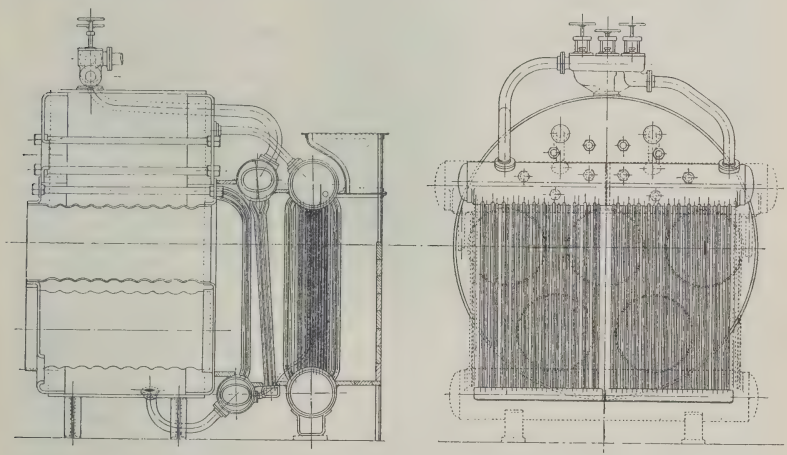


FIG. 33.—Typical Howden H.P. Boiler.

9. The water tubes are subjected only to the comparatively low temperature of the combustion chamber gases, without radiation effect from the fuel. They are, therefore, from the point of view of temperature, in the same condition as the middle and back row tubes of a water tube boiler, so that trouble by overheating should not occur and the life of the tube should be correspondingly prolonged.

10. Any mud introduced with feed water or scale would be deposited in the lower water tube drum, which is not subjected to flame, and where it could be readily removed.

11. The boiler is readily accessible for cleaning and inspection, both in combustion chamber and internally.

The 'Spencer-Hopwood' Patent Boiler.

This, a vertical boiler with inclined water-tubes (manufactured by *Messrs. Spencer-Hopwood Ltd., Scotswood, Newcastle-upon-Tyne*), has the tubes spread immediately over the fire, making it extremely rapid in steam raising); 100 lbs. pressure per sq. in. can be obtained from cold water in the boiler in thirty-five minutes from the time of lighting the fire. A number of mandoors of the McNeil type are provided opposite both ends of the nest of tubes, giving easy access to them and also to the entire interior of the boiler for cleaning and renewal purposes.

The standard 'Spencer-Hopwood' patent boiler will evaporate from 25 to 50 per cent. more steam than a vertical cross-tube boiler of the same dimensions.

This boiler is also made in a squat pattern for use where head room is limited, and most modern railway breakdown cranes are fitted with it.

Recently a new and improved 'Spencer-Hopwood' patent boiler has been put on the market, for standard pattern sizes Nos. 16 to 24. These have a much deeper nest of tubes than before, resulting in a higher thermal efficiency and still smaller space required for the output. In the smaller sizes this is done by arranging the major axes of the manholes in a vertical direction instead of the usual horizontal one, and using more than two, thus deepening the nest of water-tubes 33 per cent. In the larger sizes there are two or three rows of manholes, which enables the nest to be deepened as much as 100 per cent. Further, by the use of more than two manholes per row of manholes, the width of the tube nest can be increased to anything desired, enabling these boilers to be made up to 6 ft. diameter and over, to evaporate up to 5,000 lbs. of steam from and at 212° F. per hour, so that they can be installed to do the work of a small Lancashire boiler and to give as high an efficiency, without the floor space, without the brick setting, without the chimney, and without the necessity of taking hours to raise steam.

Where superheated steam is required, a superheater can be supplied for all sizes, which is arranged in the uptake.

Stock boilers are built for working pressures up to 120 lbs. per square inch.

Evaporation of 'Spencer Hopwood' Patent Boilers.

Standard Pattern for General Purposes.						Squat Pattern for Railway Breakdown Cranes and other Purposes where Headroom is limited.					
Number of Boiler.	Evaporation lbs. per hour. Natural Draught.	Evaporation lbs. per hour. Induced Draught.	Height.	Diameter.	Approximate Weight.	Number of Boiler.	Evaporation lbs. per hour. Natural Draught.	Evaporation lbs. per hour. Induced Draught.	Height.	Diameter.	Approximate Weight.
			Ft. Ins.	Ft. Ins.	Cwts.				Ft. Ins.	Ft. Ins.	Cwts.
2	261	300	5 3	2 6	18	2	257	320	4 0	2 6	17
3	376	480	6 3	2 9	21	3	336	460	4 6	2 9	19½
4	453	550	6 6	3 0	25	4	418	550	4 9	3 0	24
5	537	650	6 9	3 3	32	5	480	600	5 0	3 3	26½
6	625	760	7 6	3 6	36	6	545	690	5 0	3 6	29
8	748	960	8 0	3 9	44	8	656	880	5 6	3 9	34½
10	903	1,160	9 0	4 0	52	10	736	970	6 0	4 0	39½
12	1,053	1,360	9 9	4 3	60	12	867	1,160	6 9	4 3	49
16	1,500	2,000	10 9	4 9	77	14	1,055	1,400	7 6	4 6	56
18	1,700	2,200	11 0	5 0	94	16	1,235	1,650	8 3	4 9	70
20	2,000	2,700	11 6	5 3	125	18	1,400	1,860	9 0	5 0	85
22	2,350	3,100	12 0	5 6	130	20	1,550	2,100	9 0	5 3	94
24	3,000	4,000	12 6	6 0	160	22	1,700	2,500	9 0	5 6	100

The evaporations given are based on actual tests with 5 ft. of chimney only, feed water at 60° F., steam at 100 lbs. pressure per sq. in , and good steam coal.

Induced draught is obtained from a steam jet included with the boiler mountings.

These boilers fired with oil evaporate 25 per cent. to 30 per cent. more than the (natural draught) figures given above for coal.

SPECIFY
SPENCER-HOPWOOD PATENT WATER-TUBE VERTICAL BOILERS

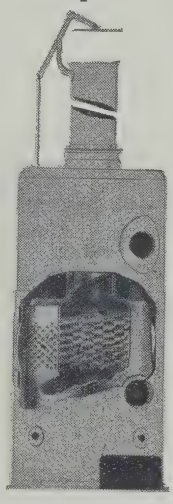
IF you are intent on obtaining the maximum results from a Donkey Boiler which costs the least to run and the least to maintain, read the following eleven points about the Spencer-Hopwood Vertical Boiler:

I.
Costs no more than other boilers of the same output.

II.
Raises steam from cold in HALF-AN-HOUR day after day without any injurious effects.

III.
Occupies less floor space than any other boiler for the same output.

IV.
Has no rivet heads subject to high flame temperatures.



V.
Is the most accessible for cleaning and inspection.

VI.
The tubes are immediately accessible for renewal, inspection, or internal cleaning, by removing the easily-handled mandoors at the side.

VII.
The tubes can be cleaned on the fire side with a lance while the boiler is working.

VIII.
33% Overload Capacity.

IX.
Lowest Maintenance Costs.

X.
Highest Evaporation for size.

XI.
Greatest Heating Surface for size.

SPENCER-HOPWOOD PATENT WATER-TUBE VERTICAL BOILERS ARE USED ALL OVER THE WORLD.

SOLE MAKERS

SPENCER-HOPWOOD, LIMITED

Scotswood Works, NEWCASTLE-UPON-TYNE

'Phones: Central 7301-2-3 *'Grams:* Spenhop, Newcastle-upon-Tyne

25, VICTORIA STREET (South Block), S.W.1

'Phones: VICTORIA 1575 and 4043 *'Grams:* Spenhop, Sowest, London

YARROW
PATENT
WATER TUBE BOILERS
FOR
LAND AND MARINE PURPOSES.

YARROW GLASGOW

THE YARROW BOILER IS FLEXIBLE AND
CAPABLE OF THE HIGHEST OVERLOADS.

FOR ALL FORMS
OF
FIRING.

YARROW BOILERS

The 'Sulzer' Water-tube Boiler.

This boiler (*Sulzer Bros., London*), shown in fig. 34, is constructed in the single and double bank type. Its main features are:—

Perfectly straight and nearly vertical tubes, preventing 'hogging,' and minimising the amount of deposit on same, resulting in the intimate contact of the tubes with the products of combustion, therefore larger heating surface than would be obtainable with slanting water-tubes.

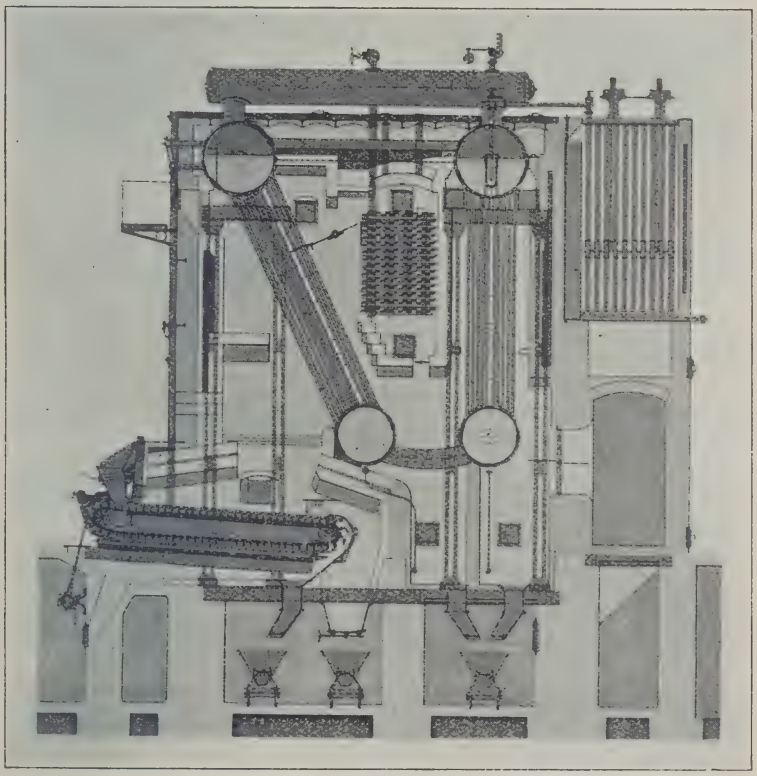


Fig. 34.—Sulzer Double-bank Four-pass Vertical Water-tube Boiler.

The tube-plates are shaped so as to reduce the strain on the material to a minimum.

Free suspension of the boiler permitting free expansion, thus avoiding leakage at the joints.

Brisk circulation of water promoted by upright tubes and suitably arranged return tubes, steam bubbles escaping freely to the upper drum.

Perfect control of the amount of superheat.

General simplicity in design, and therefore great reliability.

THE YARROW LAND BOILER.

This boiler (manufactured by *Messrs. Yarrow & Co., Ltd., Glasgow*), fig. 3, consists of three or more drums, including a saturated steam drum of large capacity, which is connected to each of the water-drums by straight tubes. The drums are of circular section, built up of pressed steel parts, the shell generally being in one ring, so that no intermediate circumferential joint is required. The drum ends are of partial spherical form, and no stays are necessary. For higher pressures, as in the case of Yarrow Boilers adopted in many recent installations, the drums are made from hollow forgings, either entirely seamless, or with riveted joints only where separate ends are attached. The tubes in both the superheater and the boiler are expanded into thickened tube plates and bell-mouthed at their ends. The tube plates on the inside of the drums are, where necessary, recessed out in way of the tube holes, to enable the expanding of the tubes to be perfectly carried out. The Yarrow boiler is supported by the steam drum being slung from a strong steel structure, which leaves the tubes and water-drums perfectly free to expand independently.

As one of the water-drums is placed at the front of the furnace and another at the back of the furnace, the tubes joining these drums to the steam-drum are adjacent to the furnace. It is well known that, owing to direct radiation, by far the greater part of the heat utilised in a boiler is transferred in the tubes nearest the fire. From the above it will be realised that as so large a proportion of the heating surface is adjacent to the fire, the total heating surface of the Yarrow boiler for the same output and efficiency can be considerably less than is usual.

Another characteristic of the Yarrow boiler is the very rapid circulation through the tubes due to their being straight and nearly vertical.

An outstanding feature of this boiler is the complete absence of any form of baffle or obstruction in the path of the gases where dirt could collect. This feature can be readily seen from fig. 36, which shows the cross section of a Yarrow boiler with superheater as arranged when the flue gases are discharged into a common flue at the back of the boiler.

The Yarrow type of superheater consists of a single mild steel drum into which inverted **U** tubes are expanded. Suitable internal divisions are fitted in the drum so that the required number of passes through the **U** tubes can be obtained. This form of superheater is self-draining.

The Yarrow boiler is often arranged with a damper in the flue above the side of the boiler on which the superheater is situated (see also description of Yarrow marine type boiler, page 2399), and by means of this damper all the gases can be made to pass up the side of the boiler remote from the superheater. This arrangement forms a convenient method of controlling the degree of superheat which can be reduced almost to a minimum without causing a serious falling-off in the steam-raising capacity or efficiency of the boiler.

In order to clean the interior of the boiler it is only necessary to remove the manhole door from each drum end (each is secured by two bolts). A man has ample room in the saturated steam drum for operating the tools for cleaning the inside of the tubes. For cleaning the outside of the tubes where this cannot be conveniently done from the furnace, access doors are provided in the side and end casings.

Owing to the great height of the furnace and to the large area between the tubes in the furnace rows, or, in other words, the low velocity of gases passing out of the furnace, the Yarrow boiler is found to remain comparatively clean after long periods of steaming. This, coupled with the fact that there are no places on which dirt can lodge, are the chief reasons why the efficiency of the Yarrow boiler does not fall off as in some other types.

For Electric Power Stations where peak loads have to be provided for, the Yarrow boiler is particularly suitable, as it cannot be damaged by too quickly increasing the rate of evaporation, and it can be forced to almost any extent without fear of damage. It is sometimes fitted with feed-water economisers or air-heaters, and sometimes with both. The general design of the boiler lends itself to making a compact and accessible unit when combined with these adjuncts.

THE YARROW WATER-TUBE BOILER.

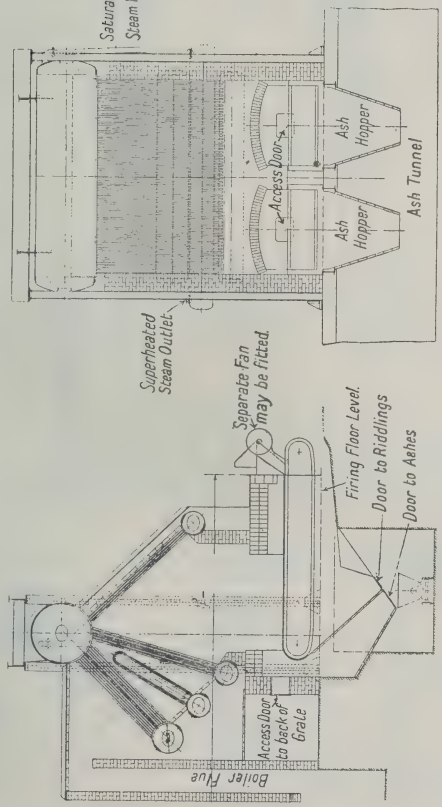


FIG. 35.

BOILER SETTING.

In ordinary practice, as developed by long experience, a boiler (figs. 36 37, 38) is set upon two longitudinal walls, parallel to one another, and distant apart over half the diameter of the boiler. On these walls are laid specially formed seating blocks (fig. 39) of firebrick, the seating edge of which should not exceed 3 inches in width, and may advantageously be convex in

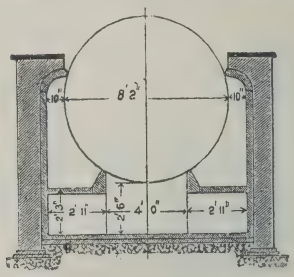


FIG. 36.

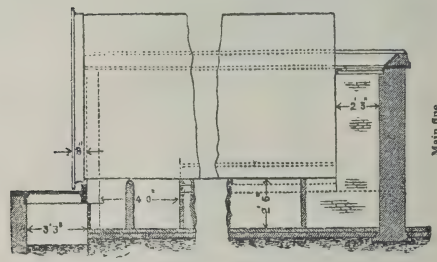


FIG. 37.

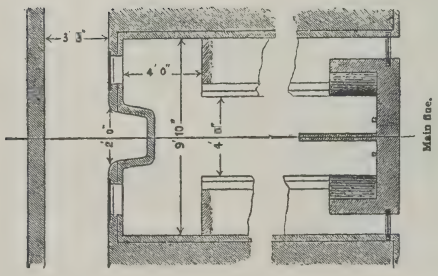


FIG. 38.

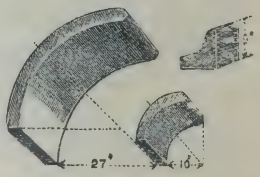


FIG. 39.—Flue Covers and Seating-block.

profile so as to touch the boiler plate along a line only. Where the ring seams cross the seating blocks, a rectangular gap should be cut across the seating for purposes of inspection, and specially formed wedge pieces ought to be provided to close these gaps. These are removed for inspection and at once replaced.

Properly designed fireclay blocks for the seating and downtake allow for expansion and contraction, and minimise air leakage. This is important, as air leakage often accounts for 5 per cent. of the coal bill. Lancashire boilers can be made to work practically airtight.

Foundations.

In preparing the foundations, a matter of great importance is that good drainage should be ensured, as it is found that serious damage often occurs to boilers owing to dampness which arises from the foundation, such outside wasting not being caused by leakages from the boiler itself, but simply from being built either on wet ground or where surface water is allowed to accumulate. Care should also be exercised that the foundations are built in hard well-burned bricks.

It is often advisable to concrete the whole area under the boiler and flues.

It is frequently specified that the internal parts of the flues should be lined with firebricks. For boilers of the Lancashire type, experience shows that this is an unnecessary expense. Good quality hard bricks are satisfactory, the only firebricks used being the specially shaped seating blocks in contact with the boiler shell and those for the bridge at the back of the grates.

Where in contact with the boiler shell, the bricks must be set in fireclay. Elsewhere ground mortar (three parts of ashes to one of lime) may be used.

The boiler should be set having a fall of about 1 in 240 towards the front end, so as to ensure its being thoroughly drained of its contents when it is emptied for its periodical cleaning. The top of the side flues also should be made up by flue covers (fig. 40) of fireclay, in such a form as generally to allow a flue of about 10 inches at each side of the boiler, the flues themselves being of such dimensions as will allow of a man getting thoroughly round for the purpose of cleaning and examining the boilers from time to time.

The following materials are required to seat a boiler 7 feet in diameter by 30 feet long :—

Firebricks and bricks for flue lining, including breakage	**6,000**
Downtake covers, 6 inches wide	14
Seating blocks	52
Flue covers, 12 inches wide	64
Fireclay and ground mortar	**3 tons**
Common bricks for external walls	Quantity varies according to thickness.

Mortar for laying up firebrick is too often made of fireclay alone. This is very bad practice for the reason that a mortar made of fireclay alone has no permanent body, and after it has been subjected to heat for some time it crumbles from between the bricks. A satisfactory mortar for this purpose may be made by mixing ½ ton of fireclay and 1 ton of firebrick dust (crushed firebrick) to the proper consistency, for each 1,000 firebricks laid. The brick dust, or crushed firebrick, should be the size of a pea and smaller. This mortar, when set, forms a compact refractory binding that is as strong as the brick itself.

Leaky Boiler Settings.

Leakage through boiler settings of brick can be lessened by giving the walls two coats of Portland cement wash and then one coat of cold water paint to brighten them up. Or the brick settings may be sealed by applying cement with a covering of canvas pasted over and then painted.

Another method is to build with mortar in which a certain quantity of cement has been mixed, the amount to be gauged by the tenacity of the mortar, which must not hold bricks so that they will be destroyed if it is found necessary to take the brickwork down : a 5 per cent. cement will give this, but may be increased or decreased ; the greater the percentage of cement the tighter the setting, but with the above disadvantage.

Washing the bricks with the liquid cement when building will give best results ; the brickwork should then be coated with the thin pure cement liquid afterwards, the cement being used as a wash.

(*Horobin.*)

Some Modern Improvements.

When preparing the foundations for a boiler or battery of boilers, it is essential that the position chosen be carefully examined, for if the situation is at all damp or there is a liability of the surrounding ground getting soaked with surface water or floods, a proper system of drainage must be adopted For this purpose an experienced engineer should be consulted. Also where the ground is of a sandy or shaley nature it may be necessary to resort to piling, and as there is no general rule to follow, experience can alone decide the laying out of the work. It is essential that there should be no sinking or warping of the foundations, which not only hastens the formation of cracks in the settings, but induces strains in the pipe work and causes other serious inconveniences.

Most boiler makers supply a general plan of the brickwork setting, which is usually of the old type, and shows broad-faced blocks and semi-circular covers in contact, with two or three courses of brickwork on top. It is far better, however, to have one's own carefully prepared drawing made for the specific installation, prepared by some firm of repute which makes a specialty of this class of work and approved by the engineers and the Boiler Insurance Co.

After the foundations are laid and brought up to the level ready for commencing the boiler setting proper, it is well sometimes to allow several days to elapse to ensure that there will be no further settlement, especially where concrete has been employed.

In the ordinary type of Lancashire boiler the path of the gases, after leaving the boiler flues, is along the bottom flues to the front, and then dividing to the side flues they pass thence into the main flue through separate dampers. There are many instances in which the gases pass first through the side flues and then along the bottom flue to the main. This system is not to be

recommended, as the hottest gases should first pass under the boiler. As to the dimensions of the flue passages, these are a compromise, small flues giving difficulty when inspecting, while large flues add too much useless surface. The back downtake should not exceed 2 ft. 6 ins. from back to front, the sides tapering from the width over the furnace tubes to about 3 ft. 6 ins., and not exceeding 4 ft., with a depth of from 3 ft. to 4 ft., depending upon the size of the boiler. It is usual now to give a greater width in the downtake in order to accommodate, if necessary, a superheater without requiring too much reconstruction. The side flues are made with vertical sides giving 9 to 12 ins. clear from brickwork to boiler plates at the centre line of diameter, and the floor level being about on a level with the bottom of the boiler gives a convenient chamber for an inspector to crawl through, while providing ample space for the accommodation of soot.

With the Paxman 'Economic' type of boiler (page 1581) the arrangement is somewhat different. The gases pass from the main iron flues into the back chamber where they divide right and left and return through the internal boiler tubes into two front smoke boxes from whence they pass downwards to two underneath flues separated by a mid feather upon which the boiler rests. These flues meet at the back, and the products of combustion pass out through a single damper or dampers to the main flue.

A number of firms make special firebrick blocks for boiler setting. These include mid feather and side seatings, large and small segment blocks for the top of the side flues and back chamber, besides flat blocks and other shapes less commonly employed but often required. However well the brickwork, as constructed on the older plan, may be cemented together, the constant movement of the boiler under steam and the changes of temperature soon begin to cause cracks and crevices to appear, especially between the boiler shell and the brickwork. The result is a loss of draught and short circuits in parts of the flue causing a considerable reduction of thermal efficiency. Further, the blocks being cemented in position the work of the boiler inspector, who has to thoroughly examine the boiler plates on the outside, is not an easy one, and to remove the permanent brickwork entails a very considerable amount of trouble and expense.

Cushioned Seating Blocks and Covers.

The 'Pearson Patent' Asbestos Cushioned Seating Blocks and Flue-Covers have been specially designed to remedy sources of waste caused by (*a*) corrosion of the plates, due to the presence of moisture in the brickwork; (*b*) short circuiting of the hot flue-gases; (*c*) leakages of cold air into the flues; and at the same time to be readily removable and replaceable when an inspection has to be made.

The arrangement, as shown by figs. 40 and 41, consists of a specially prepared cushion of hard asbestos, compressed into a groove formed on the face of the blocks, in such a way that the boiler-plates are completely insulated from all contact with the blocks themselves.

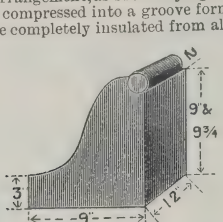


Fig. 40.—Patent 'Asbestos Cushioned' Seating Block.

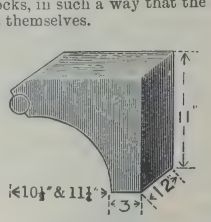


Fig. 41.—Patent 'Asbestos Cushioned' Flue-Cover.

The plates are thus secured against the rubbing caused by the expansion and contraction of the boiler, and also against any risk of corrosion through contact with damp brickwork or blocks which alone has been the cause of very many serious explosions.

Owing to the non-absorbent nature of the material, and also to the high pressure to which it is subjected, the hard asbestos-cushion does not take up moisture, as is the case with ordinary clay blocks, consequently there is no possibility of corrosion. No corrosion has ever been found in boilers set with these seatings, and boilers can be inspected which have been running for several years. In addition to this advantage, the presence of the compressed asbestos cushion ensures an absolutely gas-tight joint between the boiler plates and the blocks, so that not only is there no risk of any short circuiting of the flue-gases, but (an even more important consideration) no danger of cold air finding its way into the flues from the outside, there being no necessity to fill up large spaces with ashes or sand in order to ensure air-tightness. The arrangement gives a specially large heating surface. Under the old system, 10 per cent. of the external surface of the boiler was covered up by, and in contact with, brickwork; while in a boiler set with the Patent Asbestos Cushioned Blocks, less than 1 per cent. of the external surface is covered, and at no point are the seating blocks or covers in contact with the boiler plates.

The 'Patent Asbestos Cushioned Seam-Inspection Blocks and Covers' (figs. 42 and 43) are designed to meet the requirements of the Board of Trade and the insurance companies. The Board of Trade now require a proper inspection at least once in fourteen months, and the removal

of the brickwork, wherever necessary, is compulsory. These blocks enable complete examination of the ring-seams, without any need to remove more than the block or cover opposite the seam. This means a great saving of time and expense at each inspection. The lower half of the block is built and bonded into the side wall of the flue, rendering it a part of the permanent structure, and enables the upper half to be removed with utmost ease, admitting daylight to the flues and

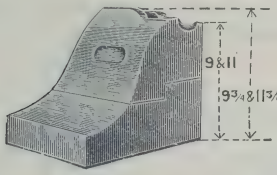


FIG. 42.—Patent 'Asbestos Cushioned' Seam-Inspection Seating Block.

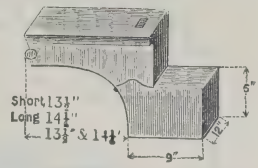


FIG. 43.—Patent 'Asbestos Cushioned' Seam-Inspection Flue Cover.

facilitating inspection. The upper part of the block turns slightly on the knuckle joint, and is thereby safe from cracking under boiler expansion. Handholes are inserted in upper half of each cover for lifting purposes.

It is probable that no part of a boiler setting of the old types gives more trouble than the back downtake. The constant expansion and contraction of the boiler causes the blocks or slabs, used to cover in the downtake flue, to crack and fall down into the flue. The 'Pearson Patent Interlocking Downtake Arch' (fig. 44) is an arrangement of great simplicity and reliability. It consists of an arch, of the required span, built up of interlocking bricks, so shaped that while

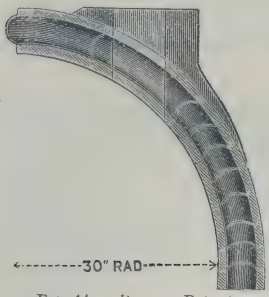


FIG. 44.—'Pearson Patent Interlocking Downtake Arch.'

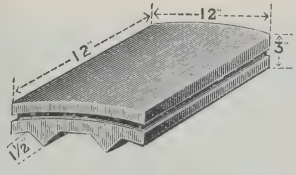


FIG. 45.—'Angular Contact Slab.'

they are as easily built up as an ordinary brick, when once in place they cannot be dislodged, and however much the arch may give to the movements of the boiler, no displacement of the brickwork can occur. Each set is fitted with two manhole blocks with lids, by means of which easy access is obtained to the flues, without disturbing the brickwork.

For the covering in of the boiler the 'Angular Contact Slabs' (fig. 45) may be used. In these coverings, advantageous use is made of the well-known fact that an envelope of stationary air is one of the best non-conductors of heat known.

The covering slabs are quite distinct, and have no connection with the flue-covers, the compressed asbestos cushion preventing any leakage of flue-gases: consequently while radiation of heat from the boiler is prevented, there is also no possibility of the temperature under the slabs being raised by the infiltration of hot flue-gases.

The covering slabs form a surface hard enough to be worked on, and can easily be removed for the inspection of the boiler plates at any time, and as easily replaced, the covering being in no way damaged by the removal.

Tests have shown the efficiency of this method of covering, there being a difference of over 90° C. between the temperature under the covering and the external air temperature.

The 'Liptak' Flat Furnace Arches.

(Liptak Furnace Arches, Ltd., 38 Victoria Street, London, S.W. 1.)

The Double Suspension system of constructing Flat Furnace Arches (fig. 46) as patented and supplied by Liptak Furnace Arches, Ltd., provides a renewable working face with ample protection for the ironwork.

The blocks of the working face are readily renewable, as the lower blocks are suspended from

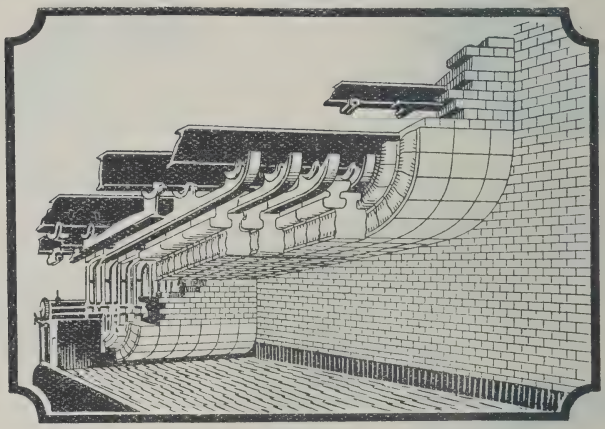


Fig. 46.

an upper arch of refractory material, so that if the whole of the lower blocks were burned away there would still be refractory protection between the furnace heat and the ironwork.

The arches have been supplied in large numbers in this country and in widths up to 26 ft. between walls.

Liptak Furnace Arches, Ltd., are also makers of air-cooled refractory walls for combustion chambers, and single suspended arches for industrial furnaces.

Securing Furnace-Linings.

Fig. 47 shows a simple method (Wildish Patent Bricks) of effectively securing furnace-lining firebricks, in which the fastening bolt is completely protected from the hot gases. The bolt head is of triangular shape, and in the direction of the long axis of it is a saw-cut in the bolt end. An ordinary pattern nut is fitted. The bolt head is introduced into a slot in the brick, which

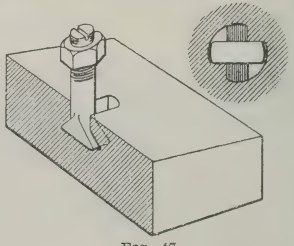


Fig. 47.

slot is enlarged in the centre of the brick to form the two quadrantal recesses (shown in small fig.). By rotating the bolt through an angle of 90 deg., it is locked into the brick, and can be prevented from turning whilst the nut, which is, of course, outside the boiler casing, is being screwed up. These bricks are said to have been put through extensive trials and to have proved quite satisfactory in service.

Bramwell Automatic Expansion Joints for the Fronts of Boilers.

When a boiler which is seated on brickwork, such as a Lancashire boiler, expands, the brickwork is crushed to a certain extent. In time slight crevices appear where the boiler is in contact with the brickwork. At the front of the boiler these crevices admit cold air into the side and bottom flues, lowering the flue gas temperature and reducing the efficiency of the boiler.

The Bramwell expansion joints are designed to stop this leakage without interfering with the structure of the boiler or its working in any way.

An 'L' section casing is fitted to the brickwork round some two-thirds of the circumference of the boiler, and an airtight joint is made between this casing and the brickwork.

Inside this casing is a ring of asbestos, which is made to maintain an airtight joint with the boiler shell by means of a wire rope, passing all round inside the casing, pressing against the back of the asbestos, and held tight by weights acting through a lever and pulleys. A joint is then made between the asbestos and the casing by screws at intervals, which draw the asbestos against the casing, tight enough to make a joint, but not tight enough to prevent the wire rope forcing the asbestos forward when the boiler contracts.

The whole device is flexible, and not affected by any irregularity of the circumferential expansion of the boiler, nor is the joint broken by the longitudinal expansion. Moreover, the sliding joints are made between metal and asbestos, and a fixed joint only is made with the brickwork.

The second part of the joint is an airtight door over the blow-off chamber. The door is split vertically to provide for the blow-off pipe, and is screwed down against asbestos packings when in position. It can readily be removed for inspection, but when in place the door completely excludes cold air, not only from the bottom flue by stopping leakage, but also from the surface of the boiler shell in the chamber.

The Eddystone.

(*Eddystones Ltd., Caxton House, Westminster, S.W.* 1.)

The Eddystone is a patented appliance for improving the operation of boilers of the Lancashire type. It consists of a central boss attached to which are helicoidal vanes (fig. 48), and is fitted in the boiler flue 5 or 6 ft. behind the fire-bridge.

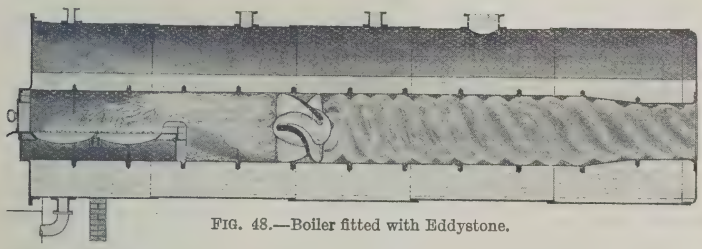


Fig. 48.—Boiler fitted with Eddystone.

It is made of refractory material, and is supplied in parts which are fitted together in place and jointed with refractory cement.

It causes the gases to take up a swirling motion, by which, and by the incandescent state of the Eddystone, combustion is made more complete. Also the motion of gases over the furnace walls improves the transmission of heat, and dust and grit are swept out of the flue tube.

The improved heat transmission is evidenced by the reduction in temperature of the furnace gases at the down-takes which has followed the installation of this appliance.

Fusible Boiler Plugs.

There are six primary causes for the rejection of fusible plugs, which may be roughly divided into two classes. Under the first class, rejections are included due to mechanical defects of casing material. The second class includes rejections due to lack of purity in the tin filling. The impurities in the latter case may be those either present in the original tin or those introduced during the manufacture of the plug.

The following conclusions have been reached as to the precautions to be taken in the manufacture of fusible plugs :

1. The pig tin should be at least 99·7 per cent. pure, containing not more than 0·1 per cent. lead, or 0·1 per cent. zinc, which are the requirements of the Steamboat Inspection Service.

2. The casing should be of bronze, an alloy the major constituents of which are copper and tin. Small amounts of zinc and lead increase the ease of casting and machining, and are not objectionable if not present in greater amounts than in the following compositions:

	I.	II.
Copper	88	87
Tin	10	7
Zinc	2	5
Lead	—	1

3. The pot or crucible for melting the tin should not be used for melting other metals, thus doing away with the liability of contaminating the good tin when these are not thoroughly cleansed.

4. Casings should be tinned on the inside with the same grade of tin used for filling, but the tin left over from this process should not be added to the filling to be used. Zinc-chloride flux may be used, although hydrochloric acid is preferred, though no flux need be used during the filling process.

5. The casing should be preheated to not above 250° to 275° C. (482° to 527° F.), and tin should be poured at a temperature not above 275° to 300° C. (527° to 572° F.).

(*'Bulletin of the American Institute of Mining and Metallurgical Engineers.'*)

In order that fusible plugs may fulfil the important function of safeguarding boiler flues from collapse in the event of shortness of water, it is essential the following conditions should be complied with:—

1. The fusible plug should be of an efficient and approved type. Common plugs are of little or no service.

2. The fusible plug should be kept clean both on the fire and water side.

3. As the nature of the fusible alloy is affected by prolonged service, the fusible alloy should be renewed after it has been in service for twelve months. Spare caps should be kept to facilitate this operation and the used cap returned for refilling.

4. When used plugs are refilled, the old fusible alloy should be removed and the surface of the shell scraped bright, and properly tinned and refilled with new alloy. Fusible plugs should not be refilled with used metal.

5. The fusible alloy must be a properly graded mixture such as research and experience has shown to be necessary. (*'Vulcan,' Oct.* 1920.)

CHIMNEY AND FLUE DIMENSIONS.

Apart from any consideration of production of draught, the height of a chimney must be such that the smoke and noxious gases are discharged at a level at which they will not be a nuisance. The minimum height is usually fixed by the bye-laws of local authorities.

For relation between draught in inches of water gauge and chimney height see page 1543.

The area of the chimney depends on the amount of coal burned per hour on the grates of the boilers which it serves, and on the velocity of the gases in the chimney, and their temperature.

The gas velocity again depends on the draught pressure, and on the frictional resistances in the flues and chimney. Thus the area of the chimney for a given size of boiler or battery of boilers should take into account many factors, some of which are not capable of exact determination. This accounts for the empirical character of the rules given by various authorities, and for the variations in them.

For typical conditions, say about 600° F. chimney temperature, flues not abnormal in length, and of area about 20 to 40 per cent. greater than chimney area, and 25 to 28 lbs. of coal burned per sq. ft. of grate area per hour, the following rules give suitable areas:—

H = height of chimney in feet; A = area at top of chimney in sq. ft.; G = grate area.

$$A = \frac{k\,G}{\sqrt{H}};$$

where $k = 1{\cdot}5$ for one boiler, 1·0 for two, 0·8 for three, 0·7 for four, 0·5 for five or more.

At 28 lbs. coal per sq. ft. of grate area per hour this corresponds to

$$A = \frac{k\,W}{28\sqrt{H}};$$

where W = lbs. coal burned per hour.

For a 100-ft. chimney, and $k = 1{\cdot}4$,

A = 0·5 sq. ft. per 100 lbs. coal burned per hour.

TABLES OF SIZE OF CHIMNEYS FOR BOILERS.

Diameter. Inches.	Area A in sq. feet.	E.	Height of Chimney in Feet. 50	60	70	80	90	100	110	125	150	175	200	225	250	300	Side of Equivalent Square Chimney.
			Pounds of Coal burned per Hour.														
18	1·77	0·97	115	125	135	145											16
21	2·41	1·47	175	190	205	220											19
24	3·14	2·08	245	270	290	310	330										22
27	3·98	2·78	325	360	390	415	440										24
30	4·91	3·58	420	460	500	535	565	595									27
33	5·94	4·48		575	625	665	705	745	780								30
36	7·07	5·47		705	760	815	865	910	965	1,020							32
39	8·30	6·57			915	980	1,040	1,095	1,145	1,225							35
42	9·62	7·76			1,080	1,155	1,225	1,290	1,355	1,445	1,580						38
48	12·57	10·44				1,555	1,650	1,740	1,825	1,945	2,130						43
54	15·90	13·51					2,135	2,245	2,360	2,515	2,755	2,975					48
60	19·64	16·98					2,680	2,825	2,965	3,160	3,460	3,740	4,905				54
66	23·76	20·83						3,470	3,640	3,880	4,245	4,590	5,905	6,265			59
72	28·27	25·08						4,175	4,380	4,670	5,115	5,525	7,000	7,425	7,825		64
78	33·18	29·78							5,190	5,535	6,060	6,560	8,180	8,680	9,150	10,025	70
84	38·48	34·76							6,070	6,470	7,090	7,655	9,465	10,040	10,580	11,590	75
90	44·18	40·19								7,480	8,195	8,850	10,835	11,490	12,115	13,270	80
96	50·27	49·01								8,560	9,380	10,135	12,290	13,045	13,750	15,060	86
102	56·75	52·23								9,720	10,650	11,500	13,855	14,695	15,490	16,965	91
108	63·62	58·83								10,460	11,995	12,960	15,500	16,440	17,330	18,985	98
114	70·88	65·83									13,425	14,500	17,240	18,285	19,275	21,115	101
120	78·54	73·22									14,930	16,130	21,000	22,275	23,480	25,720	107
132	95·03	89·18									18,185	19,645	25,130	26,655	28,090	30,775	117
144	113·10	109·72									21,760	23,505					128

Other formulæ are:

(*a*) $$A = \frac{W}{14\sqrt{H}}$$

(evidently for lower rate of combustion than the above, as it would agree with $k = 1{\cdot}5$ and rate of combustion 21).

(*b*) Area in sq. in. $= \dfrac{\text{lbs. coal per hour} \times 12}{\sqrt{H}}$ . . . . . (*Bourne*).

or $$A \text{ (sq. ft.)} = \frac{W}{12\sqrt{H}}$$

(*c*) Area in sq. ins. = 1·5 per lb. coal per hour . . . (*Bourne*).

or $$A = \frac{1{\cdot}5}{144} W$$

(corresponding to $k = 1{\cdot}5$ if H = 100 and combustion rate = 14·4).

(*d*) Area in sq. ins. $= \dfrac{\text{Area of grate}}{\sqrt{H} \times 1{\cdot}58}$ . . . . . (*Elswick*).

or $$A = \frac{1}{1{\cdot}58} \frac{G}{\sqrt{H}}$$

(corresponding to $k = 0{\cdot}63$).

(*e*) Kent gives formulæ which correspond to the following:—

$$\text{Effective area } E = A - 0{\cdot}6\sqrt{A};$$

$$W = 16{\cdot}65\, E\sqrt{H}: \qquad D = 13{\cdot}54\sqrt{E} + 4;$$

$$E = \frac{0{\cdot}06\, W}{\sqrt{H}}; \qquad S = 12\sqrt{E} + 4;$$

where,

D = diam. of round chimney in inches ; S = side of square chimney in inches.

The table on page 1611 has been calculated from the foregoing formulæ of group (*e*).

(*f*) Sometimes the area of the chimney top is taken as $\frac{1}{8}$ to $\frac{1}{10}$ that of fire grate, the height not being taken into account.

Flue areas should be about 20–40 per cent. greater than chimney area.

For stability of chimneys see Section XII, page 521.

SUPERHEATERS.

Separately fired superheaters are not often used ; the superheaters are usually integral with the boiler in the case of water-tube boilers, or placed close to the boiler, in the downtake flue, in the case of the Lancashire type.

The steam may pass through the tubes in parallel—single-flow—or some of the tubes may be in series with the rest—multiple-flow type. The multiple-flow type gives a higher rate of flow of steam in the tubes for the same tube surface. The heat transmission is thus increased, and a higher degree of superheat obtained. There is, of course, a slightly greater drop of pressure in the multiple-flow type.

The superheat added to the steam depends upon :—

1. The heating surface per pound of steam per hour.
2. The temperature difference, gas to steam.
3. The velocity of the steam in the superheater.
4. The velocity of the hot gases past the tubes.
5. The pressure of the steam.

Factor 2 is very difficult to predetermine accurately, and hence superheaters are still mainly designed by rule-of-thumb. Modern superheaters are frequently so made that the steam velocity can be altered, thus permitting an adjustment if an incorrect estimate has been made. There are two principal methods of altering the velocity : by plugging up tubes, or removing them, and by rearranging the proportion of tubes in parallel to those in series. The first method is of practically no value.

Factors 2, 3 and 4 change with the amount of steam used, as with the load. The effect of load on factor 2 (temperature) is more marked the farther the superheater is from the grate,

and is very marked if in the downtake of a Lancashire boiler for instance. Factor 2 is also very much dependent on the normal rate of firing and the quality of coal used. Thus with poor coal a superheat of 100° F. at full load may fall to 30° F. at half load. If the normal rate of firing is high or the quality of coal good, the change of loading has less effect.

Superheaters are designed chiefly with regard to practical considerations, such as standardisation of tube lengths, convenience of access to individual tubes or elements, and ease of repair.

Heating Surface of Superheaters.

As a rough approximation, the heating surface may be calculated on the assumption that the heat transmission is at the rate of 5 B.Th.U. per hour per sq. ft. of heating surface per °F. of the difference of the arithmetical mean temperatures of gas and steam.

If

W = steam generated in lbs. per hour;

H = heat required to superheat 1 lb. of steam (see Table IV, p. 1535)

T = mean gas temperature;

t = mean steam temperature;

then heating surface in sq. ft.

$$= \frac{W \times H}{(T - t) \times 5}$$

For example:—Let W be 6,000 lbs. per hour, the pressure 160 lbs. per sq. in., and superheat 200° F. The saturation temperature is 371° F., and final steam temperature 571° F., so that $t = 471°$ F. H (from Table IV) = 110·B.Th.U.

Assuming a Lancashire boiler, with the superheater placed in the downtake, the value of T may be 950° F., the heating surface $= \frac{6{,}000 \times 110}{(950 - 471) \times 5} = 276$ sq. ft.

The ratio of the heating surface of the superheater to that of the boiler varies very widely, even for boilers of the same type. It is difficult, therefore, to give typical figures. The following are taken from examples of actual practice, the ratio being expressed as a percentage of the boiler heating surface (excluding the economiser).

The ranges shown do not necessarily give the limiting values.

Lancashire boiler, superheat	150° F.	17 to 24 per cent.
,, ,, ,,	200° F.	25 to 28 ,,
Water-tube, Babcock . . .		15 to 40 ,,
,, ,, Stirling . . .		13 to 15 ,,
,, ,, Yarrow . . .		12 to 28 ,,
Locomotive		about 22 ,,

Boilers with superheaters require a smaller heating surface to give a desired efficiency.

Even if the amount of heating surface is calculated with due allowance for the effect on the heat transmission rates of the actual steam and gas velocities, it will be found that different rates of steam flow do not greatly alter the calculated heating surface. It is important to recognise, however, that at low steam rates the tube temperature becomes higher.

Sectional superheaters are arranged so that individual tubes or small groups of tubes may be readily removed without putting the whole superheater out of action.

(See also page 1552.)

The McPhail & Simpson Patent Superheater.

(McPhail & Simpson, Wolverhampton.)

The headers and tubes of these superheaters (fig. 49, p. 1614) are of solid drawn wrought steel.

The simple construction, and high-class material and workmanship used in their construction, render the superheaters absolutely steam tight, and at the same time ample means are provided for getting at the tubes should they by any chance require attention. This is made possible by the provision in the headers of the patent stoppers opposite the ends of each tube, whereby easy access to the tubes is obtained for expanding, examination, etc., and any individual tube can be

THE McPHAIL & SIMPSON PATENT SUPERHEATER.

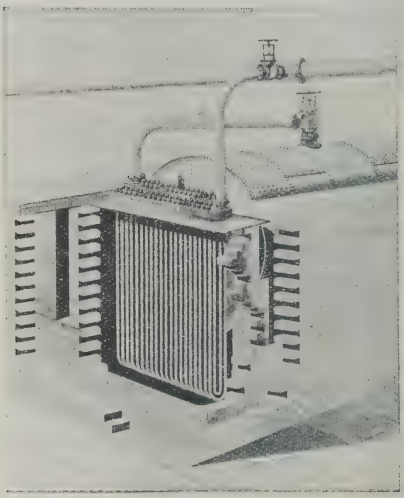


FIG. 49.

THE 'GALLOWAY' SUPERHEATER.

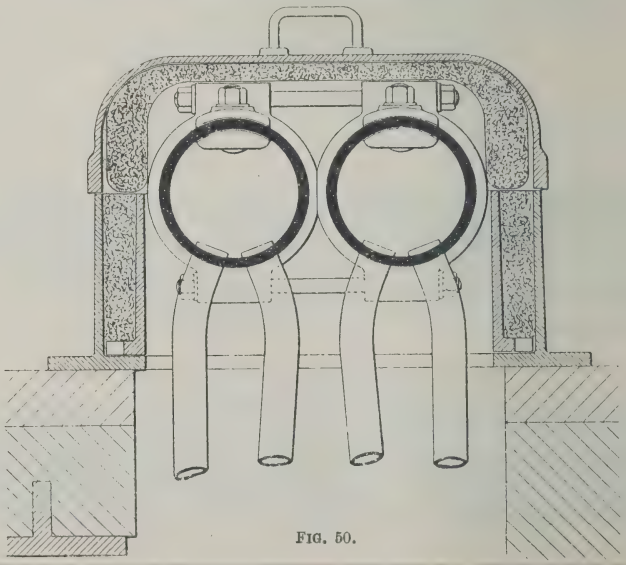


FIG. 50.

removed and replaced in a very short time, without disturbing the other tubes or pipe connections and without breaking a large joint.

There are no studs and no large joints used in these superheaters.

The substantial nature of the tubes entirely prevents warping.

The superheaters are adaptable to any type of boiler—Lancashire, Cornish, Babcock—and all types of water-tube boilers, both land and marine, and for independent firing.

Flooding of the superheaters, or the use of dampers, is not recommended.

Galloway Superheaters.

(Galloways, Ltd., Knott Mill Ironworks, Manchester.)

THE GALLOWAY PATENT MULTIPLE-FLOW SUPERHEATER.

This superheater (figs. 50 and 51) is made entirely from mild wrought steel. All parts are of circular cross-section and therefore of maximum strength. There are three headers—two short ones, with the inlet and outlet branches, and one long one closed at both ends. The headers are rigidly stayed together to prevent movement and vibration, and are connected by the tubes, which are of seamless steel. The tubes are in groups of four or five, each group commanded by an internal cap so as to facilitate inspection and cleaning. The steam passes from the short inlet header through the first set of tubes into the long header, and then through the second set of tubes to the short outlet header. The headers (fig. 50) are carried in a double-walled box resting on the brickwork and covered by a cast-iron cover, thus minimising radiation losses and protecting the superheater from weather and damage.

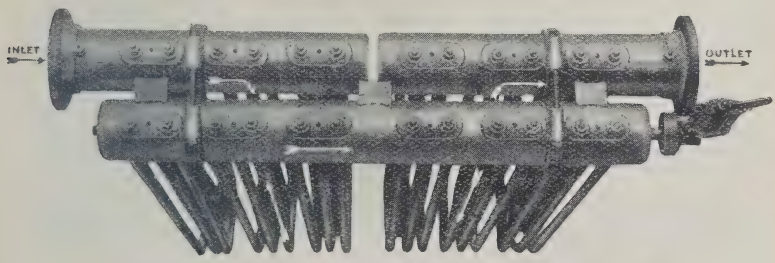


FIG. 51.—Plan view of Superheater showing direction of Steam Flow.

The joints between the superheater and the main steam piping are vertical, thus enabling the superheater to be removed for inspection without disturbing the steam pipes. If it is desired to work with the superheater removed, a straight connection or cross-over pipe is inserted in its place.

The superheat usually attained is 150° F. to 200° F. with ordinary firing.

THE GALLOWAY PATENT SECTIONAL WROUGHT STEEL STEAM SUPERHEATER.

To meet the demand for a superheater with tubes arranged so that they may be readily and quickly detached without disturbing the other tubes, *Galloways Ltd.* have designed and patented a sectional type superheater of simple and durable form.

The sectional type superheater is for use in the downtake flue at the rear of Lancashire and Galloway boilers, or for independent firing.

As shown (fig. 52) the superheater consists of two horizontal steel headers placed parallel to each other. A series of holes are drilled and faced for joints on the upper walls of the headers, to which the small steel boxes or hollow connecting chambers are secured by means of two studs of ample strength.

The superheater tubes are solid-drawn weldless steel of loop type or **U** shape, and the outside diameter of the tubes is $1\frac{1}{16}$ ins. The ends of the tubes, which are straight and not bent over (thus maintaining full thickness and strength), are expanded into the under walls of the connecting chambers, usually in groups of three.

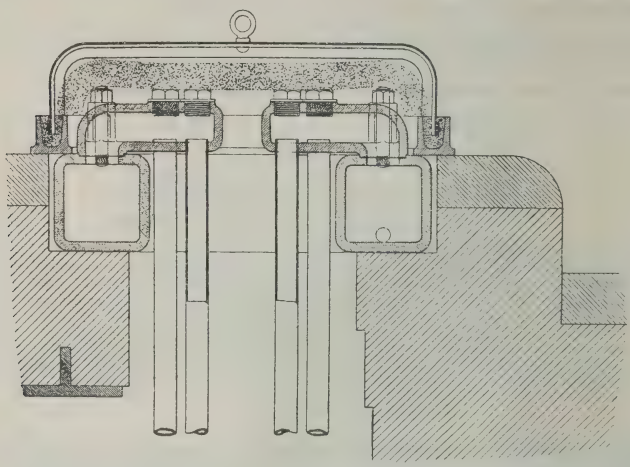


FIG. 52.

The upper walls of the connecting chambers are provided with screw plugs disposed in alignment with the ends of the several tubes, so as to give free access to the end of any tube for the purpose of examination or expanding without interference with or obstruction from the studs securing the connecting chambers to the headers.

Any of the groups or elements of superheater tubes, which are interchangeable, can be readily removed and replaced, or alternatively 'blanked-off,' by simply unscrewing the four securing studs of the two connecting chambers, and without disturbing any of the other groups of tubes or the main steam headers.

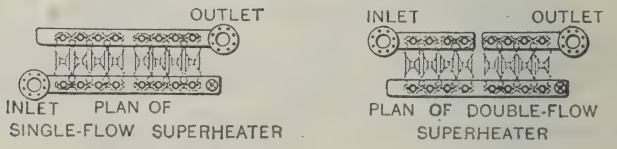


FIG. 53.

The superheater (fig. 53) is made either single-flow or double-flow, the latter type being adopted for higher degrees of superheat.

The single-flow type has headers of equal length and the steam has only a single pass through the tubes.

The double-flow superheater is similar in construction, with the exception that one of the headers is divided in its length to form two separate headers.

The steam is passed from the short inlet header, by way of the first nest of tubes, into the long header, and then through the second nest of tubes into the short outlet header.

The Babcock & Wilcox Standard Integral Superheater.

(Babcock & Wilcox, Ltd.)

The Babcock & Wilcox Patent Steam Superheater (fig. 54) consists of solid-drawn steel tubes bent into **U** shape, and connected at both ends with boxes or manifolds (F and H), one of which (F) receives the natural steam from the boiler through the dry pipe (N) and inlet pipes (D). The other (H) receives the superheated steam after it has traversed the superheater tubes (G), delivering it through pipes (J) and cross-box (K) to the stop valve (L) placed above the cross-box. Handholes are placed in the manifolds opposite the ends of the superheater tubes to facilitate inspection.

The superheater is placed in a position where there is practically no deteriorating condensation of the gases, and where the temperature is sufficiently high to ensure the steam receiving from 100° to 300°F. of superheat, or more if desired. It is not subject to the immediate action of the fire, as the furnace gases must first pass through the front part of the boiler, which comprises a considerable heating surface. Assuming the boiler to be in regular work, and the firing even, no

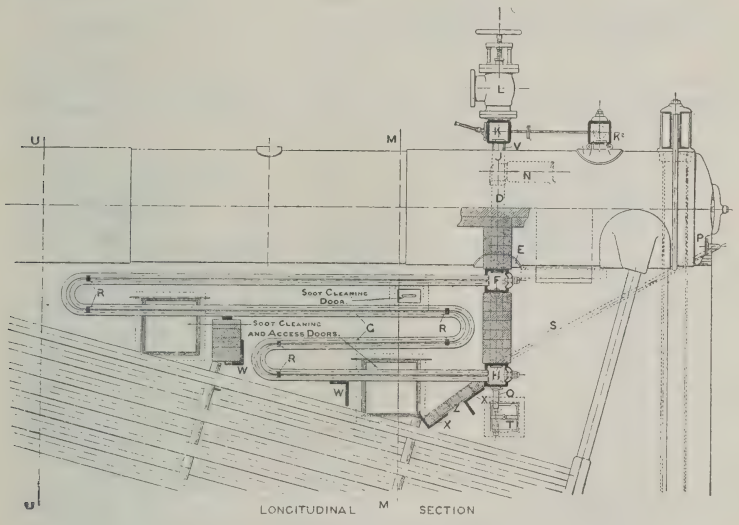


Fig. 54.

great fluctuations in temperature can take place where the superheater is fixed. It is readily accessible for examination, and for the renewal of tubes. There are no flanged joints, all the tube joints are expanded. Expansion is provided for by the tubes being free at one end, and by the manifolds not being rigidly connected with each other. Prevention against over-heating during steam raising from cold water after cleaning, etc., is ensured by the arrangement for flooding with boiling water, and using the superheater as part of the boiler-heating surface.

The flooding arrangement consists in a connection with the water space of the boiler drum; by means of the larger of two cocks (B) and connecting pipe (S) water can enter the lower manifold (H) and fill the superheater to the water level of the boiler. Any steam formed in the superheater tubes is returned into the boiler drum, through the collecting pipes (D), which, when the superheater is at work, convey saturated or natural steam into the upper manifold (F). Prior to opening the stop valve (L), when a superheater has been flooded, the water is drained away from the manifolds by the pipe (O) and the smaller cock (C). A sight glass (A) is attached to the outlet of the small cock to show when all the water has been blown out.

Very considerable economy is obtained by the use of superheated steam, and present practice in Central Station work is to use steam of the high temperature of 700° F.

The John Thompson Patent Sectional Superheater.

(John Thompson (Wolverhampton) Ltd., Wolverhampton.)

This superheater (fig. 56, p. 1619) consists of a pair of mild steel cylindrical headers connected by two groups of U or W tubes. The tubes have tapered ends machined to gauge and fitted into milled holes in the headers, making a metal-to-metal joint, and are held in position by a right- and left-hand screw stud. The advantages of this method of fixing the tubes are, perfect interchangeability and speed and ease with which any tube can be removed or replaced for any purpose, as every tube can be taken out without disturbing the headers, pipe connections, or brickwork.

The steam passes through the hot gases four times, consequently the velocity of the steam is increased and great efficiency obtained.

This superheater is adaptable to all land boilers, and is suitable for the highest pressures.

The 'M.L.S.' Superheater.

(The Superheater Company, Limited, London and Manchester; 'Schmidt' and 'Robinson Superheater Patents.)

The superheater, fig. 55, consists of solid drawn steel tubular headers, to which the elements are attached. These headers are reinforced internally with special strengthening bars into which the studs for the element connections are secured. The superheater elements are manufactured

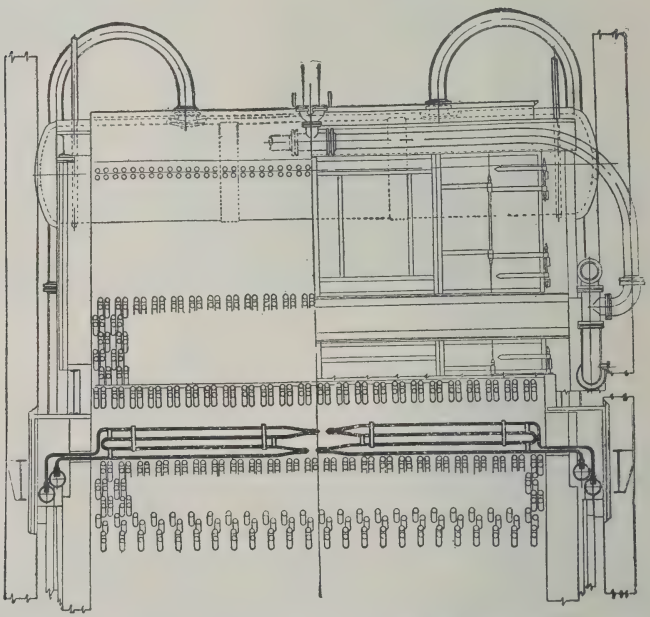


Fig. 55.—'M.L.S.' High Degree Superheater (*Inter-Deck Design*).

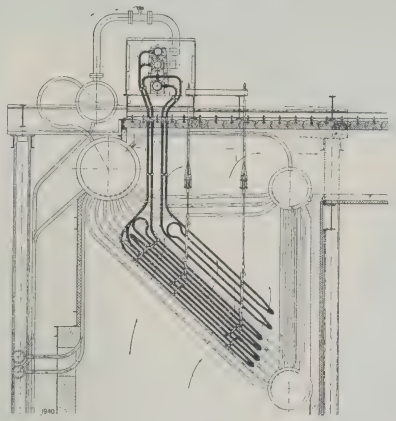
from high quality solid cold-drawn steel tubing, and they are connected to the headers by means of ball joint attachments, which latter have been adopted as standard by the manufacturers for their various types of superheaters.

"M.L.S." SUPERHEATERS

RECENT CONTRACTS

Barking
Billingham
Brighton
Brimsdown
Bolsover
Bwana M'Kubwa
Bury St. Edmunds
Collie
Derby
Grimsby
Hackney
Hammersmith
Hull
Leeds
Leicester
Mufulira
Neasden
N'Changa
Perak
Port Kembla
Rotterdam
Scarborough
Sheffield
Walsall

666 LBS. PER SQ. IN. 825° F.



"M.L.S." Single Pass Multiple Loop Intertube Superheater operating in the largest high pressure plant in Great Britain.

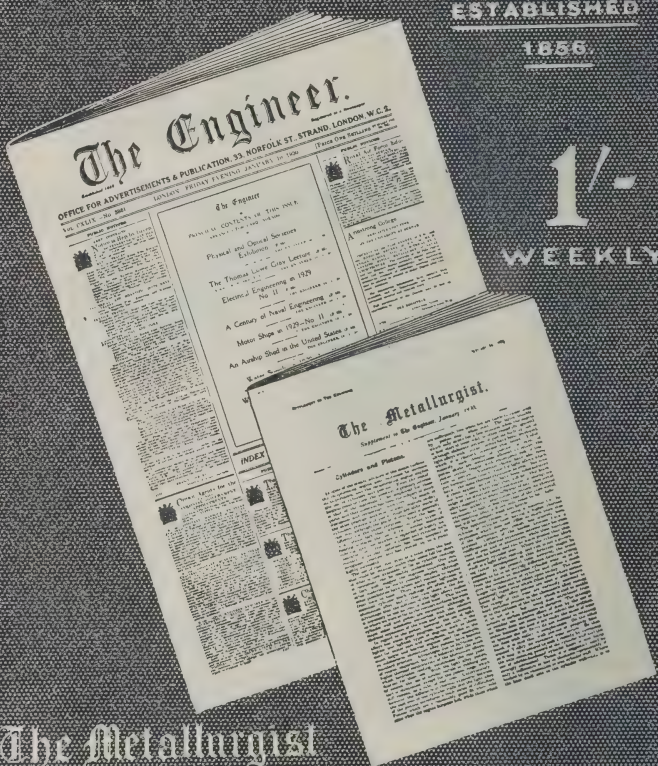
SINGLE PASS MULTIPLE LOOP CONSTRUCTION —AN EXCLUSIVE "M.L.S." DEVELOPMENT— IS AN ESSENTIAL REQUIREMENT OF HIGH DUTY SUPERHEATER DESIGN.

The SUPERHEATER COMPANY LIMITED
BUSH HOUSE, ALDWYCH, W.C. 2
Works :—Trafford Park, Manchester

The Engineer.

ESTABLISHED 1856.

1/- WEEKLY



The Metallurgist
IS A FREE MONTHLY SUPPLEMENT
to The Engineer,
EDITORIAL and PUBLICATION OFFICES:
28 ESSEX STREET, STRAND, LONDON, W.C.2.

ANNUAL SUBSCRIPTION RATE £3 5 0

F. 1619

THE 'JOHN THOMPSON' PATENT SECTIONAL SUPERHEATER.

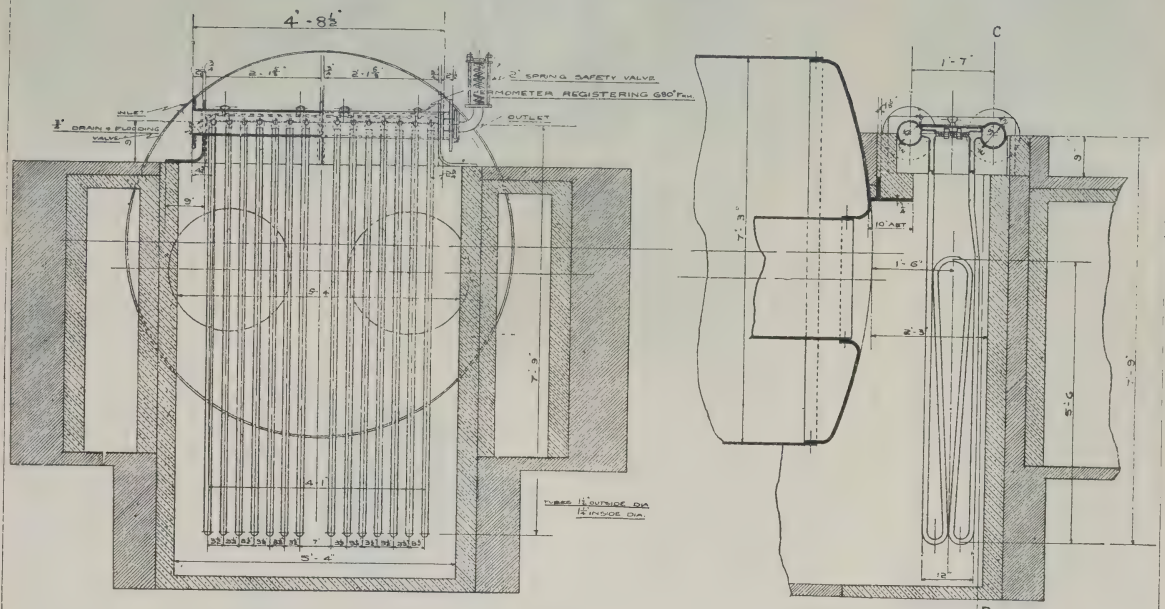


FIG. 56.

The 'M.L.S.' Superheater—*cont.*

The balls formed on the element inlet and outlet ends are forged integrally with the tubing itself and seat in cone recesses in the headers, so that metal-to-metal joints are provided without any form of auxiliary packing. One of the particular features and advantages of these types of superheaters is in the construction of the return bend of the element tubing. These also are machine forged from the element tubing itself, so that the final result is a return bend occupying minimum space but absolutely integral with the element tubing, and no oxy-acetylene or other form of welding is employed. By this method of manufacture, instead of there being a stress or weakening of the metal at the bend, as must occur when the tubes are bent to a small radius, the material is actually reinforced and strengthened. An additional advantage of this type of construction is that it enables superheater elements to be manufactured with the necessary number of loops to give a large heating surface without occupying any more space than the ordinary type of 'U' loop tubes generally employed. This enables the superheater to be placed in the most advantageous position in the boiler to give a constant desired superheat under fluctuating loads.

The 'Dee' Superheater.

(Dee Superheaters Ltd., Netherton, Worcs.)

Two distinct types of superheaters known as 'Dee' Patent Superheaters, are manufactured by Dee Superheaters, Ltd., Bishton's Bridge, Netherton, Worcs., the Expanded-Tube and the Union-Jointed types.

The name 'Dee' is given owing to the headers in the superheaters, being 'D' shaped.

The Expanded-Tube type can be arranged either with a single header or with two headers, and in the latter case they can be arranged parallel and level with or placed one above the other. The tubes are arranged in groups, and each group has one hand-hole (fig. 57), which comes in the upper flat part, enabling a more efficient joint to be made than can be obtained with a curved surface. The inner cover and stud is a stamping, and the cover is formed with a spigot which fits into the hole. Owing to the curve of the header access is given to a large number of tubes. It allows of such an arrangement of the tubes that many tube ends as arranged can be easily expanded through the one hand-hole without having to resort to any special appliances.

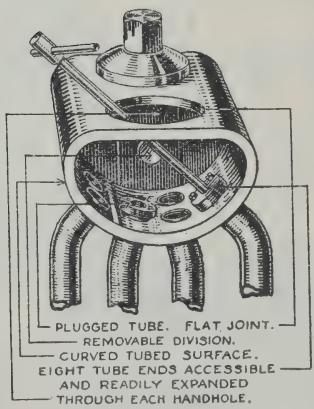


Fig. 57.

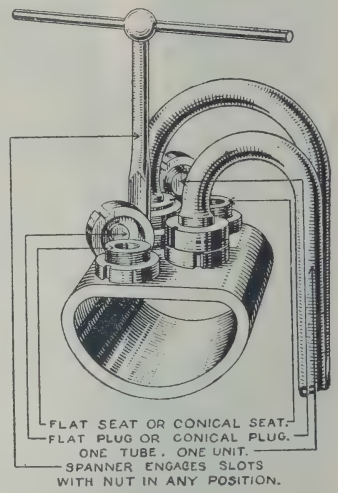


Fig. 58.

When an exceptionally high degree of superheat is required two headers are employed.

The Union-jointed tube type superheater consists of two headers of 'D' section connected together by the tubes through which the steam passes. Each of the tubes (fig. 58) is entirely independent of and interchangeable with any other, so that handling is easy and the quantity of stock tubes may be a minimum. The joint of each end of a tube is made independently, which

overcomes the difficulties experienced and skill required in fixing the tubes in those types where pressure has to be exerted centrally between the ends of the tubes. Two types of union joints are provided, one of which has a conical seat, and the other a flat seating, for use in connection with a jointing material.

The following describes the methods of applying the superheaters to the various types of boilers. For Cornish boilers, where the downtake in which the superheater is placed is of necessity small, the single-header type is used, as it takes up very little space and leaves ample room for cleaning and inspection. In the centre of the header is placed a patent sectional division, which is of wedge type in three parts and can be fixed in position through the hand-hole, thus acting as a partition, and the steam entering the branch on one end of the header passes down one end of the tubes and up the other end of the outlet branch. For Lancashire boilers, where the space available in the downtake is small, the same type is used, except that two divisions are placed in the header, so that the steam passes four times through the tube-heating surface.

The 'Dee' Patent superheaters are also suitable for installing in conjunction with water-tube, economic (Dry-back), vertical, etc., types of boilers, or may be used as independently fired units.

De-Superheaters.

The 'Drayton' De-Superheater.

(The Drayton Regulator and Instrument Co. Ltd., West Drayton, Middlesex.)

The body of this De-superheater consists of a vertical cylindrical shell, the size of which is determined by the amount of steam to be handled, its pressure, and the maximum temperature reduction required.

The superheated steam enters the shell towards the top of the body and passes down an annular space formed between the outside of the outlet pipe which extends downwards into the body from the centre of the top and the inside of the shell itself.

The de-superheating water, which should be supplied at least 15 lbs. per sq. in. in excess of the pressure of the superheated steam, passes through a fine strainer and two pressure-operated valves into the spray ring, which distributes it to the spray heads situated on the inside of the shell. A by-pass taken from between the two pressure operated valves enables the de-superheater to be placed under hand control.

Passing through the spray heads, the de-superheating water becomes a fine mist, travelling up the de-superheater body and meeting the superheated steam in contra-flow. The mist cools the incoming steam and itself becomes steam. The mixture then passes over baffles situated in the lower portion of the shell, when it is dried, and finds its way to the entrance of the outlet tube. Grit and other solid impurities are retained by the mud stool fixed to the bottom of the body and the clean, dried, de-superheated steam rises up the outlet tube, past the thermostat of the 'Drayton All-Metal Regulator' and into the service mains.

The pressure chamber of the first pressure-operated valve is connected to the inlet branch of the De-superheater. As the valve is opened by pressure, it is impossible for water to pass to the sprays unless steam is passing into the De-superheaters. On the other hand, the opening of the steam admission valve to the De-superheater automatically opens the water admission so that no superheated steam can pass out of the De-superheater. The second pressure-operated valve is under the control of the 'Drayton All-Metal Regulator' in the outflow from the De-superheater. The 'Drayton All-Metal Automatic Temperature Regulator' is of the relay type and operates as follows: Water or air at a pressure of 15–20 lbs. per sq. in. is brought to the thermostat head through a needle valve and strainer. This relay medium has free passage through the head to the pressure chamber of the controlled valve, where it compresses a metal bellows and holds the valve open. With increasing temperature the metal tube of which the thermostat consists expands, and at a predetermined point this expansion is caused to open a small relief valve to the passage through which the relay medium passes. The opening of the relief valve causes a dissipation of the pressure in the pressure chamber of the controlled valve, which moves towards a close position. By this means the amount of water passing to the spray heads is controlled by the temperature of the de-superheated steam passing into the service mains, and adjustments necessitated by fluctuation in load are made immediately.

By means of a simple device in the regulator head, it is possible to set the thermostat to operate at any desired temperature. When these fluctuations are very rapid and when the de-superheated steam is to be kept within fine limits, the 'Drayton All-Metal Regulator' is replaced by the 'Drayton Steady.' This Regulator operates on the relay principle and is fitted with a self-check action which positively prevents hunting.

An alarm signal is provided in case the temperature rises too high through failure of the water supply.

Every De-superheater is equipped with all necessary accessories such as safety valve, pressure gauge, dial thermometer, water level gauge, and steam trap of latest and most suitable type.

The 'Crosby' De-superheater.

This arrangement is designed to automatically reduce the pressure and temperature of high superheated steam, by spraying the steam with water after it has passed through a reducing valve.

The use of this de-superheater enables steam which has been generated in modern plants to be used in connection with the older prime movers, process plant, auxiliary engines, etc.

For reducing the pressure, the 'Crosby Patented Relay Operated Single-Seated Balanced Valve "A"' is used, which is fitted between the high-pressure steam main and de-superheater, and this will deal successfully with any quantity of steam from the minimum to the maximum, without hunting; the device has a cast steel body, with special heat-resisting nickel alloy valve and seat, and furnished with a balancing piston and gradual opening 'V' ports. The valve is operated by a hydraulic relay by means of a chain led over suitable guide pulleys and connected to the lever of the balanced valve. The relay consists of an hydraulic cylinder mounted upon a diaphragm chamber, which is in communication with the low-pressure steam main. The load on the diaphragm is balanced by means of a weighted steel yard, which is connected by a bridle to a small 'Pilot Valve,' and any movement of the diaphragm, due to a slight variation above or below the predetermined pressure in the controlled or low-pressure main, causes this pilot valve to admit water to the top or bottom of the piston, as the case may be; the movement of the piston operates the balanced valve in the steam main, controlling the reduced pressure within very close limits, irrespective of the volume of steam passing or variations in the high-pressure main. The low-pressure steam now passes by means of a connecting pipe to the inlet of the de-superheater chamber, this piping and the de-superheater being protected from an excess pressure by Crosby Pop Safety Valves having steel bodies and nickel alloy parts, the number depending upon the pressure and size of the main.

For temperature reduction, the de-superheating chamber consists of a mild steel vessel with all seams electrically welded and fitted circumferentially with spray pipes and nozzles which are generally arranged in four rows, each row of sprays being staggered and connected by means of a separate ring pipe, which is connected to cam-operated water supply valves. A stop valve is provided for each ring of sprays for isolation purposes. The chamber is fitted with a man-hole of the McNeil type, and provision is made for connection to a steam trap for draining any excess of water.

The control valve is so arranged that the water supply to the rows of sprays is automatically admitted in consecutive order by means of cam-operated valves. No. 1 valve, which is connected to the bottom row of sprays, opens first and No. 4, which is connected to the top row, last. The water supply is shut off in the reverse order, No. 4 which is connected to the top row of sprays first, and No. 1 last, so that it is impossible for high-temperature steam to escape through the slots in the internal pipe without first passing through a film or cloud of water spray, which automatically reduces its temperature. The low-temperature or de-superheated steam now passes through slots to the outlet of the de-superheating chamber, impinging on its way upon the expansion member of the thermostatic control valve. This element operates a valve which controls the flow of water through a small nozzle, thus varying the pressure in the connecting pipe between the thermostatic control valve and the diaphragm chamber of the special hydraulic relay, and on the diaphragm itself, so that the pilot valve is operated, causing the hydraulic piston to move the cam-operated valves, thus throttling or increasing the water supply to the sprays. In brief, any slight increase in the predetermined figure will cause the hydraulic relay to admit more water to the sprays, or if the temperature falls slightly, the water supply to sprays is throttled, thus keeping the temperature within very close limits.

The water supply for the sprays, valves and hydraulic relay must be at a pressure of 50 lbs. in excess of the low-pressure de-superheated steam. This may be taken from the feed main through a strainer and reduced by means of a valve down to a suitable pressure, or, in large plants, a separate pump is sometimes installed which is in turn controlled by a Crosby Pump Control. A relief valve is provided between the hydraulic reducing valve and the cam valves to protect same against any excess pressure, and a back pressure valve is fitted to prevent steam passing into the hydraulic main should the water supply be cut off for any reason.

SCALE IN BOILERS.

A coating of scale on boiler plates somewhat increases the resistance to heat transmission from flue gases to water. The resistance on the water side of plates, however, is only a small part of the total resistance, and even if it be multiplied several times by the presence of scale, the effect on the total resistance is small, and the efficiency of the boiler is only slightly reduced.

It is not its effect upon efficiency which renders scale objectionable, but its effect in altering the distribution of the temperature gradients from gas to water through the metal and the scale. It leads to an increase in the metal temperature which may reach a dangerous amount.

The oft-repeated statement that $\frac{1}{10}$ in. of scale requires the expenditure of 16 per cent. more fuel, and that greater thicknesses require enormously increased fuel expenditure, is quite wrong.

In regard to the increase in temperature of the plates as scale accumulates, all authorities are in agreement, and practical experience confirms the fact. Stromeyer states that with clean plates the plate temperature is only 400° F., with a water temperature of 380°, when the usual quantities of heat per sq. ft. of heating surfaces are being transmitted. With $\frac{1}{8}$ in. of scale he states that the temperature of the plate may go up to 700° F., which, of course, may result in strains being set up which are highly dangerous. Especially is this so if deposits of oil in addition to scale are allowed to accumulate on the water side of the tube, and even a very thin coating of grease may result in the boiler furnace being brought down.

Actual experience with Lancashire boilers in industrial concerns confirms the statement made by Stromeyer that, taking the boiler and economiser as a whole, the drop in efficiency will not be more than 5 per cent. with an average accumulation of $\frac{1}{8}$ in. thickness of scale. It will be found also that as the boiler becomes more scaled the outlet water from the economiser will increase in temperature, showing that this portion of the plant is absorbing the heat rejected by the unclean boiler surfaces.

Eberle's investigations (*Z.d.V.D.I.*, October 27, 1928) show that thin deposits sometimes cause over-heating while in other cases thicker deposits do not. The danger from scale depends on its density as well as its composition. Silicate scales are relatively porous, sulphate scales dense, and carbonate scales are intermediate. Some silicate scales have 20 to 30 times the thermal resistance of dense carbonate or sulphate scales.

Scale Formation.

Some waters, when heated in a boiler, form a soft, non-adherent sludge, which is easily removed by 'blowing down' the boiler, *i.e.* by discharging the sludge from an outlet in the base of the boiler. Other waters form a hard, adherent deposit known as *scale*. It is the latter which is so objectionable and even dangerous, and its prevention is the chief object of most methods and appliances used for treating water for use in boilers.

Scale (as distinct from sludge) appears to be largely a crystalline product, and if crystallisation can be prevented it is seldom that a hard scale will be produced. Consequently, by precipitating the substances which cause hardness in a liquid containing colloidal matter in suspension, the formation of hard scale may be minimised and possibly prevented completely (see below).

Prevention and Removal of Scale.

Scale is *prevented* from forming (1) by allowing only soft water to enter the boiler, though a water with 3° of hardness will seldom do harm, and is usually regarded as soft; (2) by passing a current of electricity from an iron bar immersed in the water to the boiler-plate. The hydrogen evolved at the latter prevents the scale from adhering, and forms a soft sludge; (3) by painting the interior of the boiler with a viscous adhesive to which the scale will not adhere (this method is regarded as dangerous); and (4) by adding a colloidal material to the water so as to coat the particles as they are precipitated; thus they form a soft sludge which will not form a hard scale.

Scale is removed (*a*) by chipping with a hammer and chisel, or a mechanically-operated chisel or cutter; (*b*) by grinding with a mechanically-operated carborundum wheel; (*c*) by adding caustic soda or other chemical (such as 'Boilerine,' p. 1626) to the feed-water, care being taken not to use any substance in quantities sufficient to damage the boiler. The addition of acids is highly objectionable, and so are substances of an oily or greasy nature, which would adhere to the boiler-plates; (*d*) electrolysis produced by passing a direct electric current from a series of iron bars immersed in the water to the boiler or other vessel containing it. When the current is properly regulated, small bubbles of hydrogen leave the surface of the boiler and loosen the scale, which then forms a soft sludge, easily removable by 'blowing' or 'washing.'

Sludge or *soft scale* may be removed by 'blowing down,' *i.e.* by admitting it and a portion of the water in the lower part of the boiler into the drains.

If hard water is fed into the boiler, the latter should be 'blown down' daily, and it should be cleaned and the scale removed at suitable intervals. Some engineers make a practice of cleaning out the boilers at the following intervals:

Vertical or Loco-type, after 150–300 hours use.
Water-tube and Marine dry-back boilers, after 500 hours use.
Two bottom rows of tubes in Babcock boilers, after 300 hours use.
Lancashire, Galway, and Cornish boilers, after 1,000 hours use.

These figures are not reliable guides because the thickness to which scale may be allowed to form before being cleaned off depends on the rate of heat transmission. At higher rates cleaning must be more frequent for two reasons:

(1) Because with a given thickness of scale the temperature of the metal is higher, and hence safety is reduced;

(2) Because the scale forms more rapidly. Some engineers favour low rates of transmission in order to save the expense of frequent cleaning.

Scale in a boiler may be prevented from forming by three methods:

(1) By softening the water so as to remove the scale-forming constituents.

(2) By adding to the contents of the boiler some substance of a colloidal or mucillageanous nature which will coat the particles of scale-forming material as they are formed, and so will prevent them from forming a hard scale. Instead they will form a sludge or loosely adherent scale (see Boiler Compounds, on. p. 1626).

(3) By passing a current of electricity through the water in such a manner that the evolution of hydrogen at the boiler surface prevents the adhesion of the scale-forming particles. This method is seldom used.

Corrosion.

The corrosion of boilers may be either internal or external, and as the causes are not identical, they are best considered separately.

Internal Corrosion in boilers and pipes may be due to (*a*) acid in the water, especially if of peaty or moorland origin; (*b*) acid formed by the decomposition of salts present when the water is heated. (Magnesium chloride in boiling water tends to form magnesia and free hydrochloric acid); (*c*) oxygen dissolved in the water; and (*d*) electrolysis, *i.e.* chemical changes brought about by a current of electricity.

It may be prevented by:

(1) Adding an alkali so as to neutralise the acid; chalk, milk of lime or sodium carbonate (soda) being generally used.

(2) De-aerating or de-gassing the feed-water (see p. 1639) so as to remove any soluble oxygen, which is liable to cause corrosion.

(3) Inserting plates or bars made of zinc, and connecting them metalically with the boiler plates. This sets up electrolysis, the zinc being corroded or 'wasted' instead of the boiler. The zinc should be so placed that no part of the boiler-plate is more than 6 ft. distant from a piece of zinc in direct metallic contact with it.

(4) Counter-electrolysis, *i.e.* by passing a direct electric current from anode bars in the boilers to the heating surface. A small current is ample (1 ampere per 300 sq. ft. of boiler surface being ample, and often excessive). This method is superior to (3) in so far as the current is controllable.

(5) Coating the surface of the boiler-tubes or drums with a substance which will protect it. Usually the coating is effected automatically, by adding the protecting agent to the water. With a substance such as 'Apexior' (*J. Dampney & Co., Ltd., Newcastle-on-Tyne*), one application a year is usually sufficient.

The Cumberland Electrolytic Process for Preventing Scale and Corrosion.

In the process an electric current is passed from anodes placed in the boilers to the heating surfaces.

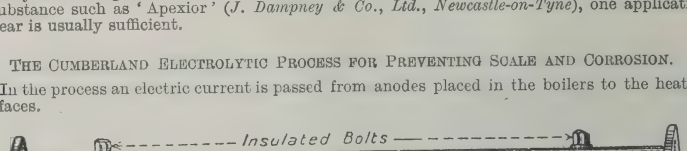


Fig. 59.

This process, whilst primarily intended for the prevention of corrosion, when applied to boilers not only arrests corrosion, but, owing to the mechanical action of the hydrogen given off from the negative surface, will loosen scale and precipitate it in the form of sludge, which is easily washed or blown out.

Where only the question of corrosion has to be dealt with in boilers, it is found that considerably less current is required than 1 amp. per 300 sq. ft.

Fig. 59 shows the type of anode as fitted to economiser tubes. In economisers of the Green, Lowcock, or Clay Cross type it is necessary to replace the usual 4-in. pipes with others of 8 ins. diameter, otherwise the process cannot be installed.

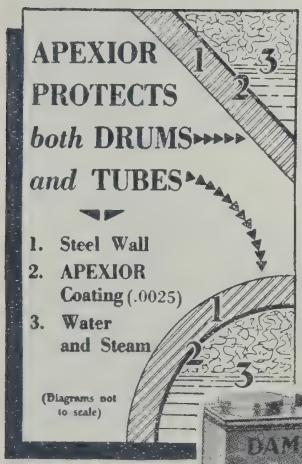
The External Corrosion of boilers is usually due to:

(1) Contact with *ashes*, particularly if they are hot.

(2) Contact with acid substances, including wet *coal* which contains pyrites; the latter may oxidise and form sulphuric acid. This is particularly liable to happen with vertical boilers on locomotive cranes, the space left between the boiler and the rear of the coal-box being easily filled with spilled coal, which soon corrodes the boiler-shell. It may be avoided by covering the space and preventing the coal from coming into contact with the hot boiler-shell.

(3) Electrolysis in the form of leaking currents from electric light or power cables. This is a rare cause, but it does occur occasionally.

APEXIOR



a dividing wall protecting your boilers from corrosion

IT IS impossible for boiler tubes and drums to corrode when they are protected from all contact with air, steam, and water.

Such protection is afforded by APEXIOR. It is a protective coating, hard, continuous, inert, which places a dividing wall between all internal boiler surfaces and the water and steam which flow through them.

It can be quickly and easily applied to the water and steam side of tubes and drums when they are laid off. A single application gives protection for approximately one year, and greatly reduces maintenance costs and losses from pitting and corrosion.

APEXIOR checks pitting and corrosion in old boilers—prevents them from developing in new ones.

FREE on Request 10th Edition Booklet—"Power and More Power." (Land or Marine Edition). Nat Physical Lab. Report on Heat Transmission

Dept. E.Y.B.

J. DAMPNEY & COMPANY LIMITED.

PROTECTIVE COATINGS APEXIOR REGISTERED FOR METAL SURFACES

BRITANNIC WORKS, NEWCASTLE-UPON-TYNE

THE IRONMONGER

HARDWARE AND METAL TRADES ADVERTISER.

ESTABLISHED 1859.

PUBLISHED WEEKLY

28 ESSEX STREET, STRAND, LONDON, W.C.2.

For over seventy years The Ironmonger has been the leading newspaper published in the interests of buyers and users of hardware, metals and mill-furnishings.

To the factory owner and his buyer it is of equal interest, as in addition to voluminous information as to prices, etc., it publishes regularly reliable and informative articles on such subjects as :—

Losses in Belt Driving
Cutting Tools
Bolts and Rivets
Drills and Drill Grinding
Electric Furnaces
Heat Transmission
Boiler-House Efficiency and Economy

All this information is obtainable for the cost of a year's subscription, £1, which sum includes a copy of The Ironmonger Diary and Hardware Buyers Guide, published each December.

To anyone interested a specimen copy of The Ironmonger will be sent on application accompanied by trade card.

Treatment of Feed-Water.

The water supplied to boilers may be either hard or soft, but as the former produces an objectionable and even dangerous *scale* its direct admission to the boiler should be avoided. If hard water is suitably treated, it can be softened sufficiently to be rendered harmless.

In addition to the substances which make a water hard, soluble salts or vegetable acids may be present in objectionable proportions.

The treatment to which a feed-water should be subjected, therefore, depends on the substances present in the water, and the proportions of these are best ascertained by a chemical analysis. It is seldom sufficient to make one or two isolated tests; a complete analysis by a competent chemist is desirable. Partial tests may be used later to ascertain whether the water has changed to a serious extent.

Rain-water, *Moorland-water* and *Artesian-water* from sand rocks are usually free from salts, but may be faintly acid. In the latter event, a suitable amount of sodium carbonate (soda) or lime should be added, to ensure neutrality or slight alkalinity.

Condensate water, obtained by condensing steam from the boilers, is usually quite free from any objectionable constituents, and needs no treatment. It is, however, important to ensure that no oil is present in it and can enter the boiler. If there is any risk of oil, a suitable oil-separating plant should be installed.

Hard water must be treated according to the nature and extent of the hardness. As explained on p. 1422, temporary hardness is best removed by the addition of a suitable quantity of lime, caustic soda, or baryta to the water, or by the Permutit process (p. 1627). Permanent hardness, on the contrary, is best destroyed by the addition of sodium carbonate (soda), though if caustic soda or baryta is used sodium carbonate may be unnecessary. Permanent hardness is also destroyed by the Permutit process (p. 1627). A chemical analysis will show which of these chemicals should be used and the amount of each which should be added.

So far as the cost of chemicals alone is concerned, lime and soda are the cheapest softening agents, but when the cost of preparing the milk of lime and of cleaning the softening plant are taken into account, caustic soda, with some waters, is cheaper than lime. Hence, the cost can only be estimated when a chemical analysis of the water, and a knowledge of the type of plant used, or proposed to be used, are available.

Highly saline waters or those made saline by the addition of softening agents, cannot be treated by any commercial means to remove the salts. The best procedure with such waters is: (1) Make as much use as possible of condensed steam and rain water, so as to reduce the amount of saline water to a minimum. (2) Blow out a proportion of the water in the boiler each day or several times a day, so as to keep the concentration of the salts as low as possible. (3) Regulate the addition of softening agents so accurately (*e.g.* by frequent tests of the water) that no serious excess of such agents enters the boiler. Simple tests for this purpose can easily be applied, and they save wastage of chemicals and premature destruction of the boiler.

External Treatment.

The most satisfactory treatment of any feed-water for boilers is effected outside the boiler in a separate 'water-softening' or 'treating' plant. By this means, the matter rendered insoluble by the treatment can be kept out of the boiler, and much inconvenience and expense can thereby be avoided.

Plants are now available for adding the necessary chemicals automatically, the only attention required being to see that the plant is working properly, and to supply it with the necessary chemicals. Several plants for this purpose are described on pp. 1627–1638.

The insoluble matter produced by the treatment may be separated by allowing it to 'settle out' or by filtration or by a combination of both these methods. The filters may be packed with sponge, wood-wool, cloth, sand, crushed rock, coke, sawdust, kieselguhr, or a special type of filter, devised by Hele-Shaw, in which the liquid is forced between parallel thin sheets may be used.

It is not necessary to allow only clean water to enter the boiler, but as much of the insoluble matter as is reasonably possible should be separated in the treating or softening plant.

Internal Treatment.

Many waters may be softened by heat alone, so that with care a boiler in which only temporarily hard water (*e.g.* that from a limestone well) is used, may be worked without producing any serious amount of scale. There is always a risk, however, that if any part of such a boiler becomes overheated, the formation of a hard scale may be serious, so that softening is desirable.

Many waters may be softened by adding lime or caustic soda and sodium carbonate (soda), to the contents of the boiler, but this is objectionable as the amount of thick slurry produced interferes with the working of the boiler.

Instead of the use of lime or soda, other substances may be added to the boiler. These are known as *boiler compounds*, and the composition of most of them is regarded as a 'trade secret.' They are sold in two forms: as a powder and as a fluid.

Some boiler compounds are definite chemical substances, such as caustic soda, sodium carbonate, aluminium sulphate, alumino-ferric, sodium oxalate, sodium tannate, and graphite;

these are all sold in the powdered form. Other boiler compounds are more complex, and may be made definitely for the purpose, whilst still others are waste materials of a starchy nature, which have been concentrated to a thick, viscous fluid, which may, or may not, contain soda or some other softening agent. A few of these boiler compounds are definitely useless as softening agents, and any success they appear to obtain is not due to their softening of the water (as they have no power to do this), but to their preventing the formation of a hard scale.

Graphite appears to have a wholly mechanical action (see below), the precise cause of which is not understood. It may be used in addition to other boiler compounds, as its action is different from them, and is unaffected by them.

The essential constituents of the best boiler compounds are :

(*a*) One or more softening agents which will precipitate the scale-forming materials before they can form a scale, and will convert them into harmless insoluble sludge, with no adherent properties.

(*b*) A colloidal or mucillageanous material (*e.g.* peat bark extracts, extracts from spent tan, and sulphite liquor from paper mills), which will surround the scale-forming particles as soon as they are formed, and will prevent them from adhering to each other or to the boiler, except as a soft pasty mass which forms a sludge and not a hard scale.

It is not really essential to have both these kinds of constituents in the same compound, though it is by no means unusual.

Boiler compounds sold in the form of powder usually consist almost wholly of softening agents of the (*a*) type, but liquid boiler compounds are chiefly of the (*b*) type, though they may contain some softening material.

In some instances, where skilled supervision is not available, a scale-preventor of the (*b*) type with no softening power may be less objectionable than the excessive addition of soda to a boiler-water, because, with soda there may be so great a concentration of soluble salts as to cause priming and other troubles, as well as raising the boiling point of the water unnecessarily.

Where adequate supervision is available, the best method of preventing scale is to soften the water and remove all insoluble matter from it before it enters the boiler.

Boilerine.

(*Boilerine Ltd.*, 885A-897 *Old Kent Road, London, S.E.* 15.)

This is a viscous fluid, consisting chiefly of a carbon compound which has no action upon iron, steel, or other metals, but which has the property of rendering all water impurities innocuous and incapable of forming scale, while it also works its way through the interstices of the scale, the rate of this percolation depending on the porosity and degree of compactness of the same, and when the preparation reaches the underlying plate of the boiler it there diffuses and insinuates itself behind the incrustation, causing it to crumble and flake, so that when the boiler is opened it can readily be removed. The preparation as a preventive of scale differs from the old style of boiler composition, inasmuch as it absorbs the gaseous carbonic anhydride from the water, forming an intermediate salt which throws the scale-forming salts out of solution, suspending them in the water in a non-crystallisable and powdery state. Incrustation is thus prevented. Steam generated from sea water containing 'Boilerine' is free from ammonia. Boilerine is not only a preventive, but also a disuniter of incrustation in boilers; it is absolutely non-volatile, and does not affect the purity of the steam produced from water treated with it; it does not discolour the water in the boiler, and it will not cause priming. It is guaranteed free from arsenic, acids, and poisonous substances, and is a neutraliser of grease and acids.

The boilers should be blown down and treated with Boilerine *daily*, as by this system greater efficiency is obtained.

'Foliac' Boiler Graphite.

(*Graphite Products Ltd.*)

The use of this material enables scale in a boiler to be easily and safely removed, a wire brush being used for the purpose. The graphite does not attack the metal of the boiler, nor, it is claimed, does it cause the metal to be attacked; it is not affected by any acid in the water or by the heat generated in the boiler. Particles of the graphite work through the minute fissures existing in the old scale and gradually penetrate between the scale and the metal. The scale thus loosened may be rapped off or removed with regular cleaning tools with little trouble. If the scale is very hard and thick it may require as long as three or four months for graphite to loosen it, but once removed the scale can never again adhere firmly to the metal as long as the graphite treatment is continued. The graphite also becomes thoroughly intermixed with new scale as it forms, rendering it soft and friable.

The graphite is introduced in the form of fine powder with the boiler feed, about half a pound being used per day for a 250-h.p. boiler.

Precautions with Treated Water.

It is not necessary to soften feed-water for boilers to zero hardness, as a hardness of 3–4° (corresponding to 3 grains of calcium carbonate per gallon of water) is not usually objectionable. With water containing magnesium chloride, however, a sufficient excess of soda or lime should be present to prevent the evolution of hydrochloric acid derived from magnesium chloride.

Where zero hardness is regarded as important, it is usually cheapest to remove most of the hardness by lime and soda, and the remainder (not more than about 5 grains per gallon) by means of a zeolite such as 'Permutit.'

It is very important not to permit an excessive concentration of soluble salts in the water in the boiler, as this may cause local overheating, which may have serious consequences.

Highly saline water also tends to produce priming, and it wastes fuel by having an unduly high boiling point.

An excess of caustic soda in a boiler may cause corrosion of the brass fittings and should be avoided.

Embrittlement of Boilers.

Failures of boiler plates due to 'embrittlement' have recently been investigated, and it is stated that the evidence points to concentration of caustic soda or of carbonate of soda in minute cracks on the surface of the plates in the vicinity of rivets as being the source of the trouble. It is the association of the soda compound and stress that appears to lead to a dangerous condition.

The degree of concentration which may lead to 'embrittlement' is not yet established, and, indeed, whether the presence of the soda is the true explanation of embrittlement at all is controversial. In view of the possible risk, however, it is advisable that when caustic soda has been used to assist in the removal of scale or for the prevention of corrosion when a boiler is out of service, it should be thoroughly washed out before the boiler is again put to work; also the concentration of caustic soda or of carbonate of soda due to softening reagents should not be allowed to become sufficient to give pronounced alkalinity.

Water Softening Plant.

Water Softening by Means of Zeolitic Exchange Process.

('Permutit' Process—United Water Softeners, Ltd.)

The Permutit system comprises filtration through a bed of the granular base-exchange material termed 'Permutit,' which has the property of giving up its sodium base to any water which is passed through it in exchange for any calcium or magnesium contained in the water.

This action takes place irrespective of whether the calcium or magnesium in the water is in the form of carbonate or sulphate, *i.e.* quite independent of whether the hardness is temporary or permanent.

There are two varieties of Permutit, the synthetic or artificial (Permutit A) and the treated and standardised natural product (Permutit B). Both are rapid in softening and regeneration, but Permutit B is more rapid in its action in either direction than any other base exchange material, either natural or artificial.

Such a method affords a means of presenting to the hard water a large supply of the softening element in readiness to be drawn upon by the water itself in exact proportion to its quantity and hardness; in other words, Permutit contains, in large excess, the active softening element, which, however, is effective in precisely the degree required to completely effect the softening of the water, and no more.

This action continues for so long as there remains in the Permutit active material (sodium) available for exchange; the total softening capacity of Permutit being represented by the multiple of the hardness and the quantity of any water, so soon as this quantity of water of the specified hardness has passed through the bed of Permutit, the sodium will become replaced by calcium or magnesium (or both), extracted from the water; thereafter, no further effect takes place either in respect of the Permutit, or of any further water passed through it. A fresh charge of the sodium is then transferred to the Permutit by flowing through it a weak solution of common salt (NaCl), when a similar exchange action takes place.

In this stage revivifying, or in its technical term 'regeneration,' takes place, by an exactly similar process. The bases of the hard salts (calcium or magnesium) are replaced in the Permutit by the sodium contained in the salt solution; and the calcium and magnesium combine with the chlorine of the salt forming the chlorides of these bases, which as a clear liquor may be run into any drain. Any slight excess of salt solution remaining amongst the granular particles of the Permutit after complete regeneration has been effected is washed out by a short rinse with clean water.

Permutit A can be regenerated in about one hour, while Permutit B is capable of being completely regenerated in 15 minutes. This rapidity of regeneration makes it practicable to use one unit practically continuously throughout the twenty-four hours.

A complete cycle of operations has then been completed, when hard water can again be passed through the filter, softening being complete as before, and continuing until the maximum quantity for which the filter is designed has again been treated.

During this to-and-fro softening action, the Permutit itself suffers no damage or wastage, it being merely a vehicle for the zeolytic exchange action which is the peculiar property of the substance. There is, therefore, afforded an extremely simple, though entirely scientific means for completely softening water of any hardness without any deposit of solid matter; any necessity for settlement and subsequent sludging away of precipitates, as was the case in the previous lime and soda systems, is thus completely obviated.

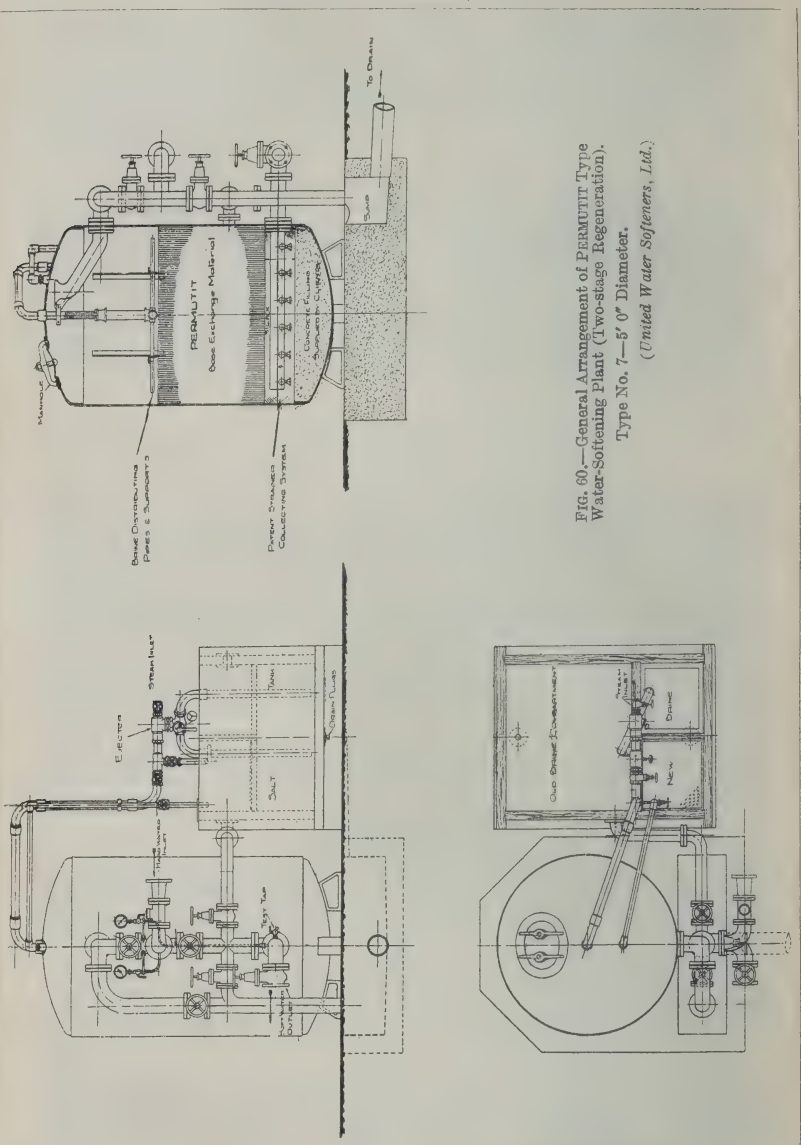


FIG. 60.—General Arrangement of PERMUTIT Type Water-Softening Plant (Two-stage Regeneration).
Type No. 7—5' 0" Diameter.
(*United Water Softeners, Ltd.*)

Another important feature attaching to the 'Permutit' system is the valuable means which it affords for treating water as it passes through the mains under pressure; and it has thereby become possible to treat any supply for industrial or domestic purposes as it enters the building in which it will be used, and to deliver the same in a completely softened condition to storage tanks in the top of the building, without any break in the pressure or consequent necessity for repumping.

Absolute uniformity in the final water, as well as elimination of any necessity for skilled control of the treatment by a chemist, are also features of importance attaching to the Permutit system.

In cases where temporary hardness preponderates heavily, and the difference in cost of lim and salt is also in favour of the lime treatment, a Permutit filter may with advantage be arranged to supplement a preliminary lime treatment.

Elimination of Iron and Manganese from Water Supplies.

By means of a special type of Permutit, prepared and oxidised by the use of manganese salts and permanganate, all traces of iron and manganese can be eradicated from a water supply, the treatment being effected under pressure and without calling for any extra pumping.

Large installations for some of the leading public water supplies throughout Europe have been laid down with conspicuous success for this purpose, as well as for the softening of the water as above described

The Permutit system constitutes practically the only one of certainty and success for the removal of manganese.

General Design of Permutit Plant.

The type of plant utilised for effecting Permutit treatment of water is of a standard design and is shown by fig. 60.

A mild steel cylinder of such proportions as are requisite, accommodates the proper quantity of Permutit.

Provision is made by means of suitable pipe connections and valves for agitating the Permutit bed by means of a flow in the opposite direction to that of filtration. By this means any uneven action due to air pockets or channels in the bed of the Permutit is prevented.

In the case of single units, the filter carries above it a shallow tank for dissolving the salt for regeneration—which is usually arranged to be effected once in each day.

In the case, however, of multiple-unit plants, these overhead salt tanks are dispensed with; and a common tank in steel or ferro-concrete provided, in which the salt solution is made up for the entire battery of the filters.

A complete plant of this description is in operation for both iron removal and softening at the Hooton Pumping Station of the West Cheshire Water Company.

Plants upon similar lines have been installed by a very large number of the leading industrial concerns both in England and on the Continent, as also by the Municipalities of Breslau, Dresden, and other leading cities.

THE KENNICOTT TYPE 'K' WATER SOFTENER.

(*Kennicott Water Softener Co., Ltd., Imperial House, Kingsway, London, W.C.* 2.

The Kennicott Type 'K' Plant is so designed as to be very simple to operate, and at the same time extremely accurate. The plant is absolutely positive in its action, and there are no oscillating parts to get out of order, and no valves or cups to get made up with lime. Fig. 61 (p. 1630) shows the general construction of the plant, the action of which is as follows:—

The calculated quantity of chemicals required for a day's supply are run into the chemical tank B, which is filled to a definite level. The hard water enters the plant by way of the pipe E into the hard-water box F, whence it flows through an adjustable slot over the water wheel D and out of the chute G at the bottom of the water wheel casing into the top of the downtake H. The incoming water, passing over a water wheel, provides all the power required for operating the plant. It drives the main shaft, and this shaft drives two vertical shafts, one in the downtake and the other in the chemical tank B. The shaft in the chemical tank B has a 4-bladed agitator at the bottom, and this keeps the chemicals constantly and homogeneously mixed. The shaft in the downtake has an agitator fixed to it, and this ensures that the chemicals added are thoroughly mixed with the water, a most essential feature. In order to proportion the chemicals, a small quantity of hard water from the hard-water box F is by-passed to the dividing box I, where the small quantity of by-passed water is divided, a portion running into the regulating tank K, and the rest down the drain. The dividing box is set so that the amount of water run into the tank K fills up this tank in a definite number of hours, generally sufficient for a day's run, and as this tank fills, the float riding on the water in this tank gradually rises and at the same time lowers the weir L in the chemical tank B, the float in the tank K being connected with the weir in the tank B by means of a flexible cable passing over two pulleys. It will thus be seen that the chemicals are added in an absolutely definite proportion to the quantity of water entering the plant. Immediately the chemicals come into contact with the water, the whole is thoroughly mixed by means of the mechanical agitator in the downtake, and the water passes down to the bottom of the downtake H, and finally flows upwards in the direction of the arrows, at a very slow rate, and is finally filtered at the top of the plant.

KENNICOTT TYPE 'K' WATER SOFTENER.

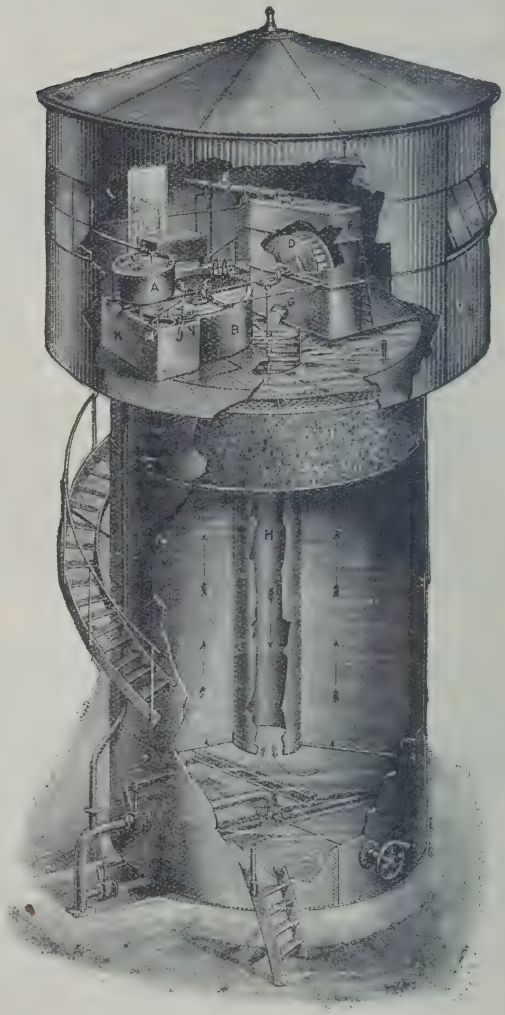


FIG. 61.

Any mud contained in the water, and the bulk of the precipitated hardening salts, settle on the bottom, and it is very necessary that this sludge should not be allowed to accumulate, otherwise the moving water will very quickly wash it up into the filter. By means of the *Kennicott Patent Revolving Sludging Gear*, however, the whole of the sludge settled on the bottom can be removed, the action of the sludging gear being very similar to a vacuum cleaner.

No Kennicott plant is made with less than three hours' settling, this being the minimum time that should be allowed for the complete precipitation of the hardening salts. With all Kennicott plants the mixing tank A is supplied. This tank is sufficiently large to hold a full day's supply of chemicals, so there is always a charge in readiness when the proportioning tank B is empty.

As the main tank always stands full of water, the clear soft treated water discharges from the overflow at the side of the main tank at a point above the filter as fast as the water to be treated enters the top of the downtake 'H.'

The machine is continuous and wholly automatic in its operation. The flow of the water furnishes all the power necessary for mixing the chemical reagents with the water and for operating the mechanism.

Every installation is designed for the work it has to perform, and to suit the site it must occupy. The chamber can be made either cylindrical or rectangular in shape. When a supply of exhaust steam is available, great economy can be obtained by installing a Kennicott Patent Type 'K' combined Heater and Softener.

The operation of this type of plant is exactly similar to the Kennicott Cold Water Softener described above. In addition to the chemical proportioning gear, there is fitted a very substantial heater, fitted with removable trays, over which the water passes. Steam is admitted at the bottom of this heater after passing through an oil separator, and passes upwards through the water. This raises it to boiling point, a considerable proportion of the hardening salts being at the same time deposited in the trays, which are from time to time removed for cleaning purposes. Practically the whole of the hardness originally present in the water is by this system removed, and at the same time a soft feed water of high temperature is available for boiler purposes.

Plants of the latest Type 'K' design of all sizes from 500 to 150,000 gallons per hour capacity are actually in operation.

The Kennicott Water Softener Co. also manufacture a Zeolite Water Softener which will reduce water, irrespective of its initial hardness, to zero degrees. This softener works under pressure, and the zeolite employed is a natural mineral which has been used for over ten years without showing any appreciable sign of deterioration.

The Kennicott Water Softener Co. claim that this zeolite is the most efficient and most durable zeolite procurable.

The Royles Automatic Water Softener and Purifier.

(*Reisert System.*)

In this apparatus, fig. 62, p. 1632 (*Royles, Ltd., Irlam, near Manchester*), the bicarbonate of lime is precipitated by hydrate of lime (lime water), and the sulphate of lime by soda. The softened water is then filtered and discharged.

The apparatus, which acts entirely automatically, consists essentially of an automatic lime saturator and decanter B, a soda chamber D, a water-distributing tank E with a lime-slaking division A and soda-mixing division C, a reaction chamber F, and an automatic self-cleansing filter K.

The *Lime Saturator* B requires no stirring contrivance, and lixiviates the lime completely; and owing to its upward widening shape, it turns the saturated lime water into a clear fluid.

By a constantly uniform water supply through the micrometer valve 3 from the distributing tank E, which is conducted downwards through the central pipe O, the lime paste at the bottom is stirred up and thoroughly impregnated, and rises, at first partly with the water, until the rapidity of the rising water diminishes so much, owing to the upward widening shape of the saturator, that the undissolved lime particles are no longer able to follow, so that the saturated lime-water, clarified, then leaves the saturator B at the top, and is carried into the reaction chamber F by way of the mixing pipe E^1.

As lime possesses the property of dissolving in a definite proportion in water until the latter is saturated, it is possible to continually keep a uniform stream of saturated lime-water supplied as required, for the precipitation of bicarbonates of lime and magnesia. Upon this uniform addition of lime depends the degree of purification of the water.

In the *Soda Apparatus* D, a quantity of soda (which, unlike lime, is very soluble), one gallon of water at 60° F. dissolving up to 1¼ lb. of dry sodium carbonate that will suffice for one or more days, is dissolved in the chamber C and let down into D through cock Q.

The action of the soda apparatus is based on the fact of the soda solution having a greater specific gravity than water, and thus the water flowing from the distributing tank E through the small micrometer valve 1—which is adjusted in accordance with the amount of soda required—into the soda chamber D, remains always on the surface of the soda solution (no mixing taking place), and displaces the same, it being carried through the small pipe from the bottom upwards, and into the mixing pipe E^1, and finally into the reaction chamber F.

The water-distributing tank, E, is supplied with the hard water to be treated, by pipe and cock as shown; it is also provided with an overflow pipe.

It is further provided with three micrometer valves at an equal height, viz.—(1) for the supply to soda tank; (2) for the supply of untreated water to the reaction chamber; and (3) for the supply to the lime saturator B. By arranging these three valves at an equal height the quantities of water flowing from them are always proportionate in the event of less water being supplied to the apparatus than corresponds to the capacity; or in the event of the water supply ceasing altogether, they cease simultaneously.

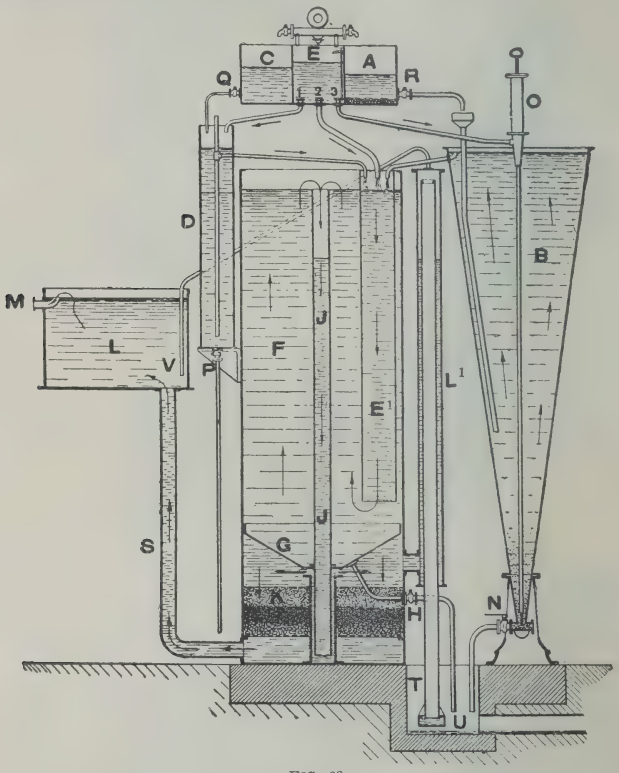


FIG. 62.

A *Lime-Slaking Division*, A, is arranged in the distributing tank. In it, once a day, the required quantity of lime is slaked by means of a supply of water obtained from the water inlet pipe. The lime paste is then conveyed in bulk to the bottom of the lime saturator, B, through the short depending pipe illustrated, by opening cock R. The *Reaction Chamber* F thus receives through the mixing pipe E¹, the untreated water, lime-water, and soda-water, which remain in this chamber until the chemical reactions have been accomplished.

The continuous inflow of the water and chemicals above described overflows *via* the pipe J into the filter chamber K, following the course of the arrows downward through the filter bed and upwards through pipe S into the tank L, the rest flowing away, at M, to reserve or for use. The height of the column of water in the pipe J will vary with the resistance of the filter bed, and as the latter fouls, the water will rise higher and higher in J.

Similarly it will rise to an equal and corresponding height in the Syphon pipe L¹, and as soon as the resistance through the filter bed reaches such a point as to cause the water to over-

flow this pipe, it forms a syphon and instantly reverses the current through the filter bed, drawing backwards the clear water held in the small tank L, thus thoroughly and automatically cleansing the filter, the sediment being discharged at T into the gully U. The small tank L is proportioned to hold sufficient water to thoroughly flush the filter K, and as soon as the tank is empty air enters *via* the pipe V, and destroys the syphonic action.

The filter then resumes its normal working, and so continues until the fouling is again such as to cause the pipe L to overflow, when the cleansing action is repeated. The loss of water in reflushing is very trivial. The filtering material rarely requires renewal.

The Lassen-Hjort Water Softening and Purifying Plant.

(United Water Softeners Ltd.)

The outstanding feature of the Lassen-Hjort patent automatic water softener, figs. 63 and 64 (p. 1634), is the device it embodies for the separate and exact measurement, in small, absolutely definite, and predetermined quantities, of both hard water and softening reagents. This device

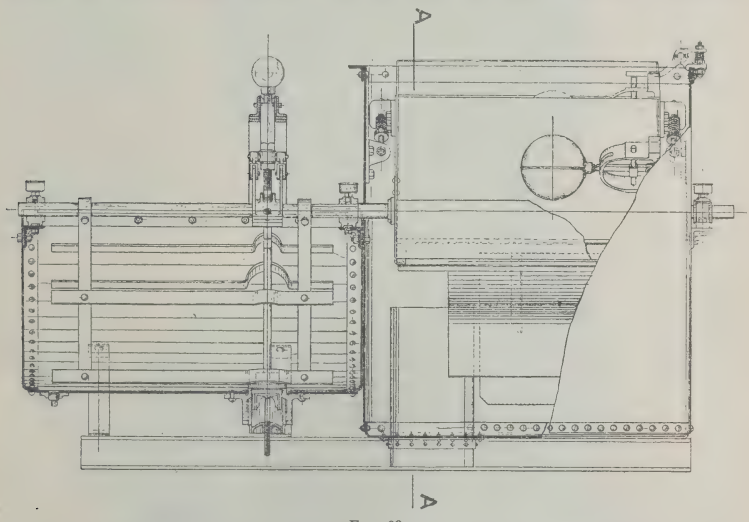


Fig. 63.

(termed the Lassen-Hjort Patent Measuring Apparatus) renders it possible to treat water or effluents of any character with any reagent or combination of reagents, in rigidly proportionate quantities, which can be varied at will and adjusted with the utmost precision.

The hard water to be softened is fed into the plant by a pipe which alternately fills each of the compartments of a two-chambered tipper oscillating on a shaft carried in bearings. When one of these compartments is full of water the disturbance of equilibrium causes the tipper to over-balance, and, by doing so, to discharge its contents of water into the tank in which it is suspended: at the same time the other compartment of the tipper is brought under the orifice of the inlet pipe and filled in its turn with hard water, to be discharged in the same manner when full. A counter affixed to the tipper shaft registers the number of oscillations, so that the quantity of hard water to be treated is thereby accurately measured.

An important feature of the apparatus is the recently introduced patent locking gear, by means of which the oscillating receiver is prevented from tipping until it contains a predetermined quantity of water. This ensures absolute precision in the measurement and delivery of the water to be softened.

The apparatus, fitted with this locking gear, has been tested as a water meter pure and simple, at the National Physical Laboratory, Teddington, and certified to be correct within 1 per cent.

At each discharge of water from the tipper into the tank in which it works, a corresponding amount of water is displaced from this tank through a standpipe and shoot into the reaction chamber of the softener, and here it receives at the same moment the requisite charge of chemical

LASSEN-HJORT WATER SOFTENER.

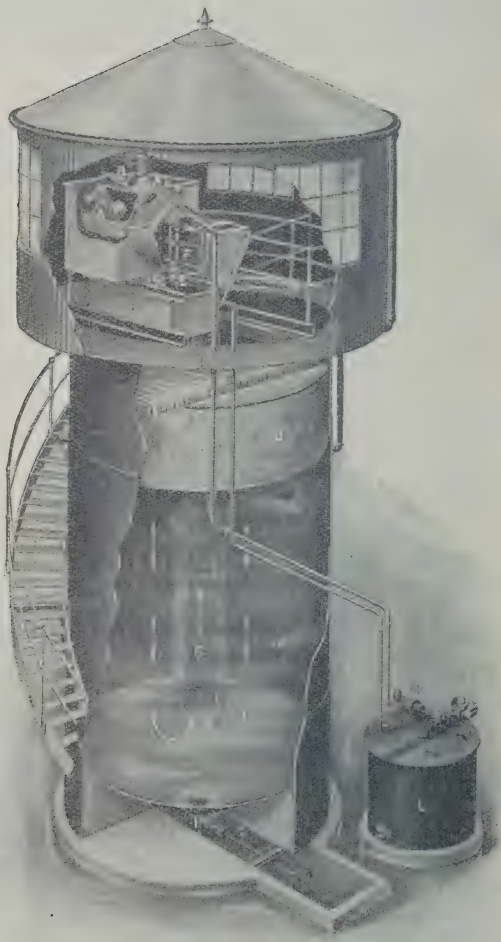


FIG. 64.

solution from the semicircular chemical container affixed to the side of the tipper tank. This is effected by means of a ' positive discharge valve ' (patented) placed in the bottom of the chemical container, which is opened at each movement of the tipper, and caused to deliver into the reaction chamber the exact amount of softening reagent (in the majority of cases a mixture of lime and soda ash) required to soften it to the guaranteed figure.

The valve can be adjusted (with great ease and over a wide range) to discharge any specified quantity of reagent required by the volume of water in the tipper compartment, so that the plant can be easily regulated while working without any necessity for altering the strength of the chemicals or changing the solution in the container. The precision with which the chemicals and water are measured and mixed makes for speedy completion of the reaction.

A special feature of the tipper gear is the fact that all the working parts are in sight, whilst the whole gear is simpler (it is claimed) than that of any other water softener plant in use.

The average results obtained with the Lassen-Hjort plants show a hardness of the water after treatment, of 2° Clark, while the residual alkalinity (i.e. excess of lime and soda remaining in the water after the conclusion of the softening process) averages only 1·5 grain per gallon, or one and a half parts in 70,000 parts by weight, i.e. practically no excess of alkalinity.

The plant being primarily designed for adding lime automatically to water in exactly measured quantities, is very suitable for carrying out the Houston ' excess lime ' process of water sterilisation.

The Lassen-Hjort process has been adopted by Messrs. Babcock & Wilcox, who supply it throughout the world under license from the makers.

Many thousands of Lassen-Hjort plants are at present in use.

See also ' Purification of Boiler-Feed Water,' No. 4, 1921, *Proceedings Inst. Mech. Engrs.*

The ' Neckar ' System of Feed-water Treatment with Continuous Boiler Water Return Device.

(*Neckar Water Softener Company*, 96 *Victoria Street*, *London*, *S.W.* 1.)

In the Neckar system the particular feature is a combination of selected preliminary treatment with control of boiler water quality.

Referring to fig. 65 the crude water is first treated in the reaction tank and then passes to the boilers softened and slightly alkaline.

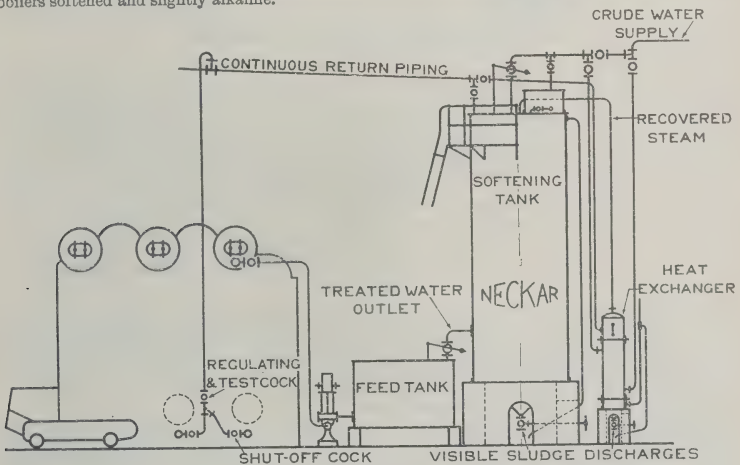


FIG. 65.

The softening plant is operated hot, with selected treatment by soda ash and such other chemicals as the crude water may render desirable.

By means of the continuous return device, the essential feature of the system, a small amount of boiler water is withdrawn, thus preventing high concentrations of any kind. This water is returned to the reaction tank so that neither water nor heat is lost.

Quite apart from water softening the system provides for control of salinity. The boiler water which must be discharged for this purpose is led to the heat exchanger where a portion is recovered by flash action as distilled water and the bulk of the heat in the remainder transferred to the feed water.

It is claimed that by thus dealing with the feed-water problem as a whole complete freedom from deposits in the boilers is ensured and also steady steam generation by avoiding high concentrations of any kind, while the usual losses through blowing-down to waste are avoided.

Filtrator Process for Prevention of Scale.

The use of 'colloidal' substances to prevent scale formation has already been referred to (p. 1625).

A recently introduced patented system for carrying out such treatment is that of Messrs. Filtrators, Ltd.

The Filtrator is an apparatus having for its object the prevention of the formation of scale in steam boilers, economisers, and condensers, and the removal of scale existing previous to its application.

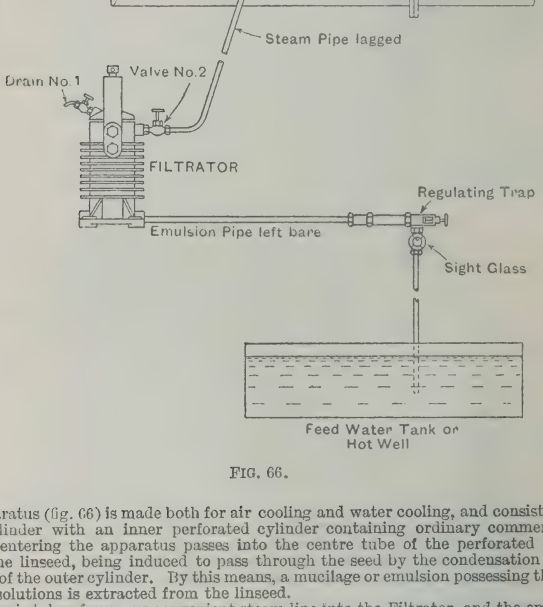


FIG. 66.

The apparatus (fig. 66) is made both for air cooling and water cooling, and consists of an outer cast-iron cylinder with an inner perforated cylinder containing ordinary commercial linseed. Live steam entering the apparatus passes into the centre tube of the perforated cylinder and permeates the linseed, being induced to pass through the seed by the condensation taking place on the walls of the outer cylinder. By this means, a mucilage or emulsion possessing the properties of colloidal solutions is extracted from the linseed.

The steam is taken from any convenient steam line into the Filtrator, and the emulsion is led through a regulating trap of such a type that a constant flow of the linseed mucilage is maintained into the feed water. This mucilage, or emulsion, when leaving the Filtrator, is very little different in its consistency from ordinary water, and freely mixes with either cold or hot water in the feed tank so that the colloidal character of the solution is automatically maintained as long as the Filtrator is in operation.

A very important point about this method of installing the Filtrator is that it permits of the economisers being treated to keep them free of scale as well as any number of boilers being served from one central installation.

There are other methods of arranging the Filtrator, such as attaching direct to the boiler, but the arrangement suggested above is recommended.

It is well known that the cause of formation of scale in steam boilers consists mainly in the precipitation of different solid matters, when the water is heated to the temperatures characteristic of each, and also from the chemical reactions which take place in the water. When these matters are precipitated they adhere to the surface of the boiler and form scale, but when the mucilage

from the linseed is introduced into the water in sufficient quantity it coats over all the scale-forming particles with a thin film, thereby preventing crystal growth (scale), and also preventing adhesion to the boiler shell. It must be obvious that if scale is not formed a sludge will be formed, and this requires periodically blowing down. The amount of blowing down does not exceed that required under ordinary working conditions.

Old scale will also be gradually removed. The emulsion penetrates the cracks which always exist on the surface of old scale; it thereby reaches the heated metal, when it expands, loosens the old scale, and this is gradually torn away by the circulating water.

The action of the mucilaginous emulsion is a physical one, no chemical reactions whatever taking place, so that this process is entirely harmless to the metal of the boiler and all fittings. By this physical method it is possible to treat any type of water, including sea water, and it is of great value where the water is of varying hardness or salinity.

Bare metal is not obtained with the Filtrator, but a protective scale coating which may reach, but will not exceed, 1-32 in. in thickness is formed in Lancashire or other fire tubular type boilers. In water tubular boilers, there is a maximum thickness of 1-64 in. of hard scale which forms a protective coating on the shell of the boiler, and over prolonged periods of steaming, scale is removed as soon as it reaches this maximum thickness. All scale which is formed is of a non-adhering character, and therefore the boiler metal surfaces are always open to water percolation.

One of the very interesting features of the treatment of the water in the boilers is that when marine boilers have been fitted with the apparatus, no corrosion or pitting has been found inside the boilers, notwithstanding that sea water only is used as 'make up.'

The lubricating oil that sometimes gets inside the boilers, owing to the impossibility of entirely eliminating it from the feed water by means of oil filters, has not been found in boilers fitted with Filtrator Apparatus. The most acceptable explanation is that the emulsion prevents the tiny globules of oil from agglomerating and forming film on the water's surface. It appears that the oil remains in this case in a finely divided state in suspension in the boiler water.

The Filtrator is applicable also to all types of condensers and water circulating systems generally.

As the husk of the seed remains intact throughout the process, and the condensed steam can extract only the matter soluble in water, no oil can possibly be extracted from the linseed.

The 'Dionic' Method of Water Testing.

(Eversheds & Vignoles, Ltd.)

This method (Digby & Bigg's patent) is based on the fact that the electrical conductivity of water varies with the degree of hardness, or impurity in solution.

The essential units of the apparatus are a glass tube containing the water under test, together with a hand-driven direct-current dynamo and a conductivity meter.

A current is passed between electrodes, which are sealed at a fixed distance apart in the tube, whilst automatic compensation for temperature is effected by an adjustable thermometer which varies the effective cross-sectional area of the liquid path.

A direct reading of the conductivity is obtained, which affords information as to :—

(*a*) Whether condensers are leaking, and the extent of the leakage.

(*b*) The presence and degree of priming, if any.

(*c*) The control of oil-eliminating and water-softening plants.

(*d*) Feed-water.

A recording instrument of this type is now available. A new Dionic instrument is described in *The Engineer*, p. 350, Sept. 26, 1930.

The Crockatt Simplex Patent Electric Salinometer.

(W. Crockatt & Sons, Ltd., Glasgow.)

This is a direct reading instrument showing the amount of salt or other impurity in water, in grains per gallon, or other convenient measurement. An instrument board (fig. 67) is fitted on the control platform, and has on it the indicating instrument, warning lamp and relay for each point to be tested, interlocking switches and a warning bell or buzzer.

The lighting supply is coupled to a master switch on the board, and leads are run to the compensator boxes (fig. 68) which contain the detector cell and compensating thermometer. One of these boxes is fitted for each condenser or feed line in which it is desired to measure the impurity.

A continuous stream of water flows through the thermometer pocket and the cell, and automatic correction is made here for varying temperature of the water.

The cells are, normally, all in circuit through the warning lamps, and the glow of any one of these indicates the presence of salt. This can immediately be measured by switching over that circuit to the indicating instrument. The switches for this purpose are all interlocked so that only one cell can be switched into circuit with the instrument at a time, the others being all in circuit through the lamps.

In the event of a large influx of salt or other impurity at any point, the corresponding relay closes, and a bell or buzzer gives audible warning to the engineer.

The Admiralty pattern instrument is entirely self-contained, the water being led directly to the

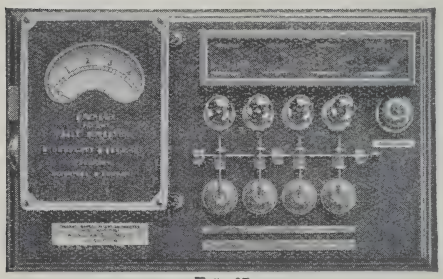


FIG. 67.

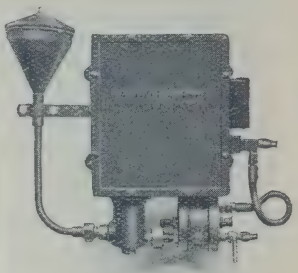


FIG. 68.

thermometer pocket and cell, which are attached to the instrument case. This pattern indicates from zero (pure distilled water) to ·5 grains of chlorine per gallon, on a scale measuring over 4 ins. long, but the calibration and the range can be made to suit the special conditions of each case.

Removal of Gases from Feed-Water.

The dissolved oxygen in feed-water may be reduced in amount in various ways. One method is to use a closed feed system, the condensate being kept from contact with the atmosphere,

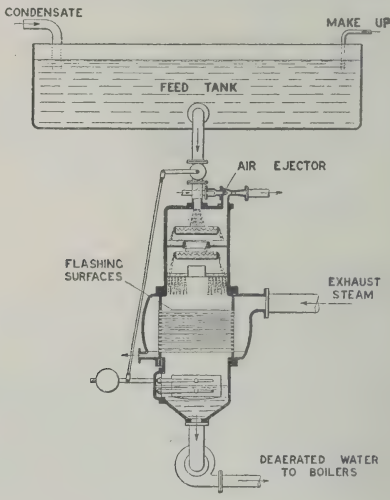


FIG. 69.

and the make-up feed-water being passed through the main condenser in which it is exposed to the vacuum and thus gives up part of its dissolved gases. Alternatively, a special de-aerating vessel may be included in the feed circuit. In the de-aerator the feed is finely divided or sprayed

by means of nozzles or by being made to fall in small streams over trays or through perforated plates. The de-aerator is either connected to the condenser or is provided with an air ejector so that the feed-water in its state of spray is exposed to a vacuum.

Fig. 69 shows one form of the 'Allen' de-aerator, in which the action of the vacuum in removing the dissolved gases is assisted by the spray falling upon steam-heated pipes placed across its path. The feed is thus heated as well as de-aerated in the same vessel.

A chemical method is exemplified by the Kestner De-gassing Plant.

THE KESTNER DE-GASSING PLANT.

(Kestner Evaporator and Engineering Co., Ltd.)

Fig. 70 shows diagrammatically one form of this plant.

The de-oxidising of the water is effected in the central portion of the vessel, which is filled with finely divided steel specially prepared both in respect to chemical composition and physical condition, so as to be readily oxidisable.

The apparatus is so proportioned that after passing through the de-gassing medium the water contains no oxygen capable of injuriously affecting the boiler, economiser, or pipe connections.

The water inlets (1) and outlets (2) are so arranged on the vessel that the flow of water may take place either upwards or downwards.

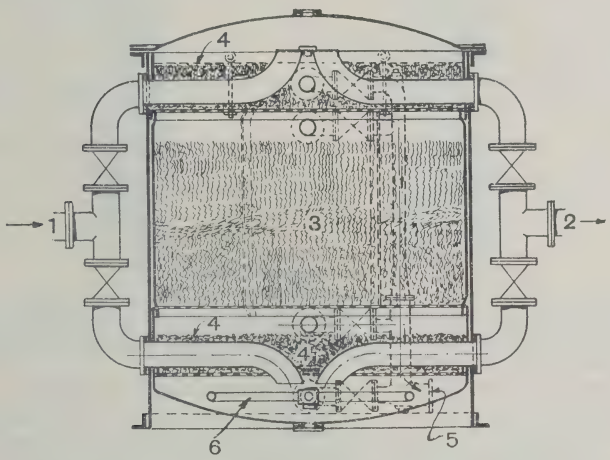


FIG. 70.

By arranging for a periodical reversal of the flow, portions of the material, which are most rapidly oxidised with the flow in one direction, are rested when the flow is reversed. During this period of rest a chemical change in the film of oxide takes place, the film being partially converted into a different form of oxide, which is easily detachable from the material.

A steam injector (6) is provided on the vessel through which steam is injected, causing sufficient turbulence to detach the oxide from the material. The flow of water carries the oxide through the filters (4), which retain the oxide, the cleansed water being returned to the feed reserve tanks. The operation of cleaning requires only a few minutes and involves no waste of water.

By changing the direction of flow every 24 hours, the de-gasser is kept in proper working condition, and will continue to function satisfactorily until the deoxidising material requires renewing.

For make-up water containing the usual proportion of dissolved oxygen, the makers state that the consumption of material amounts to about one ton for 100,000 tons of water. When the condensate is returned in a closed circuit, in which one of these de-gassers is included, a consumption as low as one ton of material for 1,000,000 tons of water may be obtained.

The Weir 'Optimum' De-aerator.

(G. & J. Weir, Ltd., Cathcart, Glasgow.)

In this de-aerator the feed water is sprayed through nozzles, meeting steam from a chamber below in which the water is collected and subjected to violent ebullition by incoming steam. The rapid boiling of the water, combined with the heating action of the vapour on the spray at reduced pressure, causes the air in the water to be liberated; the air is exhausted by means of an air ejector which maintains a vacuum in the de-aerator and assists in the de-aerating action.

The 'Neckar' De-aeration System.

(Neckar Water Softener Company, 96 Victoria Street, Westminster, S.W. 1.)

The removal of dissolved gases from the feed water is effected in the Neckar De-aerator by exposing the feed-water in the form of spray to a vacuum; the vacuum is produced and the gases extracted by an ejector, which is usually operated from the continuous return system described on p. 1635.

There is an expansion chamber in the return system, in which, due to a drop in pressure, part of the hot water is converted into steam, and this steam serves to operate the ejector.

The feed-water is pumped to the spray valve at the top of the de-aerating tank by means of a centrifugal pump, and arrangements are provided by which, if the feed demand of the boiler is small, the water in the de-aerator is recirculated.

The small quantity of boiler water returning to the softening plant through the continuous return line is sufficient to provide the steam for the ejector, so that no steam from external sources is required.

The plant can, however, also be operated independently of the continuous return system referred to.

A special feature apart from simplicity is the provision of about half an hour's supply of de-aerated water in cases of emergency.

FEED-WATER HEATING.

If heat which would otherwise be partially or wholly wasted be applied to heat the feed-water before it enters the boiler, an obvious saving is effected, since the amount of heat required to generate each pound of steam in the boiler is reduced. Even if the feed heating is carried out by means of steam withdrawn at low pressure from the engine or turbine, the gain by feed-heating may more than counterbalance the expenditure of steam.

The exhaust from auxiliaries is frequently used with advantage for feed heating.

Apart from questions of economy of fuel, feed heating is beneficial, as it reduces the temperature variations in the boiler, and thus also reduces the temperature strains. Also preheating the feed-water facilitates the removal of impurities and reduces the deposits in the boiler.

Assuming that heat which would otherwise be wasted, is used for feed heating, the saving may easily be calculated, as shown in the following example:—

Steam pressure 160 lbs. per sq. in.

Heat to generate 1 lb. steam from water at 60° F.

$$= 1203 - (60 - 32) \text{ B.Th.U. (see Steam Tables, p. 1655)}$$
$$= 1175 \text{ B.Th.U.}$$

If feed is heated to 200° F., the heat to convert 1 lb. of this hot feed into steam

$$= 1{,}203 - (200 - 32) \text{ B.Th.U.} = 1{,}035 \text{ B.Th.U.}$$

The saving is $\frac{1{,}175 - 1{,}035}{5} \times 100 = 11{\cdot}9$ per cent.

In general, if t_1 t_2 are the temperatures of the feed-water before and after heating, and H the total heat of the steam, whether saturated or superheated, the saving is

$$\frac{t_2 - t_1}{H - (t_1 - 32)} \times 100, \text{ per cent.}$$

The Row Feed-Water Heater.

(*Royles, Ltd., Irlam near Manchester.*)

In Row's Patent Feed-Water Heater, figs. 71 and 72, a peculiar form of indented tube is used, which it is claimed is *twice as efficient as an equal amount of plain tube surface*, giving great *compactness* and *efficiency*. The tube is also elastic in the direction of its length, thus avoiding liability of fracture or breakage when fixed between tube plates.

FIG. 71.

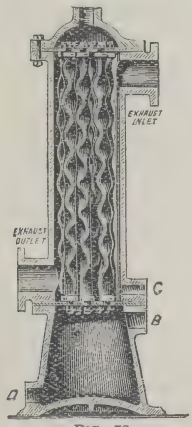


FIG. 72.

The Heater (fig. 72) shown is intended for good water—i.e. water not liable to scale—in which case the water is sent *through* the patent tubes. For bad water the order is reversed, i.e. the exhaust is sent through the tubes and the water circulates in the casing.

The Drayton Regulator and Instrument Co., Ltd., West Drayton, manufacture spirally corrugated tubes (fig. 73) fitted with plain copper ends for fastening to tube plates. These

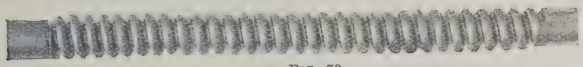


FIG. 73.

tubes possess over 60 per cent. more surface than plain tubes of the same mean diameter, and are particularly useful where space for feed-water heaters is limited.

The 'Hudson' Economiser or Exhaust Steam Feed Heater;

This heater, which is also a steam condenser, is of simple construction, and having no tubes, is claimed to cost nothing for renewal of parts.

The manufacturers state that the special features of this apparatus are :—

1. It extracts the grease from the exhaust steam *before* the latter comes in contact with the cold feed, and thus purified steam is obtained. This is important, as it is said to be impossible to perform the operation of separating the grease from the water afterwards without incurring considerable and constant expense.

2. The *purified steam* comes into direct contact with the *cold* feed (which enters the top chamber of the machine through a patent spray), the water being boiled, a large proportion of the carbonates and other impurities are thrown down, and these are easily removed through the manhole. Thus softened water may be obtained, *absolutely free from oil*, at a temperature of from 200° to 210° F. suitable for boiler feed, and, if required, a large surplus of hot softened water for washing and various other purposes may also be obtained by condensing the whole of the exhaust steam.

In many cases this softened water (free from oil) is being used for cask-washing in breweries, also in public baths, laundries, dye works, mineral-water factories, ice-making works, etc.

By use of this economiser, *incrustation* is entirely arrested before the feed-water enters the boiler, and generally the use of a metal brush may be substituted for the costly chipping of scale from the interior of the boiler shell. In this way a large saving is effected, and the apparatus by reason of this fact alone is of great value to steam users.

A report on *economy trials* by Professor T. H. Beare, M.Inst.C.E., showed an economy of fuel of 18·8 per cent. by the use of the economiser.

Automatic Temperature Regulation.

In heating feed-water it is essential to prevent the water being heated to boiling point, as at that point difficulties arise with feed pumps and injectors. By the use of automatic temperature regulators the predetermined feed-water temperature may be prevented from being exceeded. Automatic temperature regulators can be divided into two main groups: (1) Self-contained regulators producing their own operating energy for actuating valves, dampers, and such like; (2) indirect acting regulators in which the heat impulse merely brings into play an outside operating medium, which may be water under pressure, oil under pressure, compressed air, or electricity.

Self-contained regulators can work by vapour pressure obtained from a liquid with low boiling point, or by positive liquid pressure of an expanding liquid with a boiling point lying above the temperature for which the regulator is used. Vapour pressure regulators are limited to temperatures at which the vapour pressure is not too high, whilst liquid type regulators are limited to temperatures for which liquids with suitably high boiling point can be obtained. For temperatures above the limits of the vapour and liquid self-contained type, indirect regulation must be resorted to. In indirect regulators using air or liquid under pressure for the main regulation, a very small pilot valve only has to be actuated to control the main pressure, which can generally be effected with a very slight heat impulse, obtainable from a thermostat composed of two dissimilar metals. For temperatures which metals will not withstand without serious deterioration in a very short period the heat impulse must be obtained from electric pyrometers, which through at least two relays may operate either motors or powerful solenoids.

The 'Drayton' V.T. and M.S. Automatic Temperature Regulators.

(*The Drayton Regulator and Instrument Co., Ltd., West Drayton.*)

Fig. 74 shows the 'Drayton' V.T. Regulator, which is operated by the pressure of a volatile liquid in the bulb, acting through capillary tube on to a seamless metal bellows.

A spring opposing the action of the bellows determines the temperature at which the regulator will operate, and by varying the adjustment of the spring, a regulating range of 40° F. can be obtained.

Fig. 75 shows the 'Drayton' M.S. Air Operated Regulator, which is suitable where control within close limits is desired.

In this regulator the pressure of a volatile liquid or inert gas contained in the bulb acts on a diaphragm, which actuates a pilot valve controlling the supply of compressed air to the main diaphragm operated valve.

The principle of this regulator is shown by fig. 76.

The Drayton Regulator and Instrument Co., Ltd., also manufacture regulators suitable for operation by water instead of compressed air, and time temperature regulators in which the temperature adjustment is operated by clockwork, so that any desired temperature curve may be reproduced.

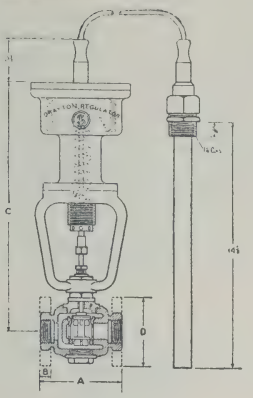


FIG. 74.—'Drayton' V.T. Automatic Temperature Regulator.

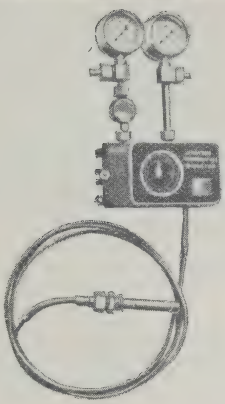


FIG. 75. 'Drayton' M.S. Automatic Temperature Regulator.

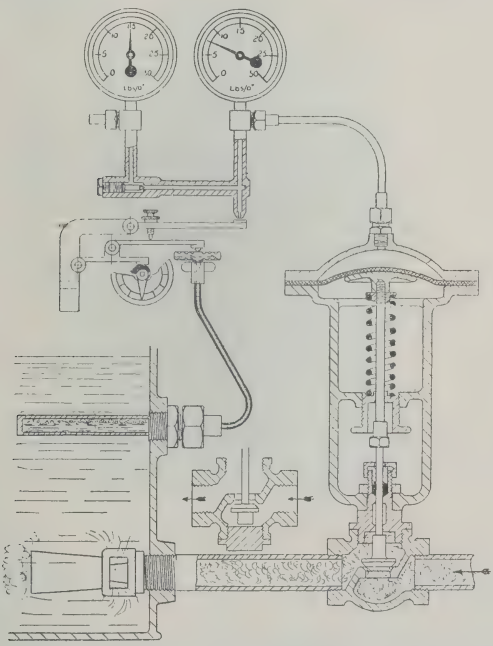


FIG. 76.—'Drayton' operating principle of M.S. Regulator.

ECONOMISERS.

General Notes.

The term 'economiser' is usually applied only to those feed-water heaters in which the heat is obtained from the boiler flue gases before they pass to the chimney.

The standard pattern for pressures up to 250 pounds per square inch is represented by the Green and similar economisers.

For the latest turbine practice, with pressures of 350 to 450 pounds per square inch, steel pipes are used, and it is becoming usual to design the economiser as an integral portion of the boiler unit. In this sense there is no economiser, but only an equivalent addition to the boiler heating surface. An alternative method is to use cast-iron pipes of moderate surface, sufficient to heat the water to about 200° F. A rotary pump then delivers the water into the boiler proper. The aim of this arrangement is cheapness and the use of standard plant.

Economiser surface can decrease almost inversely as the gas and water velocities, and in fact follows very much the same laws as for boiler and superheater surface.

When using very cold feed, below 90° or 100° F., it should be preheated to 100° F. by live or other steam, before passing to the economiser, or the pipes will usually sweat, and serious external corrosion take place.

The 'Green' Economiser.

The 'Green' Fuel Economiser, fig. 77 (*E. Green & Son, Ltd.*), consists of a series of sections or rows of cast-iron tubes $4\frac{3}{16}$ ins. in diameter and 9 ft long, placed in an enlarged part of the main flue, the tubes being fitted on their external surface with slow-moving scrapers. The sections are from 4 to 12 tubes in width, and the usual number of tubes is 96 per Lancashire

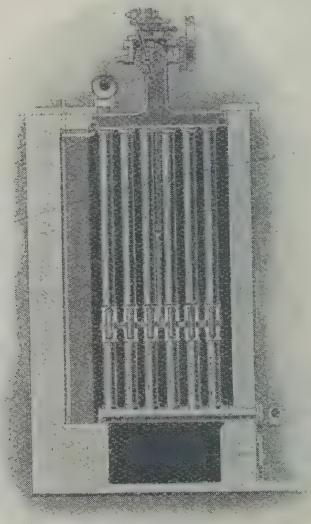


Fig. 77. The 'Green' Economiser.

boiler. When of large size the economisers are divided into a series of groups of 72 or 96 tubes each. The temperature of the waste gases may be reduced from 650° F. to 350° F., and the feed-water heated to a temperature of from 225° to 300° F. It is not advisable to feed an economiser with water at a lower temperature than 100° F. When the feed-water is colder than this, a small pipe should be led from the heated feed into the entering water, thus raising the temperature by mixture. This prevents external corrosion of the tubes, and improves the action of the scrapers.

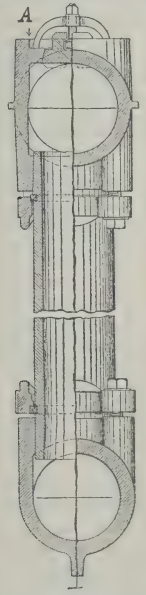
The weight of the economiser is about 1 ton for each eight tubes. Its water capacity is 6¼ gallons per tube, and its heat-absorbing surface 10 sq. ft. per tube. The general rule is to provide 4 tubes per ton of coal per week (56 hours). Another rule is to provide 1 tube for each 3 indicated horse-power, but a more accurate standard by which to calculate the size of the economiser required for a given boiler plant is the evaporative capacity of the boilers.

This economiser is claimed to effect a saving in coal varying from 15 per cent. to 20 per cent. in ordinary practice.

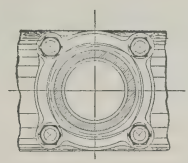
Green's Patent Ringstay Economiser

(for High Boiler Pressures).

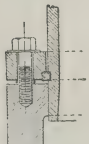
Extreme boiler pressures have made necessary certain alterations in the design of the standard economiser, which, although suitable in every way for normal pressures, does not provide for the higher pressures now met with. The chief alterations are: The staying of a certain number



Sectional View through Headers.



Plan of Ringstay Joint.



Details of Ringstay Joint showing Set-screw.

FIG. 78½

of socket joints in each section; a reduction in the diameter of the internal lid; a reversion to the circular type of top header, in place of the flat-topped rectangular design.

The Ringstay Joint, fig. 78, consists of an external ringplate and split ring threaded on to the end of the tube. The ring is held firmly to the shoulder of the tube by the ringplate, which is secured to the header by four set screws. Direct-pull tests have demonstrated that each of these joints is capable of sustaining a load of over twenty tons, irrespective of the friction of the tapered joint.

The lids are of the usual internal conical design, but of smaller diameter than the standard internal lid. The lid hole 'A' is stepped, so that if a tube requires replacing the metal can be more readily reamered out, the original lid being replaced by one of external type.

The lid holes are of sufficient diameter to enable the tubes to be cleaned, and as the headers are of the extended type, every tube can be inspected without breaking the top branch pipe joints.

One master lid is fitted into each top header.

The headers are of circular design, thereby ensuring a greater resistance to extreme internal pressures, and conforming in this respect to the best practice.

All the component parts are carefully examined, tested, and gauged before being used.

Green's Patent Tri-Tube Horizontal Economiser.

Green's Patent Tri-Tube Economiser was originally designed for erection in a horizontal position over modern types of water-tube boilers. In the space of a few years, however, it has become the standard type of economiser for modern high-duty, high-pressure boiler plants, and is installed either above or behind the boiler as necessary.

The economiser is constructed of cast iron of remarkably high tensile strength, while the patent Ringstay Joint increases by 100 per cent. the holding power between tubes and headers. The great resistance of cast iron to corrosive influences is well known and appreciated, and constitutes one of the chief reasons why Green's Tri-Tube Economisers are almost invariably specified for new installations. The success attained by this type of economiser is evident from the fact that it has been adopted at some of the largest electricity undertakings in the world.

Description.—The whole economiser, with standards and connecting bends, can be arranged to rest on the boiler supports, or, if desired, on independent joists, and can be fitted inside existing casing or provided with separate casing of Green's design. The sections are of such dimensions that they can be installed to suit almost any requirements and are interchangeable.

Sections.—Three tubes pressed into a front and back header constitute one section. and each is supported independently on vertical standards provided with projections on which the headers lie flat. There is ample room for expansion between adjacent headers and between the header ends and the standards.

Circulation of Water.—The number of sections in height or width of the economiser is decided by the dimensions of the boiler-casing. The water may pass through the bottom row of tubes, return along the next, and pass through each successive layer of tubes in series, or, on the other hand, it may enter the lowest two or three rows simultaneously, return along the next two or three in parallel, and so on throughout the remaining rows, thus putting the economiser partly in series and partly in parallel, as may be desired.

Each header is in itself horizontal and of small dimensions, consequently the temperature of the water from end to end is uniform and the expansion regular throughout the casting.

Baffling of Gases.—In modern high-efficiency boiler plants the economiser should be arranged to give as nearly perfect contra-flow conditions as possible, *e.g.* gases and water should flow in opposite directions. The special construction of Green's Tri-Tube Economiser readily permits of this.

Economiser Data.*

The present standard design of economiser has been practically the same since the first inception, and it would seem to be a very difficult matter to suggest any real improvement on the original design.

The economiser pipe is usually of a standard dimension of 9 ft. length and 4 $\frac{5}{16}$ ins. external diameter, being constructed to suit the boiler pressure and tested by hydraulic pressure to, say, 200 lbs. over working pressure. Each of these standard pipes has approximately 10 sq. ft. of heating surface, holds 6$\frac{1}{4}$ gallons of water and weighs $\frac{1}{8}$ ton. The power required to drive the scrapers is very small, being, for example, about $\frac{1}{2}$ h.p. for a set of 96 tubes. Cast iron is the most suitable material for the tubes, at any rate up to 300 lbs. steam pressure, as steel, wrought iron, copper, brass, etc., will not resist so well the corrosive action of the hot gases.

Makers recommend an installation large enough to hold one hour's supply of feed water in the pipes. Thus if a boiler is evaporating 650 gallons per hour, each tube holds 6$\frac{1}{4}$ gallons, and the number of economiser tubes would be then 104. When the evaporation is not known it is customary to allow 4 pipes for every ton of coal burnt per week of 54 hours, or one pipe for every 3 i.h.p. developed.

Out of 250 boiler plants (see page 1556) only 155 were fitted with economisers, the other 95 having no means of utilising the waste heat from the boiler flues. In these 155 plants, the exact average saving obtained was 11·4 per cent. of the coal bill. In actual practice, therefore, only about 17 per cent. of the plants fitted with economisers were saving 15 per cent. or over of the coal bill, whilst over 30 per cent. were saving less than 10 per cent., the true average for the whole 155 plants being 11·4 per cent. as stated.

---

* Abstract from 'Exact Data on the running of Steam Boiler Plants.' (D. Brownlie, B.Sc. Hons. (Lond.), F.C.S., J. Compston, and H. W. Royse; *Engineering*, November 1, 1918.)

The average figures from the 155 plants added together showed a beneficial rise in temperature of the feed-water of from 115 deg. F. before the economisers and 241 deg. F. after, *i.e.*, an average rise of 126 deg. F., equivalent to 11·4 per cent. saving. The corresponding fall in the flue gas temperatures from the 155 plants were 581° F. before the economisers and 389° F. after.

The reason the average figures for the actual saving are only 11·4 per cent. is not the fault of the economiser as an appliance, but is due to the fact that the economical generation of steam on scientific lines is not understood, and economisers generally are not installed on correct lines. The economiser is an absolutely essential part of any up-to-date boiler plant, and the saving obtained should be 17½ per cent. to 20 per cent. under average conditions.

Other advantages of the economiser are the reduction of the strain on the boilers by the provision of a high-temperature feed water preventing unequal expansion and contraction, and also that there is always a large reserve of hot water at hand for any sudden demand for steam. Further, the evaporative capacity of the boilers is greatly increased.

The chimney, as a draught producer, is an unscientific and wasteful method of creating a draught, depending for its efficiency on a great waste of heat. If this waste is utilised to the full by the economiser then the draught is curtailed and it is impossible to burn the necessary amount of coal to maintain the evaporation, in spite of the assistance of the economiser. There are many boiler plants that are forced, for this reason, to open slightly the by-pass dampers and allow a portion of the hot gases to go straight up the chimney to waste, so as to maintain a draught to burn the amount of coal required to keep up steam. The mere installing of economisers on rule-of-thumb lines may therefore in some cases be of little real advantage. This difficulty is, however, of little or no importance in the case of mechanical draught.

The water delivered into the economisers should not be over, say, 5 deg. to 6 deg. hardness, when all encrustation troubles will be at an end.

Thousands of economiser pipes have been rendered useless in a few years because of corrosive action. One remedy is to ensure that the feed water does not enter the economiser at a temperature of less than 100 deg. F., by providing an automatic circulating arrangement to take continuously a small portion of the water from the hot end and mix it with water from the cold end.

TABLE SHOWING RESULTS OF WORKING WITH ECONOMISERS.

Division according to Saving Obtained.	No. of Plants.	Average Temperature of Feed-water.		Average Temperature of Flue Gases.		Draught in Inches. W.G.		Steam Press. Gauge.	Total Evaporation on Plant per Hour.	No. of Tubes on Plant.
		Before.	After.	Before.	After.	Chimney Base.	Side Flue or Down-take of Boilers.			
		deg. F.	deg. F.	deg. F.	deg. F.			lbs. per sq. in.	lbs.	
No economiser at all .	95	—	—	—	—	—	—	—	—	—
Less than 5 per cent. .	14	134	172	568	371	0·85	0·41	80·0	17,616·0	228
5 per cent. to 7½ per cent.	15	123	194	576	423	0·61	0·35	79·0	15,340·0	203
7½ per cent. to 10 per cent. . . .	23	119	261	549	374	0·74	0·33	100·0	16,630·0	231
10 per cent. to 12½ per cent. . . .	33	118	243	583	403	0·88	0·39	106·0	29,723·0	401
12½ per cent. to 15 per cent. . . .	46	110	262	584	379	0·99	0·41	99·0	19,760·0	280
Over 15 per cent. . .	24	170	284	610	375	1·35	0·60	119·0	23,497·0	392
Total plants .	250									

Some large economisers are now made with tubes 11 ft. long.

For high-pressure plants two feed pumps may be used, one being interposed between the economiser and the boilers. Thus the economiser does not have to stand the full boiler pressure. By this means also an economiser weakened by corrosion may be retained in service where otherwise it would have to be scrapped.

For the pump between economiser and boilers a turbine pump driven by a steam turbine is suitable.

AIR HEATERS FOR BOILERS.

The preheating of the air supply to boiler furnaces has recently attracted attention, and the system has been applied in several power stations.

The air is heated by flue gases, but the recovery of heat from these gases is not the only gain, for in addition the effect of preheating the air is to increase the furnace temperature, increase the completeness of combustion of the fuel, and increase the efficiency of the boiler.

It is claimed that each 20° to 30° F. of increase of air temperature will result in at least 1 per cent. saving of coal, and that an efficient air heater is superior to an economiser and less expensive. Having regard to the desirable temperatures of feed-water and of air supply, it is possible that the best plan may be to recover part of the heat of the flue gases in an air heater and part in an economiser.

The Howden-Ljungstrom Air Preheater.

(*Howden-Ljungstrom Preheaters* (*Land*), *Ltd.*, *Caxton House*, *Westminster*, *S.W.* 1.)

The Ljungstrom Air Preheater is used to recover the heat in the waste flue gases leaving boilers or furnaces and to impart this heat to the air required for the combustion of the fuel. It consists essentially of a rotating regenerator built up of steel plates contained within a cylindrical steel casing. The regenerator incorporates a large number of heating elements corrugated and undulated in such a manner as to obtain a high rate of heat transfer. The rotor, which is driven through a rack and pinion by a small electric motor absorbing about ½ B.H.P., is contained in a steel shell which, by means of partitions, is divided into two separate chambers on both sides of the rotor. The flue gases pass on one side of the partition and through the rotor, and are discharged into the corresponding chamber of the shell on the other side of the rotor. The air passes in the opposite

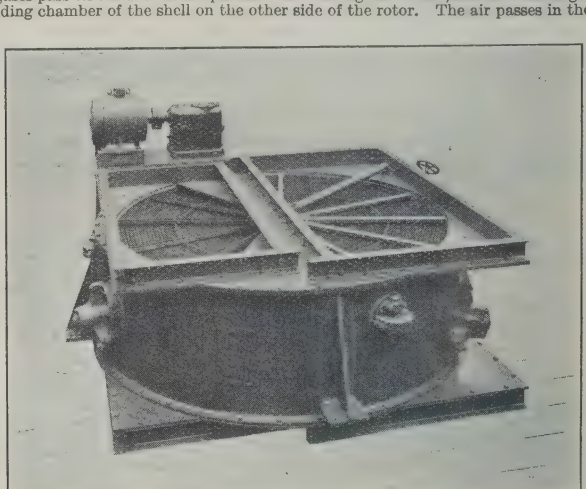


Fig. 79.

direction on the other side of the partition and through the rotor. The heating elements of the rotor pick up the heat from the flue gases on the gas side of the preheater, and as the rotor slowly revolves the elements pass over to the air side, where the heat is given up to the cold air.

The Ljungstrom Air Preheater is thus constructed as a continuous regenerator, and with this combines contra flow action. The design is such that positive and effective cleaning of the heating surface can be achieved while the preheater is in operation. The steam soot blowers are arranged radially one on each side of the rotor, and the whole operation of cleaning can be carried out in about 30 secs. during one revolution. The ease with which the elements may be cleaned ensures the efficiency of the preheater being constantly maintained.

Fig. 80 shows a Ljungstrom Air Preheater in which the rotor is visible through the air and gas openings on the top, while at the same time the soot blower and the driving motor

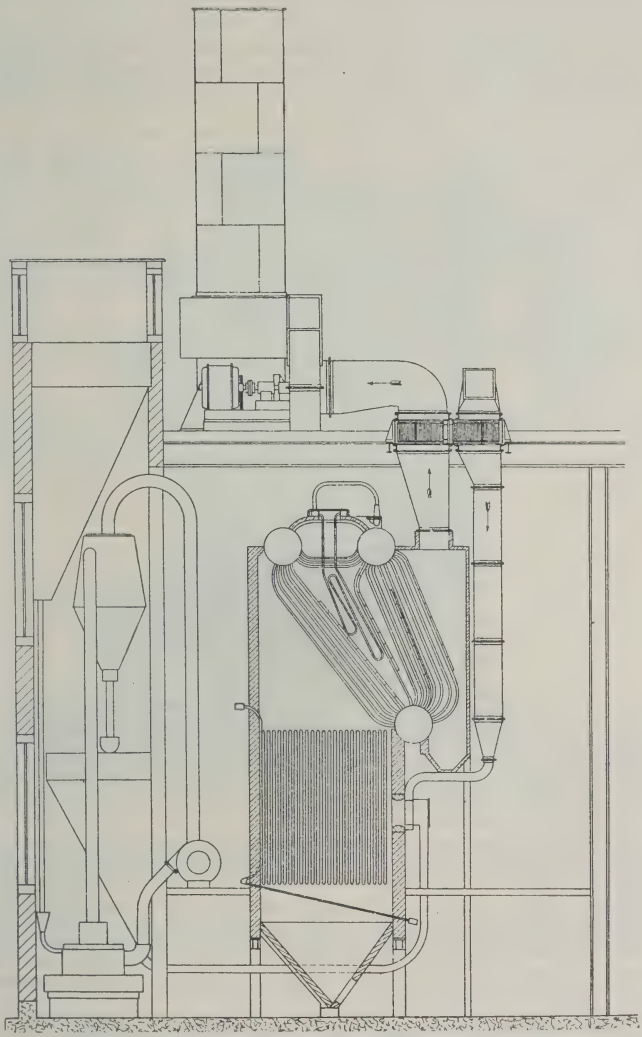


Fig. 80.—Arrangement of a Ljungstrom Air Preheater as the only Heat-Recovery Apparatus.

may also be seen. The rotor is fitted with a hard steel race ring which rests on three carrying-rollers, and which is kept central by three guide rollers. The housings of two of the carrying rollers and one of the guide rollers can also be seen in the photograph.

The regenerative contra flow principle not only enables high efficiencies to be obtained, but also permits low gas exit temperatures to be safely used. Troubles due to corrosion are almost unknown in the Ljungstrom Air Preheater, because the drying action of the fresh air passing over the same surfaces as those traversed by the gases makes corrosion extremely unlikely. The regenerative principle also provides a deterrent to corrosion as it prevents the temperature of the heating elements from falling below the mean of the air and gas temperatures even should a deposit of soot be permitted to form on these.

In the unlikely event of corrosion actually taking place in the heating elements of the Ljungstrom Preheater, it causes absolutely no leakage between the air and the gas, the only effect being a slight reduction of the efficiency due to a smaller heating surface being available. Even with the heating surface corroded, the Ljungstrom Preheater will, therefore, continue to operate satisfactorily, and the replacement of the heating surface may be deferred until a convenient opportunity occurs. This replacement can be carried out quickly and easily without disturbing the casings, and without the use of any special tackle, while the cost of the new heating elements is only a small fraction of the cost of the preheater.

A typical arrangement of the Ljungstrom Preheater as applied to a large pulverised fuel-fired boiler is shown in fig. 80. This shows the small size of the preheater, capable of reducing the temperature of the flue gases from 675° F. to 265° F., and supplying air at 515° F.—the preheater having an efficiency of 68·5 per cent.

The Ljungstrom Preheater is shallow, the design compact, and the length of the heating elements small. In spite of the high efficiency, the space occupied is negligible.

The 'Green' Rectangular Tube Air Heater.

This Air Heater has been designed to meet a demand for a heater of *light construction*, *economical proportions*, *maximum efficiency*, and *long life*. It consists of a number of elements grouped together *end to end*, *side by side*, and *one above the other*, in such a manner as to provide for a maximum amount of heat transmission with a minimum loss of pressure and draught.

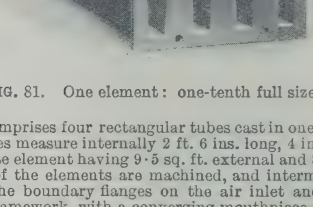


Fig. 81. One element: one-tenth full size.

Each element, fig. 81, comprises four rectangular tubes cast in one piece with a flange 10 in. × 5½ in. at each end. The tubes measure internally 2 ft. 6 ins. long, 4 ins. deep and 1 in. wide, and are 1½ in. apart, the complete element having 9·5 sq. ft. external and 8·33 sq. ft. internal heating surface. The flange faces of the elements are machined, and intermediate pairs of flanges are held together by bolts. The boundary flanges on the air inlet and outlet sides are bolted to suitable airtight sectional framework, with a converging mouthpiece opposite each tube in order to prevent undue eddying of the air (see fig. 83).

Each element weighs approximately 80 lbs; this is heavier than a heater of the same surface constructed of thin sheet-iron plates, but when it is remembered that the life of thin sheet-iron plates is only about one-tenth of the life of a cast-iron heater, a little extra weight is a decided advantage.

The space occupied by one element is slightly under 1 cu. ft.; consequently, a heater of sufficient capacity to supply the requisite quantity of air at a high temperature to the grate of a water tube boiler, need not take up more than 3 or 4 feet head-room, casing included, as the whole of the top of the boiler—being otherwise waste space—can be covered with these elements.

Accessible over the whole of the top surface of the boiler, a shallow heater can be inspected and cleaned with the greatest ease.

The elements are so arranged in the flue or uptake that the gases transversely sweep the flat surfaces of the tubes, which are arranged at 2½ ins. centres with, as already stated, 1½ in. spaces between adjacent tubes. The air, therefore, passes through the tubes in streams 4 ins. × 1 in. cross section, and the gases pass outside the tubes in streams 1½ in. in width.

The gases and the air to be heated can be made to pass either straight through the heater or through the several portions of it in series, according to the temperature and volume of air

required, and the weight of the gases at disposal. The design ensures, however, that both air and gases pass through the heater to the best advantage and with the least possible loss in pressure.

Fig. 82 shows a heater arranged immediately over a water tube boiler. Owing to the small amount of head-room taken up, it can be very easily cleaned, hand holes being arranged both in the top and sides of the casing.

The blowing off of the soot from a heater of this description is neither a lengthy nor a difficult process.

Cleaning doors in the side of the gas chamber admit hand lances for soot blowing by means of compressed air or steam.

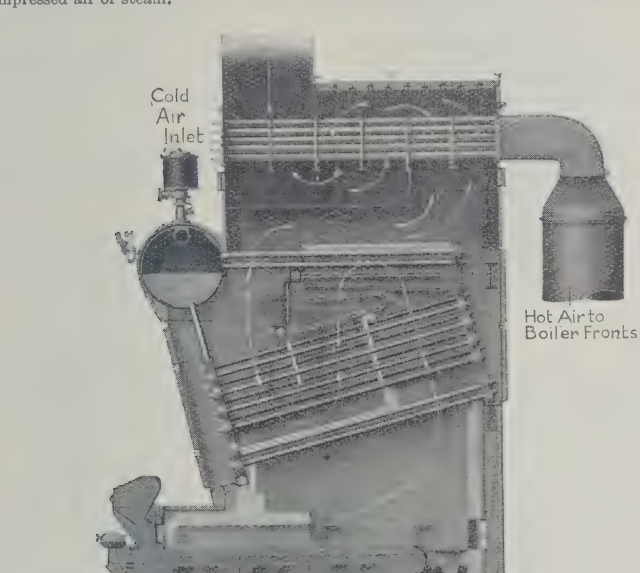


FIG. 82. 'Green' Rectangular Tube Air Heater over a Water Tube Boiler.

If desired, cleaning can be done from outside by means of fixed or revolving steam or compressed air pipes, so arranged that jets will play on or sweep practically the whole of the surface.

Hand cleaning holes can be provided in the top of the casing to give direct access to the upper part of the heater.

A large access door is recommended for entrance to the top of the heater should occasion necessitate.

The heater has no moving parts, and is therefore highly suitable for power stations where the number of necessary auxiliaries requiring close attention has become a serious problem.

THE GALLOWAY 'SUPERMISER.'

(Leek's Patents.) *(Galloways Ltd., Manchester.)*

The 'Supermiser' (figs. 83 and 84) is a combined air heater and economiser, that is, it serves to preheat the air for combustion and at the same time to heat the feed water, the heat for both these purposes being derived from the flue gases.

The apparatus contains a number of pairs of concentric tubes; the hot flue gases pass through

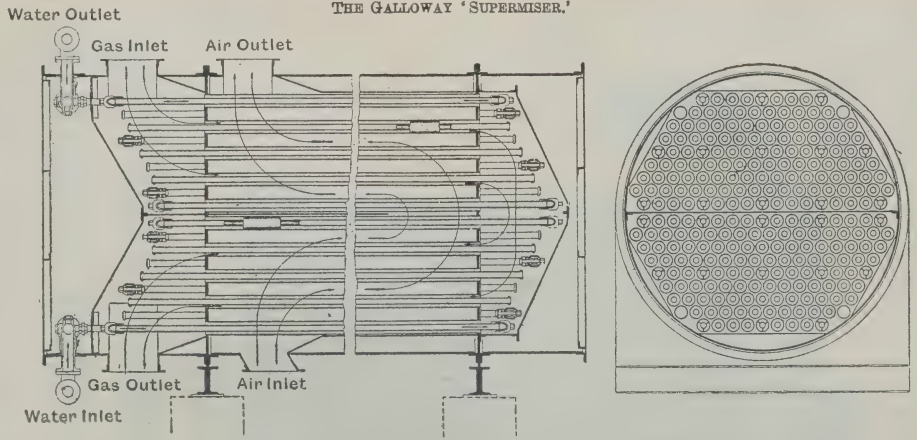


FIG. 83.—Section through 8 ft. 6 in. diameter 'Supermiser.

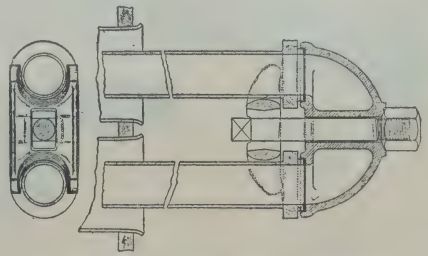


FIG. 84.—Section through header of 'Supermiser.'

the annular spaces between the inner and outer tubes, the air surrounds the outer tubes and the water flows through the inner tubes. The gas flows in the opposite direction to that of the air. The gases thus give up heat simultaneously to the water and to the air.

The rates of flow are intentionally high so that the rates of heat transmission are high, and thus the heating surfaces required are comparatively small, and the whole appliance is compact.

The tubes are carried between tube plates in a horizontal steel shell. The tubes are expanded into the tube plates at one end, but are merely a close fit at the other so that expansion and contraction due to temperature differences can take place freely.

The water tubes are supported inside the air tubes by thin radial fins parallel to the tubes, so designed as to offer minimum resistance to the gas stream, and are connected in series by headers of cast steel or of malleable cast iron.

The headers can be quickly detached from the tubes. Ample opportunity for inspection and cleaning and, if required, removal of the tubes is thus provided.

There is a horizontal division plate about the centre of the Supermiser, which extends from the tube plate at the front end (the end at which are the gas and air branches), but which at the other end leaves sufficient passage-way for the air to pass from the lower to the upper half. The courses of the air, gas, and water are clearly indicated in the illustration.

The details of construction are carefully thought out, particular care being given to the avoidance of sudden changes of section of the gas path. This ensures that there shall be no ' dead pockets ' where dust and soot can be deposited. At the end inspection doors give ample access to the end boxes ; the headers and tubes can be fully exposed for inspection and cleaning.

The high speed of the gas through the tubes not only gives high rates of heat transmission, but prevents the deposition of the solid particles contained in the gas.

In large Supermisers the water divides into two or more circuits, in each of which the tubes are connected in series.

The high gas and air speeds involve considerable resistance in the gas and air passages and fans are provided to suit ; mechanical draught would in any case be necessary on account of the low temperature at the chimney. Results have proved, however, that the economies due to the Supermiser amply repay for the power and cost of the fans.

The gas fan is placed near the gas outlet, producing a suction and delivering the gas to the main flue through which it passes to the chimney.

The air fan takes air from the atmosphere, and delivers it under pressure to the Supermiser. The heated air, still at an inch or so of water gauge, is led through a sheet metal duct, properly heat insulated, to the front of the boilers, and enters the furnaces below the grates, still at slight pressure. The pressure is regulated by dampers at the boiler fronts, so that above the grates it is slightly below atmosphere, and thus the firedoors can be opened without risk for purposes of inspection or for hand firing.

No brickwork setting for the Supermiser is required. It may be placed over the flues or carried on a light steel staging.

The results obtained in practice have been very striking, the gas being cooled to a temperature below 250° and the overall efficiency of the boiler plant with an ordinary Lancashire boiler raised to over 80 per cent., which the makers are prepared to guarantee in all ordinary cases.

The possibility of corrosion of the tubes due to deposition of moisture has been considered. It must be borne in mind that owing to the uniform distribution of the gas streams, pockets of low velocity and low temperature are avoided, and hence the *average* temperature of the gas at exit, namely, say 250° F., is the lowest temperature of the gas at any part of the apparatus. Thus deposit of moisture is unlikely, and as a matter of fact has not been found in practice.

The high efficiency of the plant is accompanied by increased capacity. In cases where a boiler battery is overloaded, the installation of a Supermiser may relieve the situation and save the provision of an extra boiler.

Life of Boilers.

Lancashire and Cornish boilers last about 30 years, although there are cases in which Lancashire and Galloway boilers (with dirty, but not very scaly, water) have lasted for 50 years. Water-tube boilers (subject to a good deal of re-tubing) may be taken to last 20–25 years. Marine boilers, if properly treated and taken care of, should last at least 20 years under forced draught, and 25 years under natural draught.

The life of any boiler depends, naturally, upon the degree of care with which it is handled.

BOILER INSPECTION.

Boilers must be inspected once in every fourteen months. The owner or user may be made responsible for the effects of an explosion.

To prepare a boiler for inspection the manhole and mudhole doors should be removed, the blow-off cock closed, the steam valve locked to prevent tampering and danger to cleaners and inspectors, the hot feed valve locked, the bars and bridges removed from the furnace, and the flue doors removed and the hearth pit plates lifted away.

The flues and boiler plates in the flues should be swept clean of soot and dust.

The amount and nature of the internal scale should be observed, and the boiler thoroughly cleaned inside and scale removed.

The special points to be observed by the inspector are the condition of the plates along the seatings and all points of contact with bricks, the dryness of the flues, the absence of lime mortar near the plates, the dryness of the top covering, leakages at seams, circularity of furnaces, blisters, signs of overheating. Internally: grooving at plate junctions, around the angle irons of shell or furnace tube connection to end plates, at the root of angle irons or flanged seams or other points of stress or flexure, tautness of gusset stays, easy slackness of longitudinal stay bolts if present, condition of plates as to cleanliness, corrosion &c., condition of low-water appliances, scum troughs, feed pipes, gauge openings, and generally all points bearing on the safety of the boiler that may by some unlikely and hard to conceive contretemps lead to mishap of any sort.

In old boilers the openings in the shell should be looked to in order to ascertain the weakening due to removal of so much plate, and the degree of strength given back by added rings or other reinforcements. It is also desirable to note the quality of the attendance and workings of the boiler periodically, and to test the accuracy of the steam gauge and working of the safety valves.

The foregoing remarks apply more particularly to shell boilers. For water-tube boilers, the caps of the water tubes must all be removed, and each tube examined and seen through for scale deposit, and as far as possible externally for blisters. Locomotive-type boilers are not easy to inspect, should never have riveted shell seams below water line, and must be largely judged by the appearance of such parts as are visible. Every few years the shell should be examined internally, the fire tubes being removed for the purpose. The accessibility of a boiler for complete inspection ought to be a considerable factor in determining the choice of a boiler, and more particularly so when the water is dirty or corrosive in quality.

Experience is necessary to make a good inspector. He must possess judgment and sense of proportion, and be able to order a boiler not to be worked until further notice, without it being possible for further enquiry to show that there were not good grounds for his fears.

BOILER EXPLOSIONS.

The following table gives the description of the boilers which have exploded or collapsed during the forty-four and a half years from June 1882, when the first Boiler Explosion Act came into operation, to December 1926, and the causes of the explosions and collapses.

Type of Boiler.	Land Boilers.		Boilers Afloat.		Cause of Explosion or Collapse.	Land Boilers.		Boilers Afloat.	
	No. of Explosions.	No. of Collapses.	No. of Explosions.	No. of Collapses.		No. of Explosions.	No. of Collapses.	No. of Explosions.	No. of Collapses.
Plain cylindrical .	71	0	0	0	Corrosion . .	127	169	15	18
Locomotive and portable	47	47	0	3	Grooving . .	31	5	0	0
Cornish . .	43	70	0	0	Excessive pressure	50	34	4	9
Vertical, with internal firebox	31	175	10	39	Outlets frozen .	84	4	0	0
Lancashire . .	27	39	0	0	Failure of stays .	15	16	0	4
Galloway . .	0	10	0	0	Deficiency of water	38	79	1	80
Chimney . .	1	6	0	0	Structural weakness	25	31	1	4
Marine . . .	1	0	7	104	Fractures . .	18	2	0	0
Return flue . .	3	1	7	10	General deterioration	9	2	4	0
Water tube . .	72	0	5	0	Seam rips . .	11	0	0	0
Multitubular, externally fired	1	0	0	0	Deposit . .	35	12	2	40
Miscellaneous . .	35	6	0	3	Not ascertained .	3	4	2	4
Hot water . .	114	4	0	0					
	446	358	29	150		446	358	29	159

(The British Engine, Boiler, and Electrical Insurance Co., Ltd.)

BOILER LEGISLATION.

The Boiler Explosions Acts, 1882 (45 & 46 Vic. ch. 22) and 1890 (53 & 54 Vic. ch. 35) provide for inquiry into boiler explosions. The Acts extend to the whole of the United Kingdom. The term 'boiler' means any closed vessel used for generating steam, heating any liquid, or receiving steam for heating, steaming or boiling, excepting boilers used exclusively for domestic purposes or in the service of his Majesty. On the occurrence of a boiler explosion, notice must be sent within 24 hours to the Board of Trade by the owner, user, or person acting for him. This notice to state the name of premises or works, postal address, day and hour of explosion, number of persons killed and injured, description of boiler, purpose used for, particulars of failure, working pressure, by whom last inspected, and by whom insured.

The Act of 1890 makes the Act of 1882 applicable to sea and mine boilers.

BOILER FITTINGS.

Injectors.

To Compute the Diameter of Throat and Volume of Discharge.

d = diameter of throat in millimetres; V = volume of water required per hour in gallons; P = pressure of steam in pounds per sq. in.

$$d = \cdot 709 \sqrt{\frac{V}{\sqrt{P}}} \qquad V = 1 \cdot 985\ d^2 \sqrt{P}.$$

Example.—The pressure of steam is 200 lbs., and the volume of water required to be delivered per hour is 1,010 gallons.

Then $\cdot 709 \sqrt{\frac{V}{\sqrt{P}}} = \cdot 709 \times 8 \cdot 45 = 6$ mms.; and $1 \cdot 985\ d^2 \sqrt{P} = 1 \cdot 985 \times 36 \times 14 \cdot 142 = 1{,}010$ gals.

The above particulars apply to all properly proportioned live steam injectors.

The Penberthy (Regd.) Patent Injector.

The 'Penberthy' (Regd,) Injector (fig. 85) is a perfectly automatic arrangement; by this is meant that when the injector is working and forcing water to the boiler if the current of water be suddenly broken by any cause, such as a sudden jar or jolt in the case of a traction engine or road

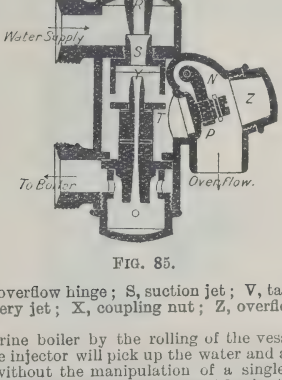


Fig. 85.

R, steam jet; O, plug; N, overflow hinge; S, suction jet; V, tail pipe; P, overflow valve; Y, delivery jet; X, coupling nut; Z, overflow cap.

roller, or in the case of a marine boiler by the rolling of the vessel throwing the sea-cock out of the water for a moment, the injector will pick up the water and again establish the current to the boiler automatically, *i.e.* without the manipulation of a single valve or the least attention from an attendant. It is this quality which has made the 'Penberthy' (regd.) so popular among

marine engineers and which has led to its adoption by the great majority of the traction and threshing engine manufacturers.

The 'Penberthy' (regd.) delivers water to boiler at 160° to 212° Fahr., according to temperature and steam pressure; but 200° to 211° can always be delivered at nearly all pressures over 50 lbs. by throttling the suction valve and delivering the minimum capacity.

The injector is made in fifteen sizes, from ½ in. to 2½ ins., and it is also supplied in four distinct styles to suit various positions. The maximum quantity delivered is from 80 to 4,200 gallons per hour according to size, and it is applicable to boilers up to 600 h.p. Sizes AA to CC are especially supplied for traction, portable, and ploughing engines It is most efficient at the pressures of 50 to 80 lbs. (when with the valve wide open it will lift 20 ft.), although it will work at a lower pressure and also up to as high as 180 lbs. under favourable conditions. It is suitable for all classes of boilers. A *low-pressure* non-lifting pattern is supplied for steam pressures below about 20 lbs.; these injectors will start at from 1 to 3 lbs. steam pressure where water is taken from a barrel or tank at, or just above, the level of the injector. With water pressure of 20 lbs. they will start at from 10 to 12 lbs. low steam, but with two globe valves in water-supply pipe the pressure can be reduced and injectors started at correspondingly lower pressures.

A special pattern (Nos. 328, 332) also is being made in ½ in., ¾ in., and 1 in. sizes, for very high pressures—namely, up to 225 lbs. No. 328 AA. is used extensively for steam wagons and tractors. Nos. 428 and 430 (½ in. and ¾ in.) are also made for the latest type Super Steam Wagons, for pressures up to 250 lbs.

For pressures to 240 lbs., and for hot-feeds, the Auto-positive type is supplied. This type has fine working parts, and combines the features of both the positive and the automatic injector.

Over two million 'Penberthy' (Regd.) Injectors have now been supplied.

POWER AND CAPACITY OF 'PENBERTHY' (REGD.) INJECTORS.

Size.	Horse-power.	Pipe Connections.	Capacity per Hour. 1 to 3 ft. lift, 60 to 110 lbs. Steam Pressure.	
			Maximum.	Minimum.
O	3 to 6	½ inch	60 gallons	35 gallons.
OO	4 „ 8	⅝ „	80 „	45 „
A	8 „ 16	½ „	135 „	70 „
AA	12 „ 22	⅝ „	180 „	100 „
B	17 „ 32	¾ „	260 „	140 „
BB	20 „ 45	¾ „	360 „	180 „
C	40 „ 65	1 „	475 „	250 „
CC	45 „ 80	1 „	600 „	325 „
D	50 „ 100	1¼ „	800 „	425 „
DD	75 „ 135	1¼ „	1,000 „	525 „
E	100 „ 180	1½ „	1,400 „	740 „
EE	115 „ 255	1½ „	1,900 „	850 „
F	160 „ 320	2 „	2,400 „	1,275 „
FF	200 „ 400	2 „	3,000 „	1,600 „
GG	375 „ 600	2½ „	4,200 „	2,150 „

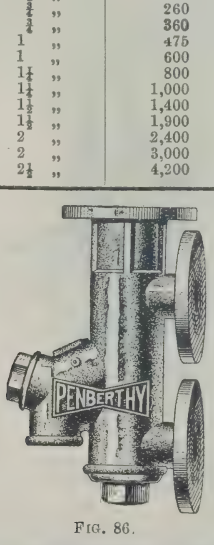


FIG. 86.

Owing to the increasing use of flange connections in this country, especially on fine works, the 'Penberthy' Regd. Injector is now made up with flanges as shown in fig. 86. The working

qualities of the flanged injector are only slightly different from those of the screw-nipple machine, except that by the different construction of the body the high working pressure is brought up to 180 to 185 lbs. The injector with flanges is connected and operated in precisely the same manner as the injector with screw nipples. It is made in all sizes except ¼ and ⅜ in.

TABLE OF FLANGED 'PENBERTHY' (REGD.) INJECTORS.

Size No.	Pipe Connections.	Diam. of Flanges.	Thickness of Flanges.	* Gallons of Water delivered to Boiler per Hour on 100 to 120 lbs. Steam Pressure, 3 ft. lift.	Horse-power.
	Ins.	Ins.	Ins.		
228	½	2 9/16	⅜	180	12 to 22
230	¾	3 5/16	7/16	355	20 „ 45
232	1	4 5/16	½	600	45 „ 80
234	1¼	5½	½	1000	75 „ 135
236	1½	5½	9/16	1900	115 „ 255
238	2	6 5/16	9/16	3000	200 „ 400
240	2½	7 5/16	11/16	4200	375 „ 600

(W. H. Willcox & Co. Ltd.)

* These capacities can be reduced 40 per cent. by regulating water supply valve.

WILLCOX'S EJECTOR STEAM-JET PUMP OR WATER RAISER.

This ejector, fig. 87 (*W. H. Willcox & Co. Ltd.*), will lift water 20 ft. with any steam pressure above 25 lbs., and 25 ft. with pressure at 40 lbs. or over. On pressures above 80 lbs.

FIG. 87.

steam must be throttled to get best results on a lift. It will also elevate 50 ft. at 60 lbs. pressure and 65 to 100 ft. on pressures of 50 to 100 lbs., by placing the ejector within 12 to 18 ins. of water supply.

TABLE OF WILLCOX'S EJECTOR WATER RAISERS.

Size Nos.	1	2	3	4	5	6	7	8	9	10
Pipe connection, steam . ins.	⅜	½	¾	1	1	1¼	1½	2	2	2½
„ suction and delivery „	½	¾	1	1¼	1½	2	2½	3	3½	4
CAPACITY PER HOUR.										
40 to 65 lbs. Steam, 3 ft. lift, gall.	240	500	840	1,350	1,950	3,500	5,700	9,500	13,600	18,400
20 „ 40 „ or 65 to 100 lbs. „	235	450	700	1,300	1,850	3,000	4,350	8,160	12,400	17,100
40 „ 65 „ 50 ft. elevation „	120	250	420	650	975	1,750	2,500	4,750	6,800	9,200
40 „ 65 „ 25 ft. „ „	180	375	625	950	1,450	2,600	3,750	7,200	10,200	13,800
VERTICAL LIFT.										
40 to 75 lbs. Steam . . ft.	23	25	25	25	25	25	25	25	25	25
25 „ 40 „ or 75 to 100 lbs. „	20	22	22	22	22	22	22	22	22	22
Horse-power of Boiler . .	2 to 6			8 to 12			15 to 20		25 to 30	

The ejectors are also suitable for exhausting air from pumps to create vacuum, and are largely used for this purpose.

Notes on Injectors.

A very frequent cause of trouble is that by one means or another the injector becomes overheated and refuses to lift its water. This may be due to steam leaking past the check valve into the body of the injector, and then passing to the suction pipe, so that the latter becomes too hot. Under these conditions it is impossible to make the injector work. Another cause of overheating is sometimes to be traced to the proximity of the fitting to the hot boiler shell, which allows heat from the latter to pass to it by conduction and radiation. In the first case the remedy is to fit a well-made check valve, combined with a plug cock, between the injector and the boiler. This should be done, even if the former is designed with a self-contained back-pressure valve. Further, the plug cock should be shut off when the injector is not in use. In the second instance, it sometimes happens that an injector is employed which is fitted with its own back-pressure valve, and it is then bolted directly to the boiler. This is very bad practice, as a single valve is not dependable in the first place, and a further point is, that the injector is much too close to the hot boiler. This is a fruitful cause of trouble. Also, should anything go wrong with the injector, it is necessary to blow down the boiler before it can be put in order; whereas, with a combined check valve and plug cock, the latter can be closed, and the injector removed under steam.

An injector will not work satisfactorily if supplied with wet steam; therefore, in order to ensure that this is as dry as possible, it should be taken from the highest available point in the boiler.

Another defect sometimes arises through the check valve not having sufficient lift, or on account of some sort of obstruction in the delivery pipe. The effect of this is that the injector will lift the water, but on account of the greater pressure due to the obstruction against which it has to work, it will be unable to force the water into the boiler, which will consequently find an escape through the overflow instead.

In some cases the water supply to the injector is taken from a tank placed beside the boiler on the boiler-house floor, the supply to the tank being from an external source, and controlled by a ball cock with a float attached, so that the level of the water in the supply tank is constant. This is a very satisfactory arrangement. In other instances the source of supply may be some little distance away, and connected to the injector by means of pipes. Further, if the water is taken from a well the level may vary between considerable limits. Hence, when purchasing an injector, it is of the utmost importance that the manufacturer should be given all possible data, such as the greatest static head and the length of pipe with the number of bends, so that he can obtain the total suction lift and supply an injector to suit the conditions.

Much trouble will be experienced if the feed-water is allowed to become hot. The lower the temperature the better. There are many so-called hot-water injectors, but even these will not prove very satisfactory if the temperature of the feed-water is allowed to rise too near to the maximum temperature specified by the makers. In order to prevent as much as possible any rise in temperature, it is of the greatest importance that the overflow from the injector, should not be allowed to drain back into the feed-tank. An overflow pipe should be fitted that will lead this hot water clear of the feed supply. Such a pipe should not be closed, but should enter an open cup, similar to the overflow of the circulating water in a gas-engine jacket. It is then under constant observation of the man in charge, and if any irregularity occurs it is quickly detected.

Occasionally an injector will refuse to work, due to a leak in the suction pipe. If this defect is present it may be located by fixing a plug in the suction pipe and opening the steam supply to the injector, when the position of the leak will be easily detected.

Lastly, dirt and grit are a source of trouble at times. To prevent this as far as possible, a straining rose should be fitted to the suction pipe. This will not exclude all grit, and occasionally small grains of solid matter suspended in the feed-water will find their way through and lodge on the internal surface of the injector. Also if the water contains soluble salts of magnesium and calcium, it becomes heated on passing through the injector, and some of these salts are deposited, forming incrustations which may cause the injector to refuse to work, either because it has become totally choked up, or else small particles of the incrustations may break away and become lodged under the back-pressure valve or check valve and prevent its closing, when the injector will refuse through overheating.

It is always advisable to keep a spare injector in stock, and if the water contains a considerable amount of impurity, instructions should be given to change the injector at regular periods. The length of time that one should be allowed in continual service must be governed by experience with the particular conditions under consideration. In order to loosen the deposit the injector may be left to soak in diluted hydrochloric acid.

If the above very easily followed points are given attention, injectors will be found most reliable, and few complaints will be heard.

(*'Mechanical World.'*)

SAFETY VALVES.

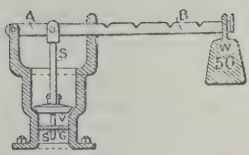


FIG. 88. B, lever; A, short arm of lever; SS, stem; V, valve; G, guide; W, weight.

WEIGHT ON SAFETY VALVES.

If A = area of valve in sq. ins., P = pressure in lbs. per sq. in., l_1 = distance of valve from fulcrum, l_2 = length of lever, w_1 = weight of lever in lbs., l_3 = distance of centre of gravity of lever from fulcrum, w_2 = weight of valve and stem in lbs.

Then W, the weight required at the end of the lever

$$= \frac{P \times A \times l_1 - w_1 \times l_3 - w_2 \times l_1}{l_2}.$$

SAFETY VALVE AREAS.

Since the steam production of a boiler is sensibly the same at all pressures, and since the density of steam increases very nearly as the pressure, and since steam escapes at an outlet at practically constant volume, it appears that safety valves, like boiler stop valves, may be reduced in size as pressure increases. Ordinary valves have a very small lift. Special valves in which the steam reaction is utilised to increase the lift will give a much greater opening. If a valve lift a height equal to a fourth of its diameter the area of opening is equal to the valve area, and no greater lift would give greater output.

It is not sufficiently well known that boilers working at a low pressure require larger safety-valves than boilers at a high pressure. Thus, for example, a boiler working at 15 lbs. by the steam gauge requires three times the area of safety valve opening that is required by a boiler at 75 lbs. This is the more important when it is remembered that in a boiler which is safe only at low pressure, an additional 15 lbs. pressure doubles the strain, whilst in the boiler at 75 lbs. a rise of 15 lbs. adds but a fifth to the stress on the plates. This point as to the safety-valves on old low-pressure boilers is too little known or recognised.

Safety-valve areas should be inversely as the absolute pressure.

In well-proportioned dead-weight safety valves it is usually expected that if a pressure of 60 lbs. per sq. in. opens the valve, a pressure of 70 lbs. will keep it sufficiently open for the escape of all the steam produced.

The former Board of Trade rules are largely followed, and as they are more detailed, bringing out many points of importance, extracts are given below, also extracts from the rules of Lloyd's and Bureau Veritas.

BOARD OF TRADE RULES FOR SAFETY VALVES.

The area per square foot of fire-grate surface of the locked-up safety valves for boilers with natural draught should not be less than that given in the table (page 1661) opposite the boiler pressure intended, but in no case should the valves be less than 2 ins. in diameter.

When ascertaining the fire-grate area, the length of the grate should be measured from the inner edge of the dead plate to the front of the bridge, and the width from side to side of the furnace on the top of the bars at the middle of their length.

[*Note.*—The table corresponds to the formula $A = \frac{37{\cdot}5}{P + 15}$, where A = safety-valve area in sq. ins. per sq. ft. of grate, and P = pressure in lbs. per sq. in.]

Where forced draught is used, the area should not be less than $\frac{A}{20}$ × estimated coal consumption in lbs. per hour per sq. ft. of grate.

The safety valves should be fitted with lifting gear, so arranged that the two or more valves on any one boiler can at all times be eased together, without interfering with the valves on any other boiler. The lifting gear should in all cases be arranged so that it can be worked by hand either from the engine-room or stoke-hole.

Care should be taken that the safety valves have a lift equal to one-fourth their diameter; that the openings for the passage of steam to and from the valves, including the waste-steam pipe, have each an area not less than the area according to the table; and that each valve box has a drain pipe fitted at its lowest part.

In the case of lever valves, if the holes in the lever are not bushed with brass, the pins must be of brass; iron and iron working together must not be used.

Too much care cannot be devoted to seeing that there is proper lift, also that free means of escape for the waste steam are provided, as it is obvious that unless the means for escape of the waste steam are ample, the effect is the same as reducing the area of the valves or putting an extra load upon them. The valve seats should be secured by studs and nuts.

At least two valves should be fitted to each boiler.

The springs and valves must be so cased in that they cannot be tampered with.

Provision must be made to prevent the valves flying off in case of the springs breaking.

The requisite safety-valve area must be cased in and locked up in the usual manner of the Government valves.

The size of the steel of which the springs are made must be in accordance with that found by the following formula.

$$d = \sqrt[3]{\frac{L \times D}{C}};$$

where,

L = load on the spring, in pounds;
D = diameter of the spring—from centre to centre of wire—in inches;
d = diameter, or side of square, of the wire in inches:
C = a constant, being 8,000 for round and 11,000 for square steel.

The springs must be protected from the steam and impurities issuing from the valves.

When valves are loaded by direct springs, the compressing screws must abut against metal stops or washers, when the loads sanctioned by the surveyor are on the valves.

The springs must have a sufficient number of coils to allow a compression under the working load of at least one quarter the diameter of the valve.

Spring-loaded valves are tested under full steam and full firing for at least twenty minutes with the feed-water shut off and stop-valve closed.

When the valves are of novel design, and the accumulation of pressure exceeds 10 per cent. of the loaded pressure, the case has to be specially reported upon by the surveyor.

Lloyd's Rules for Safety Valves.

(Abridged.)

At least two safety valves are to be fitted to each boiler, arranged so that springs and valves are cased in, that valves cannot be overloaded while steam is up, that the valves can be lifted by easing gear and turned round on their seats, and that valves cannot lift out of their seats if the springs break.

Vertical boilers with 100 square feet or more of heating surface to have two safety valves each not less than 1·5 in. diameter; those with less than 100 square feet heating surface one valve not less than 2 ins. diameter.

All safety valves of one boiler may be fitted in one chest connected direct to boiler by passage having cross-sectional area not less than half the aggregate area of the valves. Valve chest to be drained. The minimum aggregate area of safety valves of a boiler, coal- or oil-fired, in square inches

$$= \text{total heating surface of boiler in square feet} \times \frac{C}{p + 15},$$

where

p = pressure in lbs. per square inch;
C = 1·25 for boilers using coal;
C = 1·5 for boilers using oil and for those with closed stokehold forced draught.

Waste-steam pipe to have area not less than 1·1 times area given by above formula.

Safety valves to be set under steam. During test of fifteen minutes with stop valve closed and full firing conditions accumulation of pressure is not to exceed 10 lbs. per square inch.

Bureau Veritas Rules and Regulations for Safety Valves.

At least two spring safety valves of an approved design must be fitted to each main boiler.

The total area of the safety valves is given by the formula

$$A = \frac{8{\cdot}7}{\sqrt[3]{(P - 18)^2}},$$

where

A = sectional area of safety valve in sq. ins. per sq. ft. of grate surface; P = working pressure in lbs. per sq. in. No safety valve to be less in diameter than 1¼ in.

When forced draught or oil fuel is provided for, the area is to be increased in proportion to the increased evaporative power of the boilers.

Suitable arrangements and gear to be fitted in connection with the safety valves, whereby they may be lifted from the deck as well as from the stoke-hole floor.

BOARD OF TRADE RULES AND REGULATIONS FOR STEAMSHIPS.

SAFETY-VALVE AREAS. (*Abridged Table.*)

Boiler Pressure.	Area of Valve per Square Foot of Fire-grate.	Boiler Pressure.	Area of Valve per Square Foot of Fire-grate.	Boiler Pressure.	Area of Valve per Square Foot of Fire-grate.	Boiler Pressure.	Area of Valve per Square Foot of Fire-grate.
Lbs.	Sq. Ins.	Lbs.	Sq. Ins.	Lbs.	Sq. Ins.	Lbs.	Sq. Ins.
15	1·250	62	·487	109	·302	156	·219
16	1·209	63	·480	110	·300	157	·218
17	1·171	64	·474	111	·297	158	·216
18	1·136	65	·468	112	·295	159	·215
19	1·102	66	·462	113	·292	160	·214
20	1·071	67	·457	114	·290	161	·213
21	1·041	68	·451	115	·288	162	·211
22	1·013	69	·446	116	·286	163	·210
23	·986	70	·441	117	·284	164	·209
24	·961	71	·436	118	·281	165	·208
25	·937	72	·431	119	·279	166	·207
26	·914	73	·426	120	·277	167	·206
27	·892	74	·421	121	·275	168	·204
28	·872	75	·416	122	·273	169	·203
29	·852	76	·412	123	·271	170	·202
30	·833	77	·407	124	·269	171	·201
31	·815	78	·403	125	·267	172	·200
32	·797	79	·398	126	·265	173	·199
33	·781	80	·394	127	·264	174	·198
34	·765	81	·390	128	262	175	·197
35	·750	82	·386	129	260	176	·196
36	·735	83	·382	130	·258	177	·195
37	·721	84	·378	131	·256	178	·194
38	·707	85	·375	132	·255	179	·193
39	·694	86	·371	133	·253	180	·192
40	·681	87	·367	134	·251	181	·191
41	·669	88	·364	135	·250	182	·190
42	·657	89	·360	136	·248	183	·189
43	·646	90	·357	137	·246	184	·188
44	·635	91	·353	138	·245	185	·187
45	·625	92	·350	139	·243	186	·186
46	·614	93	·347	140	·241	187	·185
47	·604	94	·344	141	·240	188	·184
48	·595	95	·340	142	·238	189	·183
49	·585	96	·337	143	·237	190	·182
50	·576	97	·334	144	·235	191	·181
51	·568	98	·331	145	·234	192	·181
52	·559	99	·328	146	·232	193	·180
53	·551	100	·326	147	·231	194	·179
54	·543	101	·323	148	·230	195	·178
55	·535	102	·320	149	·228	196	·177
56	·528	103	·317	150	·227	197	·176
57	·520	104	·315	151	·225	198	·176
58	·513	105	·312	152	·224	199	·175
59	·506	106	·309	153	·223	200	·174
60	·500	107	·307	154	·221		
61	·493	108	·304	155	·220		

WATER GAUGES.

The breakage of a water-gauge glass in high-pressure steam boilers is fraught with danger, especially in places where the attendants are working in confined places from which it is difficult for them to escape, and any appliance which prevents the serious consequences arising from this cause is one of the most important and desirable safety fittings that can be attached to a steam boiler.

The Klinger High-Pressure Water Gauge.

(*Richard Klinger, Ltd.*, 120 *Southwark Street, London, S.E.* 1.)

The latest type of this well-known gauge (fig. 89) is suitable for working pressures up to 1,000 lbs. per square inch, the great advantage of which is that it can be entirely dismantled

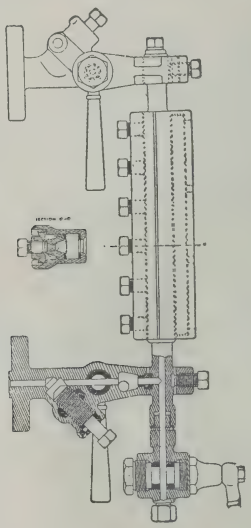


Fig. 89.

whilst the boiler is under steam. This is made possible by the inclusion of an auxiliary plug, which can be screwed right home, thus shutting off the water and steam.

It is impossible to make a mistake in reading the gauge as the water shows black and the steam silver.

These gauges can be supplied tipped forward at an angle of 30° for Babcock & Wilcox Stirling, Vickers, and Thompson water-tube boilers.

The tendency to increase the steam pressure on modern steam plant well over 500 lbs. per square inch has necessitated the construction of absolutely reliable water-level indicators which ensure a perfectly safe reading of the water level in high-pressure boilers working under pressures from 500 to 2,000 lbs. per square inch.

A sectional drawing, fig. 90, describes such a gauge constructed for 2,000 lbs. steam working pressure, and the chief feature of this type of gauge is the construction of the gauge body, which is designed in such a manner that the main joint for which the

The World's Best Jointing

"Klingerit"

The Original

JOINTING

Withstands

5,500

lbs. pressure per sq. inch.

550° C.

superheat

This well-known high pressure jointing has now been on the market for more than 30 years, and its unfailing reliability and economical qualities have gained it a reputation unequalled by any other.

It has successfully withstood the most exacting tests and conditions of modern steam and chemical plants, and is eminently suitable for hot oils, acids, alkalis, and other reacting chemicals.

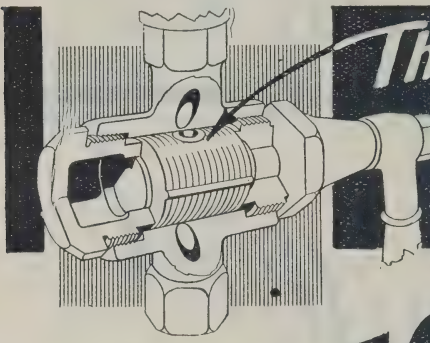
Klingerit is only genuine if stamped on one side of the sheet 'KLINGERIT'

IN SHEETS AND READY-CUT JOINTS

RICHARD KLINGER LTD.

120 SOUTHWARK STREET, LONDON, S.E.1.

Telephone: Hop. 1384 Telegrams: KLINGERIT, Boroh, London.



This feature eliminates all cock Troubles

KLINGER'S

SLEEVE PACKED COCKS

Whatever your problem there is a KLINGER sleeve packed cock to give you every satisfaction. Largest bore made 1″ dia.

Cocks that are troubleproof—that eliminate the old wasteful method of packing with asbestos—simple, reliable and with extraordinary long wearing qualities, that is the wonderful principle embodied in all **KLINGER SLEEVE PACKED COCKS.** For this sleeve, constructed of highly resilient material effectively prevents leakage under the highest pressures and temperatures obtainable, and gives a cock that has been proven indestructible, efficient and economical.

RICHARD KLINGER LTD.
120 Southwark Street, London, S.E.1.
Telegrams : "Klingerit, Boroh, London"
Telephone : HOP 1384

INSTALLED IN POWER PLANTS ALL OVER THE WORLD FOR EVERY PURPOSE UNDER ALL CONDITIONS.

F. 1663

gauge relies upon for its steam tightness is now removed to the front, and away from the corroding action of water and steam inside the gauge body. Such gauges have already

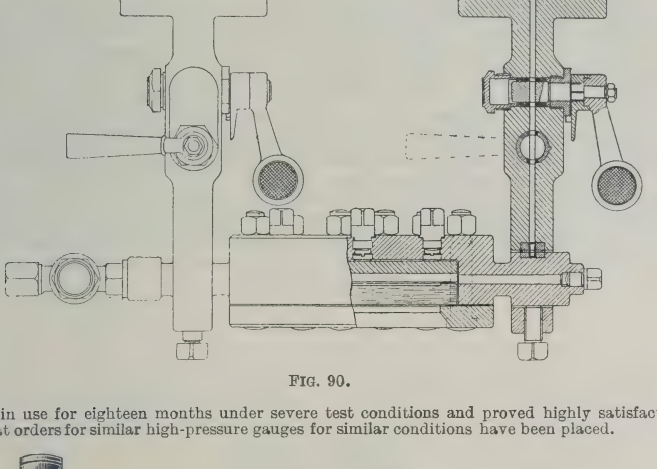


FIG. 90.

been in use for eighteen months under severe test conditions and proved highly satisfactory, repeat orders for similar high-pressure gauges for similar conditions have been placed.

FIG. 91.

HULBURD'S CARTRIDGE SIGHT GLASSES.

(*Patent.*)

This (fig. 91) is a neat and efficient device for protecting and rapidly removing and replacing sight glasses in water gauges and sight-feed lubricators. In Hulburd's make of gauge frames and lubricators the glasses are packed at leisure in metallic guards or sheaths, each thus forming a self-contained 'cartridge' from which the system derives its name. In the event of breakage of a glass the complete cartridge, containing the broken fragments, is easily withdrawn while under steam, and a new one readily inserted in its place. The metallic guard or cartridge case enables all splinters of glass to be removed without danger to attendants, and permits the immediate screwing up of the gland nuts without the possibility of fracture, the metallic cartridge case receiving the gland pressure and completely protecting the glass tube within it. Suitable windows are provided in the cartridge case of ample size to afford an unobstructed view of the glass tube and contents, and in the case of gauges the back is arranged as a water level indicator of special design, which shows with peculiar distinctness the height of the water column and makes it impossible to mistake an empty glass for a full one.

'WOODITE' PACKING.

The material known as 'Woodite' (*The Woodite Co.*) is of a special secret composition to resist the action of high or low pressures, whether of steam or water, and is impervious to acidulated or other chemically impure fluids. 'Woodite' is made to any shape or size and of a quality or flexibility to suit the service to which it is to be applied. It has been extensively employed for conical gauge-glass rings, packing rings, pump valves, hydraulic packings, gaskets, etc., with most satisfactory results. The chief advantages of the Woodite conical gauge-glass and packing rings for steam users are their ease of application or renewal, the maintenance of a perfectly tight joint, the entire avoidance of unnecessary pressures on the glasses or rods, which causes breakages in the former and excessive friction in the latter, and the elimination of the dangers incident to steam or water passages becoming choked in the case of gauge glasses. On large slow-moving boiler feed-pumps these conical packing rings have shown excellent results and economy, working at 185 lbs. steam pressure with superheat, the piston rods being $3\frac{5}{8}$ ins. diameter.

Ordinary valve spindles packed with these packings will always be found free to move, no matter how long the valve has been idle.

Defects in Water Gauges.

As ordinarily fitted, the gauge is usually attached directly to the boiler and provided with three taps, one connecting to the steam space, the second to the water space, and the third controlling the discharge to atmosphere. The latter should always discharge inside the boiler-house, where the effluent can be seen, and not be as it sometimes is, with a mistaken idea of cleanliness, coupled to a waste pipe, led outside, where the evidence of its action is not visible when the attendant opens the tap. It is of value to have duplicate gauges, as the cost is comparatively trifling, and one gauge serves as a check on the other. All three taps should be tested independently, that is to say, the bottom tap should be shut while the top one is opened and *vice versâ*, the top one shut while the bottom is opened, so as to give a thorough 'blow through' of the steam and water connections to the boiler. Unless this is done there is no certainty of the gauge being in working order, while there is risk of the top and bottom connections becoming choked and causing a false reading. More than one serious accident has been traced to defects in the taps themselves, owing to the holes in the plugs of the taps not coinciding with the thoroughfare of the fitting, so that when the position of the handle suggests the tap is open it is in reality nearly closed. Another cause of false readings is the blocking of the passage in the glass tube through the indiarubber packing ring working over the ends of the glass, owing to the glass being cut too short and the gauge not being provided with a nipple in the bottom of the gland to protect the glass against this risk. Having regard to the serious results which may attend overheating from shortness of water, it is advisable whenever possible to supplement the water gauge with a fusible plug, and if the boiler is a large one, with a low-water safety-valve as well. (*Mechanical World.*)

Gauge Glasses.

It is important to have the fittings exactly in line with each other, and to have them large enough so as not to cause the glass to bind or to be cramped in the least. When the fittings are not in line the glass is apt to be cramped either when putting it into the sockets in the fittings or when tightening the packing nuts. Wrenches should not be used for tightening the packing nuts around the gauge glass. When a glass is packed with the proper kind of packing and the nuts have been screwed up just tight enough to prevent leakage when both the glass and the packing have been warmed, the glass may readily be turned around in the fittings with merely the thumb and forefinger.

After putting in a new glass turn on a very little steam, leaving the drip at the bottom of the glass open so that the steam will gradually fill the glass and escape through the drip-cock. The glass is thus warmed thoroughly before the water is let into it and before it is subjected to pressure. When the glass has become thoroughly hot, open the steam valve wider, then open the water valve a like amount, and continuing to open both valves alike until fully open, then slowly close the drip at the bottom. This method not only allows the fittings to be thoroughly blown out, but enables the operator to increase the pressure on a new glass and new packing more gradually. It is best to anneal all the glasses before using them. This is generally accomplished by means of a water and steam bath, the water and the glasses being heated by steam. It is cheaper in the long run to cut glasses with a gauge-glass cutter than to attempt to cut or break them off with a broken file, or by similar means, because a slight nick in the glass at the wrong place may mean the early destruction of the glass when subjected to pressure and varying temperature. Some engineers fit a thin brass or copper thimble into the upper end of the tube, as shown in fig. 92, which prevents scale and water

Fig 92.

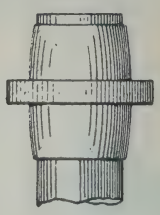


Fig. 93.

from coming in contact with the glass when blowing out the latter, and hence it reduces the eroded and pitted appearance commonly seen at the upper end of unprotected glasses. Tubes made of aluminium may be obtained of dealers in engineers' supplies which accomplish the same result.

These tubes and thimbles, it will be seen, protect the inner surfaces of the glass only, and as the glass is liable to be crowded to one side or the other in the packing nut as well as in the fitting itself, it is a good plan to also protect the glass on the outside.

A simple means of protecting the glass on the outside is also obtained from dealers in gauge glasses. The arrangement consists of a rubber sleeve adapted to slide over the end of the glass, as shown in fig. 93. The glass cannot come in contact with the socket in the fitting, neither can the packing nut touch the glass when the packing is screwed up. The combination of the thimble and the latter device prevents the rapid wear of the glass at the top, and also retains the glass clear of all metallic contacts, which has been found to be very desirable where glasses are subjected to high pressure and temperature and sudden changes in the latter.

When the rubber protector is employed, separate packing rings are not needed, because the packing forms a part of the sleeve, as shown. Annealed gauge glasses protected inside and out by this or similar means will last as long under any circumstances as it is possible to make them by any means thus far made known.

If packing 'rings' are used, then in order to secure a tight joint with the least possible pressure, thus avoiding any tendency to break or to cramp the glass due to the packing, it is necessary to obtain a form of packing that will fit snugly in the space in the nut.

(*H. E. Waverly*, '*Mechanical World.*')

The following arrangement, fig. 94, for packing is said to give very satisfactory results:—A fibre washer $\frac{1}{16}$ in. thick with an outside diameter to allow an easy fit into the gauge-cock nut, and an inside diameter equal to that of the glass, is put into each nut before the rubber gasket.

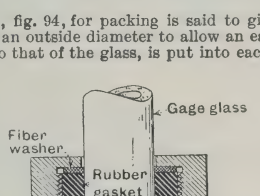


FIG. 94.

The fibre washer so fitted prevents the rubber gasket being pushed out between the glass and hole in the nut. The fibre does not scratch the glass, and therefore will not cause the latter to crack and break, as would a metal washer or a nut with too snug a hole in it.

(*F. R. Rawson*, *Power*, *July* 10, 1928.)

Superheated water acts on most varieties of glass as a violently corrosive agent, and for gauge glasses ordinary glass cannot be used. A special glass, under the brand 'Hercules,' to resist high pressure steam, is made by Joseph Tomey & Sons, Ltd., Birmingham. It is specially constructed of boro-silicate, and is made after the 'Jena' formula. The glass is practically white in colour and has withstood the following remarkable tests:—

> After the glass had been submerged in iced water for five minutes it was plunged into molten lead, and showed no signs of fracture. It has withstood 4000 lbs. hydraulic pressure, and 750 lbs. steam pressure.

Although the glass was specially manufactured to meet the requirements of high-pressure and superheated boilers, it has been used for a great number of low-pressure boilers, because at low pressure the material is practically unbreakable and is considerably cheaper in the long run than the ordinary quality gauge glass.

CUTTING GAUGE GLASSES.

A simple and reliable method of cutting gauge glasses to any desired length consists in filing a notch, then wrapping the tube with a strip of wet filter paper or paper towel on each side of the notch, leaving the space between the strips $\frac{1}{16}$ to $\frac{1}{8}$ in. wide. The strips of paper may be $\frac{1}{2}$ to $\frac{3}{4}$ in. wide and wrapped on about three layers deep. The fine flame of a blast burner is directed against the tube between the papers and at the notch for a second or two when the tube is slowly rotated in the flame. The tube should be turned rapidly enough to avoid burning the wet paper.

At the completion of the revolution the tube will break off with a clean and even cut. Short tubes for oil-sight feeds may be cut with ease by this method when it would be impossible to break them off from just a file notch. Bottles and large tubes are equally easy to cut.

In breaking a tube from a file notch, which is the universal method for small-diameter tube and rod, much depends on the nature of the file mark. Just as in cutting window glass, what is wanted is a cut and not a scratch, and in filing a notch completely around the tube, even the inexperienced is likely to make a real cut at some point on the circumference, while in his hands the single file notch may not register the cut necessary for the clean break.

Unbreakable Water Gauge.

A simple and effective form of water-level indicator which has all the advantages of the test cocks in checking the water level, but does not suffer from their disadvantages, has been patented by Messrs. Cammell Laird & Co., Ltd., and is being manufactured by Messrs. W. Crockatt and Sons, Ltd., of 64 Darnley Street, Glasgow. It consists of a perforated metal tube, and replaces the ¾-in. glass tube in an ordinary water-gauge mounting. The diameter of the holes is $\frac{1}{16}$ in. On the gauge cocks being opened, water issues from the hole below the water line in the form of a rapidly diverging white cloudy spray, well defined and readily distinguished. Steam, on the other hand, which issues from the holes above the water line, is only faintly visible and is hardly discernible. The steam jet is almost parallel and quite distinct in character from that of the water; a definite indication is therefore assured. The character of the water jet bears a distinct relation to the thickness of the wall of the tube; by reducing the thickness of the tube, the jet can be made to assume wider proportions. Good results are obtained by thinning the tube down to $\frac{1}{32}$ in. round a $\frac{1}{32}$ in. diameter hole. Much smaller holes than these, however, will give a perfectly visible jet.

PRESSURE GAUGES.

Apparatus for Testing Pressure and Vacuum Gauges.

(Brotherhood's Patent.)

This apparatus will test the accuracy of gauges at all steps on the scale instantaneously, independent of temperature variation, and does not rely on springs. A permanent accuracy

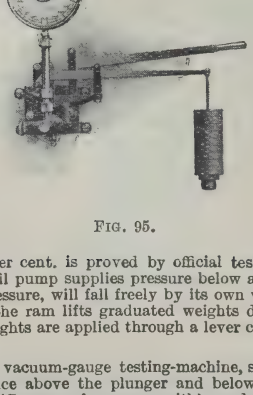


Fig. 95.

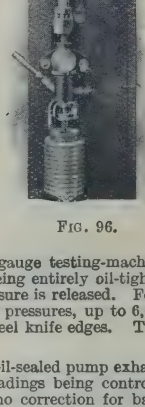


Fig. 96.

within ½ per cent. is proved by official tests. In the pressure-gauge testing-machine a hand-operated oil pump supplies pressure below a ram which, while being entirely oil-tight under the highest pressure, will fall freely by its own weight when the pressure is released. For moderate pressures the ram lifts graduated weights direct, and for higher pressures, up to 6,000 lbs. per sq. in., weights are applied through a lever carried on hardened steel knife edges. This is shown in fig. 95.

In the vacuum-gauge testing-machine, shown in fig. 96, an oil-sealed pump exhausts the air in the space above the plunger and below the gauge. The readings being controlled by the absolute difference of pressure within and without the gauge, no correction for barometric or temperature conditions is needed. In this respect the machine possesses great advantages compared with the mercury column often employed.

The machines can be made for fixing to the wall or to stand on a shelf or table, and are enclosed in dust-proof boxes.

A certificate of the National Physical Laboratory can be supplied with each machine.

THERMOMETERS.

NEGRETTI & ZAMBRA'S DISTANCE THERMOMETER.

(Patent.)

In all superheaters and boiler work, the registration of temperature is very important, and not only should the thermometer be accurate, but it should be legible and easy to read. The instrument, fig. 97 (*Negretti & Zambra*, 38 *Holborn Viaduct*, *London*, *E.C.* 1), which is claimed to effect the desired result, is actuated by distilled and purified mercury contained in a steel bulb connected by means of steel capillary tubing to a Bourdon tube to which is attached a pointer spindle. The special feature in the apparatus is the compensation of the capillary tubing, whereby any changes of temperature of the tubing itself do not affect the accuracy of the reading; an invar wire is run down the centre of the bore of the tubing so as to make the amount of mercury in the tube to be in the same proportion to the size of the steel tube as the expansion of mercury is to the expansion of steel: thus the tubing can be made red hot and any expansion of mercury will be taken up by the tube itself. Another important feature is the construction of the Bourdon tube which enables the pointer to be attached direct to the tube itself, thus eliminating all gears, hair-springs, levers, etc.

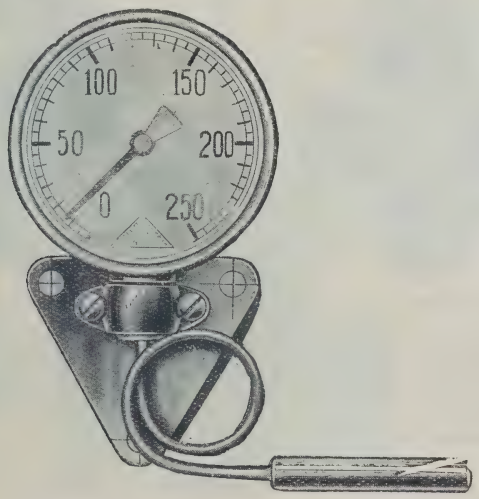


FIG. 97.

The whole arrangement combines strength, legibility, sensitiveness, reliability, and accuracy. Being a dial pattern with a pointer, it is more accurately and easily read than a liquid column in a glass tube; also it is more robust and less liable to accidental breakages such as might occur in a glass bulb instrument.

The dial may be fitted in a position where it is most conveniently read (up to 100 ft. away from the bulb), and the range of scale may be practically any number of degrees on the dial or chart up to 1,000° F.

A transmitting dial movement may be applied to a recorder to give automatic written records of temperature changes at a distance.

The patentees guarantee an accuracy to within 1 per cent. of the actual temperature.

Stop-Valves.

The Hopkinson-Ferranti Patent Stop Valve.

This valve (fig. 98) is designed to pass an amount of steam equal to the full carrying capacity of the pipe. The working parts of the valve are similar to those of the Patent Parallel Slide Valve, and are placed in the throat of a converging-diverging nozzle. The construction allows a large reduction in size, weight, and first cost, as compared with ordinary full-way valves.

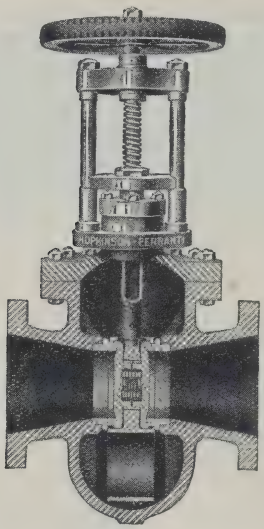


Fig. 98.

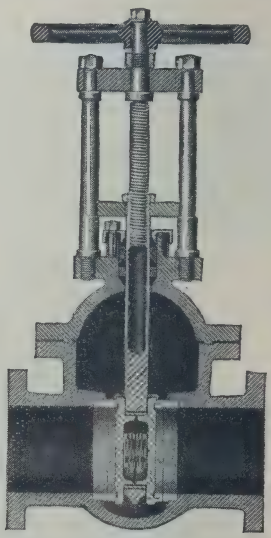


Fig. 99.

The design is such that a part of the pressure energy of the steam is converted into kinetic energy in the converging nozzle, the steam increasing in velocity and dropping in temperature and pressure. With the high velocity thus acquired the steam passes through the valve proper and enteres a diverging nozzle in which the kinetic energy is reconverted with an increase of pressure and temperature. The steam thus leaves the valve at the pressure, temperature, and velocity with which it entered the converging cone on the inlet side, the loss of pressure under normal conditions being negligible.

Hopkinson's Patent Fullbore Parallel Slide Valve.

This valve (fig. 99) is fitted with ' Platnam ' seats and discs which do not corrode or soften under high temperature of superheated or high-pressure steam.

There is no wedging action to cause unnecessary or dangerous strains or to make it difficult to open and close.

The valve when closed is held to its seat by fluid pressure and in opening and closing slides upon the seats with an equally uniform flexible pressure. The discs rotate on their axes, ensuring even wear and perfect condition of the valve faces. This valve is free from the mechanical strains usual in mushroom or wedge valves, being also fluid tight and easy to operate under varying pressures. It is fitted with Hopkinson's composite gland, which eliminates spindle gland leakage.

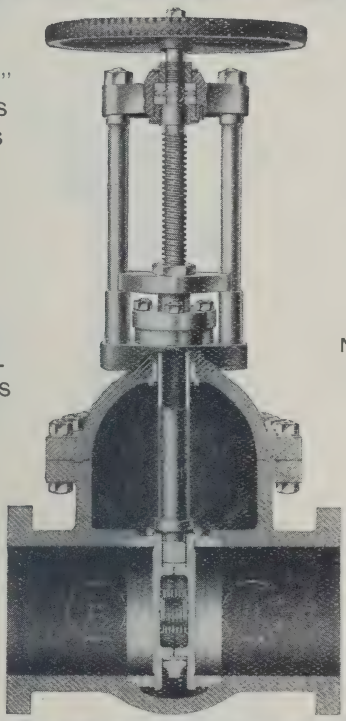
The Hopkinson-Ferranti Valve for steam is made in all the standard sizes from 4 inches to 24 inches diameter, with cast-iron or cast-steel body and cover in accordance with the pressure and temperature, and with ' Platnam ' disc and seats in all cases. Steam can be passed in either direction as required, the old pattern with unequal nozzles having been superseded. The principle is also applied to valves for water and air, and a number of special patterns are manufactured.

HOPKINSONS'

PATENT
PARALLEL-SLIDE VALVES

"PLATNAM" DISC FACES AND SEATS

FOR HIGHEST PRESSURES AND TEMPERATURES



EASILY OPERATED UNDER ALL CONDITIONS

NO WEDGING ACTION

ALL PARTS EASILY ACCESSIBLE

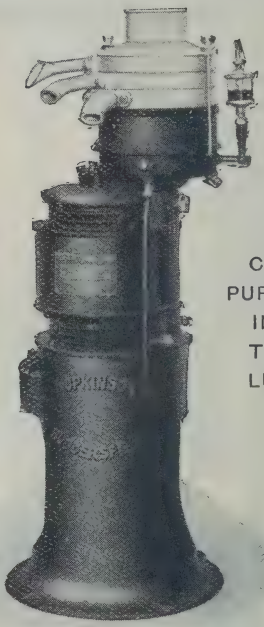
GLANDS CAN BE PACKED UNDER PRESSURE

HOPKINSONS
LIMITED,
HUDDERSFIELD.

LONDON.
34, NORFOLK STREET, STRAND, W.C.2.

HOPKINSONS'

CENTRIFUGAL
OIL PURIFIERS



MADE IN STATIONARY AND PORTABLE TYPES

FOR CONTINUOUS PURIFICATION OF INSULATING, TURBINE AND LUBRICATING OILS, Etc.

WRITE FOR DESCRIPTIVE LITERATURE

HOPKINSONS

LIMITED,
HUDDERSFIELD.

LONDON.
34, NORFOLK STREET, STRAND, W.C.2.

The 'Vee-Reg' Valve.

This valve, fig. 100, has a cylindrical sliding guide part (containing specially arranged triangular openings) that fits into the seat, the face of which is protected. As the valve is raised, there is no appreciable passage of steam or fluid until the faces are well apart, so when steam does pass

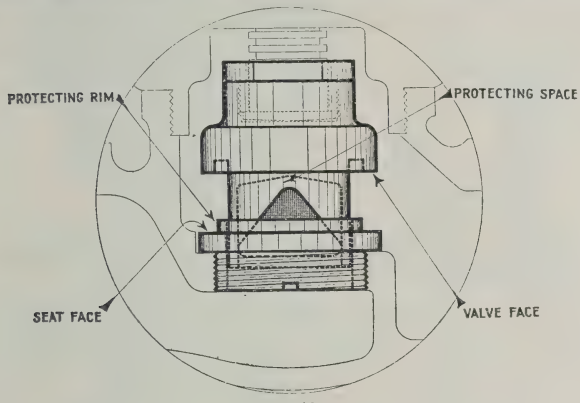


Fig. 100.

it is deflected from the faces. Thus the possibility of 'scoring' taking place is reduced to an infinitesimal degree. In addition, the triangular openings ensure a very fine control—a distinct advantage. The combination of the prevention of 'scoring' and the fine control ensures a very efficient and, incidentally, economical valve service. There are grades of these valves for all pressures from the lowest to the highest.

The Klinger Valve.

(*Richard Klinger, Ltd.*, 120 *Southwark Street, London, S.E.* 1.)

This valve (which is of quite novel design), fig. 101, consists of a piston sliding through two flexible rings, generally of asbestos. A close contact on the piston is maintained when the rings wear, by tightening a form of gland in which the rings are held, but the rings are constructed so that no amount of pressure from this direction can displace them.

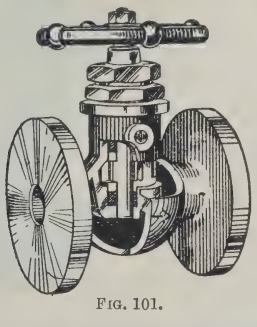


Fig. 101.

The permanent contact provided by the upper ring cuts off the working parts from the valve chest, avoids the necessity for the ordinary type of spindle stuffing box, and provides a perfectly steam-tight joint with very much less friction.

The piston when lowered into and through the bottom ring closes the valve, and is of such a length that wear due to wire-drawing action takes place on the portion that protrudes, while the closing surfaces are always kept out of the steam area.

The rings have been found in actual practice to last much longer than nickel or other seatings and, of course, they are very much cheaper.

The valves are used with superheated and saturated steam, hot oils, motor spirit, water, air, gas, and chemicals, and have been found in the various materials in which they are made to be very suitable for these duties.

The principal points in favour of the valve are : extremely low maintenance costs, simplicity of adjustment, and absolute tightness under the hardest conditions.

These Seatless Piston Valves can now be made up to 10-in. bore.

Valve Refacers.

The Simplex Patent Valve Reseating and Facing Machine.

(W. Crockatt & Sons, Ltd., Glasgow.)

(1)

This tool, fig. 102, has been designed to avoid the necessity of using the studs in the valve chest as a means of steadying. At the same time it is simpler and stronger than the machines which are secured and centred by means of a chuck operated by a scroll.

The tool is fixed in the valve chest (which may be either plain or screwed) by means of three stepped arms, pivoted at their upper ends. These arms are expanded by a cone which is depressed downwards by means of the hand nut. The action of expanding these arms both centres the tool in the valve and fixes it securely in position. This is ensured by the manufacture of the tool, the arms being turned up truly central with the spindle, after assembling. The cutters used are made of flat steel, and one end is arranged for flat and the other for bevelled seats. The simplicity of this type of cutter is found to counterbalance any slight advantage the milled type of cutter may have. Also a very much smaller number of cutters can be employed to deal with a large range of valves, as each cutter is adjustable. In order to prevent chattering of the tool, the feed is against the action of a strong spring, which keeps an upward pressure on the tool, so that feeding only takes place when the cutter is forced down by the action of the nut. This prevents the cutter following any irregularities in the face of the seat.

A simple attachment is provided when desired for trueing up the valves themselves. The valves are centred in this part of the tool and faced up by a movable tool, which can deal with either flat or bevelled valves. The arms and lower part of the body of the tool are made of cast steel, so that the machine will stand hard wear. Ball-bearings are fitted throughout, to ensure smooth working and to avoid jarring.

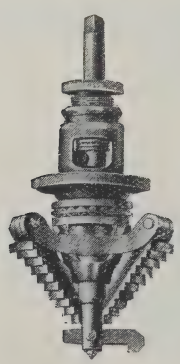


Fig. 102.

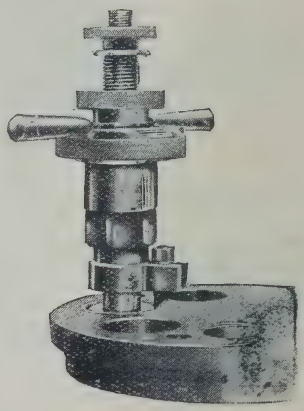
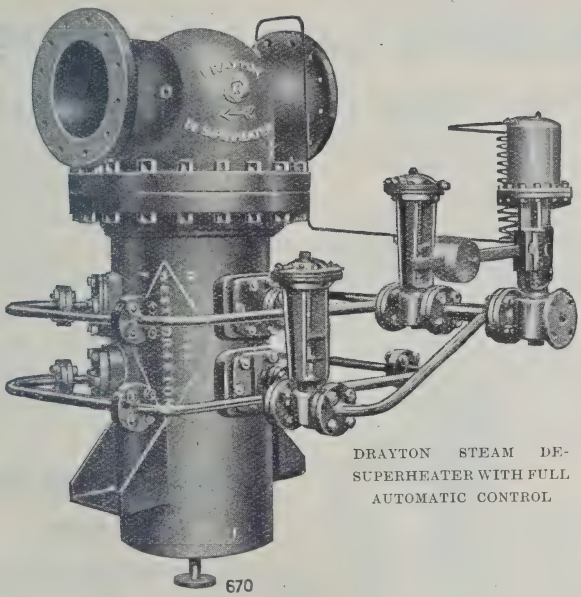


Fig. 103.

The machine is made in three standard sizes to deal with valves from $\frac{3}{4}$ in. to 8 ins. diameter, but has been supplied and can be easily adapted to do valves up to 12 ins. or more.



DRAYTON STEAM DE-SUPERHEATER WITH FULL AUTOMATIC CONTROL

670

DRAYTON AUTOMATIC REGULATORS & RECORDERS FOR TEMPERATURE, PRESSURE, HUMIDITY & LIQUID LEVEL

There are no industrial requirements for automatic control of temperature, pressure, humidity, and liquid level which cannot be adequately met from the Drayton range of manufactures. Years of specialisation in this branch of engineering have enabled the Company to develop a unique series of Regulators for all purposes.

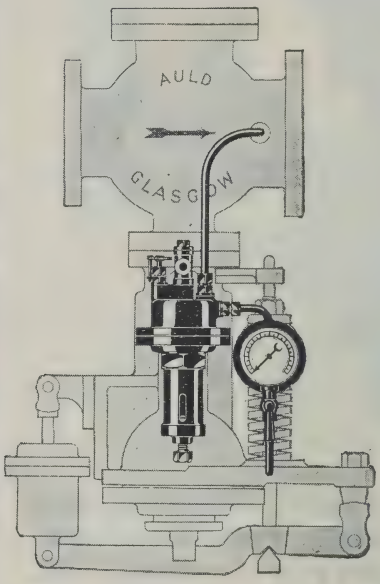
The DRAYTON REGULATOR AND INSTRUMENT CO. LTD.,

WEST DRAYTON, MIDDLESEX, ENGLAND.

F. 1670

Auld's reducing Valves

FOR ALL PURPOSES, PRESSURES AND SUPERHEATS



The valve illustrated is our latest patent RELAY OPERATED REDUCING VALVE which is specially suitable for HIGH PRESSURES and EXTREMELY ACCURATE REGULATION.

The ideal valve

for PRESSURE CONTROL to PROCESS PLANTS

Also makers of: STANDARD TYPE (CLASS A) QUITETITE and FULL BORE REDUCING VALVES

CATALOGUES ON REQUEST. TELEGRAMS: "REDUCING."

SOLE PATENTEES AND MANUFACTURERS:

DAVID AULD & SONS, LTD.

WHITE VALE FOUNDRY,

GLASGOW, S.E.

F. 1671

(2)

In order to meet the demand for a tool which will face up the seats of the group valves in Weir's and similar feed pumps, the Simplex No. 3 machine has been introduced. This tool, fig. 103, is secured in position by the expansion of three segments, which are turned concentric with the central spindle, and which are expanded hard against the inside of the bore of the valve seat by the action of the tapered end of the spindle. It can easily be adapted to flat or bevelled seats by simply changing the cutter used.

If it is desired to face up the valves in the bottom seat of the pump without drawing the complete seat, a special tool with an extended body can be supplied.

Besides being suitable for the type of valve mentioned, the tool is equally adaptable to dead-weight safety valves, or, indeed, to any design of valve in which the inside of the bore is a plain cylindrical surface. Ball-bearings are fitted to ensure smooth and easy working.

REDUCING VALVES.

These are very needful fittings in connection with steam appliances when steam has to be reduced from a high to a uniform low pressure. It is a common mistake to order reducing valves too large, which leads to their excessive wear. No reducing valve should be implicitly trusted; but wherever a rise in steam-reduced pressure is liable to bring about serious results a relief valve should follow the reducing valve.

Reducing valves which are too large for the amount of steam they are to pass are usually unsatisfactory; either they will open and close continually and chatter violently, or will stand in a position nearly closed and become steam-cut and leaky. It is more satisfactory to use two small valves in parallel, one set slightly heavier than the other, so that one will supply steam up to its maximum capacity before the other opens. Another way is to use one reducing valve equal to about one-half the maximum demand, and a by-pass of about the same capacity with a hand-operated valve in it, so that when the reducing valve reaches its capacity the by-pass valve may be opened wide and the reducing valve be closed down and make up the shortage. This, of course, requires that an attendant shall observe the pressure gauge occasionally.

Auld's Patent Reducing Valve.

(David Auld & Sons, Ltd., Glasgow, S.E.)

Figs. 104-106 represent three of the types of Auld's Patent Reducing Valves, viz., Standard, Quitetite, and Full-Bore types, each of which is suitable for certain requirements.

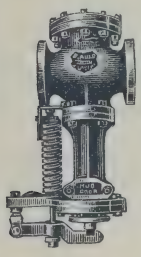


Fig. 104.

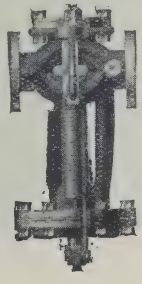


Fig. 105.

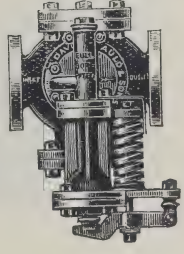


Fig. 106.

The Standard type valve is used for general purposes and is almost universal in steamships. The 'Quitetite' Reducing Valve is particularly suitable for great pressure reductions and also for heating and cooking installations. The Full-Bore type valve is desirable for dealing with large steam capacities, as in collieries, iron and steel works, etc. A further type of reducing valve is Auld's Patent Relay-Operated Steam Reducing Valve. These valves are all self-contained, operating automatically immediately the pressure is turned on; no external water or other pressure being necessary for their working.

Safety valves and pressure gauges can be fitted in combination. The advice of the makers should be obtained for guidance in the proper selection of the most suitable types for different purposes.

FEED-WATER REGULATORS.

The Crosby Patent Feed-Water Regulator.

The action of this regulator (*The Crosby Valve and Engineering Co. Ltd.*, 41–42 *Foley Street*, *London*, *W.*1) is in no way dependent upon the rising and falling of a float, and there are very few parts which are liable to give trouble and so to cause the apparatus to work in an erratic manner.

In fig. 107, A is a chamber which is put into communication with the steam space of the boiler, the steam being led in through the pipe H. A pipe L also puts the chamber in connection with the boiler, but in this case with the water space. If, for example, the apparatus is fitted to a Babcock boiler it might be placed at the side of the drum, so that the top of the chamber A is in line with the required water level. If the pipe L be then connected to the drum G, about

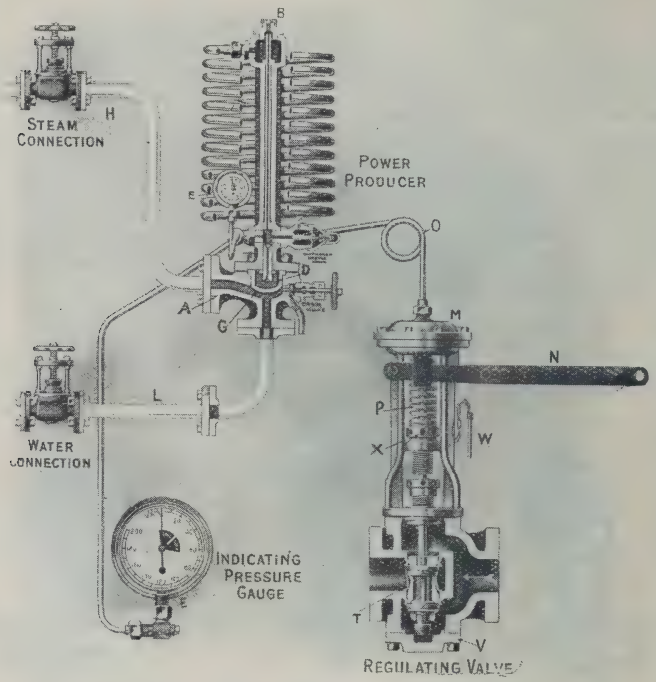


FIG. 107.

12 ins. below the water line, and the pipe H to the top of the drum, the level of the water in the chamber A will be dependent upon the level of the water in the boiler. When the water in the boiler rises, that in the chamber A will rise, and when this chamber is full of water steam will be prevented from entering from the pipe H. When the water in the boiler falls it will also fall in the chamber A and steam can enter, which causes an immediate rise in pressure in the system above the chamber A due to the latent heat transfer through the heating surface G.

The part of the apparatus above the chamber A is partly filled with water and is connected to the chamber M of a regulating valve by the pipe O, which acts as a cold water column. When the apparatus is used for regulating the supply of water to boilers working on a common steam main, the double-beat balanced valve T is connected in circuit with the feed pipe, each boiler being provided with one of these regulating valves.

There is a diaphragm in the chamber M, this being connected to the double-beat balanced valve by means of a spindle. When the water level in the boiler falls, the pressure formed in the apparatus as described below is exerted on the diaphragm, causing the valve to open against the action of the spring. When, however, the water rises, this pressure falls and the spring causes the valve to close, thus shutting off the supply of water to the boiler.

When the apparatus is first put into use, both the cocks on the boiler are closed, so that neither steam nor water can enter the chamber A. The air valve B is removed, and water is poured in. At the same time the union, or plug, on the diaphragm chamber is loosened, in order to enable the water to drive out the air in the pipe connecting the two pieces of apparatus. This having been done, the union on the chamber M is again tightened up, and more water is poured into the chamber at B when the air valve is closed. Steam from the boiler is then admitted under the partition G and the water in the system above is boiled up, the air valve B being loosened to let the air escape. This process is repeated several times, and on each occasion the apparatus is allowed to cool. In this way a vacuum is created in the apparatus above the partition G, and when a gauge to the left of the upper chamber registers a vacuum of at least 10 ins. the device is ready for use. The gauge serves for registering both pressure and vacuum, the latter being to the left of the zero mark and the former to the right of it.

The vacuum having been obtained, the two cocks on the boiler are opened, and the device commences to work. When the water in the boiler falls and steam is admitted to the chamber A through the pipe H, the heat of the steam is transmitted through the plate G. This heat boils the water in the chamber D. The pressure of the steam so generated, acting on the surface of the water in the top chamber, gives rise to a pressure above the diaphragm in the chamber M. This diaphragm is depressed, and the valve (which in a battery of boilers is arranged in the feed-pipe to each boiler) is opened, a supply of feed-water is admitted to the boiler, and in this way the water level is raised. As the water rises in the drum of the boiler, it also rises in the chamber A, and when the proper level in the boiler has been reached, steam ceases to enter from H. The chamber A then contains nothing but water. Under these conditions the water in the chamber D is no longer evaporated, and the steam condenses, the pressure in the top chamber consequently falls, and the valve closes. The supply to this particular boiler then ceases for the time being. The pressure gauge indicates whether the apparatus is working properly, and in this way the stoker is provided with a means of guarding against low water even should the device get out of order, which, however, is not at all likely to occur.

When the apparatus is working properly the pressure rises and falls slightly in accordance with the fluctuations in level in the boiler, opening the regulating valve wider or allowing it to close, thus increasing or reducing the feed supply to the boiler. It does not constantly open and shut, but regulates the feed so as to keep the water level constant within close limits. A lever is provided on the regulating valve for testing its operation or to keep the valve open if desired, and an adjustable device is provided to keep the valve always open a limited amount if required.

The principal advantages claimed for the apparatus are the following:—(1) It enables a lower water level to be maintained with perfect safety, thus increasing the steam space in the boiler and ensuring a supply of dry steam; (2) it reduces coal consumption by reason of maintaining a regular feed proportionate to the evaporation of the boiler; and (3) it ensures a greater regularity in the steam-raising, and so tends to prevent fluctuations in the pressure; (4) it will work continuously without requiring attention or repair.

A desirable adjunct to the Boiler Feed Regulator is the Crosby Automatic Feed Pump Control. This regulates the speed of the Boiler Feed Pump in accordance with the demands of the feed line to the boilers; and consists of a balanced valve in the steam supply pipe to the pump, controlled by means of a device which maintains the pressure in the feed line a constant amount above the boiler pressure irrespective of the demand. The magnitude of this excess pressure is adjustable at will.

The 'Drayton' Feed-Water Regulator.

(*The Drayton Regulator & Instrument Co., Ltd., West Drayton.*)

This regulator, fig. 108, is of the 'delayed action' thermostatic type, the lag of the thermostat allowing the stoker to adapt the firing to altered conditions of steam consumption. The all-metal thermostat is contained in a steel tube which lies at an angle outside the boiler across the normal water-level line; the highest point of the tube corresponding to the highest, the lowest point to the lowest permissible water level. The upper end of the thermostat container is connected to the steam space, the lower end to the water space of the boiler. The thermostat actuates through a magnifying lever a small relief valve, which latter is connected to the

pressure chamber of the feed valve, into which water enters at 15 lbs. per square inch pressure through a strainer and a needle valve. With rising water level the thermostat contracts, closes the relief valve, and allows the feed valve to close; with falling water the tube fills

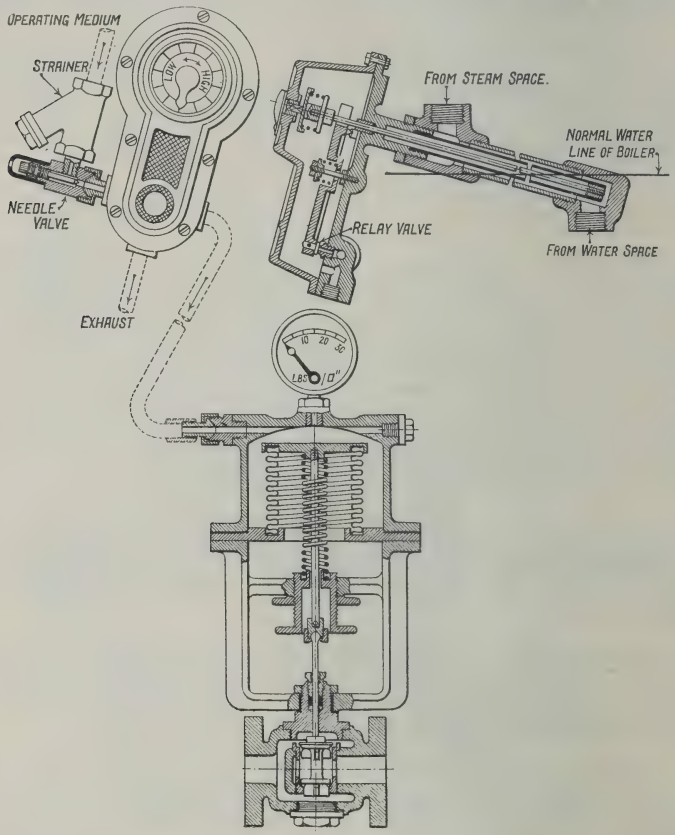


FIG. 108.

with steam, the thermostat expands, opens the relief valve, and causes the feed valve to open gradually.

THE 'THERMOFEED' FEED-WATER REGULATOR.

(Patent.)

(*Ronald Trist & Co., Ltd.*, 56 *Victoria Street, Westminster, S.W.* 1.)

The maintenance of a constant water level in a boiler cannot be too strongly emphasised, and the difficulty in keeping even an approximate level in boilers using pulverised fuel with a fluctuating load is large. The 'Thermofeed' keeps the level with a far higher degree of accuracy than can possibly be kept by hand, even with a man having no other duty than to keep his hand constantly on the feed check valve. The device is entirely mechanically operated, and being independent of temperatures or evaporations, possesses the following advantages:—

(*a*) Instantaneous in operation: (*b*) assists in preventing priming; (*c*) increases thermal efficiency.

The regulator is easily adapted to any existing boiler and methods of erecting are shown. Fig. 109 shows a 'Thermofeed' fitted to one of a battery of boilers controlling a valve placed in the branch feed pipe. Fig. 110 shows a similar machine where the valve is placed in the steam pipe to the pump. This is the method adopted where there is one boiler only fed by its own pump.

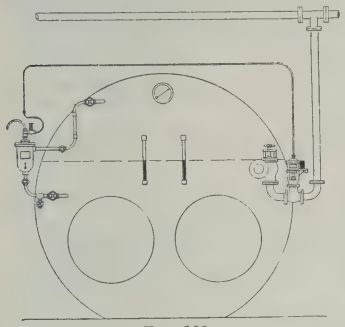


FIG. 109.

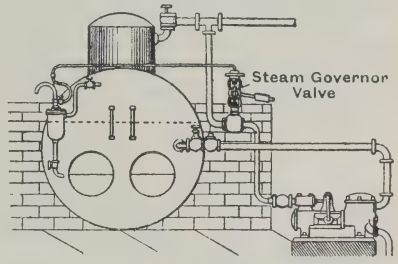


FIG. 110.

The 'Thermofeed' consists of two units—a 'Float Chamber,' Fig. 111, and an 'Automatic Valve,' Fig. 112, operates as follows:—

Within the float chamber is a float containing a small quantity of water which is converted nto steam at the same pressure as that surrounding it. This float is, therefore, non-collapsible

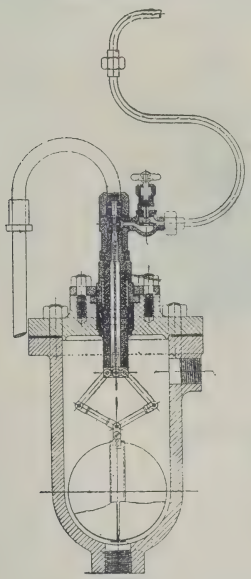


FIG. 111.

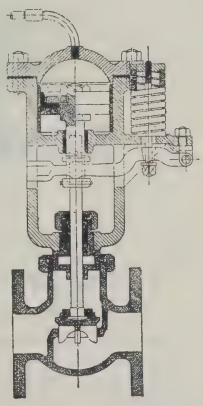


FIG. 112.

and rises and falls with the water level which corresponds to that in the boiler, operating a small three-way valve by means of toggle levers and a rod.

The small valve referred to is in a vertical position and so far removed from the water level as to be kept entirely free from any sediment or dirt.

If the water level is low the float is down, and due to the toggle levers reversing the motion, the small three-way valve is pushed on to its bottom seat. Steam, which is present under the bottom of this valve, can get no farther. As the level rises the float gradually changes over this small valve so that it is drawn on to its top seat and the bottom seat is open. Steam passes in at the bottom seat, through the copper connecting pipe between the two units, and gradually depresses the piston of the automatic regulating valve, and so closes the valve. As the water level falls the reverse action takes place, and any steam or water present on top of the piston of the automatic regulating valve is allowed to pass to atmosphere.

An adjusting nut is provided on the machine so that the actual working level may be varied slightly after the machine is erected. Provision is also made to put the machine instantly out of action by means of a small valve in the copper connecting pipe.

In addition to the advantages set out above, it is also claimed that (where all the latest devices are not installed) the 'Thermofeed' will effect a considerable saving in fuel consumption and eliminate the employment of labour.

The 'Drayton' Compensating Pressure Regulator.

(The Drayton Regulator and Instrument Co., Ltd., West Drayton.)

The Drayton Regulator & Instrument Co.'s Compensating Regulators are regulators in which the regulating device, generally a hydraulic or pneumatic pressure cylinder (which actuates valves or dampers), after it has carried out certain movements inhibits the impulse and the actuated device only continues its movement if a fresh impulse is given, *i.e.*, if the first regulation has not had the desired effect.

Different operating conditions may require considerable variations in the amount of regulation effected by the impulse, and for the purpose of setting the regulators for specific conditions, the 'Drayton' Compensating Pressure Regulator is made adjustable.

The 'Weir-Contraflo' Feed-Water Regulator.

(G. & J. Weir, Ltd., Cathcart, Glasgow.)

The action of this regulator (fig. 113) is as follows:—When the float J falls, due to a fall in the boiler water level, the needle H rises, cuts off the flow of water from the piston chamber D through

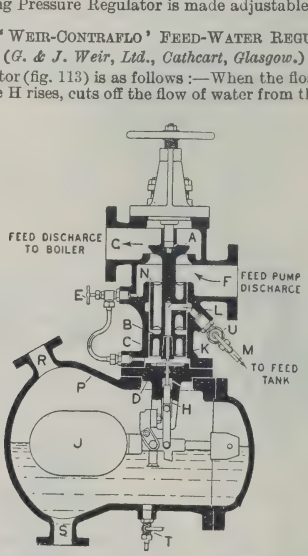


Fig. 113.—'Weir-Contraflo' Boiler-Feed Regulator.

the opening K to the feed tank, and the pressure in the chamber D rises, due to the leakage of water into the chamber through the valve E and the clearance between the piston B and the cylinder C. Immediately the water pressure in the chamber D rises above the pressure in the boiler, the valve A lifts and allows feed-water to pass into the boiler. The valve A can only rise by the amount that the needle H has been raised, because immediately the opening K is uncovered the pressure under the piston B falls below the pressure in the boiler and the valve A commences to close.

When the water level in the boiler rises, the needle valve H is lowered by the action of the float, and the opening K is uncovered and allows water to escape from the chamber D through the opening K to the feed tank. The pressure in the chamber D then falls below the pressure in the boiler, and the valve A falls to the same extent as the needle.

It will be seen that the valve A and the piston B are constrained hydraulically to move in the same direction and to the same extent as the needle H, which is operated by the float; and for any given position of the needle H the valve A is maintained in equilibrium in a corresponding position.

The working water level in the boiler varies from a top level, when no steam is being generated in the boiler, to a bottom level when the boiler is operating at its maximum rate of evaporation, and the feed regulator maintains a steady water level between the top and bottom water levels for any given rate of evaporation between no load and the maximum load.

The 'Arca' Automatic Regulator.

(*British Arca Regulators, Ltd., London, S.W.* 1.)

This regulator (which can be applied *universally* for the control of pressure, temperature humidity, electricity, etc.) is very effective for steam regulation; it is fully standardised, and consists generally of an operating hydraulic cylinder capable of exerting ample regulating force, and a very sensitive relay control which is mounted either on the cylinder or on a separate wall panel. The design is compact and is very easy to install.

Water is used as the operating medium, and the relay is so sensitive that any kind of impulse to which it is subjected, however slight, is transferred through the hydraulic cylinder into a considerable force sufficient to operate valves or other mechanisms.

The principle upon which the Arca regulator works is that of using a sensitive relay device whereby hydraulic power of any desired magnitude is called into action, and exerts a regulating force on the valves or other mechanisms whose adjustment requires to be maintained at a particular value.

The relay device contains an 'impulse receiver' which is connected to a pivoted lever in such a way as to transmit to it any impulse due to any slight change in the value required to be maintained. One end of this lever impinges on a small jet of water, and its movement is arranged to suppress more or less the free flow of water from the jet, thus causing a difference of pressure in the pipe supplying the jet.

This pressure difference is used and multiplied on a diaphragm directly connected to a small pilot valve, which in turn controls the supply of water to the hydraulic cylinder operating the regulating valve or mechanism.

The action is instantaneous, due to the fact that the relay is practically free from friction, and that a non-compressible liquid, usually water, is the medium for transmission of force to the regulator. The essential regulating feature is therefore the jet of water, which, by being partially choked or allowed to flow freely according to the movement of the pivoted lever, causes movement of the pilot valve, and thereby of the operating piston in the hydraulic cylinder.

The amount of water at any time escaping from the jet is very small, so that the water consumption is of little importance. The water pressure may vary within wide limits, and variation during working does not affect the efficiency of the regulator in any way.

The usual range of pressure within which the regulators are generally designed to operate is from 8 lbs. to 80 lbs. per sq. in. A simple and effective self-cleaning filter forms an integral part of the relay device to ensure absolute freedom from impurities that may be found in the water, but if the operating water is impure it is recommended that a coarse filter be inserted in the water supply pipe close to the regulator.

This water supply pipe is usually $\frac{1}{2}$ in. diameter.

A variation in pressure amounting to only 2 ozs. per sq. in., or in some cases a variation of $\frac{1}{50}$ in. water column, is sufficient to cause the regulator to act.

The Mumford Patent Automatic Feed Water Regulator.

(*A. G. Mumford, Ltd., Colchester.*)

The Mumford Boiler Feed Water Regulator consists of two essential parts (see fig. 116, p. 1679):—a float box, with its needle valve I, and a special check valve A, which can be fitted to any boiler as separate units; all parts of the regulator being outside the boiler, they are readily accessible for examination and overhaul. All parts of the float gear can be examined even when the boiler is under steam.

The float controls the needle valve, which regulates the check valve, thereby regulating the supply of water from the feed pump to the boiler.

The lift of the check valve is regulated as follows: The check valve is provided with a balance piston (B) at the lower end, which is machined with sufficient clearance to allow a small leakage of water past it under the pressure of the feed pump. The float and its needle valve control this leakage.

When the water level is maintained, the needle valve is open and allows the leakage water to pass freely to the feed tank, in this case the check valve is in equilibrium as regards the feed pump pressure, the boiler pressure keeping it shut. When the water level drops the needle valve closes, the pressure on the top and bottom of the balance piston then becomes equal, and the valve opens as a normal check valve until the level is restored.

The 'Arca' Compact Valve.

This valve (fig. 114) is used for small steam, air, or water regulating valves where the hydraulic cylinder is unnecessary and where only comparatively small power is required to operate the valve; it is operated by a diaphragm in precisely the same manner as the pilot valve is operated where a hydraulic cylinder is used. Great care has been taken in the design to ensure the same

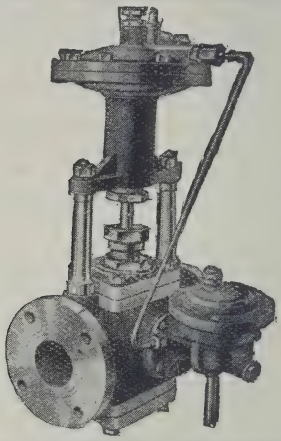


Fig. 114.

accuracy and reliability as with the cylinder type. The cost of the combined unit, embodying relay and compact valve, is of course considerably less than the cylinder type, though possessing the same sensitiveness and accuracy. It is particularly suitable for use with steam for heating purposes.

Copes Feed Water Regulator

(*Copes Regulators, Ltd.*, 231 *Strand, London, W.C.* 2.)

consists of a thermostat, the actuating element, and a balanced valve controlling flow of boiler feed water. A rigid strut connects thermostat and valve. The thermostat consists of an

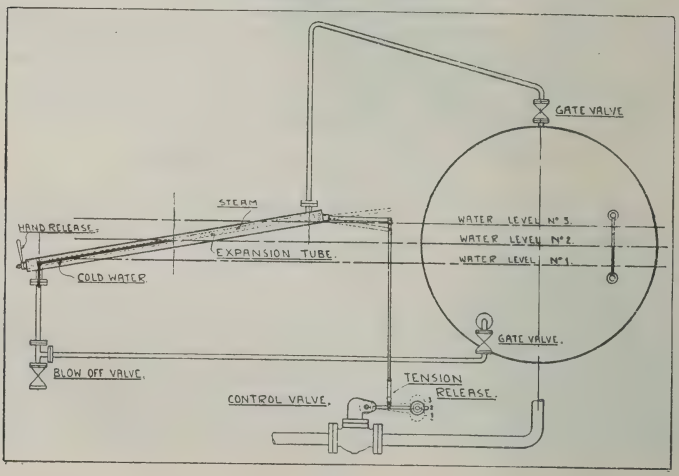


Fig. 115.

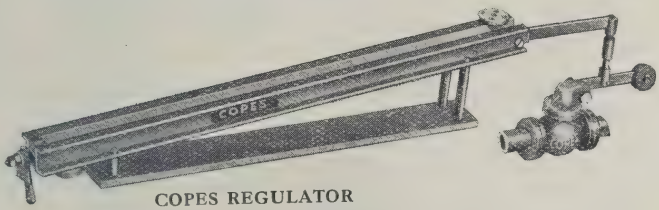
COPES BOILER FEED REGULATORS

Simple! Reliable! Economical!

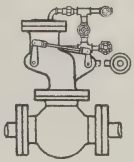
The Standard Regulator—used on

Over 33,000

Boilers Throughout The World



COPES REGULATOR
WITH TENSION TYPE THERMOSTAT



COPES DIFFERENTIAL TYPE
PUMP GOVERNOR

DESIGNED AND MANUFACTURED BY
SPECIALISTS

CONSULT OUR ENGINEERS ABOUT YOUR
BOILER FEED PROBLEMS

COPES REGULATORS LTD.

231 STRAND, LONDON, W.C.2.

Telegrams : Copereg, Estrand, London *Phone :* City 4441

F. 1678

STANDARD WORKS

Surveying as Practised by Civil Engineers and Surveyors

Including the Setting-out of Works for Construction and Surveys Abroad, with many Examples taken from Actual Practice. A Handbook for Use in the Field and the Office, intended also as a Text-book for Students. By JOHN WHITELAW, A.M.Inst.C.E. Eighth Edition, thoroughly Revised and Enlarged, by Colonel Sir GORDON R. HEARN, C.I.E., D.S.O., Assoc Inst.C.E., and late Chief Engineer, Indian State Railways. 594 pages. 297 Illustrations. Demy 8vo, cloth *Net* **16s.**

Reinforced Concrete Bridges

The Practical Design of Modern Reinforced Concrete Bridges, including Notes on Temperature and Shrinkage Effects. By W. L. SCOTT, M.Inst.C.E., Chief Engineer, Considère Constructions, Ltd. Third Edition, Revised and Enlarged. 280 pages. 140 Diagrams. 33 Photographic Plates. Royal 8vo, cloth ... ... [*Nearly ready.*

Water Supply of Towns and the Construction of Waterworks

A Practical Treatise for the use of Engineers and Students of Engineering. By W. K. BURTON, A.M.Inst.C.E., &c. Fourth Edition, Revised and Enlarged, by J. E. DUMBLETON, A.M.Inst.C.E., Author of "Wells and Bore-holes for Water Supply." In Two Volumes. Vol. I.—**Collection and Purification Works**; Vol. II.—**Works for Distribution.** With Folding Plates, Diagrams, and Illustrations. Crown 4to, cloth Per Vol. *Net* **25s.**

Water Channels

By G. HIGGINS, M.C.E., University of Melbourne, M.Inst.C.E., M.Am.Soc.C.E., formerly Lecturer on Civil Engineering at the University of Melbourne. 144 pages. Illustrated by 32 Figures, 7 Diagrams, and 16 Plates. Demy 4to, cloth ... ... *Net* **21s.**

Mechanical Handling and Storing of Material

Being a Treatise on the Automatic and Semi-Automatic Handling and Storing of Commercial Products. Third Edition, thoroughly Revised. By GEORGE FREDERICK ZIMMER, A.M.Inst.C.E., with a Foreword by Sir JOHN PURSER GRIFFITH, M.A.I., M.R.I.A., Past-President of the Institution of Civil Engineers. 820 pages, with over 1,100 Illustrations (including numerous Folding Plates) specially prepared for the Work. Super royal 8vo, strongly bound in Buckram ... ... *Net* £3 3s.

CROSBY LOCKWOOD & SON
Stationers' Hall Court, London, E.C. 4.

F. 1679

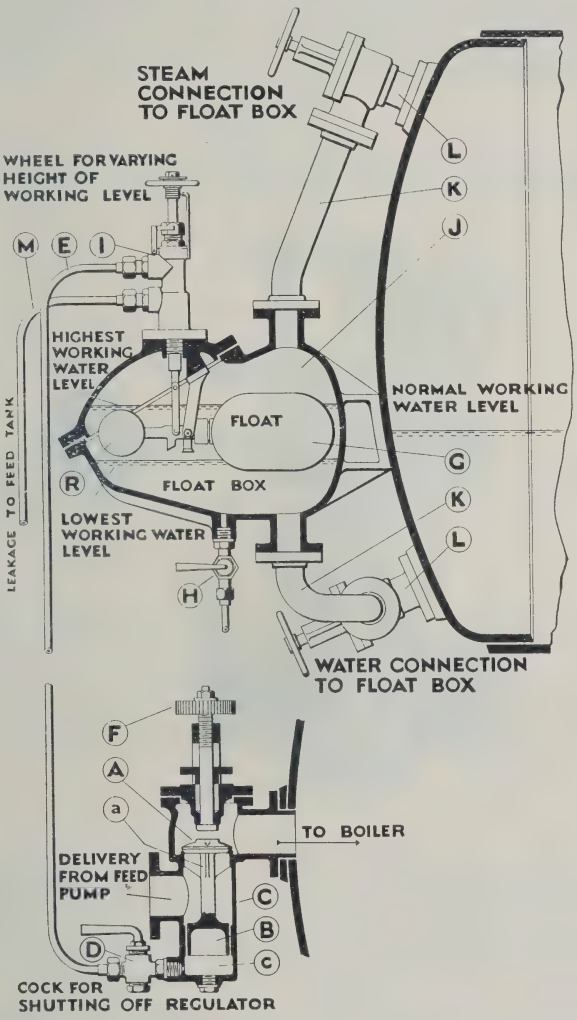

STEAM CONNECTION TO FLOAT BOX
WHEEL FOR VARYING HEIGHT OF WORKING LEVEL
M
E
I
HIGHEST WORKING WATER LEVEL
LEAKAGE TO FEED TANK
FLOAT
NORMAL WORKING WATER LEVEL
R
FLOAT BOX
G
K
L
LOWEST WORKING WATER LEVEL
H
WATER CONNECTION TO FLOAT BOX
F
A
a
TO BOILER
DELIVERY FROM FEED PUMP
C
B
c
D
COCK FOR SHUTTING OFF REGULATOR

FIG. 116.

expansion tube securely mounted between two steel channel pieces. It is so designed that the expansion tube is always in tension. The tube is welded to the flanged head and heel pieces.

The upper end of the expansion tube (see fig. 115) is connected to the steam space, and the lower end to the water space of the boiler. The same water level exists in the expansion tube as in the boiler; the upper end of the tube being filled with steam and the lower end with water. Movement of the water in either direction causes the expansion tube to act as a thermostatic relay, which actuates the feed water control valve.

The control valve is fully balanced, rugged and practically frictionless, the usual sliding stem moving in a stuffing-box is replaced by a horizontal rotating shaft.

The Copes Pump Governor.

The Copes Pump Governor is an automatically controlled balanced valve intended to be fixed in the steam supply main to the feed pump. The valve is operated by means of levers actuated by the difference of pressure between the boiler pressure and the feed-line pressure. The main steam header pressure (C) is carried to the inside of the syphon bellows. The main water-header

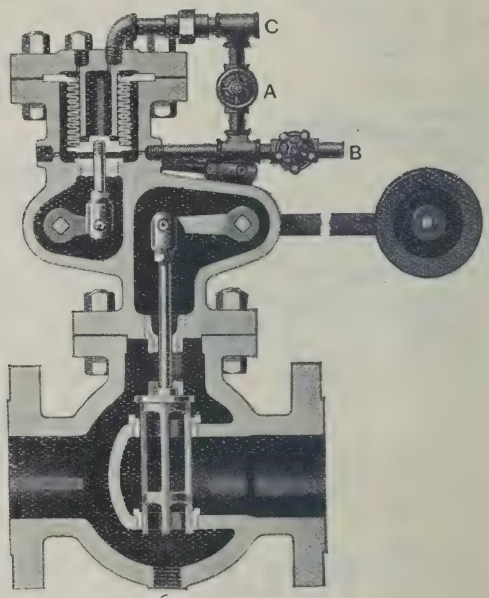


FIG. 117.—Copes Type Pump Governor.

pressure (B) is carried to the outside of the bellows. The position of the weight on the lever arm determines how much the water pressure will be above the steam pressure. The amount of water that will pass through the control valve of the feed water regulator depends on the pressure drop across the valve, as well as upon the area of the feed valve opening.

The governor maintains a fixed excess pressure between steam header and water header by varying the pump speed to correspond with the steam load on the boilers. As the steam pressure varies the water pressure should vary to correspond.

Removing Air from Boiler-Feed Lines.

In the power station of the London Hydraulic Power Company, Wapping, London, there was considerable trouble due to the collecting of air in one of the boiler-feed lines. It was overcome by the device shown in Fig. 118. The water was supplied under a pressure of 160 lbs. and the boilers were in continuous operation.

The device consists of a cast-iron body A and a floating ball B, the ball being an easy fit in the guides on the body. The ball has a wooden core 5⅛ ins. in diameter covered with rubber. Over the body is fixed a cast-iron cover in which is screwed a small nipple C. The inner end of the nipple forms a seat for the ball, and the other end is open to the atmosphere. A perforated splash plate is fitted on the cover to prevent any water that may be blown out being splashed around the plant. The separator is fitted to the pipe line at the highest point in a vertical position. When

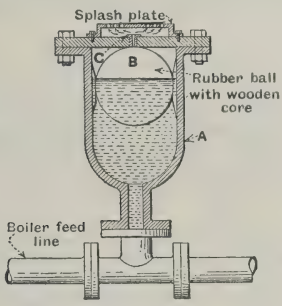


FIG. 118.

free from air, the floating ball rises with the water and seals the small orifice in the nipple. When air is released from the water, it collects under the ball and displaces the water supporting it. When sufficient air has accumulated, the ball drops away from the seat and permits the air to escape. When the air has been discharged, the ball rises again and seals the orifice.

OIL SEPARATORS.

Lubricating oil, if allowed to pass into boiler feed-water, constitutes a danger to boilers so serious that means for its separation and removal are now invariably insisted upon by insurance companies. The method generally employed is that of the mechanical separation and extraction of oil and grease from the exhaust steam before it enters the condenser.

The fitting of high-pressure boilers with superheaters has increased the difficulty of the removal of the oil, while rendering the necessity of the complete separation all the more imperative.

For *small* quantities of greasy water, finely-broken coke placed in a perforated-bottom pail and hung below the greasy drip of an exhaust pipe has been found efficient in separating oil.

THE 'BAKER' PATENT OIL SEPARATOR FOR EXHAUST STEAM.

(The Baker Oil Separator Co., Scarborough.)

This separator is designed to eliminate lubricating oil from the exhaust steam of all classes of reciprocating engines and under all working conditions, varying between 29 inches of vacuum and 75 lbs. per square inch back pressure. The sizes made have pipe connections ranging from 2-in. bore to 60-in. bore. Separation is effected by mechanical means only, the apparatus being fitted into the exhaust steam pipe between the cylinder of the engine and the condenser or other receptacle in which the cleansed steam is to be used. The makers give a definite guarantee as to the results obtained, which in actual practice means that all condensed steam can be used for boiler feeding, or any manufacturing purpose, without fear of oil contamination. The apparatus is extensively used for colliery winding, rolling-mill, and central exhaust engines, special attention being given to the strength and rigidity of all internal parts, to ensure their withstanding the heavy pulsations from these types of engines. In every instance, each Baker Patent Oil Separator is designed and manufactured to suit the particular working conditions specified at time of ordering, and is guaranteed accordingly. It should be noted that the makers' claim is to cleanse the exhaust steam, not to recover the lubricating oil, although in many of the larger installations as much as from 50 per cent. to 75 per cent. of the oil is recoverable for use again.

There are no maintenance costs in running the apparatus.

The 'Drayton' Oil Separator.

(*The Drayton Regulator & Instrument Co., Ltd., West Drayton.*)

This separator differs from other types by the absence of rigid baffles. The eliminators consist of vibrating elements which shake off the oil into an inverted cone-shaped bottom, from where it is removed in the usual way by grease trap or vacuum trap of the makers' own design.

It is claimed that one-tenth of a grain of oil per gallon of condense can be guaranteed without time limit.

BOILER CLEANERS.

The Hulburd Mechanical Boiler Cleaner.

Excellent results have been obtained with the Hulburd mechanical boiler cleaners, apparatus which, although suitable for all classes of boilers, are especially recommended for locomotives, in

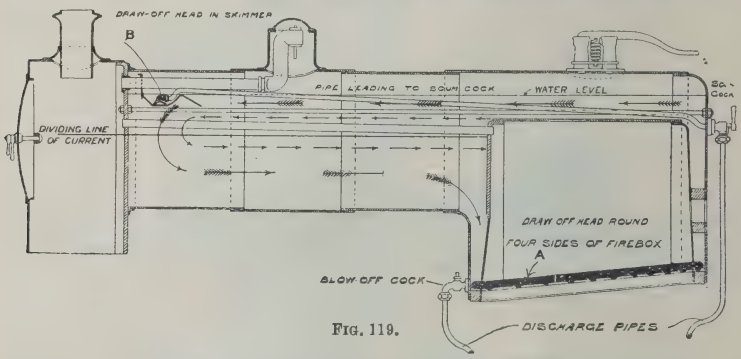


Fig. 119.

which reduced efficiency of the heating surfaces, due to formation of scale and obstruction of the water spaces by accumulations of deposit, are above all things to be prevented if possible.

As almost invariably fitted, the cleaners, fig. 119, consist essentially of two portions : one in the boiler barrel, placed in such a position near the forward end as to collect all scum-forming matter in a kind of skimmer-trough, or basin, to which it is brought by the natural circulation of the water and from which it is discharged, while running, by a suitable cock worked from the cab ; the other in the water spaces surrounding the firebox, immediately above the foundation ring, to evacuate by means of blow-off cocks the sediment which otherwise inevitably accumulates there. Both front and back portions are in the form of draw-off heads having openings or suckers on their undersides spaced so as effectively to scour the whole of the surfaces with which they have to deal, and their efficacy is such that an engine fitted with the apparatus, working on water which may previously have necessitated opening up the boiler and washing out every four or five *days*, can with the Hulburd cleaners be run 5, 6, or even 8 *weeks* without being laid off a single day for cleaning in the ordinary sense.

Among the advantages obtained by the use of the cleaners are :—

1. The immediate stopping of priming by the use of the scum cock in connection with the forward fitting.

2. The preservation of the boiler tubes, firebox sheets, stays, etc., in a clean condition through the removal of all scale-forming matter as soon as it is deposited. The cleaners' action may be advantageously assisted in some cases by the use of a suitable boiler composition, which together with other resultant solid matter is evacuated as soon as it becomes inert.

3. Immunity from leaky tubes, stays, and strained tube plates, owing to the less frequent necessity for cooling down and washing out boilers.

4. Saving in washers' wages, cost of repairs, and depreciation of boiler fabric.

5. More efficient steaming, owing to the entire boiler being kept clean, and a consequent saving of fuel.

Soot Removers.

It has been shown that the presence of scale on the water side of boiler heating surfaces does not affect the efficiency very much. The presence of 'soot' on the gas side, however, is more serious in this respect. The provision of efficient soot removers is therefore of importance.

Some modern installations are provided with permanent pipe connections, valves, and nozzles, so that jets of steam may be directed on the boiler tubes in order to remove the soot by merely operating the valves.

The Willcox-Ramoneur Boiler-Tube Soot Remover.

This apparatus, fig. 120 (*W. H. Willcox & Co., Ltd.*), consists of a 10-foot length of specially-constructed flexible Trellis Metal tubing, attached to the following parts:—*For Smoke Tubes*: a specially-designed, cone-shaped head, made in either of three sizes, viz.:—*large*—to suit tubes above 2¾-in. bore; *small*—to suit tubes up to 2¾-in. bore; *extra small*—for loco. tubes; this cone head is provided with an internal interchangeable nozzle, spiral partitions or webs, and a disc throttle for regulating the admission of air. Steam lance of finest quality steel tube. A sliding hand-hold, also protector on handling end of tubing. Lever regulating valve. Steam valve for fixing to steam pipe on boiler.

A special type is made for locomotives fitted with superheaters. This superheater type is arranged for cleaning both the fire tubes and the superheater tubes from the fire-box end, and is supplied with various cranked lances to reach the nest of tubes. For water tubes a slightly different pattern (for blowing around the outside of the tubes) is supplied.

The boiler being cleaned utilises its own steam—it does not have to be shut down.

The action of the apparatus is as follows:—On the steam valve being opened the steam passes through the flexible tube into the nozzle. This nozzle is made in various styles to suit the type of boiler and the existing steam pressure. Passing from the end of this nozzle, the steam meets with ten times its own volume of hot air drawn in from the surrounding flues of the boiler, and is thoroughly mixed with same. The combined hot air and steam strike against spiral-shaped partition webs inside the cone head, and, by the united action of the nozzle and the partitions, emerge at an increased velocity through the mouth of the cone into the boiler tubes in spiral form. The steam at this point has become completely dried by the admixture of hot air, and, further, it is now joined by an additional current of warm air from the exterior of the cone; the whole ærated mass then assumes what is best described as a 'whirlwind' movement, which sweeps with great force through the boiler tube, and drives all deposits out at the opposite end; in a similar manner to that of a projectile from a gun. As the steam is thoroughly dried in the process, there is no coagulation, and formation of sooty incrustations is avoided.

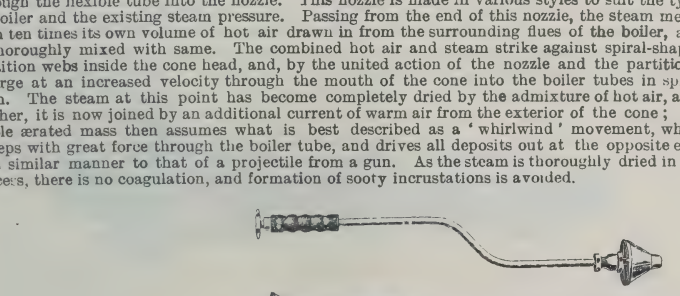


FIG. 120.

In the water-tube pattern the apparatus is fitted with a tubular end-piece, joined at its base to the form of cone head shown. The opening at the tubular end is provided with a spreader which creates a fan-shaped spray, which is directed upon the outside of the water-tubes. With

this arrangement it is possible to sweep away thoroughly, and with equal velocity, the soot and ash dust which collect on the surfaces of these tubes.

For works using both types of boilers the *'Combined Ramoneur'* is supplied. This is arranged with interchangeable heads, so that either the smoke or water-tube pattern may be utilised as required. The Willcox-Ramoneur Tube Cleaner is of British manufacture throughout.

TUBE CLEANERS.

Simplex Patent Boiler Tube Brushes and Scrapers.

(W. Crockatt & Sons, Ltd., Glasgow.)

The 'Simplex Boiler Tube Brushes' are made with short hardened steel wires which, being comparatively unyielding, are capable of removing deposits which softer brushes will not readily attack. The necessary spring of the brushes is obtained through the action of the arms, which are made of spring steel of sufficient strength to put a reasonable pressure on the tubes being cleaned. The filling of the brush plates is done in a special machine, and in such a way that it is impossible for the bristles to come out, unless they are worn quite away.

Two designs of brush are made, the 'Self-Adjusting Brush,' which depends for its action entirely on the spring of the arms, and the 'Adjustable Brush,' which also has a certain amount of spring in the arms, but which has also a central spindle with a wing nut at each end, by which the diameter of the brush can be increased within a limited range.

The 'Simplex Patent Tube Scraper' is made of drop forgings and will remove harder deposits than will the brushes. It is arranged to scrape only on withdrawal from the tubes, so that the soot, etc., is brought to the front of the boiler, and not shoved into the back ends. This tool also is adjustable by means of a nut which is easily operated as required.

The 'Simplex Condenser Brushes and Scrapers' are for use in marine work as well as on shore.

The Crockatt Simplex Turbine Tube Cleaner.

(W. Crockatt & Sons, Ltd., Glasgow.)

This appliance (fig. 121) is made in sizes from 1⅜ ins. upwards, and is driven by a turbine suitable for either air, steam, or water, at about 100 lbs. pressure. The cleaning head is detachable, so that special cutters or brushes may be attached to suit different conditions, and the universal joint enables the tool to be used in curved tubes, such as in Stirling boilers. It is suitable also for Babcock tubes, economiser tubes, and any other kind of pipes having hard scale.

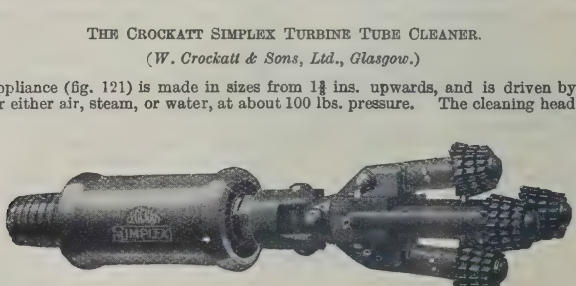


Fig. 121.

STEAM PIPES.

It is essential in steam pipe arrangements to make adequate provision for (*a*) drainage, and (*b*) expansion.

The pipes if possible should not rise from the valve on the boiler to the main steam pipe, as this causes a dangerous water pocket and, if suitably drained, only automatic steam traps should be used; these should be tested by blowing to atmosphere at least twice a day.

The best steam pipe arrangement between engine and boiler is that the boiler stop valve should stand at the top of a vertical pipe on the boiler and there should be a steady, if small, slope from that valve to the engine separator. Rupture from water-hammer occurs from failure of pipe drains, and if possible all drains should be unnecessary by design. The closing of no valve should be able to cause water to accumulate against it. Where it is impossible to avoid drainage points, these should be carefully arranged, and unduly small drain pipes must be avoided lest they choke with rust or dirt. New pipes and drains especially should be well blown through before fixing traps.

Long lengths of drain piping, from steam mains to traps, should be avoided: they are a continual source of waste, and with superheated steam the water they drain away has largely been formed by condensation in the drain piping itself. Therefore the pipes from steam main to traps should be as efficiently lagged as the main steam line.

Idle parts of the steam range should be isolated when not in use, and carefully drained, as valves often leak when shut, through scoring of faces and grit wedging in, preventing tight closing.

All live steam drains should be returned to the feed tanks.

(See also Manchester Steam Users Association Reports 1907, 1908.)

The connecting pipes from the boilers to the main steam pipe, and from the main steam pipe to the engines, should be preferably of steel, with half (not 'full,' which would form a 'water pocket') swan-neck bends, to allow for expansion.

Steam pipes between 32° and 212° F. expand by heat 0·0000111 per degree C. = 0·00000618 per degree F., if of cast iron. For wrought iron these figures become 0·0000118 and 0·00000656. Between 60° F. and a pressure of 190 lbs. or a temperature of 383° F. the expansion is about $2\frac{1}{2}$ ins. per 100 ft. The coefficient for steel is 0·000006 per degree F.

With superheated steam, pipes may expand nearly $6\frac{1}{2}$ ins. per 100 ft. of their length. At higher temperatures the coefficient of expansion is considerably larger.

In fixing pipes the expected movement when hot should, if possible, be half provided for in the opposite direction, in order to minimise stresses.

Expansion bends are best arranged to project horizontally from the line of pipe, in order to allow a free passage for water. If they stand vertically upwards they prevent water flowing, and if hung downwards they form water pockets. In both cases they necessitate careful drainage.

Pipes often vibrate under the pulsating demand of steam by an engine. This requires that they should be anchored at one or more points.

The pipes should be marked when cold opposite marks on the walls, and again when hot. Anchors should be at the points where least movement is shown. Pipes are best carried on brackets. Where they rest they should have rubbing pieces firmly attached, and they should not rest on rollers as the friction helps to check vibration.

The centre of a main steam pipe should be anchored, to divide the longitudinal expansion. Where engine and boiler room are in line and steam mains are of considerable length, a $\cap$ or Ω-shaped bend should be inserted between the engine and boiler-room. Anchor-plates should then be fixed in the centre of the engine room main and in centre of boiler-room main.

When erecting new steam pipes, attention should be given to the thorough cleaning of the pipes, as otherwise fine particles of grit, driven by the steam into the faces of the valves, cut into the seats, causing considerable trouble and expense.

Riveted pipes and flanges may be used from 6 ins. diameter upwards.

If the pipes are of riveted steel the necessary thickness must be secured for caulking. It is not the practice to make these plates less than $\frac{1}{4}$-inch; it is better to use $\frac{5}{16}$ plates. To avoid risk of rivet-heads springing off during caulking, rivets should not be less than $\frac{5}{8}$ in. in diameter.

Serviceable and reliable steam pipes may be made up of lap-welded mild steel with heavy wrought-iron flanges, the flanges being screwed on hot and the ends brazed. The flanges must be faced up to enable good joints to be made.

The radius of curvature measured to the centre of pipes should not be less in solid drawn pipes than three times the diameter of pipe; even this radius of curvature causes a reduction in thickness of 20 to 25 per cent. on the outside of the bend, and a corresponding increase in thickness on the inside. Sharper bends can, if necessary, be made by working the pipes out of two plates with a brazed joint along each side. The latter construction is more expensive than the former.

STANDARD RADIUS OF STEEL AND WROUGHT-IRON STEAM PIPE BENDS.

Bore .	Ins. $1\frac{1}{2}$	Ins. 2	Ins. 3	Ins. 4	Ins. 5	Ins. 6	Ins. 7	Ins. 8	Ins. 9	Ins. 10	Ins. 11	Ins. 12
Radius	Ins. 6	Ins. 9	Ft. 1	Ft. Ins. 1 3	Ft. Ins. 1 6	Ft. Ins. 1 9	Ft. 2	Ft. Ins. 2 6	Ft. 3	Ft. Ins. 3 6	Ft. 4	Ft. Ins. 4 6

Radius may be slightly less, but this is not advisable.

Pipes may be best bent by filling with thoroughly dry sand and plugging the ends. They will then bend without buckling.

Copper pipes are bent by being filled with melted resin, or with solder, the pipe bent cold, and the filling then melted out.

Copper pipes (although not advisable), if used for steam, should be solid drawn, and not brazed. Flanges to be screwed and brazed should be of brass in all cases. On no account should copper or brass be used for superheated steam, owing to loss of strength at these high temperatures, which to obtain the full benefit of superheat is raised to 650° or 700° F. (*Horobin.*)

Brazed copper is apt to be unreliable, and copper steam pipes ought to be wound with steel wire.

Corrugated Steel Steam Pipe Bends.*

These bends, which are adapted for resisting steam at a pressure of 400 lbs. per sq. in. superheated to a temperature of 750° F., have a series of corrugations (fig. 122) which in themselves strengthen the tube, especially as the process of manufacture thickens the wall of the tube. The corrugated pipe is formed into a bend in the usual way, and there can be no thinning of the wall in bending as all that happens is that the corrugations are somewhat opened out on the outside of the bend and are closed up on the inside.

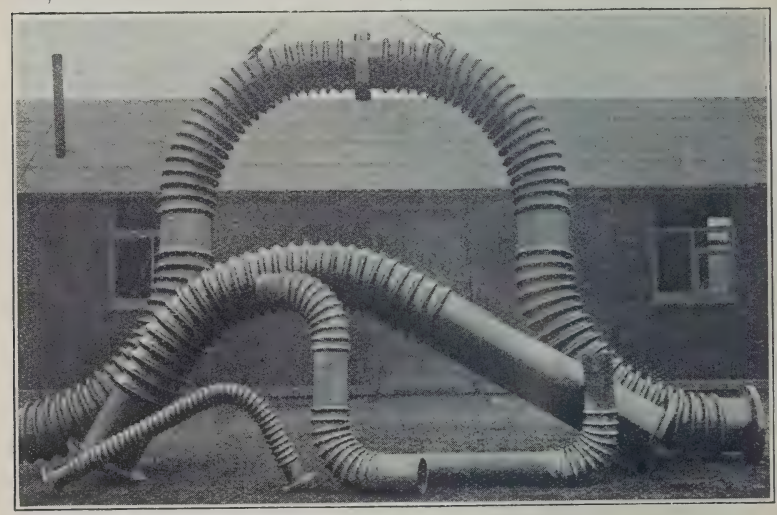


Fig. 122.

These corrugated pipes can be bent to a much smaller radius than plain pipe—in extreme cases they can be bent to the same radius as an elbow. In all cases, to stand corrugating, the tubes must be of a good soft quality of steel and must be well made, free from variations of thickness; they are usually solid drawn tubes, hot finish, 24 tons to 28 tons tensile, 25 per cent. elongation in 8 ins.

Bends made from corrugated tube are five times as flexible as similar bends made from plain tube—that is to say, if the same thrust is applied to both bends the corrugated one will move five times as far as the plain one.

An expansion bend of the lyre type made from plain tube 10 ins. bore and 138 ins. high from the centre of the main to the centre of the pipe at the crown of the bend, allows for a movement of $2\frac{1}{2}$ ins. out or in; a bend made from corrugated pipe of tne same bore can be made 55 ins. high, and gives $2\frac{1}{4}$ ins. out or in, and a similar bend 72 ins. high gives $3\frac{3}{4}$ ins. out or in, or 50 per cent. more than the much larger plain bend.

* 'Steam Pipes for Extra High Pressure and Temperature,' paper by J. A. Aiton, C.B.E. (*Transactions of the Institute of Marine Engineers*, Oct. 1926).

The severest test of the behaviour of the bends under heavy vibration has been on steam hammers. A 5-ton hammer when first started to work had connected to it a plain 5-in. steel bend and subsequently a copper bend, but this did not last more than a month, and the steam joints had to be remade, on an average, once a day. A 5-in. corrugated bend 24 ins. × 20 ins. made from stock solid-drawn hot-finish tube, $\frac{3}{32}$ in. thick, was fitted, and after working for two years was microscopically examined and showed no signs of deterioration.

Roughly speaking, the pitch of the corrugations is $2\frac{1}{4}$ ins. and each curve of the corrugation $1\frac{1}{4}$ ins. The steam friction is about double that of a plain bend.

There are no records of a corrugated pipe ever having failed in service.

Pipe Thickness.

If t = thickness in inches ; p = pressure in lbs. per sq. in.; d = internal diameter in inches.

$= \frac{pd}{6000}$, for lap-welded iron or lap-welded or solid-drawn steel, provided the thickness is not less than $\frac{1}{4}$ in.; $t = \frac{pd}{3500} + \frac{1}{4}''$, for cast iron ;

$t = \frac{pd}{6000} + \frac{1}{16}''$, for brazed copper ; $t = \frac{pd}{6000} + \frac{1}{32}''$, for solid drawn copper.

Thickness of Copper Main Steam Pipes.

Internal Diameter of Pipe in Inches.	Working Press of Steam in Boiler in Lbs. per Sq. In.				
	100 lbs.	125 lbs.	150 lbs.	180 lbs.	200 lbs.
	Thickness of Pipe in Inches.				
$1\frac{1}{2}$	·140	·143	·147	·152	·155
2	·145	·150	·155	·161	·165
$2\frac{1}{2}$	·150	·156	·162	·170	·175
3	·155	·162	·170	·179	·185
$3\frac{1}{2}$	·160	·168	·177	·188	·195
4	·165	·175	·185	·197	·205
$4\frac{1}{2}$	·170	·181	·192	·206	·215
5	·175	·187	·200	·215	·225
$5\frac{1}{2}$	·180	·193	·207	·224	·235
6	·185	·200	·215	·233	·245
$6\frac{1}{2}$	·190	·206	·222	·242	·255
7	·195	·212	·230	·251	·265
$7\frac{1}{2}$	·200	·218	·237	·260	·275
8	·205	·225	·245	·269	·285
$8\frac{1}{2}$	·210	·231	·252	·278	·295
9	·215	·237	·260	·287	·305
$9\frac{1}{2}$	·220	·243	·267	·296	·315
10	·225	·250	·275	·305	·325
$10\frac{1}{2}$	·230	·256	·282	·314	·335
11	·235	·202	·290	·323	·345
$11\frac{1}{2}$	·240	·268	·297	·332	·355
12	·245	·275	·305	·341	·366

(*Philip Dawson.*)

Lloyd's Rules for Steam Pipes.

The following is abstracted and abridged from Lloyd's Rules :

Pipes made from electro-deposition of copper on a mandril are not to be used for steam.

Copper pipes are to be properly annealed.

Copper pipes for pressures over 75 lbs. per square inch are to be solid drawn.

Pipes for pressures above 180 lbs. per square inch are not to be of copper if the diameter exceeds 5 ins.

Copper pipes are not to be used for superheated steam.
Permissible working pressures (W.P.) are as follows:

where,

t = thickness in hundredths of an inch; d = diameter of pipe in inches.

For brazed copper . . . . . W.P. $= \frac{45\,(t-3)}{d}$

For solid-drawn copper . . . . . W.P. $= \frac{60\,(t-3)}{d}$

For solid-drawn cold-finished steel . . . W.P. $= \frac{120\,(t-10)}{d}$

For solid-drawn hot-finished steel . . . W.P. $= \frac{120\,(t-12)}{d}$

For lap-welded iron or steel with or without covering straps . . . . . W.P. $= \frac{90\,(t-12)}{d}$

Pipe Hangers.

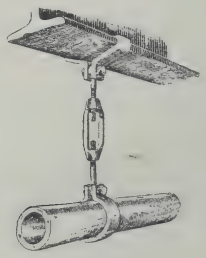


FIG. 123.

DIMENSIONS AND SAFE LOADS FOR PIPE-HANGER PARTS.

Pipe, Size. Ins.	Pipe, Thick. Ins.	Band, Width. Ins.	Clamp, Thick. Ins.	Clamp, Width. Ins.	Rod, Diam. Ins.	Bolts, Band. Ins.	Bolts, Clamp. Ins.	Safe Load. Lbs.	Length ½ Band. Ins.
2	¼	1¼	⅜	1¼	½	½ × 2	½ × 2¼	1,160	6
2½	¼	1¼	⅜	1¼	½	½ × 2	½ × 2¼	1,160	7
3	¼	1¼	⅜	1¼	½	½ × 2	½ × 2¼	1,160	8
3½	¼	1½	25/64	1½	½	½ × 2	½ × 2¼	1,160	9
4	¼	1½	25/64	1½	⅝	½ × 2	⅝ × 2¼	1,870	10½
4½	¼	1½	⅜	1½	⅝	½ × 2	⅝ × 2¼	1,870	11
5	¼	2	25/64	2	⅝	½ × 2	⅝ × 2¼	1,870	13
6	¼	2	½	2	⅝	⅝ × 2¼	¾ × 2¼	1,870	14
7	5/32	2	½	2	¾	⅝ × 2¼	⅝ × 3	2,830	16
8	5/32	2	½	2¼	¾	¾ × 2½	⅝ × 3½	2,830	18
9	5/32	2	½	2½	⅞	¾ × 2¾	⅞ × 3½	3,940	20
10	½	2	½	2½	⅞	⅝ × 3	⅞ × 3½	3,940	22
12	½	2¼	½	2½	1	⅞ × 3½	1 × 3½	5,180	25
14	½	2¼	½	2½	1⅛	1 × 3½	1⅛ × 3⅝	6,510	27
15	½	2½	½	2½	1⅛	1 × 3½	1⅛ × 3¾	6,510	30
16	½	3	⅝	3	1⅛	1⅛ × 3⅝	1¼ × 4	6,510	33
18	⅝	3	⅝	3	1¼	1¼ × 4	1¼ × 4	8,410	35
20	⅝	3	¾	3	1¼	1¼ × 4	1¼ × 4½	8,410	37
22	¾	3	¾	3	1¼	1¼ × 4½	1¼ × 4½	8,410	39
24	¾	3	¾	3	1¼	1¼ × 4½	1¼ × 4½	8,410	42

Exhaust Pipes.

These may be of cast iron and should be free from blow holes. The flanges should be faced and jointed with soft rubber rings. Exhaust pipes should be painted while under vacuum, preferably with several coats of red lead paint well brushed in until all the pores are filled. It has been suggested that exhaust pipes, especially if long, might be laid in a concrete trench under water so as effectually to prevent air leakage.

Flanges and Joints for Steam Pipes.

Pipes for steam and hot water should not be jointed with socket and spigot; this is a dangerous type of joint.

Steam pipes may be screwed into sleeves or branch pieces, but the screws should be tapered. This method is quite good for small-bore pipes which do not have to stand up to a very high steam pressure, and seems to have been adopted generally by American engineers for marine work. It makes a very neat job and is easy to cover.

Flanges should have the number of bolts increasing in multiples of 2. In marine work the usual method for spacing the bolts with regard to position of flange is as follows: For vertical and inclined flanges there should be one bolt hole spaced on either side of vertical centre line, and for horizontal flanges one bolt hole spaced on either side of fore and aft centre line of flange. The thickness of flanges for different materials to some extent should vary with the modulus of elasticity of the material, and consideration given as to shock and vibration which may take place.

The following formula gives thickness of flange with the constants as now generally adopted by the British Engineering Standards Association for marine flanges:—

$$\sqrt[4]{\frac{4\,P \times l^4}{t^3}} = \text{constant } (C).$$

C for cast iron = 11; for brass or steel = 12·5 to 13·5,

where,

P = working pressure in lbs. per sq. in.; t = thickness of flange in ins.; l = pitch between bolts or studs.

For the attachment of flanges to pipes up to 8-in. bore, the usual practice is to screw on the flanges with a vanishing thread, which should be about 11 threads per inch. These screwed-on flanges require to have collars in order to obtain a suitable depth of screw, which varies from $1\frac{1}{8}$ in. to $1\frac{7}{8}$ in. For pipes above 8-in. bore, the flanges are riveted on.

For steam pressures up to about 180 lbs. per sq. in., machine-faced flanges may be jointed simply with a ring of copper, preferably corrugated, or jointed with some form of asbestos, such as Kymbala or Tauril jointing.

In large-sized steam pipes, also small pipes for high-pressure and superheated steam, it will be found that the best joints are made by forming a channel recess round face of flange $\frac{1}{8}$ in. deep and $\frac{1}{4}$ in. wider than the diameter of bolt holes. The face of the flange projecting above this recess is scraped up by hand to suit a small hand surface plate. The channel is then filled with red lead paint or some type of soft asbestos packing, when making up joint. This form of joint is adopted universally for high-pressure steam from about 200 lbs. per sq. in. and upwards.

A cement for pipe joints that is said to be as good as red lead putty in a faced or rough flange joint, but at one-tenth the cost, consists of a mixture of ordinary pine tar and dry oxide of iron. It is stated that it does not harden so soon as red lead, and is very adhesive under pressure.

A thin gasket has less edge surface exposed to the pressure, and is therefore less likely to blow out than a thick one.

The Use of Metallic Gaskets.

The ordinary corrugated copper gaskets are approximately $\frac{1}{64}$ in. thick, and are generally used as a ring inside the bolt circle.

Some engineers recommend the use of two gaskets in order to allow for the spring of the pipe flanges when bolted up. Flanges, however, should be stiff enough not to spring sufficiently to touch at the edges when one gasket is used. The second gasket should then be omitted. The gaskets should be dipped in jointing red lead (red lead mixed with boiled oil). Metal gaskets should be coated with pipe cement, graphite and oil, or lead and oil, and bolted up before the coating has dried, so that each corrugation flattening under the pressure forces a ring of this filler concentrically around the flange, and the copper gasket practically acts as a binder. Copper gaskets should be annealed soft and pliable after corrugating.

The ideal gasket is one that covers the pipe ends as well as the flange if the pipes are faced off flush with the flange, because such a gasket will save thread leaks. If, however, the pipe does not come out flush with the flange, this cannot be accomplished with any gasket.

For jointing purposes the corrugated copper gaskets in common use have not been very successful with superheated steam. The reason lies in the fact that the annealing temperature of copper lies in the neighbourhood of 600° F., corresponding with a boiler pressure of 180 lbs. to a superheat of 220°. Corrugated gaskets, moreover, used by careless pipe fitters are always likely to give trouble owing to the danger of the bolts and flanges being overstrained. Joints of asbestos and of rubber preparations have often been used, but they last but a short time. It is fairly certain that the high temperature of superheated steam at the usual boiler pressure prohibits the use of organic material for jointing purposes if results of any permanence are desired. A satisfactory joint has been found to be one in which a gasket of thin steel of a radial width not exceeding $\frac{5}{8}$ in. has been used, the steel being soft annealed. The comparatively small area of the gasket enables great pressure to be brought to bear by bolts of ordinary size, with the result that the soft steel is thoroughly pressed into the rough surface of the flange. Such a joint will, although simple and cheap, give all the permanence that could be desired.

(*Times Engineering Supplement.*)

'Dexine.'

This material (*Dexine, Ltd.*) is a very suitable substance for making joints. It does not stick to the flanges, nor does it press out, blow out, or burn out; and it can be used over and over again. It is made in sheets from $\frac{1}{64}$th inch upwards.

Beldam's Pilot Jointing.

(*The Beldam Packing & Rubber Co., Ltd.*, 16 *Gracechurch Street, London, E.C.* 3.)

This is a compressed asbestos fibre jointing for all steam or water pressures, ammonia, acids, oils, etc. It will stand temperatures (up to 750° F.), and will not burn, blow, or squeeze out. Its tensile strength is over 3 tons per sq. in.

Joints made of this material can be opened up as often as desired, and it does not lose its elasticity.

To facilitate quick measuring when cutting, the sheets are marked in 1-in. squares.

Leaks in Steam Pipes.

Leaks in steam pipes that cannot be stopped by caulking or plugging may be stopped with a cement made by mixing black oxide of manganese with enough raw linseed oil to form a thick paste. The steam pressure should be relieved in the pipe, but the pipe should be kept warm enough to dry the oil.

Size of Steam Pipes.

The diameter of a steam pipe is usually in practice chosen to give a certain steam velocity in the pipe, 100 feet per sec. being a customary velocity for steam at ordinary pressures.

Many engineers use this speed for both saturated and superheated steam. Others vary the velocity as in the following table:

Diameter of Pipe.	Velocity, Feet per sec. Saturated.	Superheated.
Up to 3 ins.	75	100
3½ ins. to 9 ins.	90	120
10 ins. and over	90	140

For superheated steam in long, straight pipe lines 180 ft. per sec. may be used.

For steam at high pressures, 300 to 500 lbs. per square inch, useful information is given by Broido (*Amer. Soc. Mech. Engineers*, 1922, page 1199), who concludes that for steady flow in pipes over 5 ins. diameter 10,000 to 12,000 feet per min. is advisable, and states that velocities up to 15,000 ft. per min. have been used in larger pipes without undesirable effects.

It should be remembered that the higher the velocity the smaller is the pipe diameter, and hence the external radiating surface is less.

Pressure Drop in Pipe Lines.

The formula (known as the Babcock formula) usually employed to relate the flow, pressure drop, and pipe diameter is

$$W = 87{\cdot}5\sqrt{\frac{D\,(p_1 - p_2)\,d^5}{L\left(1 + \frac{3{\cdot}6}{d}\right)}},$$

where W = weight in pounds per minute; d = diameter in inches; D = density or weight per cubic foot; p_1 = initial pressure; p_2 = pressure at end of pipe; L = length in feet.

This formula corresponds to $p_1 - p_2 = 0{\cdot}0001306\,\frac{W^2 L\left(1 + \frac{3{\cdot}6}{d}\right)}{D d^5}$ and also to $h = f\,\frac{v^2}{2g}\cdot\frac{4L}{d_1}$,

where h = loss of head in feet, d_1 = diameter in feet, f = coefficient of friction of the steam in the pipe, f being equal to $0{\cdot}0027\left(1 + \frac{3}{10 d_1}\right)$; this coefficient is due to R. C. Carpenter, and is accepted by Unwin.

The following table gives, approximately, the weight of steam per minute which will flow from various initial pressures, with one pound loss of pressure, through straight, smooth pipes, each having a length of 240 times its own diameter. The table is calculated from the above formula.

TABLE OF FLOW OF STEAM THROUGH PIPES.

Initial Pressure by Gauge.	Internal Diameter of Pipe in Inches. Length = 240 diameters.												
	¾	1	1½	2	2½	3	4	5	6	8	10	12	15	18
Lbs. per Sq. In.	Weight of Steam per Minute in Pounds, with One Pound Loss of Pressure.													
1	0·92	1·79	4·80	9 4	15·6	23·8	45·5	75·0	112	209	337	497	799	1160
10	1·14	2·20	5·90	11·5	19·2	29·4	56·0	92·2	137	258	415	611	982	1430
20	1·33	2·58	6·91	13·5	22·7	34·4	65·6	108	161	302	486	716	1150	1680
30	1·49	2·89	7·75	15·2	25·3	39·5	73·5	122	180	338	545	803	1290	1886
40	1·64	3·18	8·54	16·7	27·8	42·5	81·0	133	198	372	600	885	1420	2070
50	1·77	3·44	9·23	18·1	30·1	46·0	87·6	144	215	403	650	956	1540	2240
60	1·89	3·67	9·85	19·3	32·1	49·0	93·5	154	229	429	694	1020	1640	2390
70	2·01	3·91	10·45	20·5	34·1	52·1	99·5	164	244	457	736	1080	1740	2540
80	2·12	4·11	11·0	21·6	35·9	54·9	105	172	266	480	775	1140	1830	2670
90	2·22	4·31	11·6	22·6	37·7	57·6	110	181	269	504	813	1200	1930	2810
100	2·32	4·50	12·1	23·6	39·3	60·0	114	188	281	526	850	1250	2019	2920
120	2·50	4·85	13·0	25·4	42·4	64·6	123	203	302	567	914	1340	2160	3150
150	2·74	5·33	14·3	28·0	46·5	71·0	136	223	332	623	1004	1480	2380	3480
200	3·11	6·04	16·2	31·7	52·8	80·5	154	253	377	705	1140	1679	2700	3940

(*Morley.*)

The foregoing table is based on saturated steam. For any other loss of pressure multiply by the square root of the given loss. For any other length of pipe *divide* 240 *by the given length expressed in diameters, and multiply the figures in the table by the square root of this quotient*, which will give the flow for 1 lb. loss of pressure. Conversely, dividing the given length in diameters by 240 will give the loss of pressure for the flow given in the table.

For superheated steam multiply the values in the table by $\sqrt{\frac{\text{density of superheated steam}}{\text{density of saturated steam}}}$

—or *approximately* $\sqrt{1 + \frac{1}{\cdot 0015\,t_s}}$, where t_s = degrees of superheat.

The loss of head due to getting up the velocity, to the friction of the steam entering the pipe, and passing elbows and valves, will reduce the flow given in the table. The resistance at the opening, and that at a globe valve, are each about the same as that for a length of pipe

equal to 114 diameters divided by a number represented by 1 + (3·6 ÷ diameter). For the sizes of pipes given in the table these corresponding lengths are, approximately :—

$\frac{3}{4}$	1	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	4	5	6	8	10	12	15	18
20	25	34	41	47	52	60	66	71	79	84	88	92	95

The resistance at an elbow is equal to two-thirds that of a globe valve. These equivalents—for opening, for elbows, and for valves—must be added in each instance to the actual length of pipe. Thus a 4-in. pipe, 120 diameters (40 ft.) long, with a globe valve and three elbows, would be equivalent to $120 + 60 + 60 + (3 \times 40) = 360$ diameters long; and $360 \div 240 = 1\frac{1}{2}$. It would therefore have $1\frac{1}{2}$ lbs. loss of pressure at the flow given in the table, or deliver $(1 \div \sqrt{1} = \cdot 816)$, 81·6 per cent. of the steam with the same (1 lb.) loss of pressure.

In place of the calculations on the table given in the preceding paragraphs, the following nomographic chart (based on the formula, p. 1691) may be used, and provides a very convenient and sufficiently accurate method of determining pipe sizes, pressure drops, etc.

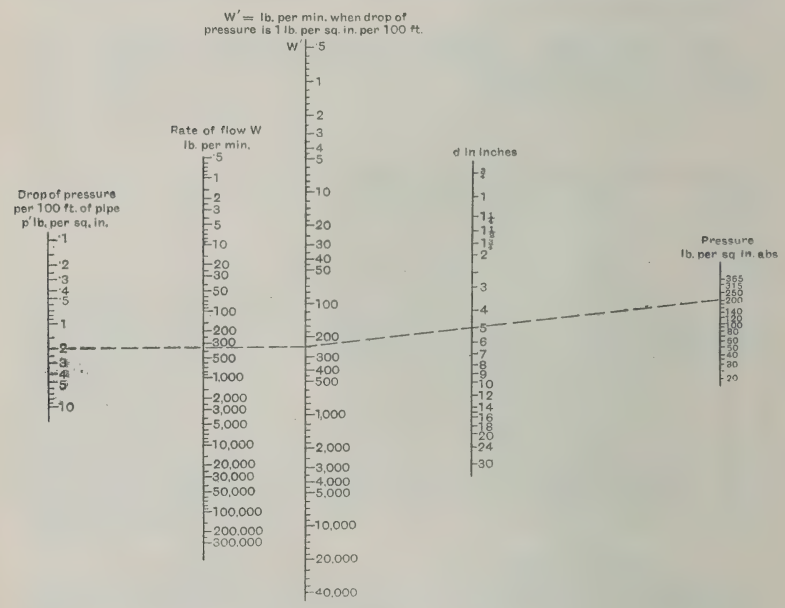


Fig. 124.

Method of use of chart :

(*a*) To find diameter for given quantity and pressure drop, join W and p' and produce to cut scale of W′, join point so found to pressure scale at right, read required diameter at intersection with scale of d.

(*b*) To find quantity for given pipe and pressure drop, reverse above process, finding first W′ and then W.

(*c*) To find pressure drop for given pipe and flow, join points on scales of pressure and d, produce to find W′, join W′ to W and produce to cut scale of p' at required value.

Flow of Air and Steam in Pipes.

The Babcock and other formulæ for the flow of steam have recently been reviewed and compared with the results obtained by improved experimental methods.

The conclusions of the investigation made by McAdams and Sherwood (Amer. Soc. Mech. Eng., October 1926) are that for air the best formula is:

$$P_1 - P_2 = \frac{0{\cdot}13 f L Q^2 P^2}{P_a D^5 T}$$

$$\text{or } Q_1 = 2{\cdot}77 \sqrt{\frac{D^5 (P_1 - P_2)}{f L P_1^2}}$$

and for steam:

$$P_1 - P_2 = 0{\cdot}0484 \frac{f L G^2}{D^5 w_a}$$

$$\text{or } G = 4{\cdot}54 \sqrt{\frac{D^5 w_a (P_1 - P_2)}{f L}}$$

where

P_1, P_2 are initial and final pressures in lbs. per sq. in.;
P_a = average pressure, lbs. per sq. in. abs.;
Q_1, Q = cub. ft. air per minute at P_1, P, respectively;
D = inside diameter of pipe in inches;
L = equivalent frictional length of pipe in feet;
P = length of straight pipe + 2·5 D × number of elbows;
T = air temperature, ° Fah., abs.;
G = lbs. steam per minute;
w_a = lbs. per cub. ft. of fluid at average temperature and pressure;
f = friction coefficient.

f is given by charts from which the values in the following table are extracted:

Values of $\frac{G}{D}$ for Steam, or $\frac{QP}{DT}$ for Air.	Steam.	Air.			
		Iron and Steel Pipes.		Copper and Brass Pipes.	
		300°	70°	300°	70°
0·10	—	0·0113	0·0104	0·0094	0·0088
0·20	—	0·0098	0·0090	0·0080	0·0074
0·40	0·0097	0·0085	0·0079	0·0067	0·0062
0·60	0·0086	0·0078	0·0072	0·0061	0·0056
0·80	0·0081	0·0074	0·0068	0·0056	0·0052
1·00	0·0077	0·0070	0·0065	0·0053	0·0049
1·50	0·0070	0·0065	0·0060	0·0048	0·0045
2·00	0·0066	0·0062	0·0059	0·0045	0·0043
4·00	0·0060	0·0059	0·0058	0·0041	0·0038
10·00	0·0059	0·0058	0·0058	—	—
20	0·0056	0·0058	0·0058	—	—
30	0·0056	0·0058	0·0058	—	—
40	0·0055	—	—	—	—

Loss of Heat from Uncovered Steam Pipes.

The large amount of heat lost from bare steam pipes, and the consequent waste of coal, is frequently not fully appreciated. One square foot of uncovered pipe surface, the pipe containing steam at 100 lbs. per sq. in., will dissipate about 3 B.Th.U. per hour per degree of temperature difference between the steam and the atmosphere, or about 850 B.Th.U. per hour at ordinary

atmospheric temperatures. The steam condensed per hour due to this square foot of bare pipe will be 0·96 lb. In a year of 3,000 working hours the steam wasted will amount to 2,880 lbs., corresponding to a coal consumption of about 410 lbs.

A single 9-ft. length of 6-in. pipe, if uncovered, will waste by heat loss about 2·5 tons of coal per year.

80 to 90 per cent. of this loss may be avoided by the use of suitable coverings, the expense of which is amply counterbalanced by the saving in coal.

It is important that flanges, valves, etc., as well as the body of the pipe lengths, should be covered.

The loss of heat per hour per sq. ft. of surface, in the case of bare pipes, depends principally upon the difference of temperature between the pipe—*i.e.* practically, that of the steam in the pipe—and the surrounding atmosphere, but also is influenced to some extent by the diameter of the pipe, its position—whether horizontal or vertical—the presence of air currents about the pipe, the nature of the pipe surface, and the velocity of the steam in the pipe.

Complete data on all these factors are lacking.

It is usually assumed that the loss is proportional to the difference of temperature, but careful experiments show that the rate of loss, B.Th.U. per sq. ft. per degree F., increases at the higher temperature differences. Thus from 1·95 at 50° difference, it becomes 2·4 at 150°, 3·26 at 300°, 4 at 400°, and 5·26 at 460°. (*McMillan.*)

There is little experimental data on the effect of pipe diameter, but it appears that the loss per sq. ft. decreases as the diameter increases, and this is in accordance with theory.

The following table shows the heat loss from pipes covered with various thicknesses of 85 per cent. magnesia composition, and the approximate loss from uncovered pipes is given for comparison.

LOSS OF HEAT FROM STEAM PIPES, WITH STEAM 300° F. ABOVE ATMOSPHERIC TEMPERATURE.
LOSS IN B.TH.U. PER HOUR PER SQUARE FOOT OF PIPE SURFACE.

	Pipe Diameter.					
	1 in.	2 ins.	4 ins.	6 ins.	8 ins.	12 ins.
Bare pipe	1100	1070	1010	945	920	880
Magnesia covering 0·5 in. thick	340	280	250	240	235	230
,, 1·0 in. ,,	260	200	170	155	148	140
,, 1·5 ins. ,,	220	165	145	120	115	105
,, 2·0 ,, ,,	195	140	110	100	92	87
,, 3·0 ,, ,,	170	120	89	75	71	66
,, 4·0 ,, ,,	—	110	75	65	59	52
,, 5·0 ,, ,,	—	—	66	58	52	46

(*Based on tests by McMillan, 'Trans. Am. Soc. M.E.'* 1915.)

The 'efficiency' of the covering in preventing loss of heat is taken as the ratio (heat loss from bare pipe — loss from covered pipe) ÷ heat loss from bare pipe. The table shows that this 'efficiency' varies with the thickness and the pipe diameter, hence to compare different covering materials, the comparison must be made under the same conditions of thickness, etc., if the comparison is to show the relative insulating properties.

The coverings may also be compared on the basis of cost per foot run—*i.e.* taking a greater thickness in the case of the cheaper materials.

The relative properties of various pipe coverings are shown in the following table, the thickness being about 1·4 ins., diameter of test pipe 1 in., steam pressures in pipe 50, 100, and 200 lbs. per sq. in.

RELATIVE VALUE OF PIPE COVERINGS.

Material.	Weigh per foot run, lbs.	Insulating Efficiency per cent.		
		At 50 lbs. per sq. in.	At 100 lbs. per sq. in.	At 200 lbs. per sq. in.
Sectional cork	1·0	85	86	87
Slag wool	2·6	83	84	—
Cellular asbestos	2·4	81	82	83
Silk waste	2·3	85	85·5	86
Hair felt and asbestos	2·7	83	84	85
Sectional magnesia	1·6	79	79	—
Plastic magnesia	2·2	77	77	—
Diatomite (plastic)	2·2	79	79	—
Kisol and magnesia (plastic)	2·9	77	78	79
Kisol and hair felt (plastic)	3·0	75	77	78
Slag wool and composition (plastic)	1·5	75	76	—
Mica (plastic)	2·6	73	75	—
Asbestos composition (plastic)	3·9	69	71	72
Thinner Coverings.				
Hair felt 0·5 in. thick	—	79	—	—
Magnesia-filled rope 0·7 in. thick	0·8	61	64	—
Asbestos rope 0·7 in. thick	1·2	47	53	—

(*Bradley.*)

BOXES FOR PIPE FLANGES.

In estimating the loss of heat from covered pipe lines, the effect of the boxes covering the flanges in adding to the total area of radiating surface must be allowed for.

For typical flange boxes, assuming one box at each 10 feet of pipe line, the additional area may be taken as about 7 per cent. *i.e.* to find the heat radiated, calculate the pipe surface in square feet, as if there were no flanges, add 7 per cent., and multiply by the appropriate figure from the table on page 1691.

NOTES ON COVERINGS FOR PIPES AND BOILERS.

As well as having good heat insulating properties, it is important that pipe coverings be durable. A material which easily disintegrates under vibration will not long remain an efficient insulator, nor will a material which cannot stand high temperature. For example, hair felt is a good insulator, but will not stand contact with pipes carrying high pressure or superheated steam.

Tests with locomotive-type boilers, first uncovered and then covered with 2 ins. of 85 per cent. magnesia, showed in the second case 2·5 per cent. additional evaporation for 13·8 per cent. less coal, or a total coal saving of 16 per cent.

(*'Chem. and Met. Eng.,' Feb.* 16, 1921.)

Recent tests at the National Physical Laboratory have shown that 85 per cent. magnesia (that is, 85 per cent. hydrated magnesium carbonate and 15 per cent. asbestos) begins to calcine at 450° F., and that at 600° F. the decomposition is considerable. At the high steam temperatures now coming into use in large power stations (pressures up to 375 lbs. per sq. in. and temperatures up to 700° F.) it is necessary to place next the pipe a material unaffected by heat. About ½ in. of slag wool or of special composition may be used, then a layer of 85 per cent. magnesia or other good non-conductor, and then an outer layer, ½ in. thick, of hard-setting composition to bind the whole together.

NOTES ON LAGGING.

The surface temperature of a lagging, when in use, is commonly supposed by engineers to furnish a clue to its efficiency, and conclusions are often formed from the sensation of heat experienced when the hand is placed on the surface. Another test, frequently employed, is to make

a small hole in the lagging into which the bulb of a thermometer is inserted ; and it is then assumed that the material in which the lowest reading is obtained possesses a superiority in respect to insulating power. Whilst a correct measurement of the temperature of the surface is of value in conjunction with other tests, if taken alone an entirely erroneous opinion of the merit of the material may be formed.

The important factor which is not taken into account in the above rough-and-ready test is the nature of the surface in respect to its radiating power. A lagging with a rough exterior radiates heat with much greater freedom than one which is finished smooth, and the latter which is superior as an insulator, owing to retention of heat by the surface, will feel hotter than the former when touched by the hand. Hence it follows that a given non-conducting coating, if improved by painting and varnishing the surface, would appear to have suffered in insulating power if judged by its surface temperature only. To avoid such a fallacious conclusion, it is evident that the radiating power must also be taken into account, and only when this is done can a correct inference be drawn.

When the radiating surfaces are identical, the surface temperature furnishes a reliable and delicate guide to the insulating powers of different materials. A smooth metallic surface, such as aluminium paint, may be hotter than an unprepared surface, and yet radiate less.

On reference to a test in which a thermometer bulb is placed in a hole made for the purpose in the outer part of the lagging, into which it is customary to pour mercury to secure a good contact between the material and the bulb of the thermometer, it is difficult to say what the reading indicates, as the temperature recorded will be the mean of that existing between the end of the bulb and the surface ; and, moreover, the normal condition of the lagging at this part is entirely changed by the introduction of the mercury and thermometer bulb. One thing is certain : that the reading affords no clue to the surface temperature it is supposed to measure ; and in this respect the test is inferior to the application of the hand. The thermal-junction method is undoubtedly the best for obtaining a definite reading of the surface temperature : and as it is now possible to obtain cheap portable indicators for use with a junction, other so-called measurements should be definitely abandoned.

The relations between surface temperature and efficiency may be summarised as follows :—

1. When the surface of a given insulating material is treated so as to reduce its radiating power, the temperature is higher, although the heat escaping is less.

2. If two different materials possess identical surfaces, that which has the higher surface temperature, under given conditions, is the worse insulator.

A recognition of these two points will prevent engineers from being led astray by the extravagant claims frequently put forward for compositions, based entirely on imperfect surface temperature measurements. The advantage of finishing the surface with aluminium paint, instead of ordinary paint and varnish, should be recognised, as not only is the insulating power improved, but the finished work has a better appearance. The fact that such a surface feels warmer to the touch is in this instance not a proof of inferiority, but of the contrary.

(*C. R. Darling*, '*Engineering*,' *Sept.* 22, 1911.)

The holding of slag wool to pipes or boilers (to which asbestos composition has been applied) is best effected by means of wire-bound split laths, so spaced that the finishing overlay of cement may not only be keyed in between them but may also afford a means of locating any leakages, in addition to preserving a favourable density estimated at from 10 to 12 lbs. per cubic. foot.

A cheaper but decidedly less effective way of holding slag wool together is by means of wire netting in the meshes of which an overlay of plastic cement is impacted. When, therefore, disintegration takes place and the wool settles down by its own gravity, the upper surface is reduced to a mere shell of wire and cement by which the bulk is held in suspension.

Silk waste, although almost equal in efficiency to slag wool, is wanting in durability.

Hair-felt is one of the finest non-conductors known, but is quickly consumed by direct contact with heated surfaces. See also pp. 1697, 1698. (*E. Kenyon.*)

Overleaf is a record of the performance of the various insulating products for high and low temperatures which have made this trade-mark recognised throughout the world as a sign of the utmost efficiency in insulation of all kinds.

For free illustrated literature dealing authoritatively with insulation requirements write to

NEWALLS
INSULATION CO., LTD.
WASHINGTON · CO. DURHAM

N145—95

P.T.O.

F. 1696

See previous page

A RECORD

OF PERFORMANCE. FIGURES WHICH SPEAK LOUDER THAN WORDS.

THERMAL CONDUCTIVITIES OF NEWALLS INSULATING SPECIALITIES.

The following values of K., i.e. Thermal Conductivities or Coefficients of Heat Transmission, are given in English and Metrical Units for various Mean Temperatures of Insulation.

K (a) B.Th.Us. per sq. ft. of insulated surface per hour per 1° F. difference between temperature of internal and external faces of insulation for 1" thickness of insulation.

K (b) Kg. Calories per sq. metre of insulated surface per hour per 1° C. difference between temperature of internal and external faces of insulation for 1 metre thickness of insulation.

$$\text{Mean Temperature} = \frac{T1 + T2}{2}$$

Where: T1 = Internal Temperature
T2 = Surface Temperature of outside of insulation.

Insulation	Mean Temp. °F.	K (a)	K (b)
85% Magnesia and *Newtempheit	100	0.42	.052
	200	0.43	.053
	300	0.46	.057
	400	0.48	.060
	500	0.52	.065
*Nonpariel Cork	48	0.279	.035
*Nonpariel Bricks	393	0.93	.115
	618	1.10	.136
	843	1.19	.148
Blue Asbestos Firefelt	89	0.38	.047
	150	0.44	.055
	208	0.50	.062
	266	0.55	.068
	324	0.60	.074
Asbestocel	90	0.68	.084
	154	0.78	.097
	218	0.87	.108
	282	0.98	.122
	347	1.10	.136

* N.B. The above figures are compiled from results of actual tests carried out by the National Physical Laboratory, Teddington.

Normal Pipe Size.	Thickness. STD.	Thickness. Double Layer Double Standard.		Temperature of the Fluid in the Pipe. 650° F. to 800° F. Thickness of Fire Backing.	650° F. to 800° F. Thickness of 85 per cent. Magnesia.	500° F. to 650° F. Thickness of Fire Backing.	500° F. to 650° F. Thickness of 85 per Cent. Magnesia.
1	$\frac{7}{8}$	$1\frac{3}{4}$	Straight pipe and bends	2″ sectional 6 oz. sheeting covered	None	2″ sectional sheeting covered	None
			Fittings, flanges, etc.	2″ sectional 6 oz. sheeting covered	None	2″ sectional sheeting covered	None
$1\frac{1}{4}$	$\frac{7}{8}$	$1\frac{3}{4}$	Straight pipe and bends	2″ sectional sheeting covered	None	2″ sectional sheeting covered	None
			Fittings, flanges, etc.	2″ sectional sheeting covered	None	2″ sectional sheeting covered	None
$1\frac{1}{2}$	$\frac{7}{8}$	$1\frac{3}{4}$	Straight pipe and bends	2″ sectional sheeting covered	None	2″ sectional sheeting covered	None
			Fittings, flanges, etc.	1″ × 3″ × 36″ block	1″ × 3″ × 36″ block	1″ × 3″ × 36″ block	1″ × 3″ × 36″ block
2	$1\frac{1}{32}$	$2\frac{1}{16}$	Straight pipe and bends	$1\frac{1}{32}$″ sectional	4″ sectional covering $1\frac{1}{2}$″ thick	$1\frac{1}{32}$″ sectional	4″ sectional covering $1\frac{1}{2}$″ thick
			Fittings, flanges, etc.	1″ × 3″ × 36″ block	$1\frac{1}{2}$″ × 3″ × 36″ block	1″ × 3″ × 36″ block	$1\frac{1}{2}$″ × 3″ × 36″ block
3	$1\frac{1}{32}$	$2\frac{1}{16}$	Straight pipe and bends	$1\frac{1}{32}$″ sectional	5″ sectional covering $1\frac{1}{2}$″ thick	$1\frac{1}{32}$″ sectional	5″ sectional covering $1\frac{1}{2}$″ thick
			Fittings, flanges, etc.	1″ × 3″ × 36″ block	$1\frac{1}{2}$″ × 3″ × 36″ block	1″ × 3″ × 36″ block	$1\frac{1}{2}$″ × 3″ × 36″ block
4	$1\frac{1}{8}$	$2\frac{1}{4}$	Straight pipe and bends	$1\frac{1}{8}$″ sectional	6″ sectional covering $1\frac{1}{2}$″ thick	$1\frac{1}{8}$″ sectional	6″ sectional covering $1\frac{1}{2}$″ thick
			Fittings, flanges, etc.	1″ × 3″ × 36″ block	$1\frac{1}{2}$″ × 3″ × 36″ block	1″ × 3″ × 36″ block	$1\frac{1}{2}$″ × 3″ × 36″ block
6	$1\frac{1}{8}$	$2\frac{1}{4}$	Straight pipe and bends	$1\frac{1}{2}$″ sectional	$1\frac{1}{2}$″ segmental block to cover $1\frac{1}{2}$″ covering on 6″ pipe	$1\frac{1}{2}$″ sectional	$1\frac{1}{2}$″ segmental block to cover $1\frac{1}{2}$″ covering on 6″ pipe
			Fittings, flanges, etc.	$1\frac{1}{2}$″ × 3″ × 36″ block	$1\frac{1}{2}$″ × 3″ × 36″ block	$1\frac{1}{2}$″ × 3″ × 36″ block	$1\frac{1}{2}$″ × 3″ × 36″ block
8	$1\frac{1}{4}$	$2\frac{1}{2}$	Straight pipe and bends	$1\frac{1}{4}$″ sectional	2″ segmental block to cover $1\frac{1}{4}$″ covering on 8″ pipe	$1\frac{1}{2}$″ sectional	$1\frac{1}{2}$″ segmental block to cover $1\frac{1}{2}$″ covering on 8″ pipe
			Fittings, flanges, etc.	$1\frac{1}{2}$″ × 3″ × 36″ block	$1\frac{1}{2}$″ × 3″ × 36″ block	$1\frac{1}{2}$″ × 3″ × 36″ block	$1\frac{1}{2}$″ × 3″ × 36″ block

PIPE COVERING STANDARDS FOR VARIOUS SERVICES—(*cont.*)

Normal Pipe Size.	Thickness. STD.	Thickness. Double Layer Double Standard.	Temperature of the Fluid in the Pipe.	650° F. to 800° F.		500° F. to 650° F.	
				Thickness of Fire Backing.	Thickness of 85 per cent. Magnesia.	Thickness of Fire Backing.	Thickness of 85 per cent. Magnesia.
10	1½	2½	Straight pipe	1½″ segmental	2″ segmental block to cover 1½″ covering on 10″ pipe	1½″ segmental	1½″ segmental block to cover 1½″ covering on 10″ pipe
			Bends, fittings, flanges, etc.	1½″ × 3″ × 36″ block	2″ × 3″ × 36″ block	1½″ × 3″ × 36″ block	1½″ × 3″ × 36″ block
12	1½	3	Straight pipe	1½″ segmental	2″ segmental block to cover 1½″ covering on 12″ pipe	1½″ segmental	1½″ segmental block to cover 1½″ covering on 12″ pipe
			Bends, fittings, flanges, etc.	1½″ × 3″ × 36″ block	2″ × 3″ × 36″ block	1½″ × 3″ × 36″ block	1½″ × 3″ × 36″ block
14	1½	3	Straight pipe	2″ segmental	1½″ segmental block to cover 2″ covering on 14″ pipe	1½″ segmental	1½″ segmental block to cover 1½″ covering on 14″ pipe
			Bends, fittings, flanges, etc.	1½″ × 3″ × 36″ block	2″ × 3″ × 36″ block	1½″ × 3″ × 36″ block	1½″ × 3″ × 36″ block
16	1½	3	Straight pipe	2″ segmental	2″ segmental block to cover 2″ covering on 16″ pipe	1½″ segmental	1½″ segmental block to cover 1½″ covering on 16″ pipe
			Bends, fittings, flanges, etc.	2″ × 3″ × 36″ block	2″ × 3″ × 36″ block	1½″ × 3″ × 36″ block	1½″ × 3″ × 36″ block

Double layer double standard 85 per cent. magnesia sectional covering (pipes and bends) up to and including 8″ pipe. Segmental block 10″ pipe and larger. — 300° F. to 500° F.

Standard thickness 85 per cent. magnesia sectional covering (pipe and bends) up to and including 8″ pipe. Segmental block on pipe 10″ and larger. — 100° F. to 300° F.

Single ply tar paper. Two 1-in. layers hair felt fastened with 4-ply jute twine. Tar-paper 6 oz. sheeting sewed on, and painted, out-doors. Barrets 'ever jet' water pp. — Anti-Freeze. — General Service and City Water.

1-in. tar-lined wool felt covering. 6-oz. sheeting pasted on and fastened with ¾″ × 1″ staples painted outside. Hair felt and 6-oz. sheeting for fittings, etc. — Anti-Sweat. — Drinking Water.

1½″ sectional cork covering. #16 galvanised soft steel wire. 6-oz. sheeting and glue comes with covering.

Note A.—Each layer of all two-layer block coverings is to be tied on with galvanised wire, 16 gauge on pipes over 10 ins., 18 gauge on 10 ins. and under.

Note B.—The outside layer on pipe 6 ins. and larger is to be covered with ¼ in. of S.B. Conklins #00 asbestos cement finished smooth and wrapped 6-oz. sheeting, pasted on.

Note C.—All covering to be wrapped with 6-oz. sheeting, pasted on, except as noted.

Note D.—Flat blocks are known as 'John Manville fire backing,' sectional form as 'John Manville high temperature Insulation.'

(*Power, Sept. 28, 1926.*)

STEAM DRYERS AND WATER SEPARATORS.

STEAM TRAPS

Dry steam plays a far more important part in the efficiency of engines and boilers than is generally recognised, the economical efficiency of an otherwise high-class plant being frequently marred by the waste, condensation, broken joints, etc., due to wet steam.

Water carried forward from the boiler by priming, although it does no work whatever, is nevertheless heated up to the same temperature as the steam itself, and therefore represents a loss in coal proportionate to its weight. All boilers prime more or less, but priming is invariably heavy in boilers that are worked to their limit, or in any way hard pushed.

An engine may give a satisfactory diagram and yet be anything but economical if regard is had to the quantity of water contained in the steam, or to the amount of condensation in the cylinder, which the presence of water greatly increases. In the same way a boiler may appear to have a high evaporative efficiency, judged by the quantity of water it disposes of, whereas its efficiency would frequently appear in quite a different light if judged only by the amount of water it evaporated, *i.e.*, gave off as dry steam.

Dry steam itself is not a good conductor of heat, but water is, and therefore its presence in steam pipes and cylinders should never be permitted, as by acting as a conductor between the steam and the metal it sets up active condensation and absolutely prevents economical results being obtained, even in engines of the best design.

Wet steam is very deleterious in steam turbines, causing erosion and abrasion of the blading and deposition of solid matter on the blades and passages.

In addition to the above considerations, there is also the fact that water in steam pipes invariably causes leaky joints and other annoyances of a nature only too well known.

A separator should be fitted near the engine, unless the steam is superheated, in which case the separator is of questionable service, any water drained from it being probably due to the large surface of the separator itself offers for cooling by radiation.

Separators as a rule act by causing change of direction of the steam, either giving it a definite whirling path, or by causing it to pass down one side and up the other side of a vessel having in it a plain vertical baffle.

STEAM TRAPS.

Steam traps are indispensable adjuncts to steam apparatus in which water of condensation must be drained off with regularity and certainty, and without loss of steam.

A steam trap should work reliably and economically and without adjustment with considerable fluctuations of steam pressure, and working parts should be readily accessible without having to disconnect pipes.

All steam traps are either of the expansion (thermostatic) or float (bucket) type. The three main groups of expansion traps work (*a*) on expansion and contraction of metal tubes, (*b*) on expansion and contraction of either a liquid enclosed in a tube actuating a valve attached to a piston (cartridge type), or expansion and contraction of a rod other than metal, to which the valve is attached, and (*c*) on expansion and contraction of volatile liquid enclosed in a capsule or bellows (vapour type).

Float traps are made either with closed (ball) float or with open floats operating valves either directly, or through levers, or operating pilot valve only, the main valve being operated by the steam pressure.

Expansion traps on account of their small dimensions are generally preferred for draining individual steam-heated appliances (radiators, calorifiers, evaporators, etc.). When cold they are always wide open and therefore do not require venting devices.

Float traps are desirable for draining collecting drain pipes, steam mains, oil separators, and generally where sudden variations in water quantity are likely.

'The Bradford' Steam Trap.

(*United States Metallic Packing Co., Ltd., Soho Works, Allerton Road, Bradford.*)

'The Bradford' Steam Trap embodies a new principle in this class of device. It is a float trap, but with an important difference. The float does not open or close the discharge valve. It simply releases or engages a weighted lever, which is so arranged that the outlet is instantaneously opened or instantaneously closed. There is no middle position. This effectually prevents the wire-drawing of the discharge, which cuts the valve and seat and so destroys the efficiency of those traps where the opening and closing take place gradually.

The method of operation is as follows (fig. 126). The water accumulates in the collecting chamber and gradually raises the float until the latter reaches the desired point, when it trips the weighted lever, which immediately opens the discharge to its fullest extent. The water is

then forced out under full pressure until it reaches the bottom level, when the lever is again tripped and the discharge is at one closed. The outlet is always submerged, so that there can be no escape of steam.

The apparatus is made in two classes, viz. the piston-operated trap and the direct-operated trap. Fig. 125 represents the piston-operated trap, in which the discharge valve is operated by the steam acting on a piston working in a cylinder. By this method the maximum diameter of outlet opening is obtained without reference to the steam pressure. This is a most important feature, as in consequence of the unusually large outlet the trap will deal equally well with flooding and priming as with ordinary condensation. In the direct-operated trap the action is simpler, the discharge being opened and closed directly from the trip lever, but the area of the outlet is smaller.

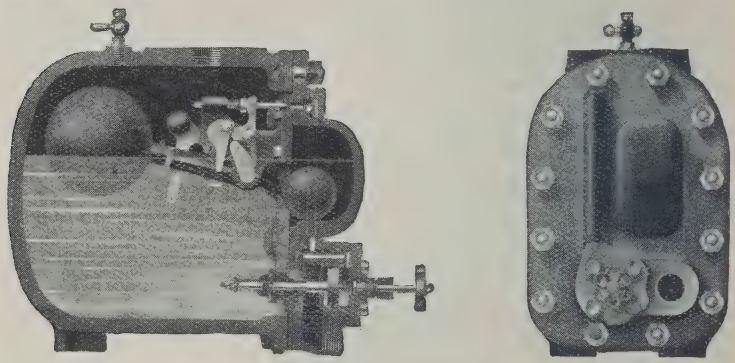


FIG. 125.—Piston-Operated Trap.

The traps will work at any pressure from 250 lbs. downwards, and are made in several sizes. The piston-operated traps will discharge from 10,000 to 160,000 gallons per hour when working at 100 lbs. pressure. In the direct-operated traps the discharge capacities range from 325 to 1,800 gallons per hour at 100 lbs. pressure. Both traps will force the discharge water 1 foot high for every pound of steam pressure.

An *eliminator*, which combines the above trap with a convenient form of separator, is also prepared, the manufacturers being The United States Metallic Packing Co., Ltd., of Soho Works, Allerton Road, Bradford.

DRAYTON HIGH-PRESSURE STEAM TRAPS.

(*The Drayton Regulator and Instrument Co., Ltd., West Drayton.*)

The 'Differential Steam Trap,' fig. 126, consists of two sections: (1) the 'Drayton Capacity Multiplier,' (2) the 'Euston' all-metal thermostatic steam trap.

The 'Drayton Capacity Multiplier' is a pear-shaped vessel having a central tube to the outlet and a nozzle situated at the highest point thereof. The entrance to the nozzle is controlled by a throttling screw with outside adjustment.

The 'Euston' steam trap contains an inner tube which forms an expanding element and carries a valve seat at its higher end. The valve is supported by a spindle passing through the trap cover and held by a thread in an exterior bridge.

Condensate and steam pass freely into the body of the Capacity Multiplier. The condensate rises evenly inside and outside the central tube by virtue of the steam pressure being transmitted to the central tube through the nozzle. The steam, having free access to the trap, expands the inner tube and the valve is held closed.

When the condensate rises sufficiently high in the capacity multiplier to seal the lower end of the nozzle, the steam is shut off from the trap; condensation causes a rapid fall of pressure in the trap and the central tube of the capacity multiplier, which permits the pressure in the main body of the capacity multiplier to lift the condensate into the trap.

The condensate which has been longest in the capacity multiplier is the first to enter the trap, and the considerable difference in temperature between this and the steam causes immediate

contraction of the inner tube of the trap and opening of the trap valve ; the condensate, therefore, is expelled through the valve, and passes to the outlet down the annular space between the inner and the outer tubes, thereby expediting the opening of the valve to its fullest extent.

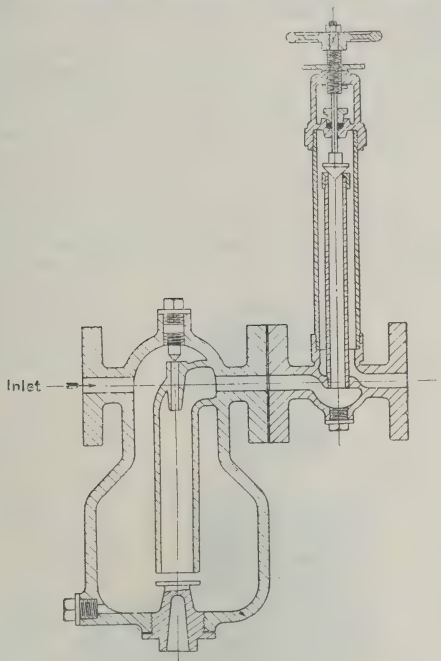


FIG. 126.

The nature of the discharge depends entirely on the amount of condensate handled. When the rate at which condensate accumulates in the capacity multiplier is slower than the rate at which it can be discharged through the trap valve, the action is intermittent ; under these conditions it will be found that the trap opens and closes sharply ; there is little or no live steam lost after the

Size	A	B	C	D	E	F
ins.	ins.	ins.	ins.	ins.	ins.	ins.
$\frac{1}{2}$	$2\frac{5}{8}$	$8\frac{3}{4}$	$5\frac{3}{4}$	$3\frac{1}{4}$	$3\frac{3}{8}$	$10\frac{3}{8}$
$\frac{3}{4}$	$2\frac{5}{8}$	$8\frac{3}{4}$	$5\frac{3}{4}$	4	4	$10\frac{1}{2}$
1	$2\frac{5}{8}$	$8\frac{3}{4}$	$5\frac{3}{4}$	$4\frac{3}{4}$	$4\frac{1}{2}$	11
$1\frac{1}{4}$	3	13	8	$5\frac{1}{2}$	$5\frac{1}{2}$	$12\frac{3}{4}$
$1\frac{1}{2}$	$3\frac{1}{4}$	13	9	$5\frac{1}{2}$	6	14
2	4	18	10	$6\frac{1}{2}$	$7\frac{1}{8}$	$15\frac{1}{2}$

trap has finished discharging. On the other hand, if the rate at which the condensate accumulates in the capacity multiplier approaches or equals the rate at which the trap valve permits discharge, the action is continuous.

When the level of the condensate in the capacity multiplier sinks to the lower end of the central tube, steam passes into the trap, closes the valve, by expanding the inner tube, and the cycle commences again. As condensate is continuously trickling into the trap, the level in the

capacity multiplier will commence to rise as soon as the flow up the central tube is broken by the unsealing of the lower end of the nozzle.

All grit, dirt, and other impurities which may come through with the steam and condensate, are retained in the base of the capacity multiplier by the mud stool. A plug is provided for draining out periodically.

The 'Drayton-Nicholson' super trap is of the type in which a float-operated lever opens and closes the small relay valve sharply and thereby controls a piston-operated main valve.

These traps are suitable for highest capacity and for any pressure, fig. 102 showing the trap for working pressure of 1,200 lbs. per sq. in.

All internal parts are made from non-corrodible material including the floats which are rustless steel balls welded in accordance with a patented process.

'Drayton' Balanced Pressure Trap.

(The Drayton Regulator and Instrument Co., Ltd., West Drayton.)

Fig. 127 shows the 'Drayton' Balanced Pressure Trap with cover and thermostatic element removed. This trap is of the thermostatic type and utilises the difference in temperature between steam and water.

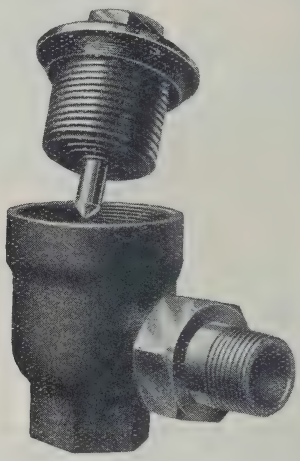


Fig. 127.

The particular feature of the 'Drayton' Balanced Pressure Trap is that the thermostatic element is so filled that the difference in pressure inside and outside the bellow chamber remains constant, no matter what the actual pressure outside the thermostatic element may be.

In place of the usual diaphragm, a bellows of large diameter is employed, and the length of the bellows is such that the valve has a lift of $\frac{3}{16}$ in.

This trap requires no adjustment for varying steam pressures, and opens and closes sharply.

General Notes on Steam-Trap Practice.

Steam traps when cold should be wide open, or air vents should be provided so that at starting all air and water are expelled from the pipes when steam commences to pass. The traps close automatically, so that there is no possible chance of water hammer and rents occurring in pipes through an accumulation of cold water suddenly receiving the impact of live steam, the cause of many accidents.

The traps should always be fixed at a point where a droop or bend in the pipe is situated and at the lowest possible point.

In the event of the drain pipe being screwed into a cylinder or pipe, great care should be taken that the drain does not rise above the inside of the vessel or pipe, otherwise perfect drainage is not possible.

It will be found that if a length of pipe is fitted between traps and vessels to be drained, so as to form a reservoir for the water, it will greatly facilitate the easy working of the trap, as the discharge water is much cooler, and enables the trap to do a larger amount of work than it would do under ordinary conditions.

Every lb. per sq. in. pressure balances a column of water 2¼ ft. high, so that in arranging for tanks, if 2 ft. of water for every 1 lb. of steam pressure is allowed for, the allowable height for placing the tank can be ascertained. In this case a non-return valve, similar to a boiler feed check valve, should be placed on the outlet side when the trap is used for lifting water, to prevent the water returning into the pipes when main steam is shut off.

When discharging into a sealed drain it is advisable to fit a three-way cock to the outlet, so that the trap can be adjusted with the cock opened to the atmosphere. There is an additional advantage in having a three-way cock fitted, as the working of the trap can be inspected at all times by simply opening the cock to the atmosphere.

By blowing the pipes through with live steam before fixing the trap, the dirt will be cleaned out, and much trouble saved in the after working of same.

The greatest precaution taken cannot always prevent grit getting into the pipes, and if same is not blown out at starting it is likely to get into the trap and become fixed in the seat or valve face. The consequence is, the trap is likely to let steam pass. Steam at high pressure cuts almost as badly as compressed air, and the longer the steam is allowed to blow through, the worse the valve is cut; when this takes place the valve should be examined, and, if cut, it should be trued up in a lathe, or if badly cut it should be renewed.

Steam issuing from the discharge of a high-pressure trap is not a proof of steam leakage, Wherever water under pressure and at a temperature above 212° is released to the atmosphere, about 1 per cent. will flash into steam for each 10° by which the original water temperature exceeds 212°.

Selection, Installation, and Upkeep of Steam Traps *

(Abstract.)

The two main factors to be considered in selecting traps are (1) the steam pressure at the trap, and (2) the amount of condensation to be handled. It should be carefully noted that the steam pressure desired is that at the trap, not the boiler pressure or the pressure at some

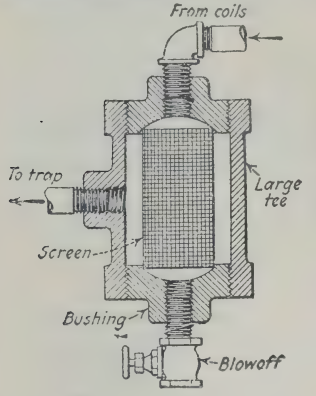


FIG. 128.

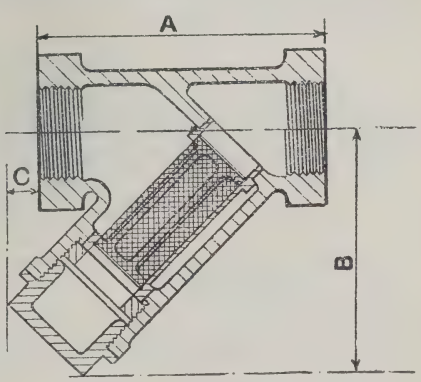


FIG. 129.

point near the trap. Where the pressure varies, the trap should have the correct opening for the higher pressure, yet it should have ample capacity to operate on the lower pressure.

The best practice is to put a separate trap on each steam coil, drying cylinder, steam main, steam separator, etc.,

When a trap is attached to a piece of apparatus using superheated steam, care must be taken that the steam is not superheated when it reaches the trap, otherwise the latter will not operate.

* E. Smiley, *Power*, July 11, 1922.

Drips on superheated lines are necessary to handle such conditions as the priming of the boilers. In such cases the connection to the trap must have enough radiating surface to ensure condensation for the trap to work on. Traps of the continuous-discharge type and many of the bucket traps require a certain amount of water in the body. If superheated steam gets into the trap, it re-evaporates the water and causes leakage and wiredrawing. On account of the tendency to wiredrawing with superheated steam and with apparatus operating at high pressure with but little condensation, a special metal should be provided in the valve and seat.

The best way to avoid trouble from dirt, scale, or sediment is to use a 'strainer' like that shown in fig. 128, which is easily made up from pipe fittings and a strip of heavy brass or copper wire screen made in the form of a cylinder, so that it will fit in the recesses of the bushing. Another convenient type of strainer is that shown by fig. 129, and made by *The Drayton Regulator and Instrument Company, Ltd., West Drayton*, which can be inserted into pipe line at the trap inlet side and can be cleaned without breaking the pipe line.

A strainer requires occasional cleaning, but that is a simple matter compared to cleaning the trap.

Many traps are provided with blow-off pet-cocks or valves, so that the traps can be blown down and cleaned out periodically. A simpler but less efficient remedy is a dirt leg in the piping in front of the trap. This can be made up with a tee, long nipple and cap. Excessive oil in the line can usually be traced to a faulty oil separator in the system.

The only way to be absolutely sure that a trap is clean is by an examination of the interior. In fact, it is good practice to inspect traps at least once or twice a year.

Although 1 lb. pressure at the trap will support 2·4 ft. of water at 212 deg., it is well to figure back pressure at the rate of about one pound for every two feet of head. This makes a reasonable allowance for friction in the pipe and fittings.

High-pressure traps should not in general discharge into sealed return pipes under a low pressure, as the breaking into steam (or flashing, as it is called) of the high-temperature water upon release will cause snapping and water hammer in the pipes. It is usually best for all traps, when the condensation is to be saved, to discharge into a vented receiver above the water line. This prevents any backing up or water hammer, and also gives an opportunity to detect severe steam leakage by an inspection of the outlet to the vent pipe leading from the trap.

A general rule on the location of a trap is to keep it below the lowest water level in the unit it is draining. This keeps the discharge line free of condensation when the pipes get cold on intermittent service, and thereby decreases the possibility of water hammer.

Traps that are outside and likely to be injured by freezing should be insulated and located as near the heating unit as possible. Their discharge lines should also be automatically drained in case they rise above the lowest water level of the trap. This prevents the condensation from flowing back into the traps when the pressure is taken off. It is of primary importance that they be accessible for periodic inspection.

ADDENDA.

Radiant Heat Transmission in Boiler Furnaces. (See p. 1542.)

In a discussion of heat transmission by radiation in boiler furnaces in *The Engineer*, May 16, 1930, a conclusion is reached endorsing the formula given originally by Hudson and modified by Orrok, viz. :

Heat in B.Th.U. per hour entering one sq. ft. of water-cooled surface exposed to radiant heat is equal to

$$CH\left(\frac{1}{1+\frac{A\sqrt{C}}{27}}\right)$$

where C = lbs. coal burnt per sq. ft. of heat absorbing surface per hour
H = calorific value of coal
A = lbs. air per lb. coal.

This formula agrees well with the results of Wohlenberg's analysis (for which see *Trans. Amer. Soc. Mech. Eng.*, 1925, 1926, 1928, 1929).

A boiler in which the heat transmission is essentially by radiation is the 'Wood' boiler, described in *Trans. Inst. Mech. Eng.*, January 1930, and *The Engineer*, January 31, 1930.

Tubes for Water-Tube Boilers and Superheaters. (See p. 1572.)

The 'Board of Trade Instructions as to the Survey of Passenger Steamships,' 1928, states :—

The minimum thickness of tubes for pressures up to 250 lbs. per sq. in. shall be determined by the following rule :

$$t = \frac{\text{W.P.} \times d}{\text{F}} + 7$$

where W.P. = working pressure in lbs. per sq. in.

d = external diameter in ins.
t = thickness in $\frac{1}{100}$ths of an in.

For the two rows of tubes next the fire and round the gaps formed in the nest of tubes for the outflow of hot gases from the fire F = 55. For all the other tubes F = 75.

Superheater tubes.—The mean thickness of the heating tubes shall be determined from the formula $t = \frac{\text{W.P.} \times d}{75} + 5$. The symbols have the meanings given above.

The recent extensions of design to plants to work at very high pressures call for the development of a new formula for high pressure and high temperature combined. For an up-to-date survey of the subject, reference may be made to a paper by Dorey, entitled, 'Tubes for High Pressure Water-Tube Boilers,' Inst. Marine Engineers, 1930 (reported in *The Engineer*, November 21 and 28, 1930). See also 'The Thickness of Boiler Tubes,' Docherty, *Engineering*, April 25, 1930.

Scale in Boilers. (See p. 1623.)

The decrease of thermal conductivity of scaled boiler tubes is indicated by the following table (from *Trans. Amer. Soc. Mech. Eng.*, 1929, page 388).

Thickness of scale inches	0·02	0·04	0·06	0·08	0·11	0·13
Reduction of conductivity, per cent. average	2·9	6·8	6·2	8·6	15·8*	6·6*

* Probably unreliable, owing to small number of tubes averaged.

Eberle states that 5·5 mm. scale influences the heat utilisation very little, not more than 5 per cent. if the scale is of low conductivity, and not more than 2 to 3 per cent. if of medium conductivity.

The Engineer of August 15, 1930, refers to report on scale issued by University of Michigan and states that porous scales are poorer local conductors than dense scales. Colloids undoubtedly exert influence on physical properties of scale, but should not be relied upon to protect boilers where slight scaling might cause failure.

Water Softening by the Zeolite Process. (See p. 1628.)

(From 'Base-Exchange or Zeolite Process, summary of existing knowledge,' Dept. of Scientific and Industrial Research, Technical Paper, 1930).

Hydrated alumino-silicates are used, which when containing a sodium base, remove the calcium and magnesium ions from the hard water, replacing them by sodium. When the action ceases to be effective, the material is regenerated by brine. Certain natural minerals, zeolites, have the required properties, but the materials are now synthetically prepared and a number are on the market.

The process is carried out under pressure in closed steel cylinders coated with protective enamel. These are half filled with zeolite, and the water is usually let in at the top. Sometimes the water flow is upwards, so that with rapid flow the resistance is reduced. The salt required for regeneration is usually about six times the weight of calcium oxide equivalent to the calcium and magnesium to be removed. It is applied as a rule as a 5 per cent. solution. Before applying it, dirt at the top of the material is removed. After the brine is pass d through, the bed of material is washed out downwards with hard water until the effluent is of zero hardness. This takes from ten to fifteen minutes, the regeneration process takes from five to twenty minutes.

The advantages of the zeolite method are :—

(*a*) It yields water of zero hardness.
(*b*) It adjusts itself automatically to variations of hardness, while the lime-soda process does not, giving either imperfect softening or excess of reagent.
(*c*) There is no precipitate to be filtered out.
(*d*) It works under pressure and thus avoids the necessity of re-pumping.
(*e*) There are no moving parts in the apparatus.
(*f*) The plant occupies less space than the lime-soda plant.

The cost of the two types of plant is about the same, but for waters of permanent hardness, the operating costs of the base-exchange system are lower ; for temporary hardness they are higher.

The disadvantages of the zeolite process are that, whereas the lime-soda process and subsequent filtering remove the salts causing temporary hardness and leave only the sodium equivalent of those that caused permanent hardness, with the base-exchange process the sodium equivalent slightly increases the original weight of dissolved solids, and the concentration of the feed water may lead to foaming, priming, corrosion due to the CO_2 evolved, and caustic embrittlement.

These disadvantages, it is said, may be overcome, but the Report states that for boiler feed water, the base-exchange process is *not* recommended for waters with much temporary hardness.

A combination of the treatment of temporary hardness by lime-soda and of permanent hardness by base-exchange is satisfactory for boiler feed and other purposes.

The process is not applicable to turbid, ferruginous, or acid waters,

The Report describes the chief methods of preparing base-exchange silicates.

SECTION XXX

PART III

RECIPROCATING STEAM ENGINES

EXPANSION OF STEAM — HORSE-POWER—STEAM CONSUMPTION—INDICATOR AND INDICATOR DIAGRAMS — ENGINE TYPES FOR VARIOUS DUTIES — WEIGHTS OF ENGINES — ENGINE DESIGN — FRAMES — CYLINDERS — BEARINGS — FLYWHEELS — GOVERNORS — PACKINGS — PISTONS — SLIDE VALVE GEARS—LINK MOTIONS

(Revised by T. B. Morley, D.Sc., M.I.Mech.E.)

Expansion of Steam.

It is found in practice that in engines using saturated steam the expansion line follows closely the hyperbolic law :—

absolute pressure × volume = a constant.

Hence it is usual to compute indicator diagrams on the assumption that the expansion is hyperbolic. With highly superheated steam the pressure falls more rapidly than the hyperbolic curve indicates.

Referring to fig. 1, the theoretical indicator diagram is A B C D O, there being no clearance and a perfect vacuum. Steam is cut off at B and exhausted at C.

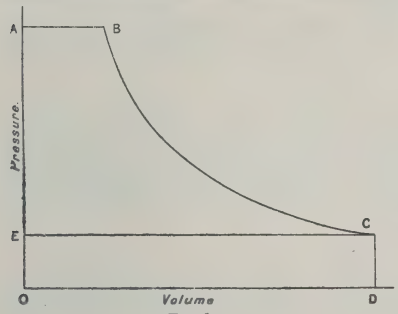


FIG. 1.

p = absolute pressure in lbs. per sq. in.;

u = volume in cubic feet;

R = ratio of expansion = $\frac{OD}{AB}$;

W = area of diagram;

P = mean effective pressure in lbs. per sq. in. (M.E.P.);

pu = a constant.

then W = ABCE + ECDO

$= p_2u_2 \text{ hyp. log } R + p_2u_2.$

$P = p_2\left(\frac{\text{hyp. log } R + 1}{R}\right) = p_2 K.$

TABLE OF MEAN PRESSURE CONSTANTS K.

The table also gives the points of cut-off (nominal), based on assumed cylinder ratios, L.P. to H.P., in the case of multiple-expansion engines.

R.	K.	Nominal Cut-off (clearance neglected) as proportion of Stroke.			
		Single Cylinder, including Uniflow.	In H.P. Cylinder of Multiple-Expansion Engine.		
			Compound. Cylinder ratio = 4.	Triple. Cylinder ratio = 6.	Quadruple. Cylinder ratio = 8.
1·25	0·979	0·8			
1·4	0·955	0·72			
1·6	0·919	0·62			
1·8	0·882	0·55			
2·0	0·847	0·50			
2·5	0·767	0·40			
3·0	0·700	0·33			
3·5	0·644	0·29			
4·0	0·597	0·25			
4·5	0·556	0·22			
5·0	0·522	0·20			
6·0	0·465	0·17	0·67		
7·0	0·421	0·14	0·57		
8·0	0·385	0·12	0·50		
9·0	0·355	0·11	0·44		
10·0	0·330	0·10	0·40	0·60	
11·0	0·309	0·091	0·36	0·55	
12·0	0·290	0·083	0·33	0·50	
14·0	0·260	0·072	0·29	0·43	0·57
16·0	0·236	0·062	0·25	0·38	0·50
18·0	0·216	0·055	0·22	0·33	0·44
20·0	0·200	...	0·20	0·30	0·40
25·0	0·169	...	0·16	0·24	0·32
30·0	0·147	...	...	0·20	0·27
35·0	0·130	...	...	0·17	0·23
40·0	0·117	...	...	...	0·20

(*Morley.*)

The referred M.E.P. of compound engines is the equivalent M.E.P. if the entire expansion were to take place in the low-pressure cylinder.

The nominal ratio of expansion is the stroke volume of the L.P. cylinder ÷ volume swept by H.P. piston up to point of cut-off, or

$$R = \frac{V_L}{a\ V_H} = \frac{\text{cylinder ratio}}{a}$$

where V_H, V_L are stroke volumes of H.P. and L.P. cylinders, and a is proportion of stroke at which cut-off takes place in the H.P. cylinder.

If C_H and C_L are the clearance volumes at each end of the H.P and L.P. cylinders, the real ratio of expansion is

$$\frac{V_L + C_L}{a\ (V_H + C_H)}$$

To determine the probable actual mean pressure (referred to the low-pressure cylinder in the case of a multiple-expansion engine) multiply the initial pressure (absolute) by K, deduct the back pressure, and multiply by a suitable 'diagram factor.'

Diagram factors depend on the type of engine, type of valves and valve gear, jacketing, superheating, speed, etc. Approximate typical values are as follows:—

TABLE OF DIAGRAM FACTORS.

Triple-expansion	·65–·75
Compound condensing, superheated	·65–·75
„ „ saturated	·7– ·8
„ non-condensing	·8–·85
Simple condensing, trip gears	·9–·95
„ „ slide valves	·85– ·9
„ non-condensing	·9–·95

Note.—In calculating the M.E.P. the clearance is ignored. If the clearance is allowed for, the ratio of expansion for a given cut-off is lower and the theoretical M.E.P. higher, necessitating a lower diagram factor, particularly for simple and compound engines. Thus, some authorities give from 0·7 to 0·9 for simple engines.

Low revolutions, large clearance, jackets, reheaters, and low compression increase the diagram factor. These special conditions must be allowed for, as in slow-speed pumping engines.

Usual Expansions in Various Types of Engines.

Single Condensing Engines	from 2 to 5 times
Two-cylinder Compound Engines	„ 10 „ 20 „
Triple-expansion Engines	„ 12 „ 25 „
Quadruple-expansion Engines	„ 16 „ 30 „
Uniflow Engines	„ 7 „ 12 „

INDICATED HORSE-POWER OF STEAM ENGINES.

P = Mean effective pressure, in pounds per square inch;
A = Area of cylinder in square inches;
L = Stroke in feet;
N = Revolutions per minute;

then I.H.P. (for a double-acting engine) = $\frac{2\ PLAN}{33{,}000}$

(See also p. 1719.)

The I.H.P. of a compound or multiple expansion engine is the sum of the powers of the individual cylinders, each calculated as above. For such engines, the 'referred M.E.P.' or 'M.E.P. referred to the L.P. cylinder' is frequently spoken of and used in designing. It means that M.E.P. which, if it acted in a single cylinder of the same dimensions as the L.P. cylinder of the compound engine and working at the same speed, would produce the same power.

The '*referred M.E.P.*' is thus equal to M.E.P. of H.P. cylinder $\times \frac{\text{H.P. cylinder area}}{\text{L.P. cylinder area}}$ + M.E.P. of I.P. cylinder $\times \frac{\text{I.P. cylinder area}}{\text{L.P. cylinder area}}$ + M.E.P. of L.P.

For a two-cylinder compound engine the middle term disappears.

Brake Horse-Power (B.H.P.).

'Brake' or 'Effective' or 'Actual' horse-power is the horse-power delivered by the engine to whatever it is driving. If the engine shaft is coupled to the driven machine—such as a generator for electric power, or a rolling mill, or whatever it may be—the B.H.P. is the power transmitted at the coupling. For a rope drive it is the power at the engine rope pulley. In all cases the B.H.P. is the indicated H.P. less the power absorbed in the running of the engine itself. This latter power—the frictional H.P.—is about 10 per cent. in good modern engines, 8 per cent. in those of the highest class, but may be as much as 20 per cent. in small engines of crude design.

Mechanical Efficiency.

The 'mechanical efficiency' is the ratio of B.H.P. to I.H.P. For high-class engines of, say, 100 H.P. and upwards, typical values of the mechanical efficiency at various loads are as follows:—

Load	Full	¾	½	¼
Mechanical efficiency, per cent.	92	90·5	88	80·5

TABLE OF STANDARD HORSE-POWER FOR DIFFERENT NATIONS.

Country.	Kilogrammètres per Second.	Baden Foot-pounds per Second.	Saxon Foot-pounds per Second.	Wurtemberg Foot-pounds per Second.	Prussian Foot-pounds per Second.	Hanoverian Foot-pounds per Second.	English Foot-pounds per Second.	Austrian Foot-pounds per Second.
France and Baden . .	**75**	**500**	529·68	521·58	477·93	513·53	542·47	423·68
Saxony	75·045	500·30	**530**	523·89	478·22	513·84	542·80	423·93
Wurtemberg	75·240	501·36	531·12	**525**	479·23	514·92	543·95	424·83
Prussia	75·325	502·17	531·97	525·85	**480**	515·75	544·82	425·51
Hanover	75·381	502·41	532·23	526·10	480·23	**516**	545·08	425·72
England	76·041	506·94	537·03	530·84	484·56	520·65	**550**	429·56
Austria	76·119	507·46	437·58	531·39	485·06	521·19	550·57	**430**

STEAM CONSUMPTION.

The ideal steam-engine cycle, adopted as a standard of comparison, is the Rankine cycle. In it the steam is fully expanded adiabatically, and there are no losses due to throttling, clearance, leakage, condensation, etc.

The following tables show the work done (in B.Th.U.) per lb. of steam, and the steam consumption per i.h.p. hour in the case of ideal engines:—

(1) when the expansion is incomplete;
(2) when the expansion is complete—*i.e* the Rankine cycle.

TABLE OF STEAM CONSUMPTION AND WORK DONE BY STEAM.

(1)

Theoretical lbs. of steam per i.h.p. hour and work done in B.Th.U. per lb. of steam, assuming adiabatic expansion from dry steam and no compression or clearance losses, to:—

(1) 25 lbs. release pressure with 16 lbs. back pressure.
(2) 10 lbs. release pressure with 2 lbs. back pressure.

All pressures are in lbs. per square inch absolute except the approximate gauge boiler pressure corresponding to the absolute initial pressure at the engine. A pressure drop of 5 lbs. between boiler and engine is assumed.

The correction given in the table is the change in theoretical steam consumption for one pound change in the back pressure, other conditions remaining unaltered.

Boiler Pressure lbs. gauge.	Initial Pressure absolute.	Back Pressure, 16 lbs. absolute.			Back Pressure, 2 lbs. absolute.		
		Lbs.	B.Th.U.	Correction per lb.	Lbs.	B.Th.U.	Correction per lb.
210	220	13·5	188	0·2	9·4	270	0·2
200	210	13·8	185	0·2	9·6	266	0·2
190	200	14·1	181	0·2	9·7	262	0·2
180	190	14·4	177	0·2	9·8	259	0·2
170	180	14·7	173	0·3	10·0	255	0·2
160	170	15·1	169	0·3	10·1	252	0·2
150	160	15·5	164	0·3	10·3	248	0·2
140	150	15·9	160	0·3	10·5	243	0·3
130	140	16·4	155	0·3	10·7	238	0·3
120	130	17·1	149	0·3	10·9	233	0·3
110	120	17·8	143	0·3	11·2	227	0·3
100	110	18·6	137	0·4	11·5	222	0·3
90	100	19·6	130	0·4	11·9	214	0·4
80	90	20·9	122	0·4	12·3	207	0·4
70	80	22·5	113	0·5	12·7	200	0·4

(*Morley*)

Table of Steam Consumption and Work done by Steam.

(2)

Theoretical steam consumption in lbs. per i.h.p. and work done per lb. of steam in B.Th.U., assuming adiabatic expansion from dry saturated steam with complete expansion. Rankine cycle.

Absolute Initial Pressure, Lbs.	I.lbs. per i.h.p. hour. Back pressure, lbs. abs.				B.Th.U. per lb. of Steam. Back pressure, lbs. abs.			
	14·7	2	1·5	1	14·7	2	1·5	1
220	12·9	8·2	7·85	7·4	198	310	324	344
210	13·0	8·3	7·9	7·45	195	307	321	341
200	13·2	8·4	8·0	7·5	192	303	318	338
190	13·5	8·5	8·1	7.5	188	300	315	334
180	13·8	8·6	8·2	7·6	184	297	311	331
170	14·1	8·7	8·3	7·7	180	293	308	328
160	14·5	8·8	8·4	7·75	175	289	304	324
150	14·9	8·9	8·5	7·95	170	285	300	320
140	15·3	9·1	8·6	8·1	166	280	295	315
130	15·9	9·25	8·75	8·2	160	275	290	310
120	16·5	9·4	8·9	8·35	154	270	285	305
110	17·2	9·65	9·1	8·5	148	264	279	300
100	18·0	9·9	9·3	8·65	141	257	273	294
90	19·1	10·2	9·55	8·85	133	250	266	287
80	20·5	10·5	9·8	9·1	124	243	259	279
70	22·1	10·9	10·2	9·4	115	234	250	271
60	24·7	11·4	10·6	9·75	103	223	241	261
50	28·6	12·0	11·2	10·2	89	211	227	249

(*Morley.*)

The steam consumption, lbs. per i.h.p. hour = 2545 ÷ work done, B.Th.U. per lb.

For pressures of 100 to 200 lbs. per sq. in. and back pressures 1 to 2 lbs. per sq. in., the steam consumption of the ideal (Rankine) engine decreases about 6 per cent. for each 100° F. of superheat.

EFFICIENCY OF ENGINES.

The thermal efficiency—of the engine alone, exclusive of boiler plant—is the ratio of the work done per lb. of steam (in B.Th.U.) to the heat required to generate 1 lb. of steam, at the pressure and temperature, at the engine stop valve from feed water at the temperature of the engine exhaust.

$$\text{Thermal efficiency } e = \frac{\text{I.H.P.} \times 42{\cdot}42}{w\,(H_1 - h_2)} = \frac{\text{I.H.P.} \times 2545}{W\,(H_1 - h_2)},$$

where,

w = steam used, lbs. per minute; W = do., lbs. per hour;
H = total heat at pressure and temperature at boiler side of engine stop valve;
h_2 = water heat at temperature of exhaust.

It is customary to express the performance of engines in steam consumption per h.p. hour, but since the heat required to generate the steam increases with pressure and temperature, the steam-consumption figure alone is not a complete criterion of the merit of the engine.

It has been proposed to state the performance of engines in B.Th.U. per minute per i.h.p $= \frac{w\,(H_1 - h_2)}{\text{I H.P.}} = \frac{42{\cdot}42}{e}$, or in 'equivalent feed,' which is the weight of steam which would be required per i.h.p. hour if $H_1 - h_2$ were 1100, thus equivalent feed $= \frac{W}{\text{I.H.P.}} \times \frac{H_1 - h_2}{1100}$.

Neither of these proposals has been widely adopted.

EFFICIENCY RATIO.

The ratio of the thermal efficiency of the actual engine to that of the Rankine cycle—or of the steam consumption of the Rankine cycle to the actual steam consumption (both having the same pressure, superheat, and vacuum (or back pressure))—is the ' efficiency ratio.'

The efficiency ratio varies with the pressure, superheat, and vacuum, as well as with the merit of the engine as regards freedom from condensation, leakage, and other losses.

Nevertheless the ' efficiency ratio ' is useful in comparing engine test results and in predicting steam consumptions.

Economy of High-pressure Working.

The consumption of steam in pounds per horse-power per hour decreases as the pressure rises. The actual steam consumptions vary according to the size and type of engine.

It is estimated that the saving resulting from pressure increase from 200 lbs. to 500 lbs., keeping steam temperature constant at 600° F. and vacuum at 28 ins., would be $1,400 per annum per 1,000 kw. (assuming boiler efficiency 70 per cent., load factor 60 per cent., and cost of 14,000 B.Th.U. coal $5 per ton). Capitalised at 17 per cent., this corresponds to a saving of $8·25 per kw. compared with a probable cost of $3·50 per kw. to work at the higher pressure.

(*J. T. Foster, in* '*Electrical World*,' *May* 11, 1918.)

Experiences with High Pressure Steam.

High pressures, 600, 1,000, 1,200 lbs. per sq. in., are being tried in power stations. A convenient method is to combine a high-pressure plant, say, at 1,200 lbs. per sq. in., with an existing plant at, say, 350 lbs. per sq. in., the steam being exhausted from the high-pressure plant at 350 lbs. per sq. in., and, after being reheated, passing to the 350-lb. turbine.

Moultrop (*A.S.M.E.*, February 4, 1929, and *World Power*, May 1929) describes operating experiences with such a system. It was anticipated that the 1,200-lb. safety-valves would have their seats destroyed after blowing once, and pilot valves were fitted to warn operators, so that they could prevent the main valves blowing. The pilot valves were found to be unnecessary.

Hand-regulation of boiler water level was unsatisfactory. Low water level was probably a cause of boiler tubes warping. Automatic feed regulation proved satisfactory.

Packing difficulties with the high-pressure centrifugal feed pumps were satisfactorily overcome.

In general, operating difficulties have been overcome and the anticipated fuel economy has been realised.

Steam Saving by Superheating.

Superheating improves the efficiency ratio, so that in practice the saving due to superheating is much greater than that calculated for the Rankine cycle.

Superheating reduces the steam consumption approximately at the rate of 1 per cent. for each 10° F. of superheat, but the rate is rather greater for the lower grades of superheat and less for higher grades.

The practical steam and fuel economy due to the use of superheated steam are about as follows, all superheats being measured at the engine stop valves :—

Approximate Percentage Reduction in Steam Consumption by Superheating. for Compound and Uniflow Engines.

Superheat, deg. Fahr.	50	100	150	200
Common Steam and Exhaust Ports . . .	12	20	26	30
Separate " " " . . .	10	16	21	25
Uniflow, about	8	12	17	21

(*Foster.*)

The following is a recent example of the saving recorded due to superheating. Ship *Minnedosa*, with reciprocating engines and L.P. turbine :

	Speed. Knots.	S.H.P.	Coal for Propulsion.	
			Tons per Day.	Lbs. per S.H.P. Hour.
Saturated steam . . .	16·33	11,500	158	1·28
Superheated steam . . .	16·53	11,650	137	1·09
Saving due to superheat, 14·8 per cent.				

(*Johnson* Inst.N.A., 1929.)

Superheat reduces the steam consumption most in simple engines.

For superheat in steam turbines see Part IV, page 1767.

The advantages of superheating in general are chiefly :—

1. Greater thermal efficiency of the engine, due to reduced condensation loss.
2. If the boiler plant is not provided with very ample heating surface the addition of a superheater increases the boiler plant efficiency and capacity apart from the engine effect.
3. As compared with saturated steam, higher steam velocities in the pipes may be used with superheated steam, and consequently the pipes may be smaller. For this reason, the percentage loss of heat in the pipe system is lower, in spite of the higher temperature.
4. Reduction in size of auxiliary plant owing to smaller weight of feed water and coal.

Degree of Superheat.

In modern practice the following are usual :—

1. General engine practice : superheat at boilers 150° F., or, say, 120° to 130° at engine. Up to 650° F. total temperature, say 260° F. of superheat at 200 lbs. pressure, can be used if the engine is designed for it.
2. General turbine practice : superheat at turbine 200°-250° F., say up to 650° F. total temperature.
3. Latest turbine practice : superheat at turbine of 250° to 300° F., say up to 750° F. total temperature, at about 350 lbs. sq. in. pressure.

In general, with flat slide or Corliss valves on the high-pressure cylinder, the lower temperatures (about 550° F. total temperature) must not be exceeded.

Characteristics of Superheat.

Condensed steam is a slight lubricant. The absence of this must be met by adequate lubrication when using superheated steam. The high temperature requires a special oil of high flashpoint. Pipe friction losses are reduced by superheating.

In reciprocating engines a later cut-off is required to develop the same horse-power. Hence the maximum power which can be developed is slightly reduced by superheating.

Very high superheats lead to growth and distortion of cast-iron turbine cylinders. Steel should be used at the high-pressure end.

Moderate superheat, about 50° or 100° F., is desirable for evaporating and similar heating purposes, especially with long pipe mains, but very high superheats are undesirable, as the steam should have little, if any, superheat by the time it reaches the heating surfaces of the evaporators.

Throttling slightly raises the superheat in steam; expansion, as in a turbine nozzle, very greatly reduces it.

Except for very moderate superheat, cast-iron and copper must not be used for pipes and valves in contact with the steam.

STEAM JACKETS.

Steam jackets are rarely adopted to-day except for slow-running engines. There is no doubt that a well-arranged jacket leads to considerable economy when the engine is supplied with saturated steam. The economy is much less marked with superheated steam, especially as jackets are usually arranged. Having regard to the extra first cost and complication, jackets are not to be recommended with superheated steam unless the conditions are exceptional.

The usual arrangement of steam supply is defective; the steam is supplied to the jacket and a trap fitted to drain away the condensed steam, but there is no through flow. Usually, on intermediate and low-pressure cylinders the jacket steam pressure is reduced to a few pounds above the steam chest pressure; except to relieve the cylinder barrels and liners of mechanical stress, there is no justification for such an arrangement; from the point of view of steam economy it is ridiculous and largely nullifies the value of the jackets. Wherever possible, the jacket steam should have a through-flow. With saturated steam the engine steam should pass through the jackets and then through a water separator before entering the cylinders.

When jackets are used with steam superheated 100° F. or more, they are unnecessary on the high-pressure cylinder.

Small and slow-running engines benefit most from jackets.

Reheaters, *i.e.* appliances for heating the steam in the receivers between the cylinders, do not appear to be useful with saturated steam.

With good initial superheat, say over 150° F., reheaters give good results.

Steam Consumption.

Typical steam consumptions, with the corresponding efficiency ratios, are shown in the following table:—

TYPICAL STEAM CONSUMPTIONS AND EFFICIENCY RATIOS.

Type of Engine.	Pressure at Stop Valve. Lbs. per Sq. In. (Gauge.)	Superheat °F.	Condensing.		Non-Condensing.	
			Steam, Lbs. per I.H.P. Hour.	Efficiency Ratio.	Steam, Lbs. per I.H.P. Hour.	Efficiency Ratio.
Simple	160	—	17 to 17·5	51 to 49	21 to 22	67 to 64
	160	150	12 to 13	65 to 61	16 to 17	78 to 74
	120	—	18·5 to 19·5	49 to 47	24 to 25	65 to 62
	120	150	14·5 to 15	57 to 55	19 to 20	73 to 70
	80	—	22 to 23	45 to 43	30 to 33	62 to 56
	80	150	16 to 17	56 to 53	24 to 26	69 to 64
Compound	200	—	12·5 to 13	66 to 64	—	—
	200	150	10 to 10·5	74 to 71	—	—
	160	—	13 to 13·5	66 to 64	17·5 to 18·5	80 to 75
	160	150	10·5 to 11	74 to 71	14·5 to 15	86 to 83
	120	—	14 to 15	65 to 61	19·5 to 20·5	79 to 75
	120	150	11·5 to 12·5	72 to 67	16 to 17	86 to 83
	80	—	—	—	25 to 26	74 to 71
	80	150	—	—	20·5 to 21·5	81 to 77
Triple	200	—	11·5 to 12	72 to 69	—	—
	200	150	10 to 10·5	75 to 71	—	—
	180	—	12 to 12·5	70 to 67	—	—
	180	150	10·2 to 11	75 to 70	—	—
Uniflow	200	—	11·8 to 12·3	70 to 67	—	—
	200	150	10 to 10·5	75 to 71	—	—
	160	—	12·5 to 13	69 to 66	19 to 20	74 to 70
	160	150	10·5 to 11	75 to 71	16 to 17	78 to 74
	120	—	13·5 to 14·5	68 to 63	21·5 to 22·5	73 to 69
	120	150	11·5 to 12·5	72 to 67	18 to 19	78 to 73

The particulars in the table refer to high-class modern engines with Corliss or drop valves, and of about 600 i.h.p. or upwards working at normal full load.

In the case of condensing engines, the vacuum has been taken as 26 ins.

The non-condensing uniflow values are subject to considerable variations.

The figures given being for engines of the highest class, for good average practice add from 5 to 10 per cent. For badly clothed or defective engines add 20 per cent. or more. For small engines add from 5 per cent. upwards according to size. The superheat is measured at the engine stop valve. The loss of temperature between superheater and engine may be from 20 to 100 degrees, according to steam-pipe conditions. Simple (alternating-flow single-cylinder) engines vary a good deal according to the cut-off. In general, a uniflow has a rather lower steam consumption than a good compound engine.

TYPICAL ENGINE TESTS.

REFERENCE No.	1	2	3	4	5	6
Type	H.T.C.C.	V.C.C.C.	—	H.T.C.C.	V.T.C.	Uniflow
Cylinder Diameter, inches	15·94 × 31·94	28 × 57	Same as No 2	26·65 × 45·3	13·7–24·4–39	30·5
" Stroke, "	48	48		47·24	72	39·0
R.P.M.	75·37	80·35	80·5	100·5	50 59	120
Stop Valve Pressure Gauge	124	170	169	146·5	175·7	116·5
Superheat, deg. Fahr.	0	117	110	171	0	142
Vacuum	27·2	25·5	25·6	27·0	27·25	27
H.P. Valves	Corliss	Corliss	—	D.B. Drop	Corliss	D.B. Drop
L.P. Valves	Corliss	Corliss	—	D.B. Drop	Corliss	—
Steam Jackets	No	No	No	L.P.	H.P., I.P., L.P.	Cyl. ends
Reheater	No	Not used	In use	No	Yes	—
Clearance Volume %, H.P.	8·5	6·65	—	5·0	—	(2)
" " %, L.P.	5·6	6·4	—	5·0	—	—
" Surface %, H.P.	310	320	—	206	—	(150)
" " %, L.P.	185	215	—	161	—	—
Nominal Cylinder Ratio	4·01	4·15	—	2·9	3·17—8·1	—
Total I.H.P.	394	1,401	1,405	1,234	576	570
I.H.P. in H.P. Cylinder %	53·2	55·7	50·0	62·0	26·2	100
" I.P. " %	—	—	—	—	32·3	—
" L.P. " %	46·8	44·3	50·0	38·0	41·5	—
Receiver Pressure Gauge	11·0	18·9	20·8	25	46·5—2·4	—
Total Piston load about H.P. lbs.	22,500	92,000	89,000	76,000	18,500	80,000
" " I.P. lbs.	—	—	—	—	19,500	—
" " L.P. lbs.	18,500	69,000	74,000	32,000	17,500	—
Cut-off in H.P. Cylinder, %	32	31	30	24	38·4	—
" I.P. " %	—	—	—	—	34·5	—
" L.P. " %	45	35	35	42	48·0	—
Dryness H.P. Cut-off %	86	103	97	114·3	—	—
" H.P. Release %	90	95	94	101·2	—	—
" I.P. Cut-off %	—	—	—	—	—	—
" I.P. Release %	—	—	—	—	—	—
" L.P. Cut-off %	71	82	90	85	—	—
" L.P. Release %	74	86	89	89	—	—
M.E.P. referred to L.P. Cyl.	27·24	28·44	28·41	32·5	26·3	33·2
" in H.P. Cylinder	58·2	67·0	60·1	58·3	57·0	33·2
" in I.P. "	—	—	—	—	22·1	—
" in L.P. "	12·75	12·53	14·2	12·36	10·9	—
Total Expansions, apparent	12·5	13·4	13·8	12·1	21·1	—
Piston Speed, ft. min.	602	642·8	644	791·2	607	780
Steam per I.H.P. hour lbs.	13·89	11·74	11·67	10·75	11·22	11·86
B.Th.U. per I.H.P. per min.*	259	226	224	218	211	236
Efficiency Ratio* %	61	69	69	70	70	68
Drive	Mill	Mill	Mill	Mill	Pumps	—
Date	1896	1907	1907	1909	1895	—
Builders	British	British	British	Belgian	U.S.A.	British
Authority	Longridge	Longridge	Longridge	Longridge	Miller	Stumpf

(*Foster.*)

* Recalculated on basis of Callendar's steam tables.—*T. B. M.*

Duty of Engines.

The duty of an engine is rarely spoken of except for pumping engines, and is then expressed in terms of the work done in lifting the water per 1,120 lbs. of steam used, or some other basis. Sometimes 1,000 lbs. of steam is used, especially in America; and sometimes coal or heat units is the basis.

Example.—Pumping engine consumes 14 lbs. of steam and 1·8 lbs of coal per water, or pump h.p. hour. The total heat added per lb. of steam is 1,125 B.Th.U.

Duty per 1,120 lbs. steam is $\frac{1,120}{14} \times 33,000 \times 60$ . . . = 158,400,000 ft. lbs.

Duty per 112 lbs. coal is $\frac{112}{1·8} \times 33,000 \times 60$ . . . . = 123,200,000 "

Duty per 1,000,000 B.Th.U. in steam is $\frac{1,000,000}{1,125} \times \frac{158,400,000}{1,120}$ = 125,700,000 "

The steam consumption of high-class slow-speed triple-expansion pumping engines is very low.

TRIPLE PUMPING ENGINE PERFORMANCE.

	(1)	(2)	(3)
Cylinder diameters, ins.	23, 43, 64	16, 30, 44	20, 36, 33
Stroke, ins.	36	36	48
R.P.M.	41·25	29·3	25·7
Pressure at stop valve, lbs. per sq. in. (gauge)	183·6	148·5	161·5
Superheat, ° F.	119	nil	34
Vacuum (corrected to 30 ins. barometer)	27·3	29·92	Not stated
I.h.p. (total)	562·5	—	267
Water h.p. (neglecting slip)	537·1	—	242·7
W.h.p. (actual)	523·9	162	—
Steam per i.h.p. hour	10·04	—	10·3
Steam per w.h.p. hour	10·78	—	11·32
Efficiency ratio, per cent.	72·9	—	—
Overall mechanical efficiency, per cent.	95·5	—	90·9
Duty in millions of ft. lbs. per 1,000 lbs. steam	183·6	163·4	192

Remarks :—(1) From official tests in 1907–8 of engine at Johannesburg waterworks. (2) Tests in 1910 at Toronto waterworks. (3) Margate waterworks. See *The Engineer*, June 3, 1921. Slip neglected.

The duty of Winding Engines is rarely measured, but is sometimes stated in terms of steam used per shaft h.p. hour, or per h.p. hour in the coal hoisted. This steam consumption increases greatly when the cages are not fully loaded and the number of winds is below maximum. Under favourable conditions compound non-condensing engines use from 30 to 50 lbs. of steam per shaft h.p. (in the coal raised) per hour. For high-pressure engines and unfavourable conditions the consumption increases to about 100 or 120 lbs., and occasionally more.

TABLE SHOWING STEAM CONSUMPTION OF WINDING ENGINES.

Colliery.	Cylinders, ins. diam. × stroke.	Boiler Pressure.	Lbs. per Shaft H.P. Hour.
Bolsover	32 × 72	—	129
Denaby and Cadeby	45 × 84	80	83
Tredegar Coal Company	18 & 28 × 48	140	177
A Lancashire Colliery	36½ × 78	60	114
Richard Evans	36 × 78	180	95
Silkstone	32 × 72	120	70
Rotherham	40 × 78	105	48
Hulton	32 × 72	80	102
Hulton	36 × 72	80	129
Atherton	22 × 52	100	58
Hickleton Main	40 × 78	100	88
Sherwood	26 × 54	127	43
Cowper	60 × 84	20	178

(*W. C. Mountain.*)

Tests of a cross-compound non-condensing engine at Sherwood colliery gave a steam consumption of 40 lbs. per S.H.P. hour. The engine cylinders were 32 ins. and 53 ins., stroke 66 ins., pressure 125 lbs. per sq. in., superheat 65° F.

For the steam consumption of turbines *see* Part IV. Steam Turbines, p. 1767.

The Indicator.

The function of the engine indicator fig. 6, p. 1716, is to produce a diagram which shall show with reasonable accuracy the pressure in the cylinder in relation to the position of the piston in its stroke, Usually the diagram is drawn on a paper wrapped round a drum, which is reciprocated by means of a cord attached to a reducing mechanism driven by the engine, so that the motion of the paper is a copy on a reduced scale of the piston motion. The pencil which draws the diagram must move parallel to the axis of the drum, so that its distance from a datum line represents the pressure in the cylinder.

For very high speeds, special forms of indicator, in which the diagram is traced out by a spot of light, and recorded photographically, are used.

The Crosby Indicator.

This indicator (sold by *Crosby Valve and Engineering Co., Ltd.*, 41–42 Foley Street, London, W. 1) has been designed for use with engines, either steam, gas, or oil. Fig. 2 shows the Crosby

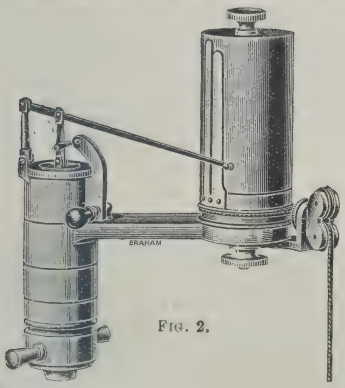


Fig. 2.

Fig. 3.

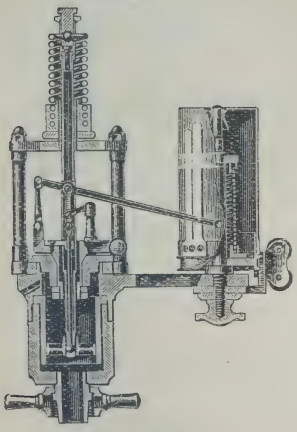


Fig. 4.

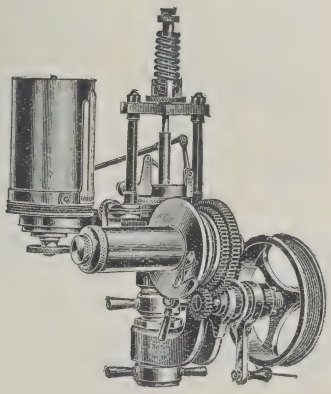


Fig. 5.

Indicator as ordinarily made, having a drum 1½ ins. in diameter, this being the correct size for high-speed work, and answering equally well for low speeds.

The special feature in the indicator is the use of a double-coil spiral spring, very carefully calibrated; this form of spring, which is shown in fig. 3, enables the strains to which it is subjected to be transmitted from the centre of the piston through a ball-and-socket joint. Each spring is made of a single piece of wire wound from the middle into a double coil, the ends of which are screwed into a head with four wings having spirally drilled holes to receive and hold them securely in place. Adjustment is made by screwing the spring in or out of this head until it is exactly of the right strength, when it is firmly brazed.

Fig. 4 shows a sectional view of the Crosby New No. 2 Indicator. This has the spring mounted outside the cylinder, where it is easily detached and replaced, and where it is removed from any effect of heat—a very necessary precaution when the indicator is used with superheated steam, or on gas engines. Another novel feature in this indicator is the shape of the piston. It is in form the central zone of a sphere, an arrangement which practically eliminates cylinder friction. The piston is connected to the pencil mechanism by a rod terminating in a ball-and-socket joint in the centre of the piston. The position of the pencil lever on the drum can be altered by screwing the spring up or down and locking it in place.

For the purpose of making the indicator of the utmost value to engineers, the Crosby New 'Continuous Drum' has been designed, and this apparatus makes it possible to obtain a series of diagrams on a roll of paper. The latest form has a novel arrangement, allowing the period between each diagram to be altered at will; the diagrams can be taken close together or farther apart as desired, and the roll supplied is 12 ft. long. Such an apparatus is of great value in ascertaining variations of load in electrical tests, and in connection with rolling and winding engines.

Another useful device, which saves having a reducing apparatus added to an engine, is the Crosby 'Reducing Wheel.' This gives a perfectly correct reduction of the stroke of any engine to the stroke of the indicator drum, and it is connected directly to the indicator and mounted on the indicator cock. The arrangement is portable and can be used on any engine.

The latest pattern includes a detent device which enables the indicator to be stopped to put on a fresh card without disconnecting the reducing wheel from the engine crosshead. See fig. 5.

To take an Indicator Card.

See that the indicator is in good order, the piston clean and free in the cylinder, all joints in the pencil gear free yet without slack. The pencil should fall by its own weight when the indicator is assembled without a spring. The spring should be chosen with reference to the pressure range, so as to give a diagram about 1¼ ins. high, or less still if the speed is high. (At very low speeds a lighter spring may be used and a larger diagram obtained.) Assemble the indicator, and see that the spring is properly secured at both ends without any slack. Couple the indicator to the cock F. Adjust the length of cord connecting the indicator drum to the reducing gear, so that there is no knock or slack at either end of the motion of the drum. At high speeds the tension of the drum spring should be considerable, so as to minimise variations in the tension of the driving cord.

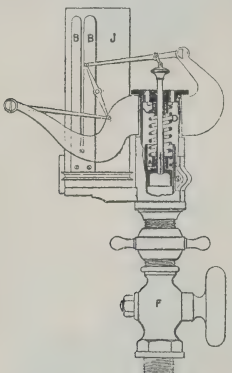


FIG. 6.

Fasten the paper on the drum J by means of the clips BB. Draw the atmospheric line by pressing the pencil lightly on the paper and pulling the drum round by the cord.

Adjust the pencil stop so that the pencil cannot be pressed too hard on the paper. The pencil point must be kept smooth and fine. A clear, fine line will thus be drawn without excessive friction between pencil and paper.

Connect the driving cord to the reducing gear, open the indicator cock, and after allowing steam to blow through for a few seconds, apply the pencil to the paper and allow it to remain in contact for a few revolutions and then withdraw it.

Shut the cock, disconnect the cord and remove the paper with the indicator diagram.

Mark the card with a reference number and write on it the particulars of speed, pressure, scale number of spring, etc., unless these are separately logged.

Scale Numbers of Springs.

The scale number of the spring is stamped upon it. For British springs, this number is the pressure in lbs. per sq. in. which will cause the pencil to travel 1 in., the indicator piston being 0·5 sq. in. in area.

For higher pressures, half-size pistons (0·25 sq. in.) may be used, and the pressure corresponding to nil. of piston travel is then twice the number marked on the spring.

Continental indicator springs are usually marked with the number of millimetres of piston travel which corresponds to a pressure of 1 kg. per sq. cm.

Indicator Piston.

In any indicator the piston should always fit the cylinder loosely enough to allow the steam to blow by it. This is very important, whether the instrument is used at high or low speeds. In regard to any errors that may result from leakage by the piston, it should be distinctly understood that leakage, unless it be sufficient to add to the atmospheric pressure above the piston, cannot affect the accuracy of the indications. Absurd devices have been even patented for keeping the piston tight, while the one thing needful is absolute freedom of motion.

In order that leakage past the piston may not accumulate pressure above the piston, the space above the piston must be in free communication with the atmosphere.

Indicator Connecting Pipes.

In general, short direct connections are preferable.

The use of a long connecting pipe may cause quite an appreciable addition to the clearance volume of a cylinder, especially in the case of small engines. Where, however, the connecting pipe has a bore of $\frac{9}{16}$ in. or $\frac{1}{2}$ in., as is usual, and suffers no contraction up to the indicator tap, and the speed is moderate, there is no appreciable error in the diagrams due to the connecting pipe being even 4 feet long, provided the bore of the two-way cock is quite $\frac{3}{8}$ in.; a bore of $\frac{1}{16}$ in. only may introduce serious error.

(*T. B. Morley*, '*Trans. Inst. E. and S.*,' 1914–15.)

THE INDICATOR DIAGRAM.

General Definitions.

The Atmospheric Line, AB, fig. 7, is a line drawn by the pencil of the indicator when the connections with the engine are closed and both sides of the piston are open to the atmosphere. This line represents on the card the pressure of the atmosphere, or zero gauge pressure.

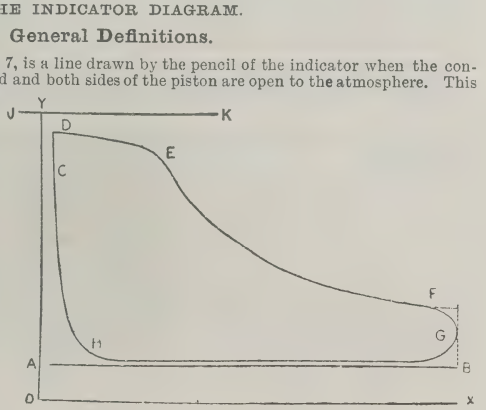


FIG. 7.

The Line of Absolute Zero Pressure, OX, is a reference line usually drawn about 14·7 lbs. by scale below the atmospheric line. It represents a perfect vacuum, or absence of all pressure.

At any considerable height above sea level the line OX becomes nearer the line AB.

The Clearance Line, OY, is another reference line drawn at a distance from the end of the diagram equal to the same percentage of its length as the clearance volume is of the piston displacement. The distance between the clearance line and the end of the diagram represents the volume of the clearance and of the ports and passages at the end of the cylinder.

The Line of Boiler Pressure, JK, is drawn parallel to the atmospheric line, and at a distance from it by scale equal to the boiler pressure shown by the gauge. The difference in pounds between it and DE shows the loss of pressure due to the steam pipe and the ports and passages in the engine.

The Admission Line is in two parts, CD, which show the rise of pressure due to the admission of steam to the cylinder by opening the steam valve (if the steam is admitted quickly when the engine is about on the dead centre this line will be nearly vertical), and

DE, which is drawn when the steam valve is open and steam is being admitted to the cylinder.

The Point of Cut-off, E, is the point where the admission of steam is stopped by the closing of the valve. It is difficult to determine the exact point at which the cut-off takes place. It is usually located where the outline of the diagram changes its curvature from convex to concave.

The Expansion Curve, EF, shows the fall in pressure as the steam in the cylinder expands and does work.

The Point of Release, F, shows when the exhaust valve opens.

FG represents the change in pressure that takes place when the exhaust valve opens; exhaust continues to the point H.

The Back Pressure Line, GH, shows the pressure against which the piston acts during its return stroke. On diagrams taken from non-condensing engines it is either coincident with or above the atmospheric line, as in fig. 7. On cards taken from a condensing engine, however, it is found below the atmospheric line, and at a distance greater or less according to the vacuum obtained in the cylinder.

The Point of Compression, H, is the point where the exhaust valve closes. It cannot be located very definitely, as the change in pressure is at first due to the gradual closing of the valve

The Compression Curve, HC, shows the rise in pressure due to the compression of the steam remaining in the cylinder after the exhaust valve has closed.

The Mean Effective Pressure (M.E.P.) is the mean net pressure urging the piston forward.

The Initial Pressure is the pressure acting on the piston at the beginning of the stroke.

The Terminal Pressure is the pressure above the line of perfect vacuum that would exist at the end of the stroke if the steam had not been released earlier. It is found by continuing the expansion curve to the end of the diagram, as in fig. 7.

Defects in Diagrams.

A very common and serious defect in diagrams arises from the stretching of the indicator cord. If two diagrams taken from the same engine at the same time show noticeably unequal lengths the cords are probably inaccurate. As a rule the stretch is greatest when the drum is pulled open. Thus one end of the diagram is widened and the other narrowed, giving incorrect M.E.P. and points of valve action.

The cord used should be as short and as inextensible as possible.

The reducing gear should give an accurate copy of the motion of the engine crosshead, and the cord from the reducing gear should run in a direction parallel to the line of motion of the crosshead.

If the diagram is very much to one side of the card, and if the near end is quite straight and vertical, the drum is probably hitting its stop at one end of its travel. Alter the length of the indicator cord.

If the expansion line is stepped in rough terraces the indicator piston is probably sticking and the M.E.P. is too high.

If the expansion is wavy the spring is too light and should be replaced by a stiffer one.

Mean Effective Pressure.

This is best determined by finding the area of the indicator diagram; dividing this by the length, so obtaining the mean height; then, multiplying the mean height by the pressure scale the mean effective pressure is the result.

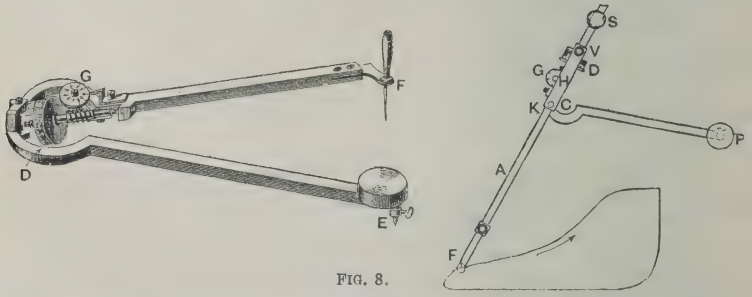


FIG. 8.

The best method of measuring the area of the diagram is to use a planimeter, which if carefully handled gives results accurate to hundredths of a square inch.

The most commonly used planimeter is of the Amsler type, and is shown by fig. 8.

It consists of two arms hinged together; one is constrained to swing round a fixed pivot, the other carries a tracing point which is made to follow the outline of the diagram. This arm also carries a roller, with its axis in the centre line of the arm, and provided with a scale and vernier by which its rotary movements can be accurately measured. As the tracing point moves round the diagram the roller partly slides and partly rolls over the paper or board on which the diagram is drawn or to which it is pinned. The readings of the scale before and after each exact circuit of the diagram are noted; their difference measures the area of the diagram on a scale determined by the proportions of the instrument.

The roller should be pivoted so as to rotate freely, but without endshake, and the surface on which it moves should be flat and unglazed. The position of the pivot point should be chosen so that the tracing point can reach freely all parts of the outline. The pivot point has a needle which can then be pressed into the paper or board.

It is advisable to check the planimeter occasionally by running it round a simple figure, such as a square or circle, the area of which can be easily calculated independently.

To Calculate the Horse-Power of an Engine.

P = M.E.P. (mean effective pressure) on piston, in lbs. per sq. in.;

A = net piston area in sq. inches. (The net area at the crank end of the cylinder = area of cylinder less area of rod. If there is a tail rod at the end remote from the crank, A = cylinder area less area of tail rod);

L = length of stroke in feet;

N = revs. per minute.

Then the I.H.P. developed on *one side* of the steam-engine piston = $\frac{\text{PLAN}}{33{,}000}$

For a double-acting steam-engine, the I.H.P. developed in the cylinder = $\frac{2\ \text{PLAN}}{33{,}000}$, where P, A, are the mean M.E.P., and mean area for the two cylinder ends.

More exactly I.H.P. = $\frac{(P_1A_1 \times P_2A_2)\ LN}{33{,}000}$,

where P_1, P_2, A_1, A_2, are the pressures and areas for the two sides respectively.

For a gas engine (single acting, single cylinder) the I.H.P. = $\frac{\text{PLAE}}{33{,}000}$, where E = explosions or impulses per minute, and depends on the cycle and governing system as well as on the revs. per minute.

For a double-acting steam-engine cylinder

I.H.P. = $\frac{\text{PAS}}{33{,}000}$, where S = piston speed in ft. per minute, since S = 2 LN.

Approximate Steam Consumption, from the Indicator Diagram.

Draw a horizontal (constant pressure) line on the diagram so as to cut the expansion line and compression line (or compression line produced, if necessary) at a pressure just above release (exhaust opening).

If

u = volume in cu. ft. of one lb. dry steam at the pressure of this horizontal line;

a = length of horizontal line between expansion and compression lines;

b = length of stroke of diagram;

then

$$\text{Lbs. dry steam on diagram per i.h.p. hour} = \frac{13{,}750\ a}{\text{M.E.P.}\ b\ u}.$$

This is often referred to as the theoretical or 'indicated' steam, the actual being less or greater according to the degree of superheat, jacketing, leakage, etc. Hence the theoretical consumption must be *divided* by a factor (which is approximately the dryness fraction at exhaust opening).

For a compound engine first find the 'indicated' steam consumption for either cylinder as above; then for the whole engine, multiply by the i.h.p. of that cylinder and divide by the i.h.p. of the engine. This result divided by the steam consumption factor gives the probable actual steam consumption of the engine.

STEAM CONSUMPTION FACTORS.

Cylinder.	H.P.	L.P.
High superheat	1·0 –1·1	0·8 –1·0
Moderate superheat	0·9 –1·0	0·75–0·95
Saturated steam	0·8 –0·85	0·7 –0·75
Badly clothed, saturated steam	0·65–0·75	0·6 –0·7

To Interpret the Indicator Diagram.

Besides the calculation of horse-power and the rough estimate of steam consumption, from the diagram, other important information can be learned from a study of the diagram, as follows :—

The diagram will show the opening and closing of the valves, as explained on p. 1717, fig. 7. If the point of cut-off is fixed, as with most slide and piston valve engines, the depth of the steam admission line DE below the boiler pressure line YK is a measure of the loss by throttling. If this is invariably considerable, it may pay to shift the eccentric so as to give an earlier cut off. With a simple slide or piston valve this would make the exhaust open earlier and the compression commence earlier. These, and particularly the latter, must be watched, as too early compression sends the compression pressure above the steam chest pressure. This is not good; although the compression line should rise nearly to the admission pressure. In the case of a reversing engine having a Stephenson, Joy, or similar link motion, the point of cut-off is varied by shifting the equivalent eccentric; and thus at early cut-off the exhaust and compression are both early, but at late cut-off the exhaust and compression both become late. Hence when studying such diagrams these changes of exhaust and compression with cut-off must always be kept in mind. The points of valve action can also be altered by changing the lap. Thus if the exhaust edge of the valve is chipped the exhaust takes place earlier and the compression later. Similarly chipping some of the lap off the steam edge of the valve makes the cut-off later and lead earlier.

If the admission line DE falls rapidly, the valve is too small or the valve travel too small.

In a multiple-cylinder engine the diagrams give the distribution of power and forces among the several cylinders. As a rule these should be roughly equal for the several cranks, but need not be equal for the two cylinders of a tandem engine. Indeed, in the latter case, the h.p. cylinder may with advantage take rather more than half the load if the steam is superheated.

Valve setting should always be checked by actual indicator cards after an engine is first started to work.

It is found in practice that the expansion line (fig. 7, p. 1717) is closely hyperbolic when using saturated steam, the two axes of the hyperbola being the absolute zero line OX and a vertical touching the diagram at CD. Hyperbolic curves can be purchased printed on transparent paper for placing over indicator diagrams. If the expansion line rises above the hyperbola as the pressure falls it indicates leakage past the steam valve into the cylinder. Similarly, if the expansion line falls more rapidly than the hyperbola it indicates either leakage past the exhaust valve out of the cylinder, excessive condensation of steam in the cylinder, or the use of initially superheated steam. If superheated steam is used, the vertical axis of the hyperbola should be about the clearance line OY in fig. 7; but the exact position depends on the degree of superheat.

The hyperbola is easily drawn more or less completely as desired when it is remembered that the absolute pressure multiplied by the volume has the same value at all points on the same hyperbolic curve.

ENGINE TYPES FOR VARIOUS DUTIES.

Each type of engine has a field, not sharply defined, in which it shows to best advantage.

Elaborate and expensive types are least desirable when coal is very cheap, the working hours few, labour unskilled, or the working conditions very rough.

Non-condensing engines consume from 20 to 50 per cent. more steam and coal than condensing engines.

Unless floor space is limited or very high speeds are desired, a horizontal engine is generally more satisfactory than a vertical. It is cheaper to install and more accessible for running and overhauling.

Steam turbines are relatively uneconomical in very small units. In the neighbourhood of 1,000 to 1,500 horse-power they surpass the best reciprocating engines, and for very large units—unless for some special purpose, as driving a reversing rolling-mill—are without serious rival. To show to advantage a turbine must be run condensing, with a good vacuum.

Where economy is at all important the ordinary simple (single-expansion) engine is at a marked disadvantage, which, however, is least with steam superheated 120° or 150°F. For most purposes the compound is the standard type, although the uniflow (a special form of single-expansion engine) is widely adopted. Triple and quadruple expansion engines are more economical than compound, but are more costly and complex. They are therefore chiefly employed on ships, for towns pumping, flour mills, and other situations where coal consumption is specially important. Their superiority over the compound engine is reduced if the steam is superheated.

The class of engine is frequently designated by the valve gear. Thus the plain slide valve gives the cheapest engine and also, other things being equal, the least economical.

The piston valve is really a circular slide valve, and gives an engine of similar characteristics. For the highest economy and class of engine Corliss or drop valves are desirable. The Corliss valve is really a circular slide valve of small diameter which rotates to and fro through a small angle in its valve chamber. The drop valve may be of the double-beat seating type or of the double-beat piston type. As a rule Corliss and drop valves are operated by a trip valve gear or other form of valve mechanism enabling the cut-off to be automatically varied to suit the load. Slide and piston valves, on the other hand, are not so suitable for cut-off governing and are not usually so arranged, but rely on throttling the steam to meet changes of load. High-speed engines almost invariably have piston or slide valves, because trip and other cut-off gears are unsuitable at high speeds of revolution.

Cut-off, or expansion, governing gives the best economy for variable loads, except at very light loads, when throttling is superior. The difference is least with highly superheated steam, which tends to equalise the steam consumptions at different loads (measured per horse-power, of course).

The superior economy of Corliss and drop valve engines is often believed to be due to the superior qualities of trip valve gears. Actually this is a very minor factor. The chief gain arises from the fact that the inlet and exhaust ports and valves are usually quite separate.

The following list shows the engine types frequently adopted for various duties:—

Duty.	*Type of Engine.*
Agriculture, occasional service	Simple, slide or piston, N.C.
„ regular service	Compound, slide or uniflow, N.C.
Cement works	Compound, Corliss or drop, uniflow, turbine, S.
Electric generating, small	Compound, high-speed vertical, Corliss, or drop, S.
„ „ medium	Compound, Corliss or drop; uniflow; or turbine, S.
„ „ large	Turbine, S.

Duty.	*Type of Engine.*
Factories, cotton mills, etc.	Compound, Corliss or drop, uniflow, geared turbine, S.
Flour mills	Compound or triple, Corliss or drop, uniflow, S.
Paper mills	Compound, Corliss or drop; uniflow, turbine, also with steam extraction, S.
Portable	Same as agriculture.
Pumping, centrifugal	Compound, usually high-speed vertical, also uniflow if at moderate speed, S.
,, hydraulic and town water	Triple three-crank vertical, slow-speed, Corliss or drop, S.
,, well pumps	Compound or triple, Corliss or drop, S.
Rolling mills, not reversing	Horizontal compound, Corliss or drop, uniflow, S.
,, ,, reversing	Two- or three-crank, piston valves, simple or tandem compound, often N.C., S.
Winding	Two-crank simple or compound (tandem usually), N.C., S.

Note.—N.C. indicates non-condensing. S. indicates superheated steam.

Engine Types, General.

As coal increases in price the general tendency must be towards the more efficient engines Thus, for agricultural, portable, and other 'occasonial' engines, superheated steam is highly desirable. Condensation is often difficult to arrange, and this makes the merit of the uniflow engine less pronounced. Compounding does not add very much to the cost and justifies itself where the total hours worked in the year reaches a reasonable figure.

For factory working, a high-class compound or uniflow engine or turbine is essential and the steam should be superheated 100 to 150° F.; similarly for electric generation, other than occasional or intermittent use.

For large electric generating sets the steam turbine alone need be considered, and should use high pressure and high superheat.

For rolling mills and winding engines a fairly high degree of superheat, 100 to 150° F., is advisable. If non-condensing the exhaust steam should be utilised in a low-pressure turbine. If this is not convenient, condensation should be employed, although it is not yet usual. In that case compounding should be adopted.

The Uniflow Engine.

The special characteristic of the uniflow engine is that the steam is exhausted through a central ring of ports; the piston has a length about 90 per cent. of the stroke, and uncovers the exhaust ports at about 90 per cent. of the working stroke (see fig. 9). The steam enters by inlet valves, usually of the double-beat seating type, placed at the cylinder ends, and, as it leaves at the middle, the inlet ends are never cooled by exhaust steam rushing over their surfaces, nor are the exhaust ports exposed to live steam. The clearance volume is 2 to 3 per cent. of the stroke volume, and compression normally begins when the piston covers the exhaust ports—*i.e.* at about 10 per cent. of the compression stroke.

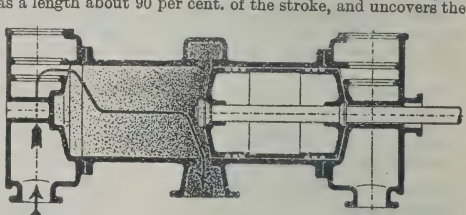


Fig. 9.

The high compression serves to heat not only the clearance surface, but the piston face, and condensation during admission is reduced to a minimum. Cyclic temperature changes in the cylinder walls are reduced, and the temperature falls gradually from the ends to the exhaust belt.

The entire expansion takes place in one cylinder, hence early cut-off is necessary.

The number of expansions is usually about 10; that is, about 10 per cent. cut-off. This is not quite so high a figure as for a typical compound engine, but owing to the absence of receiver losses and reduced initial condensation, a better economy by 5 to 10 per cent. is obtained with the uniflow, if condensing.

The full steam pipe pressure comes on the large piston, and the load on the line of parts is high in proportion to the M.E.P. This makes for weight and cost, but higher unit stresses, if based on steam pressure only, are permissible if due advantage is taken of the heavy reciprocating parts and high rotational speed to give a high inertia loading. The uniformity of turning is then equal to that of a good tandem compound engine, but is inferior at low rotational speeds.

Owing to the high compression ratio, if the vacuum falls below normal, the pressure at the end of compression would become excessive unless an efficient preventive device ('compression release gear') were provided.

The Galloway Uniflow Engine.

(*Galloways, Ltd., Manchester.*)

The steam valves are double-seated drop valves, and are placed in the cylinder covers, which form steam jackets for the valves and cylinder ends. In boring the cylinder, allowance is made for the higher temperature, and consequent greater expansion, at the ends when working. The cylinder is provided with an inspection door, by removing which the piston rings and cylinder bore may be inspected.

The novel features of the engine described are the patent shockless high-speed valve gear and the automatic compression release gear. In this valve gear (fig. 11) oil under pressure is

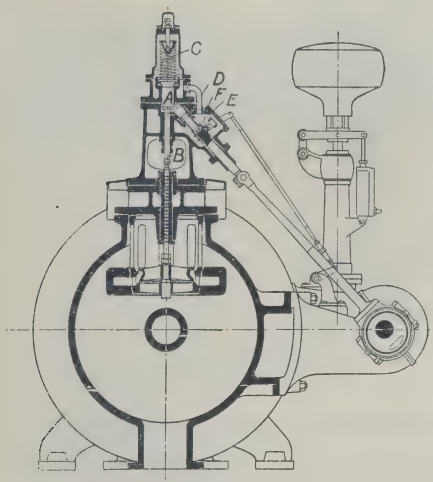


FIG. 10.—Cross Section of Cylinder.

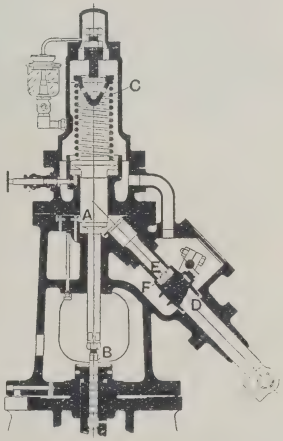


FIG. 11.—Arrangement of Oil Valve Gear.

applied below the plunger A connected to the valve spindle B and lifts the valve. An eccentric on the lay shaft operates the ram D of an oil pump in the valve bonnet. The ram is hollow, and has a ring of slots or ports E, which at certain parts of the travel may be covered by a sleeve F, the position of which is controlled by the governor. On the upstroke of the pump oil is forced into the chamber below the plunger, thus raising the plunger and the steam valve. At a point determined by the position of the governor, the ports in the hollow ram are overrun by the sleeve and the oil escapes. The valve is then free to close under the combined

action of a spring and dashpot. The valve gear is silent, compact, and takes little power to operate it.

The automatic compression release gear is shown in fig. 12. Two supplementary exhaust valves G, H, are arranged on the side of the cylinder. They are normally kept closed by springs J, K. On the engine lay-shaft is a sleeve L carrying two cams of tapering form, which are capable of longitudinal movement. Their axial position is made to correspond automatically with the vacuum by means of the spring-controlled piston O of a small cylinder P at the end of the lay-shaft, one end of this cylinder being in communication with the condenser. The

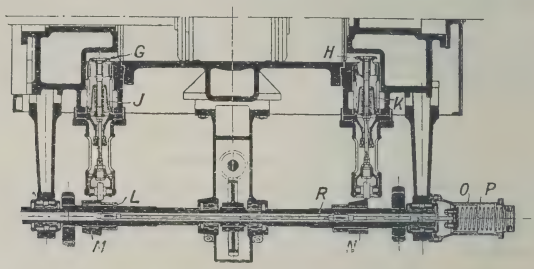


FIG. 12.—Arrangement of the Automatic Exhaust Gear.

piston operates the cams by the rod R. At normal vacuum the small end of each cam is just clear of the roller on the valve stem, and the valves are not opened. If the pressure in the condenser rises the cams move axially, so that each exhaust valve is opened during part of the compression stroke, the amount and duration of the opening depending on the cam position, and hence on the vacuum. This engine starts without vacuum, as it drives its own air pump, and the compression release gear gives full relief of compression until a vacuum is established, and then permits the compression to increase until at normal vacuum full compression is reached. Fig. 13 shows the operation of this gear

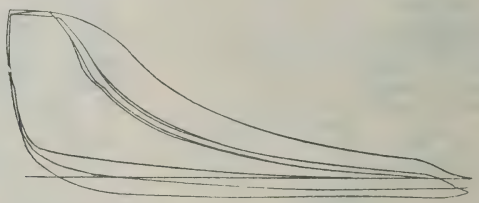


FIG. 13.—Indicator Diagrams, showing Operation of Compression Release Gear.

THE PAXMAN POSITIVE VALVE GEAR ENGINE.

The Paxman type of engine is chiefly noticeable for its simple positive valve gear, its high rotational speed, massive construction, and silent running, its sensitive inertia governor, frictionless metallic packing, and also for its low steam consumption, a result of under 10 lbs. per i.h.p. hour being obtainable under favourable working conditions.

With the Paxman valve gear, fig. 14, no dashpot is used. The valve is raised by the cam lever A, acting upon the roller B, this lever being moved by the eccentric, and the whole movement is accomplished without noise or hammering. The roller B is always kept in close contact with the cam or lever A during the travel of the valve, by means of the spring shown in the

bonnet. The cam does not move clear of the roller until after the valve is seated, and consequently, even when the engine is running at the highest speeds, the gear is entirely silent. The friction to which the valve is exposed is practically nil.

There is no tripping movement with the valve gear, but the valve is lowered gently on to its seat by means of the cam (the profile of which is so shaped as to allow this to be done rapidly) and when the valve is seated, and not until then, the cam moves clear of the roller. The system renders quite permissible the employment of such high piston speeds and revolutions (even in the case of very large engines) as could not be possible with trip-gear.

The valves are free to lift easily in the case of any abnormal pressure arising within the cylinder, and they thus act as relief valves, avoiding all danger of water shock.

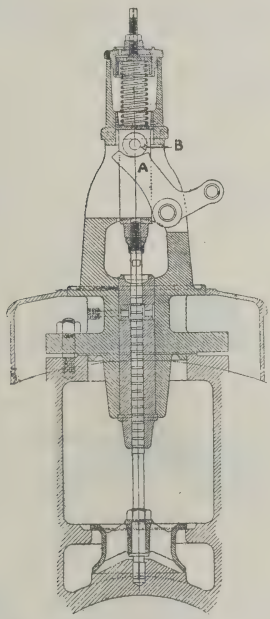


FIG. 14.

The valve spindles are grooved and work in cast-iron sleeves; by means of the Paxman system of draining, no stuffing boxes are required, which reduces the friction to a minimum, and allows the governor to be extremely sensitive while having great stability.

The governor is of the inertia type, and is placed on the side shaft; it is directly connected to the two high-pressure steam eccentrics, which eccentrics are movable on a slide block keyed to the shaft: thus the governor can alter the eccentricity, and consequently the cut-off. It regulates within an exceptionally short period, and the momentary change of speed due to the load being thrown on or off is only slightly in excess of the permanent change. Fig. 15 shows graphically the very slight alteration in speed occasioned by throwing the lead suddenly off or on the engine, the resultant change of speed being hardly perceptible. The action of the Paxman governor is immediate, and it combines extreme sensibility with a perfect stability comparable with the best turbines.

Fig. 16 shows the piston-rod packing of the Paxman effective type. By eliminating springs, friction is reduced to a minimum, and as the packing is free from wear, perfect steam

tightness is assured. The packing consists of a series of cast-iron rings which fit over the rod, and are held in cast-iron boxes which allow them to oscillate sideways, thus avoiding friction between rod and rings. There is no gland bush, the steam having direct access to the packing, but it returns to the cylinder during the period of expansion, instead of leaking through the packing. The rings form a series of chambers which are efficiently drained, any steam escaping

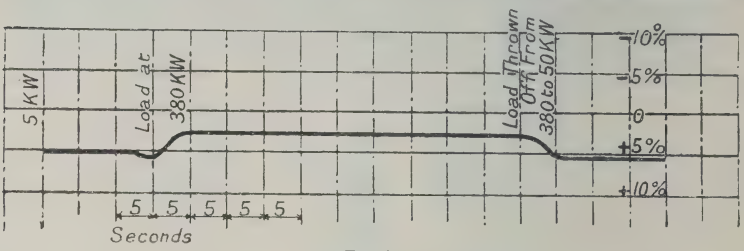


FIG. 15.

from the front chamber entering the second chamber, and there expanding. A small part of it may escape to the third chamber and undergo further expansion, and so on, but the last ring will never show the slightest leakage.

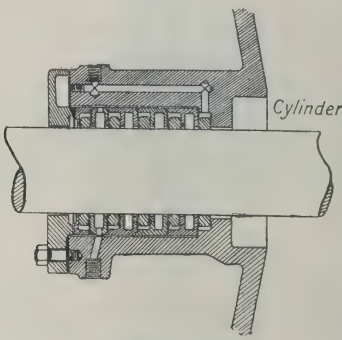


FIG. 16.

Paxman engines are made of the single-cylinder coupled compound, or tandem compound horizontal types, and for any steam pressure up to 190 lbs. at the engine stop valve.

The Paxman Positive Valve Gear lends itself to the easy conversion of existing engines to Paxman-Lentz type. In actual practice, improvements and economies in steam consumption up to 26% have been obtained with a great reduction in maintenance expenses.

ROBEY DROP-VALVE REVERSING AND OVERWINDING GEAR FOR WINDING ENGINES.

(Patent.)

The distinguishing feature of this gear, fig. 17, is its simplicity combined with absolute efficiency.

Both steam and exhaust valves are actuated by the one eccentric marked A. This is effected by the rod B actuating the rocking lever C, which is supported on a fixed bearing **D**, giving an up-and-down motion to the steam actuating lever E, and the exhaust lever F ; the latter, by means of the bell crank lever G, opening the exhaust valve at the proper moment.

The steam admission rod E by means of the well-known Robey trip-motion opens the steam valve. The rods E and F have a constant movement, and the reversing is done by varying the

position of the rod B. This is effected by moving the eccentric A (the strap and connections are omitted in the figure). This eccentric is mounted on a square block H, which is driven from a fixed plate on the layshaft.

By means of the sliding sleeve on the layshaft, and two bell crank levers, the eccentric A can be moved to the position indicated by the dotted lines, and the engine will then run in the opposite

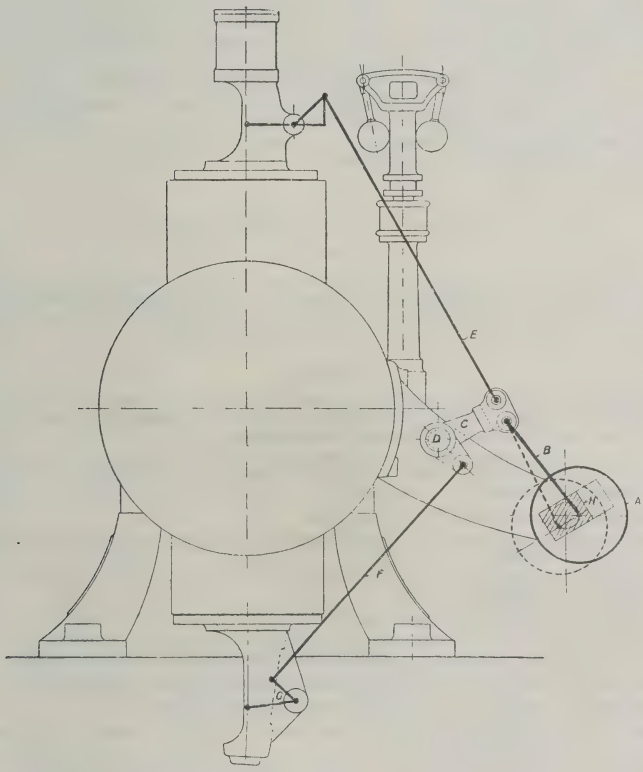


FIG. 17.—Robey Drop-Valve Reversing Gear.

direction. The sliding sleeve is connected by a simple fork connection to the reversing lever and the power necessary to move this eccentric A is very small indeed, and the lever can be worked by hand in engines up to the largest sizes.

Each end of the cylinder has an exactly similar arrangement of the gear, and the same sliding sleeve actuates both eccentrics.

When it is desired to have the gear under the control of the governor, and not by hand alone, it is necessary to have a small auxiliary eccentric on the layshaft, which is keyed on in a fixed

position, and does not vary with the direction of running of the engine. This auxiliary eccentric, in conjunction with the governor, regulates the exact cut-off in accordance with the speed of the engine.

When the governor is in the lowest position, the cut-off is at the maximum, and the driver has therefore only to turn full steam on at the commencement of the wind, as the governor will cut off the steam when maximum speed is obtained. The connection of the governor to the gear is automatically thrown out of action when the engine is reversed, thus enabling the driver to put full steam against the engine.

The steam admission valves are of the well-known Robey type, and there is no liability whatever for the valve to hold up. As a measure of precaution, however, an arrangement is fitted which closes the valve positively if it should by any means not fall, but this has never been known to come into action.

Indicator diagrams show that the lead at all points is practically constant.

In the patent overwinding gear the steam is shut off immediately at the cylinder instead of at the throttle; this renders the system particularly effective.

The overwinding gear is absolutely part of the valve gear, and has not to be driven by an independent chain or gearing of any description.

The Robey High-Speed Vertical Engine.

(Open Type.)

Figs. 18 and 19 show the latest type of high-speed vertical engine with open front. It is fitted with the Robey improved crankshaft throttle governor driven direct from the crankshaft without either gearing or belt, thus ensuring absolute reliability. The flywheel can be supplied either suitable for belt driving or of the disc type suitable for direct coupling.

Specification.

Cylinder is made of the best cold-blast iron, free from all defects, fitted with bright covers secured by studs and nuts. All stuffing boxes have gunmetal neck bushes and gunmetal glands. Cylinder drain cocks are fitted with copper-pipe connections.

Governor is of the crankshaft throttle type, dispensing with either belts or gearing for actuating governor. The speed of the engine can be varied while running by means of the adjusting arrangement fitted on the governor.

Piston is of cast iron fitted with expanding rings.

Piston Rod is of steel, coned at top end, fastened to end of piston by nut and pin, and forged solid with crosshead.

Crosshead is of forged steel, fitted with cast-iron slipper, with ample bearing surface and suitable means of adjustment.

The gudgeon pin is of case-hardened tough mild steel ground up true.

Connecting Rod is of best mild steel finished bright all over, fitted with marine large end, and small end provided with adjustment. The top is fork ended and with gudgeon pin shrunk in.

Valve Rod is of best mild steel with case-hardened eye adjustable by a small cotter.

Crankshaft is of steel turned all over and of the balanced slab type.

Standard is of special design with open front, securely attached to baseplate and cylinder, and has a removable polished pillar in front.

Crosshead Guide is tooled out perfectly true, and well fitted with oiling arrangements.

Base Plate and Bearings.—The cast-iron base plate has two adjustable bearings, fitted metal to metal, resting in bored-out carriages formed in base plate and secured by cast-iron caps, with brass and glass needle or drop-feed lubricators.

Flywheel is provided of a heavy pattern, accurately balanced, and turned on face for driving belt, or of the disc pattern with facing for direct coupling to dynamo as required.

Oiling Arrangements.—A special oil reservoir is provided at the top of engine standard with connecting pipes to all working parts.

The crank chamber is formed to collect the oil after use, which may be drawn off by means of a small cock provided at the bottom of the engine bedplate. Suitable splash plates are provided.

Extended Bedplate.—This can be made of suitable design to suit any make of dynamo.

THE 'ROBEY' SINGLE-CYLINDER OPEN-TYPE HIGH SPEED VERTICAL ENGINE.

FIG. 18.

FIG. 19.

WEIGHTS AND MEASUREMENTS OF ROBEY HIGH-SPEED VERTICAL ENGINES (OPEN TYPE).

Cylinder.		Belt Fly-Wheel.		Space Required.		Approximate Weights.				Approximate Cubic Measurement, packed, both.
						With Pickering Governor.		With Shaft Governor.		
Diam.	Stroke.	Diam.	Width.	Length over Crank.	Width over Bed.	Unpacked.	Packed.	Unpacked.	Packed.	
Ins.	Ins.	Ft. Ins.	Ins.	Ft. Ins.	Ft. Ins.	Cwts.	Cwts.	Cwts.	Cwts.	Ft.
3½	3	1 9	3	2 0	1 4½	5	7	6	8	11
4½	4	3 0	3	2 5	1 8	6	8	6½	8½	12
5½	5	3 0	4½	2 9	1 10½	7	9	8½	11	20
6½	6	3 6	5	3 2	2 0	12½	15	12½	15½	40
7½	7	4 0	6	3 7	2 5½	16	20	18	22	46
8½	8	4 0	6	3 8	2 8½	20	24	22	28	52
9½	9	4 6	8	4 3	3 7½	35	40	38½	46½	62
10½	10	4 6	9	4 6	3 6	42	50	43½	52	86
11½	11	4 6	10	5 5	4 2½	60	70	62	72	126
12½	12	5 0	12	5 10	4 5	91	101	96	108	190
13½	13	6 0	12	6 6	4 8	122	140	130	150	250
14½	14	7 0	15	7 0	5 3¾	150	170	160	180	300

THE GALLOWAY DROP PISTON VALVE ENGINE.

(*Patent.*)

The special features of this engine (*Galloways Ltd.*) lie in the steam valves, fig. **20,** which are of the drop piston type working in a slotted seat, which in turn is carried by a short vertical cylinder standing in the inlet valve of the steam chest This short cylinder is surrounded on all sides by steam at one temperature and pressure, and remains truly circular under all conditions however much the steam temperature and pressure may vary. This preserves the circularity of the valve liner and therefore its steam-tightness.

The piston valve itself is of the double-beat type fitted with 'Duplex' spring rings. The duplex rings are so arranged that they cannot fall out of the body of the valve in case of breakage.

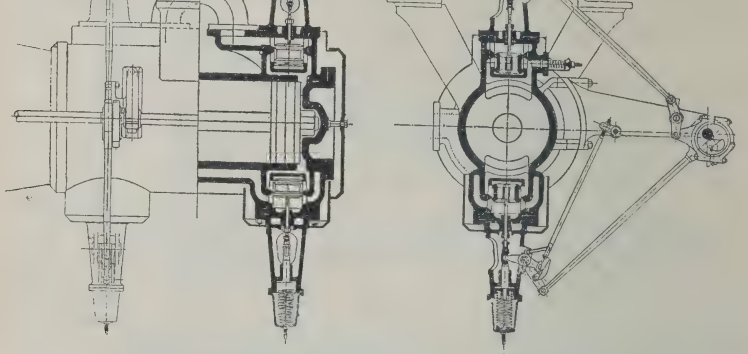


FIG. 20.—Sections through Cylinder, showing Drop Piston Valves.

There being two rings, one inside the other, bedded against each other over their whole circumference. the ring cannot be collapsed by the application of sudden pressure, as when the ring passes over a steam port. On test, the steam leakage in a 14-inch steam valve of this type proved to be a fraction of 1 per cent.

The Galloway Reversing Rolling Mill Engine.

In connection with the reversing rolling mill engines Messrs. Galloways fit a patent single lever expansion control gear. By the employment of one handling lever, both the throttle valve and the cut-off are suitably varied in a predetermined manner. This is so arranged that during the mean rolling period the engine works with variable cut-off and full steam pressure in the cylinders, thus securing maximum economy. When starting up, the gear partially closes the throttle and gives late cut-off so as to product an even steady turning at a slow speed. When the handling lever is brought to mid gear at the completion of a pass, the control valves are forcibly closed so that no steam leakage can take place. By means of this valve gear the steam consumption is claimed to have been reduced to one-half, or less, of that normally required, whilst the output of the rolling mill is increased, as the single lever expansion control gear gives very rapid action on the part of the engine.

Forced Lubrication.

Forced lubrication as a continuous or arterial system of conveying oil under pressure to all the working parts of an ordinary reciprocating engine was patented in 1890 (No. 7397) and introduced by Messrs. Belliss & Morcom Ltd. of Birmingham in their self-lubricating engine (*vide* sectional illustration, fig. 21, which shows their original design of compound engine 'C' type). The constant supply of oil under pressure prevents the working surfaces coming together, the film of oil taking an appreciable time to squeeze out, which is sufficient in a double-acting engine running at a high speed to last until the stress is reversed and the film on the other brass comes into operation; the bearing surfaces are thus prevented from coming into actual contact. In this system the engine is enclosed, and oil drawn from a reservoir in the bedplate by a pump is fed to circumferential grooves in each of the main bearings. The crankshaft is drilled, so that oil drawn from the circumferential grooves in the main bearings is conducted to similar circumferential grooves in the crankpins and thence to the crosshead pins and guides. The valve motion is lubricated in the same way, also the working joints in the governor. The oil escaping from the journals returns to the oil-well in the bed for re-use. The pump, which is of the oscillating type without loose valves or packings, is operated from the eccentric and draws its supply through a gauze strainer. The pressure of oil generally used may be anything between 10 and 25 lbs. per sq. inch.

The Belliss and Morcom 'C' Type High-Speed Engine.

The special feature of this engine, fig. 21, is the arrangement of the slide valves, requiring one eccentric and rod only. The low-pressure valve is superimposed above the high-pressure, working in the same valve chamber, the high-pressure valve taking steam on the inside and exhausting direct to the low-pressure, which cuts off on the outside edge. The cranks are necessarily set opposite to each other, the steam being admitted simultaneously to the top of one cylinder and the bottom of the other, by which arrangement the reciprocating parts are to a great extent balanced, the stress on the bearings much reduced, and a high speed of revolution made possible. A variation of speed not exceeding 3 per cent. between full and no load can be guaranteed. The governor gear is arranged so that the speed of the engine is capable of being adjusted whilst running.

A modification of this design of engine, more recently introduced, is the 'V' type (see fig. 22, p. 1732).

BACK PRESSURE AND EXTRACTION ENGINES.

Increasing attention is now being paid to the saving of fuel, which is rendered possible by the combination of power supply and heating services, the heating being effected by using exhaust steam or steam taken from the engine at an intermediate pressure. Factories using steam for heating in their own industrial processes offer special opportunities in this direction, though schemes have also been successfully carried out in which the heating of adjoining offices and houses has been effected in addition to the heating required in the factory processes.

For example, in a factory requiring 200 i.h.p. for driving machinery, and 3,200 lbs. of steam per hour at about 5 lbs. per sq. in. gauge pressure for heating purposes, the methods of independent and combined supply may be compared as follows. In the independent method an economical condensing engine would be used, taking, say, 11 lbs. steam per i.h.p. hour, or $11 \times 200 = 2{,}200$ lbs. steam per hour. The total steam is thus $2{,}200 + 3{,}400 = 5{,}600$ lbs. per hour.

In the combined system, a back pressure engine would be used, exhausting at the pressure required for the heating steam. It would require, say, 18 lbs. steam per i.h.p. hour, or 3,600 lbs. per hour. Allowing 10 per cent. for loss by condensation, 3,240 lbs. exhaust steam per hour are available for heating. Thus the steam to be generated for all purposes is 3,600 lbs., as

[*Continued on page* 1734

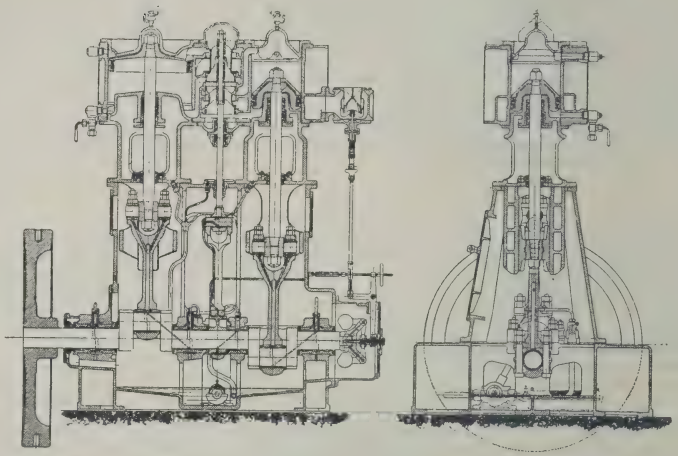


FIG. 21.—Belliss Patent Self-lubricating 'C' Type Compound Engine.

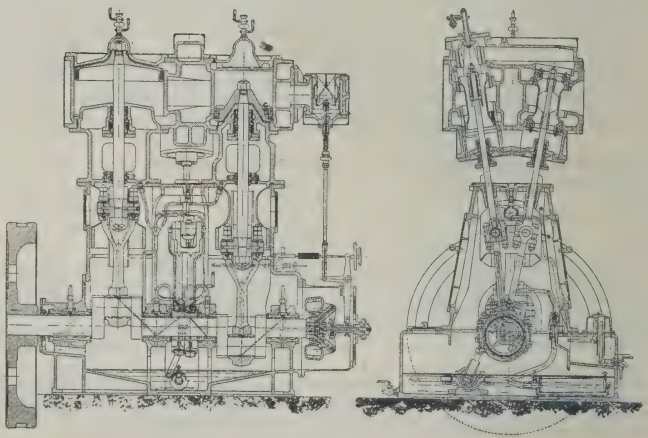


FIG. 22.—Belliss Patent Self-lubricating 'V' Type Compound Engine.

(Messrs. Davey, Paxman & Co. Ltd., Colchester.)

COMPOUND PORTABLE STEAM ENGINE ON ROLLED GIRDER FRAME (Working Pressure, 140 lbs.).

Nominal Horse-Power.	Brake Horse-Power.		Diameter of Cylinder.		Stroke. Both Cylinders.	Fly-Wheel.			Total Heating Surface.	Approximate Space required.			Approximate Weight.		Approximate Cubic Measurement Packed.
	Economical Load.	Maximum Load.	High Pressure.	Low Pressure.		Diameter.	Face.	Revolutions per Minute.		Length Overall.	Width over Wheel Caps.	Height to top of Fly-Wheel.	Un packed.	Packed.	
	H.P.	H.P.	Ins.	Ins.	Ins.	Ft. Ins.	Ins.		Sq. Ft.	Ft. Ins.	Ft. Ins.	Ft. Ins.	Cwt.	Cwt.	Cub. Ft.
8	19	23	5½	9	14	5 0	7	155	145·4	11 6	5 7	9 1	108	121	430
10	26	30	6½	10½	14	5 0	8	155	172·8	12 0	5 10	9 6	130	144	500
12	30	35	7	11¼	14	5 0	9	155	219·7	12 6	6 7	9 11	153	168	580
16	41	48	8	13	14	5 6	10	155	274·8	13 1	7 0	10 8	206	222	750
20	50	60	9	15	16	6 0	11	135	350·7	14 0	7 7	11 6	260	278	930
25	68	78	10	16	18	7 0	12	120	410·8	16 0	7 10	11 11	310	330	1,100
30	77	90	11	18	18	7 0	14	120	486·5	16 6	8 2	12 2	350	370	1,200

PORTABLE ENGINES. SINGLE CYLINDER. (Working Pressure, 140 lbs.)

Type.	Code Word.		Brake Horse-Power.		Cylinder.		Fly-Wheel.			Approximate Nett Weight in cwts.	
	Portable.	Semi-Portable.	Full Working Load.	Temporary Overload.	Diam. Ins.	Stroke Ins.	Face Ins.	Diam. Ft. Ins	Speed R.P.M.	Portable.	Semi-Portable.
L5	SPERNES	SPERRGLAS	13	16	5½	10	7	3 7	210	75	67
L6	SPERNETIS	SPERRHORN	17	20	5¾	10	7	3 7	210	94	82
L8	SPERNIMUS	SPERRSITZ	22	26	6½	12	8	3 9	200	105	93
L10	SPERPERO	SPERRUNG	27	32	7	12	8	3 9	200	112	100
L12	SPERRFORT	SPERVER	34	40	7¾	12	10	4 2	180	135	120
L14	SPERRGELD	SPESACCIA	40	45	8½	13	12	4 2	180	150	135

PAXMAN PORTABLE ENGINES. DOUBLE CYLINDER. (Working Pressure, 120 lbs.)

Nominal Horse-Power.	Brake Horse-Power.		Cylinder.		Flywheel.			Approximate Space Required.						Approximate.		
	Economical Load.	Maximum Continuous Load.	Diam.	Stroke.	Diam.	Face.	Speed.	Length.		Width.		Height.		Measurement Packed.	Weight Unpacked.	Weight Packed.
			Ins.	Ins.	Ft. Ins.	Ins.	R.P.M.	Ft. ins.		Ft. Ins.		Ft. Ins.		Cub. Ft.	Cwt.	Cwt.
8	20	27	7	12	5 0	7½	140	11	11	6	3	9	0	510	110	125
10	26	34	7¾	12	5 0	8	140	12	8	6	3	9	6	585	128	143
12	32	40	8¼	12	5 0	9	140	13	3	6	6	10	0	645	150	165
14	39	48	9½	14	5 6	9	125	13	11	6	9	10	0	740	175	190
16	46	56	10	14	5 6	10	125	14	3	7	1	10	1	875	195	215
20	57	71	11	16	6 0	11	115	15	1	8	1	10	9	940	225	245
25	72	90	12	18	6 6	12	110	16	5	8	6	11	0	1,090	255	275
30	84	110	13	18	6 6	14	110	17	2	8	9	12	0	1,310	290	315

NOTE.—The Power given in Columns 2 and 3 is the Brake Horse-Power; the Indicated Horse-Power will be correspondingly higher.

against 5,600 lbs. in the former case, and even though the whole of this steam has to be generated at high pressure and (preferably) superheated, the fuel consumption at the boiler is reduced in nearly the same proportion. The saving in fuel is thus $\frac{5,600 - 3,600}{5,600} \times 100 = 35{\cdot}6$ per cent approximately.

By means of such systems the large proportion of heat carried away and lost in the condensing water of even highly economical power plants is greatly reduced; cases are on record in which the heat utilised as work in the engine, together with heat made available for heating purposes, amounts to 77 per cent. of the heat in the steam delivered by the boiler.

In the 'intermediate steam extraction' system, which is adapted for fluctuating power and heating requirements, a compound engine is used, the heating steam is taken from the receiver between the cylinders, and only a part of the steam used in the h.p. cylinder passes to the l.p. cylinder, and thence to the condenser. By means of a pressure regulator, the intermediate pressure is allowed to vary only between fixed limits. With an increased demand for heating

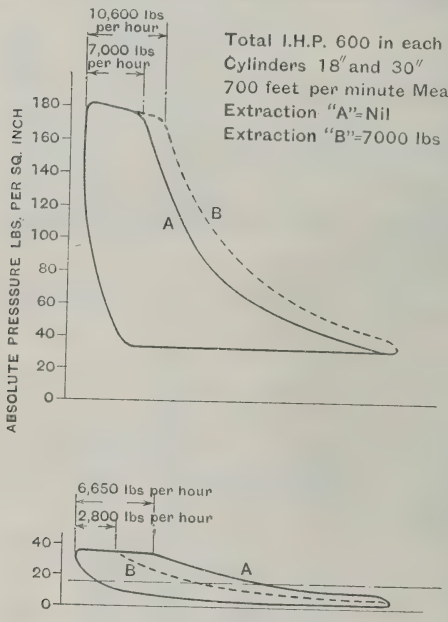


FIG. 23.

steam, the pressure falls slightly, the pressure regulator makes the l.p. cut-off take place earlier, and, if the load on the engine remains unchanged, the speed governor makes the h.p. cut-off correspondingly later; thus more steam passes through the h.p. cylinder and less through the l.p. Provision is usually made for supplementing the 'extraction' steam, when necessary, by steam taken directly through a reducing valve from the live steam main; this additional supply is also controlled automatically. The l.p. cylinder may with advantage be of the 'uniflow' type (page 1722).

On account of the variations in the distribution of the power between the two cylinders with varying demands for extracting steam, the extraction engine usually has its cylinders arranged in tandem.

This is not, however, absolutely essential, and, particularly if the extraction steam is comparatively small in quantity, the engine may be of the cross compound arrangement.

Typical indicator diagrams for an extraction engine are shown in fig. 23.

The steam quantities in each cylinder with and without extraction are marked on the diagrams, the total load being the same in both cases.

Oil in Exhaust Steam.

The objection is often raised that steam exhausted from, or extracted from, a reciprocating engine is unsuitable for use in contact with textile fabrics on account of its contamination with lubricating oil.

Vertical engines may be operated with bronze piston rings and no cylinder oil in order to avoid this objection.

The risk of trouble due to oil is, however, much exaggerated. The experience obtained in using exhaust steam from oil-lubricated cylinders in bleaching and dyeing works where it comes into contact with all sorts of fabrics and all sorts of dyes justifies the conclusion that no trouble due to oil need be feared. Not the slightest effect of oil on the fabrics has been detected in many years of working under such conditions.

An oil separator should, of course, be provided, although it will not remove absolutely the whole of the oil.

The Galloway Extraction Engine.

Fig. 24 shows a typical extraction engine as made by Messrs. Galloways Ltd., Manchester. It is of the tandem type, with a uniflow low pressure cylinder.

The inlet valves of both cylinders are of the double beat drop type and are operated by the firm's liquid valve gear (see p. 1723). The exhaust valves of the high pressure cylinder are piston valves, driven by eccentrics on the lay shaft through intermediate rocking levers.

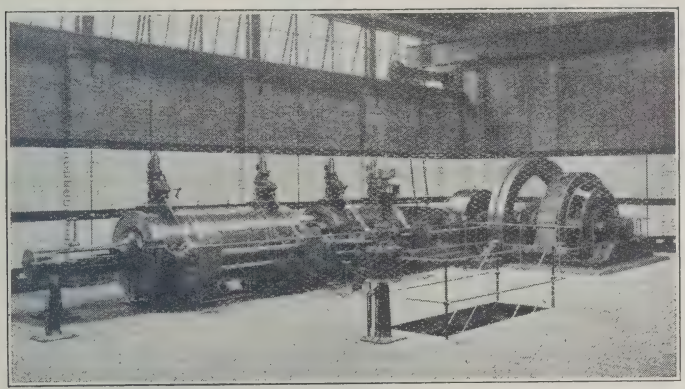


Fig. 24.—The Galloway Extraction Engine.

The engine is totally enclosed, with forced lubrication to the cylinders and principal parts and is in accordance with the best modern high-class practice.

The speed is controlled by a vertical spindle centrifugal governor, and the pressure of the extraction steam is adjusted and automatically maintained constant within fine limits by means of the pressure regulator which is seen in the foreground. The servo-motor which actuates the valve gear is in this instance below the floor; in other respects the pressure regulator is as described below.

The Galloway Steam Pressure Regulator.

(*Patent.*)

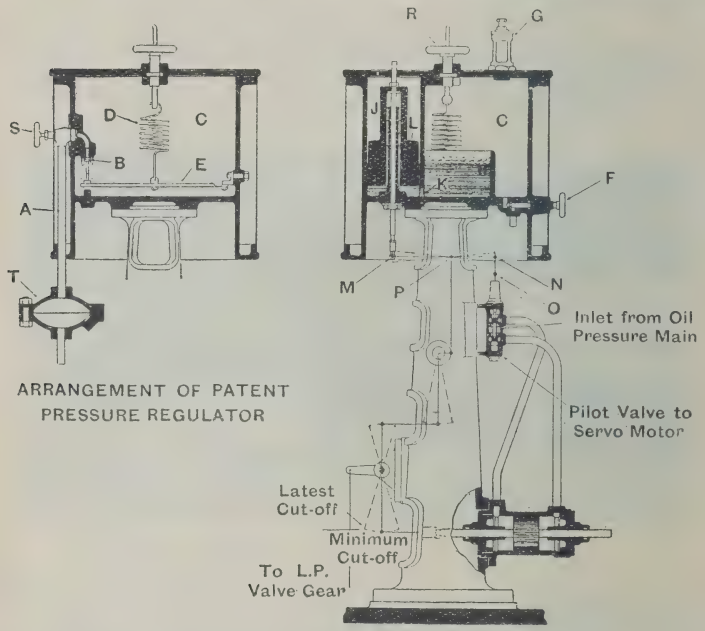
This Pressure Regulator, in the form suitable for use with an extraction engine, is shown in fig. 25.

A pipe A conducts steam from the process-steam main through a strainer T and valve S to the interior of a closed cast-iron chamber C. The entrance of the steam to the chamber is controlled by a valve B, held closed by an adjustable spring D. The tension of the spring is so regulated that at the pressure desired in the main there is a slight escape of steam past the valve B, this steam emerging from the chamber C through an adjustable leak F to atmosphere.

The rod E controlling the valve B is of stainless steel, and, to avoid friction in operation, the right-hand end is finished to a thin strip, which is secured to a lug, and bends slightly as the rod moves. The end supporting the valve has a cup engaging the rounded end of the valve spindle. The whole mechanism is practically frictionless, and responds to the slightest variation in the pressure in the main.

The steam escaping from the valve B fills the chamber C at a pressure slightly above atmospheric owing to the throttling effect of the leak F. Any variation in pressure in the main will result in a corresponding change in the pressure inside the chamber. It is this latter variation which is made use of to operate the regulating mechanism.

In the chamber is a container H filled with mercury, which communicates through a hole K with a chamber J open to atmospheric pressure. In this chamber rides a cast-iron float L, frictionlessly guided. It will now be seen that a variation in the pressure in the chamber C will act on the surface of the mercury in container H, and cause the float L to rise or fall. This movement is communicated through a lever MN to a pilot-valve of normal type, operating an oil or



ARRANGEMENT OF PATENT PRESSURE REGULATOR

FIG. 25.—The Galloway Steam Pressure Regulator.

water-servo-motor, shown towards the bottom of the section. A rise of the float L will push the pilot-valve down, causing the servo-motor to be moved to the left; this movement will, however, raise the lever MN by the pin P, and restore the pilot-valve to its normal position. There is thus a definite position of the servo-motor for any given pressure in the chamber C. The servo-motor controls the cut-off in the low-pressure cylinder.

In some combined power and heating schemes conditions arise where the demand for heating steam exceeds that available from the engine at the power at which it is called upon to work at the moment, and it becomes necessary to supplement the supply from the high-pressure main through a reducing valve. The regulator described is able to perform the extra duty of controlling this additional supply. The apparatus is also quite suitable for use independently of a heat-extraction engine for maintaining a constant pressure in a main, or for controlling a reducing valve.

ENGINE DESIGN.

The type of engine is first settled from a consideration of duty, steam consumption, price of fuel, space available, working conditions, water supply, first cost, etc.

Cylinder proportions and speed of revolution are then fixed. Typical British cylinder ratios and mean effective pressures are given in the tables below. On the European Continent the cylinder ratios adopted are usually considerably below the tabular values. That is, a larger high-pressure cylinder is adopted. This makes for economy with superheated steam, but for a two-crank engine tends towards unequal distribution of load on the two cranks which is not altogether desirable. This objection does not apply in the case of tandem engines. These remarks are illustrated by the test results tabulated on page 1713.

TYPICAL MEAN EFFECTIVE PRESSURES AT NORMAL FULL LOAD IN LBS. PER SQ. IN.

Boiler Pressure—Gauge	80	100	120	140	160	180	200
Simple Condensing, about	30	34	37	—	—	—	—
„ Non-Cond. „	35	40	43	—	—	—	—
Compound Condensing	25	27	28	29	30	31	32
„ Non-Condensing	26	30	34	37	40	—	—
Triple Condensing	—	—	—	27	29	30	30·5
Uniflow Condensing	30	33	35	37	38	39	40
„ Non-Condensing, about	26	30	34	37	40	—	—

Note.—The above are often exceeded 5 or 10 per cent.

USUAL BRITISH CYLINDER RATIOS. HORIZONTAL. CUT-OFF GOVERNING.

Boiler Pressure, lbs. Gauge	100	120	140	160	180	200
Initial Piston Press., absolute	110	130	150	170	190	210
Compound Non-Condensing	2·3–2·5	2·5–2·7	2·8–3·0	3·0–3·2	—	—
Compound Condensing	3·0–3·3	3·2–3·5	3·7–3·8	4·0	4·3–4·6	4·5–4·7
Triple Condencing :						
L.P. to H.P.	—	—	5·2–5·6	5·6 –6·0	5·9–6·3	6·0–6·6
I.P. to H.P.	—	—	2·3–2·36	2·36–2·45	2·43–2·5	2·45–2·67

Having fixed the cylinder sizes the probable steam pressures on the pistons are put down, and from these the total piston loads are obtained. From these, by allowing suitable stresses and bearing pressures the principal sizes are fixed.

Usual stresses in high-class engines are as follows:—

USUAL ENGINE STRESSES.

Crank and crosshead pins, up to.	9,000 lbs. sq. in.	Connecting rod bolts	4,000–6,000 lbs. sq. in.
Crankshaft journals	7,000–9,000 „	Main studs and bolts	5,000 „
Piston rod, net tension at cotter hole	5,000–6,000 „	Lesser „ „	4,000–6,000 „
Piston rod, gross tension (direct plus bending in body)	9,000 „	Connecting rod body, small end	3,000–6,000 „
		Shear on cotters	3,000–5,000 „
		Bending on cotters	up to 7,000 „
		General forgings	4,000–8,000 „

Cast Iron in Tension.

Simple frames, direct pull	700–900 lbs. sq. in.	Flywheel arms (C.F.)	1,500–2,500 lbs. sq. in.
Frames, with some bending	500–800 „	Cylinder barrels, new	1,200–2,000 „
Complex frames, with bending	500–600 „	„ „ usual	1,500 „

It should be noted that in calculating the above frame tensions only the direct pull is included. If bending is calculated separately and added on, very high figures are sometimes reached.

As an example, take the case of a cotton mill engine of 1,500 i.h.p. normal load, the engine to be of the cross-compound condensing type; boiler pressure 160 lbs.

The cylinder ratio we take to be 4 nominal, piston rods being ignored. We aim at equal loading on the two cranks so that of the total drop in pressure from stop valve to condenser one-fifth will take place in the low-pressure (l.p.) cylinder. Allowing 12 lbs. vacuum and losses of 5 lbs. and 8 lbs. at the steam pipe and intermediate receiver respectively, the total effective pressure drop is 159 lbs., of which 32 takes place in the l.p. cyl. and 127 in the h.p. If the engine has a stroke of 5 ft. and runs at 66 r.p.m. (higher speeds are coming into favour), the piston speed is 660 ft. per min., and the area of the l.p. cylinder for a m.e.p. of 30 lbs.

$$= \frac{1{,}500 \times 33{,}000}{660 \times 30} = 2{,}500 \text{ sq. ins.}$$

This is taken as the gross piston area, giving a diameter of 56·5 inches. The actual m.e.p. would therefore be about 31 lbs. The h.p. cylinder should then be 28·25 ins. diam., but odd dimensions are unsatisfactory, and the cylinders will be fixed at 28 and 56 inches, giving a nominal m.e.p. of 30·5 lbs., which is rather below modern practice, which tends rather to higher m.e.p. than given in the table above.

Ignoring the piston rods, the gross load on the l.p. piston is $32 \times 2{,}500 = 80{,}000$ lbs. The load on the h.p. piston is the same. Owing to the effect of the rods this value is rather high, but the difference may be ignored, and 5 per cent. added to cover variations in the load distribution, making the total piston load 84,000 lbs. Taking bearing pressures of 1,250 and 850 lbs. per sq. in. on the crosshead pin and crank pin, the projected areas of these become 67·2 and 98·8 sq. ins. respectively. We may thus fix the pin bearings (diam. × length) as 6·75 × 10 and 10 × 10. ins.

HORIZONTAL ENGINE BEARING PRESSURES.

Bearing.	Lbs. per Sq. In.	Bearing.	Lbs. per Sq. In.
Crosshead pins	1,200–1,300	Second motion bearings	100–200
Crank pins	800– 950	Crosshead slides	40– 50
Crankshaft journals (steam load)	100– 150	Intermediate piston rod slides	20– 40
Crankshaft journals (combined loads)	150– 200		

USUAL BEARING PRESSURES FOR VERTICAL HIGH-SPEED ENGINES.

Bearing.	Lbs. per Sq. In.	Bearing.	Lbs. per Sq. In.
Main bearings	200– 390	Crosshead slippers	40–50
Crank pins	400– 550	Eccentric straps	60–70
Crosshead pins	1,000–1,200		

Inertia should be allowed for in special cases.

Journal speeds are rarely taken into account, but there is at present a tendency to do so, especially in Germany. The influence of speed is complex. In many cases it improves the load which can be carried, but it also increases the heat developed per second, and where the pressures are not such as to endanger the oil film, may limit the safe load on the bearing by the tendency towards overheating. For this a rule of the form—

Pressure per sq. in. × journal speed in ft. per sec. = constant,

is used, but the 'constant' varies so widely that its value in the present state of knowledgeis limited.

We now turn to the size of the crankshaft neck. The bending 'arm' from centre of crank pin to centre of main crankshaft neck, we estimate at about 26 ins., so that the bending moment is $M = 84{,}000 \times 26 = 2{,}180{,}000$ inch lbs. The twisting moment is

$$T = 84{,}000 \times 30 = 2{,}520{,}000 \text{ inch lbs.}$$

The equivalent bending moment is

$$M_e = \tfrac{1}{2}\left[M + \sqrt{M^2 + T^2}\right] = 2{,}755{,}000.$$

Adopting a stress of 8,000 lbs. per sq. in., the modulus of the neck section is

$$Z = \frac{2{,}755{,}000}{8{,}000} = 344.$$

For a solid neck of diam. d, $Z = 0{\cdot}0982d^3$. Hence $d = 15{\cdot}2$ ins., say 15·5 ins. The length of the neck must be determined to give a suitable bearing pressure from dead weight and steam load.

The piston rod diam. is next fixed, and partly from it and partly directly from the load the proportions of the connecting rod. The piston rod is proportioned so that the stress round the cotter hole at the crosshead is about 5,000 to 5,500 lbs., whilst the total direct tension plus tension due to bending (assuming a tail support to the rod) does not exceed about 9,000 lbs. The net section of the frame will lie between about 100 and 170 sq. ins. according to type, the highest stresses being allowed where there is least bending.

The size of the crosshead slipper is determined as follows:—If the connecting rod is 5 cranks long, centre to centre, divide the total piston load by 5. The result, 16,800 lbs. in this case, is approximately equal to the maximum load on the slide due to piston pressure. To this should be added the dead weight in the case of a horizontal or inclined engine, giving, say, about 18,000 lbs. in all. Taking a pressure of 45 lbs. per square inch (less if the pressure is steady and not highly variable), gives a slipper area of 400 sq. ins.

Testing Crankshaft Alignment.

Place the crankshaft, fig. 26, on its top centre and set either a micrometer gauge or a gauge made of round iron between the crank webs at the bottom. Turn the crankshaft

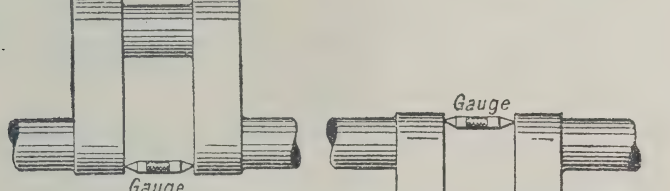


FIG. 26.

around till it is in its bottom position and gauge again. In many cases it will be found that the gauging in the two cases is different, indicating incorrect alignment. Usually, it is the bearing on the driving side that is low. (*A. Bennett, in 'Power.'*)

A simple form of crankshaft alignment indicator, of dial reading type, has been introduced by Messrs. Ruston & Hornsby, Ltd., Lincoln. This instrument is set between the crank webs, and the dial read after turning the shaft into various positions, want of alignment being indicated by differences in the reading.

(See also 'Alignment of Crankshafts and Engine Parts,' p. 1905.)

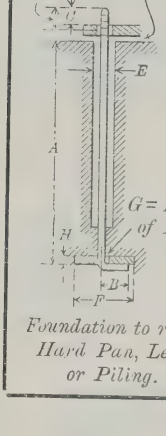
FRAMES.

Bedplates and other frames are almost invariably of strong cast iron. They should be so formed as to reduce bending to a minimum consistent with convenience for the engine attendant and reasonable cost. To secure this they should be as nearly as possible symmetrical on both sides, and also above and below the centre line. This does not mean strict geometrical symmetry, as this is very rarely attainable except as regards the two sides; but it implies an approximately equal distribution of stress. Circular bored and turned joints make for truth in alignment and often for cheapness in manufacture. Ample bearing surface on the foundations must also be allowed.

Foundation Bolt Dimensions.

For bolts 2 ins. or over in diameter, a nut is used under the washer or anchor plate, instead of the bolt being clinched, as indicated by dimension *B*.

STANDARD DIMENSIONS FOR SETTING FOUNDATION BOLTS.



Bolts. (Inches.)						Washers. (Inches.)		
Dia.	A	B	C	D	E	F	G	H
¼	—	—	—	—	—	—	—	—
⅜	—	—	—	—	—	—	—	—
½	12	2½	¾	1	2	5 by 5	¼	⅛
⅝	15	2½	1	1½	2	5 by 5	¼	⅛
¾	18	2½	1⅛	1½	3	5 by 5	¼	⅛
⅞	21	3	1⅜	1	3	6 by 6	1½	¾
1	24	3	1½	2	3	6 by 6	1½	¾
1⅛	27	4	1⅜	2½	4	8 by 8	1⅝	1
1¼	30	4	1½	2½	4	8 by 8	1⅝	1
1⅜	36	5	2½	3	4	10 by 10	2	1¼
1½	42	5	2½	3½	5	10 by 10	2	1¼
2	48	Nut	3	4	5	12 by 15	2½	1
2½	54	Nut	3⅜	4½	6	12 by 15	2½	1
2⅔	60	Nut	3¾	5	6	12 by 20	3	1½

(*'Machinery,' Mar.* 24, 1921.)

CYLINDERS.

Cylinders should be hard and strong. Apart from questions of strength, for information on which see the table of stresses on page 1737, the following important points should not be overlooked :—

(1) The clearance volume should be as small as reasonably possible.

(2) The clearance surface should be reduced to a minimum.

(3) The exhaust and inlet ports are better separated.

(4) The ports should be short, direct, and not too large. Large ports involve increased clearance surface.

(5) The covers which, however, are closely influenced by the piston details, should be free from recesses and curves.

(6) Where there are jackets the steam should be arranged to have a decided through flow, the more rapid the better.

(7) Proper relief valves and drainage should be provided.

(8) Metal bosses and projections connecting to the frames or atmosphere facilitate heat radiation, and should be kept small.

The clearance surface is specially important. Some particulars of usual practice are given in the table of tests on p. 1834, and in the following :—

TYPICAL CLEARANCE SURFACES.

Cylinder and Valves. (Piston Area × 2 = 100%.)	Surface, Per cent.	Authority.
H.P. slide valve	200	J. Stumpf
L.P. trick valve	180	,,
H.P. piston valve	300	,,
L.P. piston valve	220	,,
Double-beat drop valve	182	,,
Corliss valve	170	,,
Uniflow, double-beat valve	133	,,
H.P. Corliss valve	210–320	Foster
L.P. Corliss valve	180–300	,,
H.P. drop-piston valve	210–300	,,
L.P. drop-piston valve	200–300	,,

The port sizes are based on a steam velocity of 100 to 125 feet per second for inlet ports, and 90 to 120 feet per second for exhaust ports when using saturated steam. With superheated steam these speeds are increased to 130 or even 150 feet per second.

BEARINGS.

With very rare exceptions oil is dragged into a bearing by the relative movement of the two so-called rubbing surfaces by a kind of wiping or dragging action. To prevent loss of lubrication the oil must not be allowed to escape from the bearing except at the proper place. Ordinary crosshead pin bearings work mostly by suction. The slight pulsation of the pin in the bearing draws in oil on alternate strokes. On the return of the pressure the oil has not time to escape.

Oil grooves should be looked upon as local supply reservoirs.

Bearing clearances should not increase so rapidly as the diameter of the journal. They vary according to the class of work.

The use of white metal bearings (see page 2736) is a safety precaution in case of overheating. It also has constructive merits.

Usual bearing pressures are given on page 1738.

For the Michell bearing see page 2409.

Rubber *v.* Metal Bearings.

In many commercial uses metal is now being replaced by rubber.

On a harbour tug rubber bearings showed no wear after ten months. They stand the hard use better than bearings of metal, and speeds as high as 5,000 revolutions per minute have been reported with bearings of this type. One great advantage of rubber bearings is that they are not injured by sand or grit. Such particles imbed themselves in the rubber and work out into the grooves in the bearing, whence they are washed away by the water, the only lubricant necessary, without having scratched or cut any part of the machine. In addition, rubber bearings greatly reduce vibration and noise, and for that reason are being used more and more on yachts, house-boats, and other passenger craft.

The contrast in the wearing qualities of rubber and bronze is shown in the case of a mixing machine. Its arms of bronze lasted six weeks, and gave considerable trouble, but its arms when covered with ¼ in. of rubber were working satisfactorily after many months of use. Rubber linings have been known to increase the life of flues carrying gases containing abrasive dust, and also the service of sheet-metal chutes.

Alignment Pins.

'Alignment,' 'steady,' or 'fixing' pins are brought into service in the erection of machinery details for the purpose of ensuring that one particular piece shall be superimposed upon another piece in its precise and correct position, no matter how many times the respective pieces may be separated. The use of such pins is far preferable to fitting bolts to each and every hole; the holes for the bolts should be 'clearance' holes, *i.e.* holes through which the bolts can pass freely.

Fig. 27 shows a pin which has been used frequently in small engines, dynamo-electric machinery, and, notably, in the casings of large steam turbines, with unqualified success.

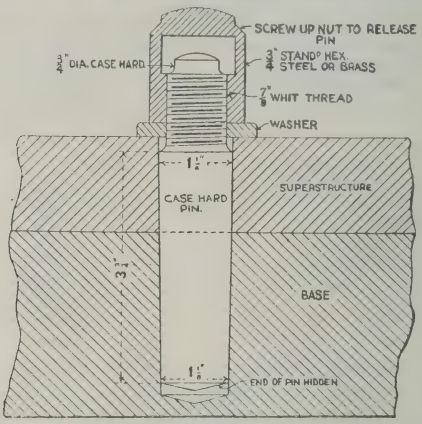


FIG. 27.

It will be observed by those familiar with the use and application of these alignment pins, that a pin such as is shown in the figure is all but imperatively necessary where it is required to fit it to a blind bottom, as represented above. ('*The Electrical Review*,' Sept. 17, 1920.)

FLYWHEELS.

A = Area of turning moment curve per revolution = Work done per revolution in ft. lbs.
a = Excess or deficiency of turning moment curve from resistance curve (usually taken as the mean steady load) = k A.
k = Coefficient of energy fluctuation.
I = Moment of inertia of flywheel in lbs. ft.^{2}
d = Diameter of gyration of wheel in feet.
D = „ rim face „ „
C = Number of complete turning-moment cycles per revolution.
G = Total weight of wheel in lbs.
W = Effective „ „ „ if placed at rim face, diameter D, velocity V.
V = Linear velocity of rim face, ft. per sec.
N = R.P.M.
X = Cyclic irregularity of turning.
Y = Angular deviation from true uniform turning.
F = Frequency of (electric) alternations per second.
p = Number of poles in alternator field magnets.
K = Flywheel coefficient = $\frac{\text{Stored energy in flywheel}}{\text{Work done per revolution}} = \frac{WV^2}{2gA}$.

Then (the suffixes $_1$, $_o$, and $_2$, referring to maximum, mean, and minimum),

$$X = \frac{N_1 - N_2}{N_o} = \frac{V_1 - V_2}{V_o} = \frac{k}{2K} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$K = \frac{k}{2X} = \frac{a}{2AX} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$Y_1 - Y_o = 28{\cdot}65\,\frac{X}{C} \quad . \quad . \quad . \quad . \quad \text{mechanical degrees} \quad . \quad . \quad (3)$$

$$= 14{\cdot}32\,\frac{pX}{C} \quad . \quad . \quad . \quad \text{electrical degrees} \quad . \quad (4)$$

$$= \frac{1719\,FX}{NC} \quad . \quad . \quad . \quad \text{electrical degrees} \quad . \quad (5)$$

The formulæ (3), (4), and (5) are based on sine curves for the turning moment, and are rarely strictly correct as regards the angular deviation $Y_1 - Y_0$, but formulæ (1) and (2) are always correct if the value of a is properly determined.

Considerable uncertainty arises as to the value of a in the case of multi-crank engines.

It is usual to specify that for alternators the angular displacement must not exceed $2\frac{1}{2}$ electrical degrees. In that case, we obtain from equation (4) the relationship

$$\text{coefficient of irregularity } X = \frac{1}{5 \cdot 73} \frac{C}{p} \quad . \quad . \quad . \quad . \quad (6)$$

In the B.E.A.M.A. standardisation rules for electrical machinery, the rule is given

$$\text{cyclic speed irregularity} = \frac{k}{6p} \quad . \quad . \quad . \quad (7)$$

where

k = engine impulses per revolution; p = number of poles.

This formula differs from (6) only in replacing 5·73 by 6, but, as already pointed out, these formulæ depend upon the form assumed for the turning moment curve, and variations are therefore not unlikely and do not indicate any error in principle.

Certain flywheel data are given in the three following tables. It will be understood that these are typical values, and that considerable variations occur in practice.

TYPICAL VALUES OF ENERGY COEFFICIENTS, k.

Engine.	Cranks.	Double or Single Acting.	Cycles per rev. C.	k	Crank Angles.
Steam engines:					
Simple	1	Double	2	0·12 –0·16	
Tandem	1	,,	2	0·11 –0·14	
Cross-compound	2	,,	4	0·06 –0·09	90
Triple	3	,,	6	0·025–0·03	120
Uniflow	1	,,	2	0·11 –0·13	
Gas engines, 4-cycle:					
Tandem, two cylinders	1	Double	2	0·13 –(0·2)	A
,, four ,,	2	,,	4	0·06 –(0·1)	90 A
One cylinder	1	,,	0·5		
One cylinder	1	Single	0·5	0·2 –2·4	
Two cylinders, tandem	1	,,	1	0·9 –1·1	360
Two cylinders, twin	2	,,	1	0·9 –1·0	360
Two cylinders, opposed	1	,,	0·5	1·3 –1·5	180
Three cylinders	3	,,	1·5	0·5 –0·6	120
Four cylinders	4	,,	2	0·1 –0·13	180
Six cylinders (tandem)	3	,,	3	0·06 –0·08	120

$k = \dfrac{\text{maximum excess energy}}{\text{work done per revolution}}$. A indicates figures in brackets, adopted to cover misfiring.

TYPICAL CYCLIC IRREGULARITIES X AND FLYWHEEL COEFFICIENTS K.

Driving.	Connection.	Engine.	No. of Cranks.	X	K
Alternators in parallel	Direct	Large	—	$\frac{1}{250}$	—
,, ,,	,,	High-speed	—	$\frac{1}{250} - \frac{1}{300}$	—
D.C. dynamos	,,	Steam	—	$\frac{1}{100} - \frac{1}{125}$	—
,,	,,	Large gas	—	$\frac{1}{150}$	—
,,	,,	Small gas	—	$\frac{1}{80} - \frac{1}{125}$	—
,,	,,	Diesel	—	$\frac{1}{100} - \frac{1}{125}$	—
,,	Ropes	Various	—	$\frac{1}{80} - \frac{1}{125}$	—
Best factory	,,	Steam	1	$\frac{1}{185}$	12
,, ,,	,,	,,	2	$\frac{1}{200} - \frac{1}{340}$	7–12
General factory	,,	,,	1	$\frac{1}{125} - \frac{1}{155}$	8–10
,, ,,	,,	,,	2	$\frac{1}{170} - \frac{1}{260}$	6– 9
Pumping, reciprocating	Direct	,,	3	—	·15–·5
Pumping, usual	,,	Triple vert.	3	—	·3

VALUES OF FLYWHEEL COEFFICENTS K FOR VARIOUS VALUES OF X AND k.

Engine.	Cranks	k	X					
			$\frac{1}{75}$	$\frac{1}{100}$	$\frac{1}{125}$	$\frac{1}{150}$	$\frac{1}{200}$	$\frac{1}{250}$
Steam :								
Double acting	1	0·13	4·9	6·5	8·1	9·8	13·0	16·3
" "	2	0·07	2·7	3·5	4·4	5·3	7·0	8·8
" "	3	0·03	1·2	1·5	1·9	2·3	3·0	3·8
Gas : double acting tandem	1	(0·2)	7·5	10·0	12·5	15·0	20·0	25·0
" " " "	2	(0·1)	3·8	5·0	6·3	7·5	10·0	12·5
Gas : 4-cycle single-acting :								
One cylinder	1	2·3	86	115	144	173	230	288
Two cylinders tandem	1	1·0	37·5	50·0	62·5	75·0	100·0	125·0
Two cylinders opposed	1	1·4	52·5	70·0	87·5	105·0	140·0	175·0
Three cylinders	3	0·55	20·6	27·5	34·4	41·2	55·0	68·8
Four cylinders	4	0·13	4·9	6·5	8·1	9·8	13·0	16·3
Six cylinders	3	0·07	2·7	3·5	4·4	5·3	7·0	8·8

NOTE.—The values of k are based on the assumption of a uniform engine load, and do not apply accurately to engines driving pumps, blowers, or even electric alternators. (*Foster.*)

Example.—Tandem compound engine of 1,200 i.h.p. normal load at 94 r.p.m., driving an alternator of 50 cycles. The deviation must not exceed two electrical degrees on either side of the mean position.

From the tables we see that k is about 0·13, and C=2.
In formula (5) we have

$$C = 2, \quad Y_1 - Y_0 = 2, \quad F = 50, \quad N = 94$$

Hence $$X = 0{\cdot}00437 = \frac{1}{229}.$$

Inserting the value of X in formula (2) we obtain K = 14·9.
The work done per revolution is

$$A = \frac{1{,}200 \times 33{,}000}{94} = 421{,}000 \text{ ft.-lbs.}$$

The stored energy in the flywheel is KA or 6,280,000 ft.-lbs.
If the flywheel is 18 ft. diam., the rim velocity is 88·6 ft. per sec., and the equivalent weight at this diameter is, therefore, 51,500 lbs. The actual flywheel weight should be about 70 per cent. in excess of this, say 87,500 lbs. This makes no allowance for the flywheel effect of the alternator rotor, which is usually considerable and need not be ignored.

For engines driving alternators, the flywheel effect should be such that the displacement $Y_1 - Y_0$ does not exceed 2·5 electrical degrees, nor should X as a rule exceed $\frac{1}{229}$. Where several units have to work in parallel the flywheel effect must also be such as to avoid resonance, i.e. the natural period of oscillation of the rotor about the position of uniform turning must not coincide with the period of any periodic impulses due to the engine. The determination of the oscillation period requires knowledge of the electrical characteristics of the alternators.

Alternator builders commonly state the flywheel quantity of their alternators in terms of GD^2, which is short for weight multiplied by the square of the diameter of gyration. If the units employed are kilogrammes and metres, the corresponding GD^2 in lbs. and feet can be obtained by multiplying by 23·7.

$$\text{Stored energy} = GD^2 \times \left(\frac{\pi N}{60}\right)^2 \times \frac{1}{2g} = \frac{GD^2 \times N^2}{23{,}500}.$$

If G, D are in lbs. and feet, the stored energy is in ft.-lbs. The stored energy at the revolutions of the machine is readily calculated by assuming any convenient diameter of gyration, calculating the corresponding weight and velocity and thence the kinetic energy. The flywheel effect is sometimes stated as T tons (or lbs.) at 1 ft. radius. The GD^2 is then T × 4 in tons (or lbs.) × feet^2. The actual diameter need not be known, which is a convenience of this method of expression. Occasionally the moment of inertia is given, which is equal to GD^2 divided by $4g$, which is 128·8 in ft.-lb. units and 39·24 in kg. metre units. (This gives the moment of inertia in 'engineers' units,' i.e. mass × length^2 ÷ g.)

SPEED OF FLYWHEELS.

The rim of a flywheel is subjected to stress on account of centrifugal force, and this stress is independent of the cross-section. For a particular material it depends only on the linear speed of the rim.

For cast-iron wheels the safe rim speed is customarily taken as 6,000 ft. per minute. For a given r.p.m. this fixes the wheel diameter.

Where higher speeds are desired with consequently lighter wheels for the same amount of stored energy, cast steel may be used, or the wheel may be made with discs of steel plate.

A steel plate wheel with wire-wound rim has been run at 15,000 ft. per minute.

Crank Connecting-Rod Mechanism

If r = crank radius; l = length of connecting rod; n = R.P.M.; θ = crank angle (measured from inner dead centre); x = per cent stroke of piston (measured from end of stroke remote from crank),

then the following relationships are nearly exact:

$$x = \frac{1 - \cos\theta}{2} + \frac{r}{8l}\left(1 - \cos 2\theta\right);$$

$$\text{piston velocity} = 2\pi nr\left(\sin\theta + \frac{r}{2l}\sin 2\theta\right) = 2\pi n r \times k_1;$$

$$\text{piston acceleration} = 4\pi^2 n^2 r\left(\cos\theta + \frac{r}{l}\cos 2\theta\right) = 4\pi^2 n^2 r \times k_2.$$

Table of Crank Angles and Factors for Piston Velocity and Acceleration.

θ = angle of crank for out-stroke.
Angle for in-stroke = 180 − θ, measured from outer dead centre.

Stroke per cent.		$\frac{l}{r}$ = infinity.			$\frac{l}{r} = 5$			$\frac{l}{r} = 4$		
Out.	In.	θ	k_1	k_2	θ	k_1	k_2	θ	k_1	k_2
0	100	0	0	1·00	0	0	1·20	0	0	1·25
1	99	11·5	0·20	0·98	10·5	0·22	1·17	10·3	0·22	1·22
2	98	16·3	0·28	0·96	14·8	0·305	1·14	14·5	0·31	1·19
4	96	23·1	0·39	0·92	21·1	0·43	1·08	20·7	0·435	1·12
6	94	28·3	0·475	0·88	26·0	0·515	1·02	25·4	0·525	1·06
8	92	32·9	0·54	0·84	30·1	0·585	0·965	29·5	0·60	1·00
10	90	36·9	0·60	0·80	33·8	0·65	0·905	33·1	0·66	0·94
15	85	45·6	0·715	0·70	41·9	0·765	0·765	41·1	0·78	0·785
20	80	53·1	0·80	0·60	48·9	0·855	0·63	48·0	0·87	0·64
25	75	60·0	0·865	0·50	55·4	0·915	0·495	54·3	0·93	0·50
30	70	66·4	0·915	0·40	61·5	0·96	0·37	60·4	0·975	0·365
35	65	72·5	0·955	0·30	67·3	0·995	0·245	66·1	1·01	0·235
40	60	78·5	0·98	0·20	73·0	1·01	0·125	71·8	1·02	0·11
45	55	84·3	0·995	0·10	78·7	1·02	0·01	77·3	1·03	−0·007
50	50	90·0	1·00	0·0	84·3	1·015	−0·10	82·9	1·02	−0·12
55	45	95·7	0·995	−0·10	90·0	1·00	−0·20	88·6	1·01	−0·225
60	40	101·5	0·98	−0·20	95·8	0·975	−0·295	94·3	0·98	−0·32
65	35	107·5	0·955	−0·30	101·8	0·94	−0·39	100·3	0·94	−0·41
70	30	113·6	0·915	−0·40	108·0	0·89	−0·47	106·6	0·89	−0·495
75	25	120·0	0·865	−0·50	114·7	0·83	−0·55	113·2	0·83	−0·565
80	20	126·9	0·80	−0·60	121·9	0·76	−0·62	120·5	0·75	−0·63
85	15	134·4	0·715	−0·70	129·9	0·67	−0·675	128·6	0·66	−0·68
90	0	143·1	0·60	−0·80	139·2	0·555	−0·73	138·1	0·545	−0·715
92	8	147·1	0·54	−0·84	143·6	0·495	−0·745	142·5	0·485	−0·73
94	6	151·6	0·475	−0·88	148·5	0·435	−0·76	147·6	0·425	−0·735
96	4	156·9	0·39	−0·92	154·4	0·355	−0·775	153·5	0·345	−0·745
98	2	163·7	0·28	−0·96	161·9	0·25	−0·79	161·3	0·245	−0·748
99	1	168·5	0·20	−0·98	167·2	0·175	−0·795	166·7	0·175	−0·749
100	0	180·0	0	−1·00	180·0	0	−0·80	180·0	0	−0·75

The positions of piston and crank for zero piston acceleration are as follows:

	$\frac{l}{r} = \infty$	$\frac{l}{r} = 6$	$\frac{l}{r} = 5$	$\frac{l}{r} = 4$	$\frac{l}{r} = 3$	$\frac{l}{r} = 2$
Stroke per cent. .	50	46·1	45·4	44·4	43·2	41·7
Crank angle (degrees)	90	80·8	79·1	76·7	73·2	67·7

Crank Effort and Turning Moment.

The crank effort is the force exerted by the connecting rod on the crank pin in the direction of motion of the pin.

The turning moment (at a particular crank) is the crank effort multiplied by the crank radius.

The total turning moment of an engine is the sum of the moments at all the cranks.

To find the crank effort: From the indicator diagram (the assumed probable diagram in the case of design) and the cylinder area, the total force exerted on each side of the piston by the steam is found for any point of the stroke, and hence also the nett steam force on the piston.

To this must be added, with due regard to algebraic sign, the inertia force, *i.e.* the force required to accelerate (or retard) the reciprocating masses. If the engine is vertical, the weight of the moving parts must also be allowed for.

Thus the force P, say, acting at the crosshead in the direction of motion is found.

To find the inertia force at any position of the piston, it is usually sufficient to find the inertia at the ends of the stroke and to find the position at which the inertia is zero, thus fixing three points on the diagram, fig. 24. A fair curve drawn through these points gives the inertia line RRR.

At the inner dead centre, the acceleration $\alpha = 4\pi^2 n^2 r\left(1 + \frac{r}{l}\right)$; at the outer dead centre $\alpha = 4\pi^2 n^2 r\left(1 - \frac{r}{l}\right)$, and $\alpha = 0$ at a position depending on the ratio $\frac{l}{r}$ (see above table).

The inertia force = mass of reciprocating parts $\times \frac{\alpha}{g}$.

Fig. 28 shows the form of a typical diagram of forces. AB is the stroke, A the inner and B the outer end. PPP represents the resultant force of the steam on the piston during the out-stroke and QQQ that during the in-stroke. RRR is the inertia line. Ordinates measured from RRR represent the forces at the crossheads in the direction of motion. (For a vertical engine, the line RRR is lowered by an amount representing the weight of the parts.)

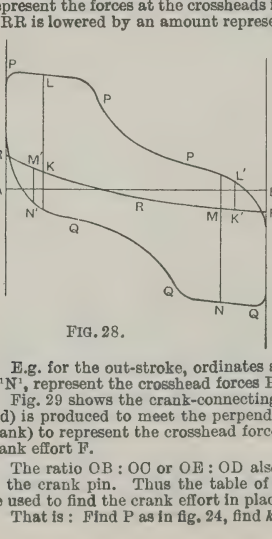


FIG. 28.

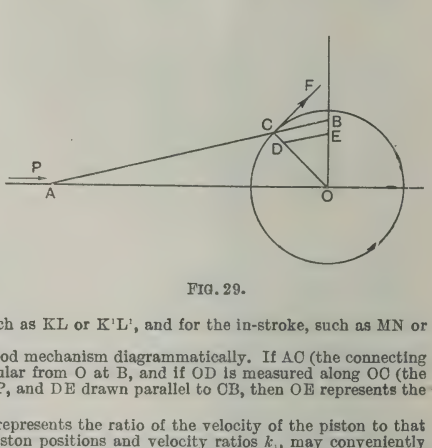


FIG. 29.

E.g. for the out-stroke, ordinates such as KL or K'L', and for the in-stroke, such as MN or M'N', represent the crosshead forces P.

Fig. 29 shows the crank-connecting rod mechanism diagrammatically. If AC (the connecting rod) is produced to meet the perpendicular from O at B, and if OD is measured along OC (the crank) to represent the crosshead force P, and DE drawn parallel to CB, then OE represents the crank effort F.

The ratio OB : OC or OE : OD also represents the ratio of the velocity of the piston to that of the crank pin. Thus the table of piston positions and velocity ratios k_1, may conveniently be used to find the crank effort in place of the graphical method of fig. 25.

That is: Find P as in fig. 24, find k_1 from the table, then the crank effort = force P $\times$ k_1.

Starting Torque of Two-Crank Engines.

Engines which have to start against load, such as locomotive engines, winding engines, etc., have usually two cranks at right angles. They should be large enough to start from any crank position with full steam pressure in the cylinders. The starting torque has minimum values at four points in the revolution occurring just after one crank has passed the position of cut-off. At these points, when the steam stop-valve or throttle-valve is opened, only one of the cylinders receives steam and its crank may be in a very unfavourable position.

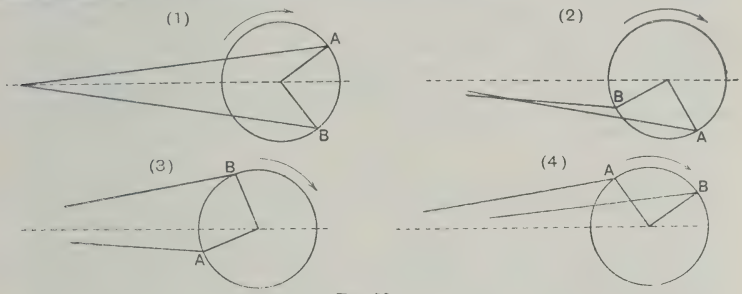


Fig. 30.

The four positions of minimum torque are shown in Fig. 30. In position (1) crank A has just passed the cut-off position on the out-stroke; in (2), B has just passed cut-off on the in-stroke; in (3), A, and in (4), B have just passed cut-off. The worst position is (1).

The ratio of the minimum starting torque to the torque produced by the steam load in one cylinder acting at full crank radius or as it is often termed 'the equivalent crank ratio,' depends on the point of cut-off. For an engine in which the connecting-rod crank ratio is 5, its values are shown in the following table:

Cut-off per cent.	Position 1.	Position 2.	Position 3.	Position 4
70	0·25	0·39	0·56	0·37
75	0·34	0·475	0·66	0·49
80	0·44	0·56	0·755	0·62
85	0·54	0·645	0·845	0·74
90	0·66	0·74	0·92	0·855
95	0·79	0·84	0·99	0·96

For estimating stresses in the shaft, etc., the maximum torque must also be known.

The combined torque due to two cranks at late cut-off reaches maximum values at four points in the revolution. Two of these points are when the cranks are both above the centre line and at 45°, and when both are below at 45°. The ratio of the torque to that due to one cylinder at full crank radius is then 1·414. Another maximum position is when crank A is at 135° from the inner and B at 45° from the outer dead centre, the ratio being then 1·214. The fourth position is with A at 135° from the outer and B at 45° from the inner dead centre, and this gives the largest ratio of all, namely 1·614.

If the cut-off is sufficiently early, the steam valve of one cylinder will close before the above positions are reached, and the steam pressure in that cylinder will have fallen due to partial expansion; the torque will thus be reduced somewhat as compared with late cut-off.

Annealing Connecting Rod Bolts.

In the annual report of *The British Engine, Boiler & Electrical Insurance Co., Ltd.*, for 1924, attention is drawn to the serious results caused by the defective annealing of the connecting rod bolts of an engine.

The Company state that the heat treating of bolts, shafts, or other parts by rough and ready methods cannot be too strongly condemned.

The malpractices, which take place when 'annealing,' have arisen owing to the prevalence of a vague idea that it is possible to restore metal which has been overstrained in service to its original condition by some form of heat treatment.

A common practice of heating parts to an indefinite temperature and for an indefinite period, and following up by protracting the rate of cooling without having any regard to the quality of the material can only lead to trouble.

The following remarks describe a treatment which is suitable for restoring a machined article, such as a connecting rod bolt made of plain carbon steel, to a state of passivity, providing the deformation caused by the overstrain has not been sufficient to set up an incipient internal flaw; in this latter contingency any form of heat treatment is only likely to intensify the defect, and the bolt should be scrapped. The recommendation is based on the assumption that the steel was initially supplied to the user in a suitable condition with regard to heat treatment, and in making it, consideration has been given to the uncertainty as to the past history and exact analysis of the bolt:—

The bolt should be heated uniformly in a specially designed muffle furnace to a temperature in the neighbourhood of 550° C., and at any rate not exceeding 600° C., for a sufficient time to enable it to reach a uniform temperature throughout. In general, with the temperature named, the actual rate of cooling adopted is not of great importance as long as it does not exceed that which takes place in still air; in certain cases, however, the maximum rate named is to be preferred, and is perhaps therefore the most suitable for a general recommendation. Whatever method of cooling be adopted, care must be taken to ensure that cooling is uniform throughout the mass of the steel. Under no consideration should the bolt be cooled on a metal floor or exposed to a draught.

GOVERNORS.

It is the function of the governor to adjust the quantity of steam to the load on the engine. The sensitiveness of the governor should depend on the nature of the load, but in all cases the governor should be as free from friction (which also applies to all the valve mechanism operated by the governor) as possible. Friction tends to coarse governing and hunting. On the other hand, a very sensitive governor, especially if of great power on an engine with insufficient fly-wheel, tends to produce hunting, which should be corrected by an oil dashpot on the governor mechanism. In governor design attention must be paid to the levers and connections so as to give a reasonably equal ratio of governor movement to speed variation at all loads.

The plain Watt fly-ball governor is rarely used, being deficient in power. When loaded with a central weight (Porter governor) it is satisfactory on typical factory engines, especially if carried on ball-bearings and knife edges throughout. The centre weight makes it somewhat sluggish in following changes of speed, and springs replace the weight in many modern governors. The springs need not be vertical.

If the governor is mounted on the crankshaft it may become bulky owing to its slow speed, but has some merits where eccentrics or valve rods have to be moved directly. Shaft governors, owing to their bulk, tend to be sluggish and to lag behind the engine when the speed changes. This inertia effect has been utilised in the inertia shaft governor, in which it is the falling behind of the governor weights due to inertia and not their outward movement under centrifugal force which operates the valve gear.

Relay Governors.

In relay governors the movement of the valve gear is effected by a motor, usually taking the form of a cylinder and piston operated by oil pressure. The movement of the piston is controlled by a small sensitive governor, usually of the centrifugal type. This governor has only to move the oil valve and can be relatively small, whilst the force exerted by the relay piston may be anything desired. The oil is supplied by a reciprocating or, more usually, a rotary pump. A simple drum or toothed-gear pump is entirely satisfactory. Any pressure can be obtained, but 50 or 60 lbs. sq. in. is a convenient value. In calculating the pump capacity allow for the maximum reasonable movement of the relay piston, and for 25 to 50 per cent. slip at the pump according to the speed, size, and pressure of the pump. These governors are common on large turbines and growing in favour for large oil, gas, and steam engines. Unlike the ordinary centrifugal governor insufficient power shows itself chiefly by slow action and not very much by a large difference in the rising and falling speeds. In short, the relay governor is, under these conditions, slow but accurate.

Governors are best driven by gears, but chains may be used and even belts or ropes if the last two are in duplicate or triplicate to minimise trouble in case of one belt or rope breaking.

Governors are best arranged to knock off at over speed, but this function may be arranged for by a simple separate governor.

Governors are often termed 'expansion' or 'throttle' governors, but these expressions refer to the valve gear rather than the governor. Expansion control is the more economical in most cases, but more difficult to arrange for except in the case of engines having Corliss or drop valves and running at moderate speeds only.

Area of Steam Passages and Valves.

The area is usually calculated from the mean piston speed. Thus if the pipe or valve area is one-tenth the piston area and the piston speed is 600 ft. min. or 10 ft. sec. the steam speed through the pipe or valve is 100 ft. sec.

Usual values for steam and exhaust valves, ports, and pipes:—

Saturated steam	100 ft. per sec.
Superheated steam	125 ,,

For drop-seated valves add about 50 per cent. at high speeds, and up to 100 and 150 per cent. for uniflow engines.

For steam turbines base on actual weight of steam, but assumed saturated, and allow 90 ft. per sec. This is equal to about 120 ft. sec. for steam superheated about 150° Fah. at the turbine. A rough rule is 1000 lbs. of steam per hour at rated load per sq. in. of pipe area.

For the main steam range, unless this is very short, base on actual volume, assumed saturated, flowing at about 90 ft. per sec.; rather less, say, 75 to 80 ft. per sec., for reciprocating engines or pipes below 4 ins. in diameter.

PACKINGS.

General Notes.

A packing may operate on two principles. It may, as its name implies, pack the stuffing-box as tight as possible, stopping every channel of escape and relying on the quality and lubrication of its materials to prevent undue friction; or it may employ accurately fitting stop-plates and rings adjusted to give minimum friction and thus permitting a very trifling leakage, which if sufficient to be noticeable is sucked away to the intermediate steam pipe or exhaust, as the case may be. All the old and many of the new packings were on the first principle. Some of the new are on the second; and for valve spindles, where friction is most undesirable, the second should always be adopted. In turbines the second is almost universal, the fitting being very slack but not casual, and the leakage considerable.

Hard-and-fast rules cannot be established as to the composition of the materials which should be used, as these depend on the actual running conditions, but long experience has shown that packing rings of a metallic structure are far superior to the older forms of soft packing; in fact, modern conditions have made the necessity of the change from the one to the other imperative.

The remaining factor in the efficiency of a packing is the design, and it is indisputable that the best results are secured when adjustments to wear and change in pressures are obtained automatically, since the care of machinery is not always entrusted to experienced and skilled workmen.

The United States Metallic Packing.

This packing is made in four distinct forms: 1, block type of packing; 2, cone or locomotive type of packing; 3, duplex type, consisting of combination of the block and cone types: 4, atmospheric-duplex type, specially designed for use on low-pressure condensing cylinders.

The United States Duplex Packing.

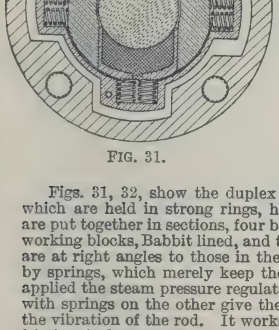


FIG. 31.

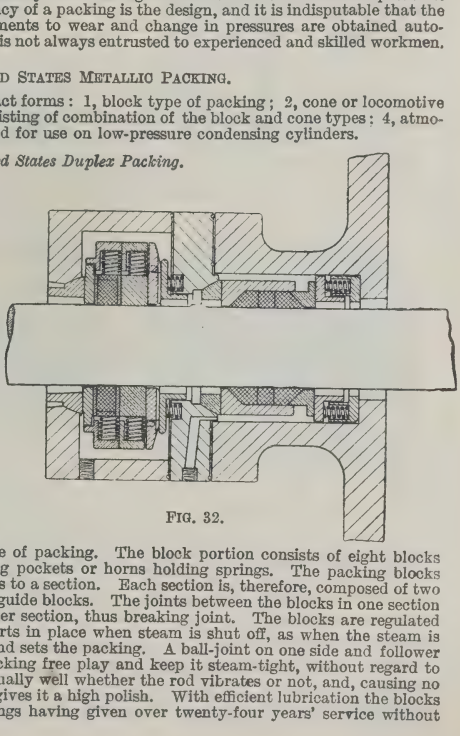


FIG. 32.

Figs. 31, 32, show the duplex type of packing. The block portion consists of eight blocks which are held in strong rings, having pockets or horns holding springs. The packing blocks are put together in sections, four blocks to a section. Each section is, therefore, composed of two working blocks, Babbit lined, and two guide blocks. The joints between the blocks in one section are at right angles to those in the other section, thus breaking joint. The blocks are regulated by springs, which merely keep the parts in place when steam is shut off, as when the steam is applied the steam pressure regulates and sets the packing. A ball-joint on one side and follower with springs on the other give the packing free play and keep it steam-tight, without regard to the vibration of the rod. It works equally well whether the rod vibrates or not, and, causing no friction, it does not wear the rod, but gives it a high polish. With efficient lubrication the blocks wear very slowly, many of the packings having given over twenty-four years' service without renewal of the blocks.

The cone portion of the duplex packing consists of three white metal rings placed in a floating cup, the interior of which is partly conical. This cup rests against ball and socket-ring, and the whole is kept in place by a follower with suitable springs, which allow the packing free play in all its movements. The whole is enclosed in a strong case, and bolted to the cylinder head with the usual stud bolts. The packing is steam setting, hence the friction is reduced at least 50 per cent. as compared with non-steam-setting packings. Drain valves are provided for removal of water, core sand, or other accumulations.

The block type of packing is used for pressures under 100 lbs. per square inch. The duplex type is employed for pressures over 100 lbs. Either packing can be arranged for working with steam superheated to 700° F.

Fig. 33 shows the cone or locomotive type of packing, which is practically similar to the cone packing described above. The white metal rings are the only parts which require renewal, and it is no uncommon thing for locomotives to run 80,000 to 100,000 miles without requiring adjustment of packing. All packings can be arranged to pass over enlarged ends.

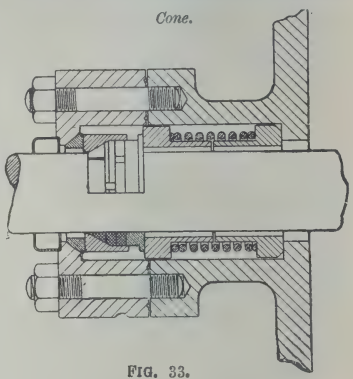


FIG. 33.

UNITED KINGDOM METALLIC PACKING.

Locomotive Packing (cast-iron type).

(United Kingdom Metallic Packing Syndicate, Ltd., 14 Cook Street, Liverpool.)

This type of packing has been specially designed to meet the requirements of consulting engineers and locomotive builders who are now favouring cast-iron blocks for packing to work against the present-day high degree of superheat.

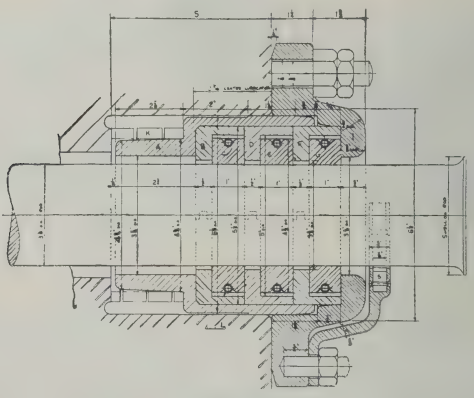
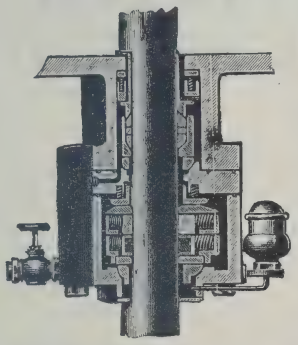


FIG. 34.

Its simplicity over other makes of cast-iron packing will be obvious from an examination of the illustration above referred to.

PACKING



for STEAM ENGINES

Increases Power by
Decreasing Friction

The most widely used
AUTOMATIC METALLIC PACKING
in the world

over 500,000 supplied

STEAM TRAPS

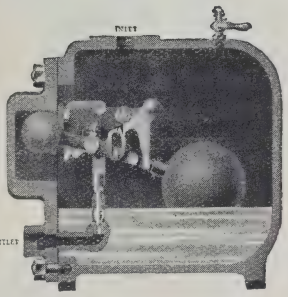
The BRADFORD STEAM TRAP

has a positive action with instantaneous opening and closing.

No wire drawing or dribbling.

FOR PRESSURES UP TO **500** LBS.

LARGE CAPACITY



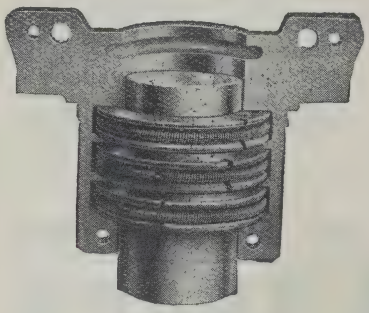
THE UNITED STATES METALLIC PACKING Co. Ltd.,
SOHO WORKS, BRADFORD.

Branch Offices: LONDON, LIVERPOOL, GLASGOW, BOLTON'
SWANSEA, BIRMINGHAM, NEWCASTLE, &c.

F. 1750

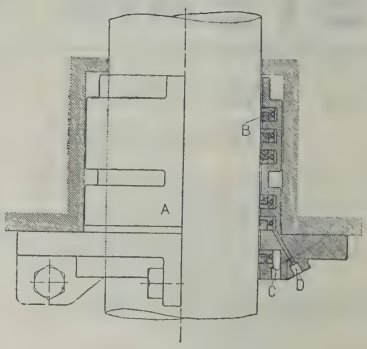
BRITIMP
BRAND
METALLIC PACKING
REGD TRADE MARK

BRITISH METALLIC PACKINGS CO. (1929) LTD.



PACKING RINGS
in HOUSING
Photograph of
Standard Locomotive
Packing.

Illustration of typical Marine Packing as applied by us for numerous conversions to the Bauerwach System.



Head Office :
7 Princes Street, Westminster, London, S.W.1.

Made at Scotswood Works, Newcastle-upon-Tyne, by Sir W. G. Armstrong, Whitworth & Co. (Engineers) Ltd.

Agents throughout the World.

F. 1751

The covers BDF, containing the wearing blocks CEG, are encased in an outer sleeve A, held in position by a powerful spring K, which adjusts the packing, and tends to take up the wear on the walls of the covers (see fig. 34). When wear does take place, the covers themselves require only a slight adjustment by refacing, and can be replaced at a very low cost as against the entire replacement of the expensive housing made in halves, a prominent feature in the old German design of cast-iron packing.

The withdrawal of the packing is greatly simplified as, on the release of the gland, the spring forces the packing from the stuffing-box.

These packings are made from the highest quality obtainable of cast iron, and the best of workmanship used in their manufacture.

A lubricating box is shown attached to the gland, but lubrication can be provided as part of the gland or separate.

'BRITIMP' METALLIC PACKINGS, LOCOMOTIVE TYPE.*

(*British Metallic Packings Co.* (1929), *Ltd., 7 Princes Street, Westminster, S.W.* 1.)

This patent packing consists of three or more rings (fig. 35), each ring forming a complete packing in itself. Each ring is in three segments, which are held together by a garter spring. The joints between consecutive segments overlap in such a way that the combined resistance of the rings to steam leakage is such that perfectly steam-tight glands are ensured. Moreover, the joints are so designed that the parts are reduced to the smallest possible number. Attention is drawn to the fact that there is an entire absence of locating pins which in the course of time are apt either to become loose or be sheared off.

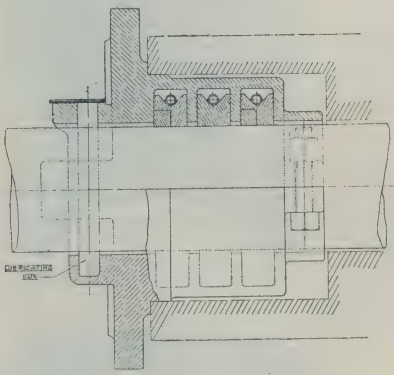


FIG. 35.

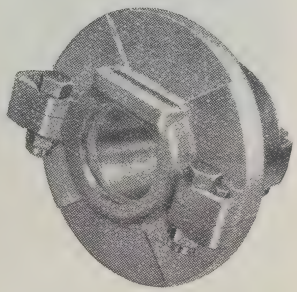


FIG. 36.

The rings are accommodated in a housing, the upper half of which is shown in section, the flanges of which are arranged to meet the requirements of the stuffing-boxes which they have to occupy, so that alterations are unnecessary when substituting for other forms of packings on existing engines. In all cases the outer circumferences of the housings are exposed to the full steam pressure; consequently a steam-tight joint between the two halves is obtained automatically, without complicated bolting or the use of jointing material.

The packing rings are made of hard special-grade cast iron, which quickly acquires a highly polished surface and therefore a low coefficient of friction. All contact surfaces between packings and housings are ground so that the packing responds to any lateral movements of the piston rod with a minimum of resistance.

A lubricating box is provided. In cases where there is dust present in the atmosphere this box is provided with a metal scraper ring, which prevents foreign matter from entering the packing.

In the illustration an external lubricator box is used, but another system is also used where the oil is introduced internally and under pressure.

* See also pp. 1199, 1793, and 1903.

'LION' METALLIC PACKING.
(James Walker & Co. Ltd., Lion Works, Woking, Surrey.)

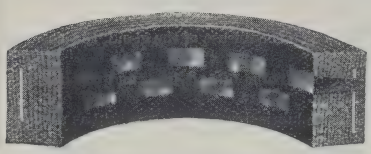


FIG. 37.—'Lion' Block Packing for Steam.

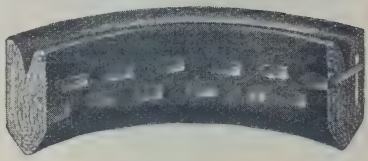


FIG. 38.—'Lion' Expanding Packing for Steam.

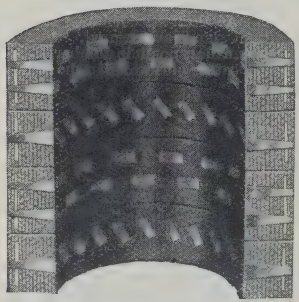


FIG. 39.—Set of 'Lion' Block Packing (in Section).

FIG. 40.—Set of 'Lion' Expanding Packing (in Section).

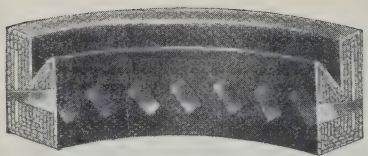


FIG. 41.—'Lion' Automatic Packing for Pumps and Hydraulic Work.

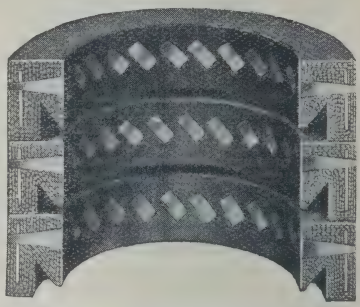


FIG. 42.—Set of 'Lion' Automatic Packing (in Section).

'Lion' Metallic Packing.

(James Walker & Co., Ltd., Lion Works, Woking, Surrey.)

Fig. 37 (p. 1752) shows the Block type of this packing. The packing is made in folds and not in layers, the fabric being taken and folded up on itself, so that the steam getting between these folds causes it to expand both to the rod and the box. This was the original type made by Messrs. James Walker & Co. Ltd., and it is still largely used.

Fig. 38 shows a more recent type of packing. This 'Lion' Expanding packing is made on an expanding principle, and in this instance, instead of the pressure acting upon the folds of cloth, there is a V recess in the top of the packing, into which the pressure finds its way and causes an expansion to the rod and the box.

It will be seen that with either of these types of packing, when there is no pressure there is no friction, and when the pressure is on, the friction is reduced to the smallest minimum on account of the internal pressure alone being required to ensure tightness, no pressure from an external gland being necessary. Both the above types of packing are used for steam purposes, and the expanding type for hydraulic work.

Fig. 39 shows a set of Block packing in section, and fig. 40, a set of the Expanding type in section.

The Automatic type of 'Lion' packing shown in fig. 41 is used principally for hydraulic work. The recess is made on the face of the packing, an arrangement which, it is claimed, is unlike any other type. The action of the Automatic packing is practically the same as that in a ram leather, and the pressure getting at the back of the lip causes it to swell out to the ram of the accumulator or pump.

Fig. 42 shows a set of packings in section; between each turn of packing there is a recess, and the pressure getting into these recesses is baffled. The Automatic packing will stand a test pressure of over 4,000 lbs. per sq. in.

One very important feature with regard to these packings is that a metallic wearing face is formed by the introduction of white metal studs or pins, in order to form in a stuffing-box the same principle of a white metal bearing that is obtained in an ordinary bearing; that is to say, there is a white metal lateral bearing. The studs are so placed that they touch the rod at all points, thus securing a complete metallic wearing surface which reduces friction, does away with the scoring of the rod, and lengthens the life of the packing.

Beldam's Patent Excelsior Metallic Rings (Duplex Type).

(The Beldam Packing & Rubber Co., Ltd., 16 Gracechurch Street, London, E.C. 3.)

These packing rings are specially adapted for superheat, the highest steam pressures, and for L.P. rods of the reciprocating and turbine combination engines.

This packing has been designed in order that the pressure of steam through the neck bushes of the cylinders shall be gradually counterbalanced and cushioned by the pressure of the receivers.

With this system the rings are maintained in equilibrium, thereby preventing any undue wear, the rod being subjected to a minimum of friction.

All the rings are self-adjusting and interchangeable.

Beldam's Patent 'V' Pilot (Tri-Bar) Packing (for Highest Steam Pressures).

(The Beldam Packing & Rubber Co., Ltd.)

This packing is constructed from a single flexible white metal bar, pure asbestos, and high quality steam-resisting rubber, and is specially designed to stand the highest pressures.

The advantages claimed are as follows:—

(1) It ensures an entirely metallic bearing surface on the rod on the slightest pressure of the gland.

(2) The single bar is so constructed that when formed in the 'V' shape the slots on the face are staggered, thus preventing any possibility of the steam escaping down the rod.

(3) These slots also serve a useful purpose in holding oil, thus ensuring the rod being well lubricated.

(4) Rods will not be scored, but, on the contrary, will be kept in perfect condition.

(5) The packing is very strong and pliable, and easily bent round the smallest rod.

Under steam this packing expands so much that the gland nuts require to be only finger tight, and the gland should enter the stuffing box the depth of one turn of packing to allow for slackening back of gland nuts.

Beldam's 'A' Serpent Packing.

(*The Beldam Packing & Rubber Co., Ltd.*, 16 *Gracechurch Street, London, E.C.* 3.)

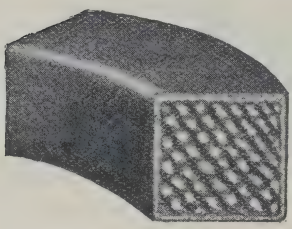


Fig. 43.

This packing, fig. 43, is for superheat and high-pressure steam, and is specially recommended for expansion glands and boiler mountings. Owing to its cross-expansion construction, it is claimed to keep a gland steam or water tight with a minimum of friction on the rod.

The packing is built of alternate layers of metal and cloth placed diagonally, thus forming—as can be seen from the illustration—a series of sliding surfaces, each of which comes into play on the slightest pressure of the gland.

The main feature of this packing is that the character of the wearing surface never varies, there being no flaking of material, as is the case with rolled-cloth packings.

Bell's Patent 'Reefer' Packing.

(*Bell's United Asbestos Co., Ltd., Southwark Street, London, S.E.*1.)

'Reefer,' figs. 44 and 45, is a high-grade semi-metallic packing made in two forms—for steam and hydraulic work. The packing is constructed with a channelled core of anti-friction white metal with side and rear cushions of Bell's Pure Asbestos Cloth (Duck for hydraulic work), proofed with india-rubber and impregnated with best quality plumbago; this form of construction gives a remarkable economy in weight; its design provides for the maximum

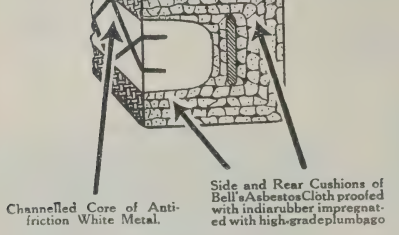


Fig. 44.

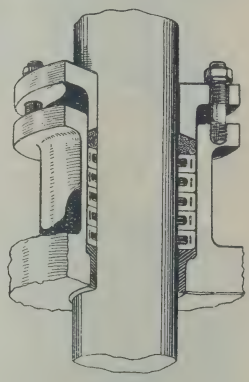


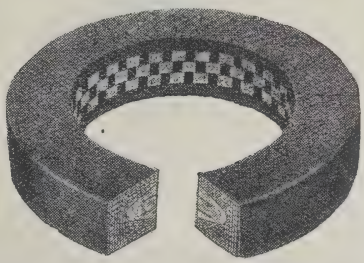
Fig. 45.

surface of anti-friction metal in contact with the piston rod, while using the minimum weight of metal required to give this surface. 'Reefer' is stated to be much lighter than any other packing of its class; it adapts itself to variations in pressure, temperature, and sudden load, and keeps the piston rods in first-class condition, giving smooth silent service with the minimum of attention. In several instances 'Reefer' has run without replacement for twelve months, while in one case a set of 'Reefer' packing rings ran 12,000 miles on high pressure work and were then in sufficiently good condition for transference to the intermediate pressure, where they gave a further spell of duty. 'Reefer' packings, both steam and hydraulic, are supplied in cylindrical tin boxes, each containing the necessary header material. All sizes from $\frac{3}{8}$ in. to $1\frac{1}{2}$ in.

"V" PILOT PACKING

For STEAM *or* WATER

Specially Designed to withstand the Highest Pressures



PATENTED IN U.K. & ABROAD

KNOWN THE WORLD OVER AS THE MOST EFFICIENT PACKING OBTAINABLE

The **Beldam** **PACKING AND RUBBER CO. LTD.**

16, Gracechurch Street, London, E.C.3.

Telephone - MANSION HOUSE 4771
Telegrams - "VEEPILOT, LONDON"

Estd. 1876

F. 1754

The Engineer.

Established 1856. Registered as a Newspaper

PROVIDES
THE LATEST AND THE MOST RELIABLE INFORMATION FOR ENGINEERS
IN ALL BRANCHES OF THE PROFESSION.

The last issue in each month contains a Supplement, **The Metallurgist,** *a journal devoted to the science of metals, ferrous and non-ferrous.*

Subscriptions

		Per annum
INLAND, YEARLY		£3 5 0
CANADIAN SUBSCRIPTIONS.	Thin paper edition ...	£2 18 6
	Thick paper edition ...	£3 3 0
FOREIGN SUBSCRIPTIONS.	Thin paper edition ...	£3 3 0
	Thick paper edition ...	£3 7 6

'THE ENGINEER' DIRECTORY AND BUYER'S GUIDE

SENT FREE ON APPLICATION.

THE LASCAR 'V' PACKING (DUPLEX TYPE).

(*Beldam Asbestos Co., Ltd., Hounslow.*)

The bearing surface of this packing, fig. 46, consists of a number of white metal sections presenting an unbroken and continuous antifriction wearing surface to the rod.

The packing is automatic in its action and allows the rods perfect freedom to move in any direction, owing to the expansive and elastic quality of the sleeve, with the result that the gland nuts need only be finger tight. It is very flexible, being easily bent round the smallest rods.

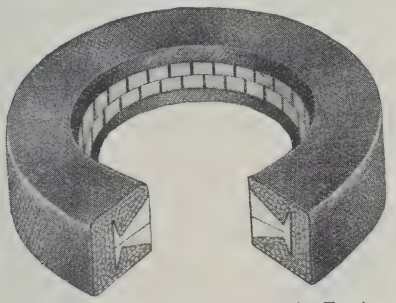


FIG. 46.—Lascar 'V' Packing (Duplex Type).

Valve Packing Tool.

A handy packing tool for quickly placing the packing around valve stems is shown in fig. 47. It is made from a short piece of steel tubing of a suitable size to enter the stuffing box and slip

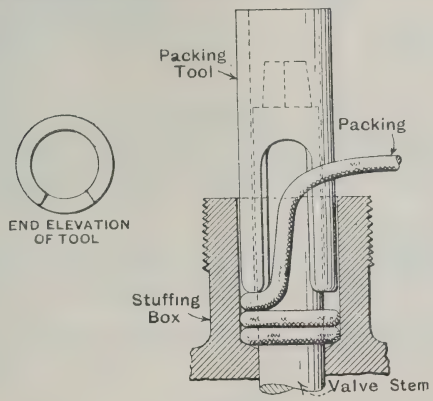


FIG. 47.

easily over the stem. An opening is made in one side of the tube through which the packing passes as the tool is rotated. In use, the end of the packing is started in the usual way and the tool is used to force the packing down until the stuffing box is filled.

(*H. L. Wheeler, Power, June* 19, 1928.)

S.E.A. Rings.

(Ronald Trist & Co., Ltd., 56, Victoria Street, Westminster, S.W. 1.)

Fig. 48 shows this type of packing, which is in ring form and is used for all mediums. The rings are so made that the heel takes the pressure exerted by screwing down the gland, while the tongue is brought into service only by the pressure of the medium.

The outstanding advantages claimed for these rings are their extraordinarily long life and minimum amount of friction as compared with other packings; they have been used for (*a*) steam up to 700° F.; (*b*) hydraulic work up to 15 tons per sq. in; acids up to 98 per cent. pure. Also for mediums which are extremely difficult to pack against, such as ammonia, hot benzol, boiling tar, tar, caustic soda, milk, and beer.

The makers have records in their files of S.E.A. Rings packed in machinery in 1910 still giving satisfactory service in 1928, and many other instances approaching this remarkable record.

The packing of a stuffing box with S.E.A. Rings is a simple matter, the only essential details required being accurate measurements of stuffing box and rod and that pressure greater than finger tight is not applied to the gland.

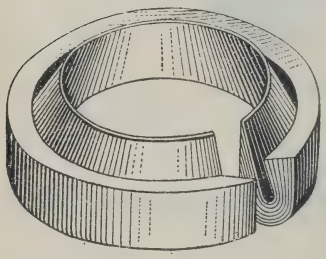


FIG. 48.

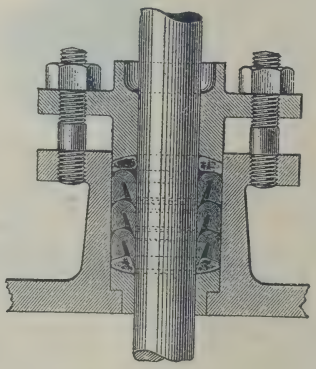


FIG. 49.

S.E.A. Rings are manufactured in two compositions, asbestos for high temperatures, and cotton for liquids and low-pressure steam, and are normally supplied cut, as shown in Fig. 48. This allows them to be slipped over the rod, and even two rings only with the cuts staggered prevent all leakage. Under certain conditions it is essential to use uncut rings—namely, when the box will hold one ring only, for all centrifugal machinery, and certain mediums of a very penetrating nature.

To prevent the tongue of the ring coming into contact with the bottom of the stuffing box a small quantity of soft packing known as 'Header' material is usually supplied, or the gland may be shaped to the contour of the crown of the ring and the neck bush to the heel of the ring. Fig. 49 shows three S.E.A. Rings packed with this Header material. Where the pressure is over ½ ton per sq. in. a shaped gland or shaped metal header is recommended.

S.E.A. Ring cups and collars, made on the same principle as S.E.A. Rings, have largely superseded cup and hat leathers in single or double-acting pistons and all machinery of this type.

PISTONS.

The Rowan Patent Piston.

In the Rowan patent piston (*John Hind & Sons, Ltd.*) absolute steam tightness and absence of friction are obtained by the use of three separate springs, one contained within each packing-ring for outward expansion, and the third lying between the two rings and keeping them in close contact with the piston flanges. These springs are so arranged as to give a *predetermined pressure*

against the sides of the cylinder piston heads (fig. 51).

Fig. 50 shows the form of these rings for use with solid

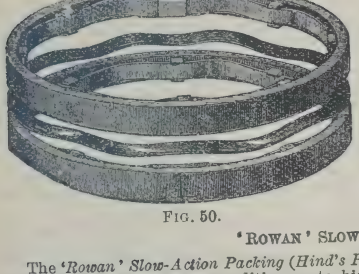


Fig. 50.

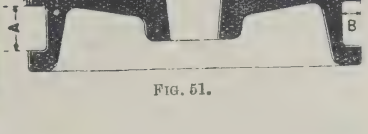


Fig. 51.

'Rowan' Slow-Action Packing.

The *'Rowan' Slow-Action Packing (Hind's Patent)* is a recent development designed to meet the requirements of modern conditions as to high-pressure, high speed, and superheat. In this packing the rings, which are of angle section, have annular grooves turned on the inner side of the flanges, into which is fitted a band of flat steel, which is coiled twice round the inside of the rings, against which it bears. Inside this steel band are two cast-iron spring rings, exerting pressure upon it, and between these is a wave-spring which presses them against the flanges of the piston-ring, thus keeping the latter in steam-tight contact with the flanges of the piston. There are no small or loose parts, the entire packing consisting of six pieces, viz.: Two angle section rings, one steel band, two cast-iron spring rings, and one wave-spring.

The rings are so designed as to give the minimum pressure against the walls of the cylinder consistent with steam-tightness, and, should wear take place, will automatically adjust themselves to the diameter of the cylinder. Should steam get behind the rings, the effect is that the coils of the steel band are pressed together, and the friction so produced absorbs the additional pressure and prevents its being applied to the walls of the cylinder. It is impossible for the steam to get between the steel band and the rings and exert pressure on the rings alone.

Similar action takes place should the rings be subject to circumferential pressure, due to over-running the counterbore or bell-mouth of the cylinder, the pressure of steam being transmitted through the rings to the steel band which is reinforced by the spring rings, when the friction between the coils of the band prevents the rings collapsing.

As these rings will not open and close so freely as the ordinary type of piston rings, they are not suitable for use in worn cylinders, and the makers do not recommend them for such cases, but where the bore of the cylinder is in good order they will give satisfaction under the most severe conditions.

The slow-action rings (Class 'S.A.') are not suitable for springing over solid pistons, and pistons which are to be fitted with these rings must be provided with junk-rings.

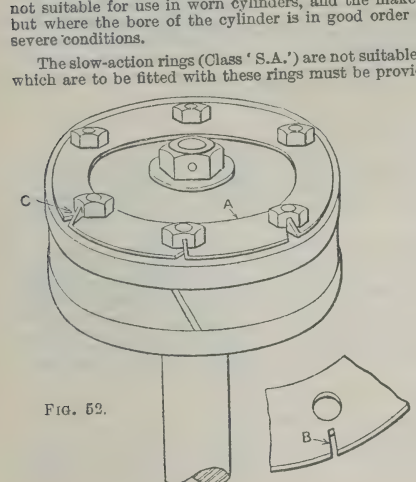


Fig. 52.

The Class 'S.A.' rings may, if desired, be arranged to fit into two separate grooves in the piston, instead of one, as illustrated. In this form it is described as Class 'S.A. Divided.'

It will be obvious from the foregoing that the principle involved renders the 'S.A.' rings suitable, not only for high-pressure pistons, but also for piston valves, in which capacity they are stated to give every satisfaction.

Nut-Lock Plate.

Fig. 52 shows a lock plate used to hold on the follower plate nuts of a steam engine piston. The circular ring, or plate A, is $\frac{3}{32}$ in. plate steel: the holes are drilled for the studs or bolts and a slot cut on the outside edge to within $\frac{1}{4}$ in. of each hole, as shown at B. The lock plate is put on and the nuts all set up tight; then with a cold chisel the tab is bent up against the flat of the nut side, as shown at C.

(*Machinery.*)

SLIDE VALVE GEAR.

(*W. C. Popplewell, M.Sc., Assoc.M.Inst.C.E.*)

The movements of an ordinary eccentric driven slide valve are shown by A, fig. 54. S and S_1 are ports leading from the steam chest to the back and front of the piston respectively; E is the port leading to exhaust. The valve itself is shown sliding on the port face. As shown by A, the valve is in mid-position; *ef* is the outside lap, and *gh* the inside lap. As the valve is moved upwards by the eccentric, it reaches a position B where *e* has come opposite *b*, and the steam port S_1 is opening to the steam chest; the amount by which the valve is open at the beginning of the piston stroke is the 'lead.' In position C the steam port is fully open; the valve then returns to D, thus shutting off the supply of steam to the cylinder. This point is 'cut off.' The movement continues until *h* reaches *c*, as at E, when communication is effected between S_1 and E. This point is called 'exhaust.' After opening S_1 fully to exhaust, the valve returns to F, where the exhaust is closed and the point 'compression' reached. This completes the cycle for one end of the valve.

The movements of slide and other eccentric driven valves are analysed by such diagrams as the 'Valve Ellipse' (II, fig. 53), 'Zeuner's Valve Diagram' (fig. 56), and several others designed for the same use. Of these the Valve Ellipse can be made to take account of obliquity of both connecting rod and valve rod, and so yield accurate results. Zeuner's Diagram makes no allowance for obliquity, and in consequence is only approximately accurate, but for purposes of design, where existing data are not available, the Zeuner is the more useful, as it enables the principal dimensions to be approximately fixed. In such a case final adjustments may be completed by using the Valve Ellipse.

Valve Ellipse.

To set this out, begin by drawing the crank circle as at I, fig. 53, and dividing it into a number of equal angles (in this case 8), the crank-pin points being 0, 1, 2, 3, 4, 5, 6, 7. Take the connecting-rod length $1, 1_1$, and set it off successively from the crank-pin positions, marking off points on the stroke line $0_1, 1_1, 2_1, 3_1, 4_1$. These are positions of the piston corresponding to 0, 1, 2, 3, 4, of the crank-pin, and are seen at II. A similar construction is followed out for the eccentric circle at III (fig. 55). This is turned through a right angle as compared with I, and the 'angle of advance' is marked $\theta°$. The distances of the valve from its mid-position, corresponding to the successive positions of the piston, are shown below the eccentric circle. These are set up as ordinates from the horizontal points on II, and a smooth curve drawn through the points so found. Thus from 1, on the line of piston travel at II is plotted an ordinate which represents the distance of the valve from its central position when the piston is at 1_1; this is the distance from ZZ to the line marked 1 at III. Next draw a horizontal line at distance *l* above $0_1, 4_1$, in II, this is the 'lap' line; also a second line at a distance *p* beyond the lap line, which is the 'port opening.' Similarly below $0_1, 4_1$, are two further lines at l_1, 'inside lap,' and p_1, 'exhaust opening.' The points where the ellipse cuts these lines represent the chief incidents in the cycle. They are B (admission), where the valve is just opening to steam; C (full port); D (cut-off); E (release); F (compression).

In the case of the back plate of a gear of the Meyer type, separate ellipses must be drawn for the edge of the plate, and the points where these cut the main ellipse noted.

Zeuner Diagram.

A few of the principal cases are given in what follows. The results are not strictly accurate, because they are vitiated by the angularity of the eccentric rod. But with reasonable proportions a design will be sufficiently near what the diagram shows. Fuller information may be found in Zeuner's treatise, or in the small book of Cowling Welch, who reduced Zeuner's methods to practice.

Reversing gears which do not employ two eccentrics usually derive motion from some point on the connecting rod. Such are the Joy gear and others, for all of which special advantages are claimed under certain conditions, though as valve movements it is doubtful if any give better results than plain link gears. The newer motions often involve considerable sliding movement of a block in an inclined link, this link being pivoted so as to reverse its inclination, whereby the valve, driven by the block which slides up and down in the inclined link, has its movement reversed or varied according to the degree of inclination either way of the link. Such valve movements are best studied by the aid of models, with variable dimensions; made from flat bars of cedar or baywood.

Simple Slide Valve.

Fig. 56 shows Zeuner's diagram of slide valve motions, simple case.

Draw NM and PR at right angles; **NM** = piston stroke.

OP = OR = valve travel each side of centre point; OA = crank when steam is admitted; OB = crank at cut-off.

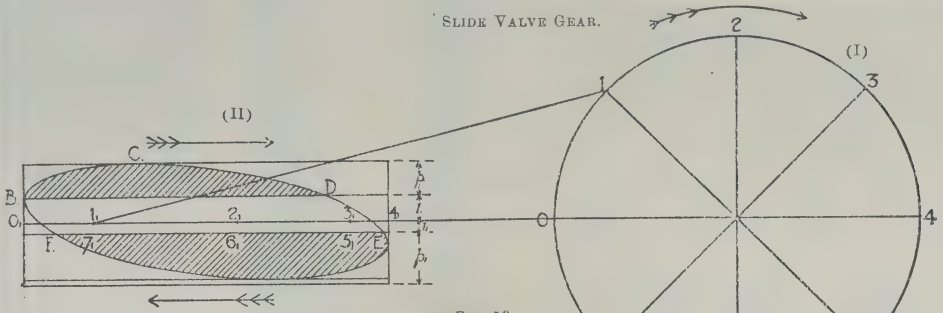


FIG. 52

FIG. 55.

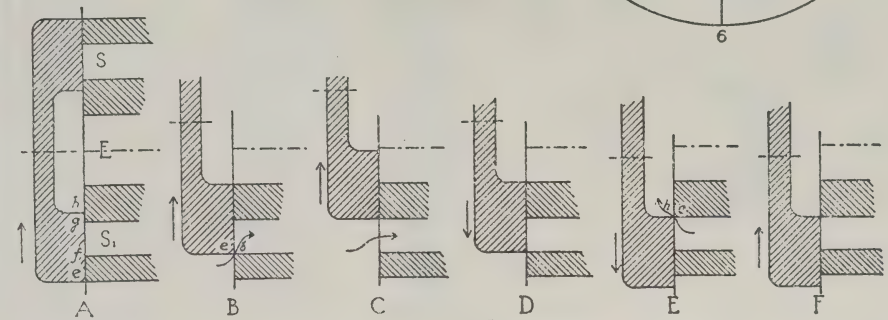


FIG. 54.

Bisect AB. Join OX, OY perpendicular to AB.

$x°$ + 90°=angular advance of eccentric; SGT=outside lap circle: cuts 'valve' circle at ST; OS=position of valve from centre when admitting steam; OT=position of valve from centre when cutting off steam; OF=inside lap; OD=crank when exhaust commences; OE=crank when exhaust closes.

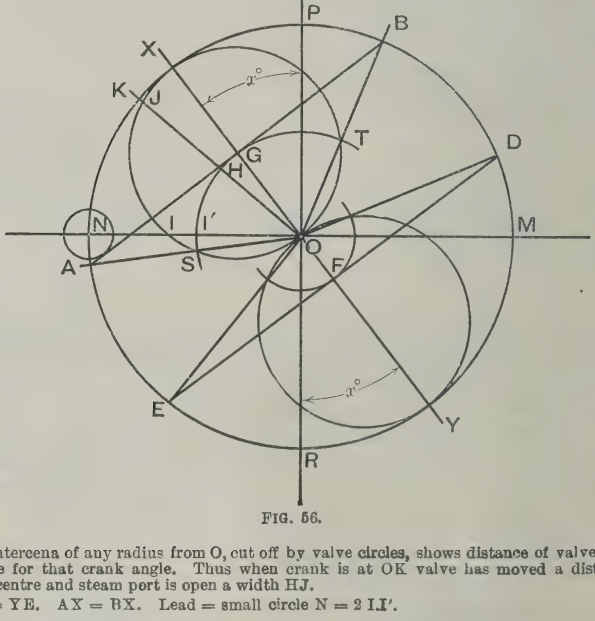


FIG. 56.

The intercena of any radius from O, cut off by valve circles, shows distance of valve from its midstroke for that crank angle. Thus when crank is at OK valve has moved a distance OJ from its centre and steam port is open a width HJ.

DY = YE. AX = BX. Lead = small circle N = 2 II′.

Double Back-Plate Cut-off.

Figs. 57 and 58 show Zeuner's expansion valve diagram, Meyer valve.

Draw AB, XY at right angles, OD at angular advance $\phi°$ = half travel of main valve; OR = crank at admission: ON = lap circle. Draw OP slightly in advance of latest opening of main valve as shown by point *t*. This gives a minimum cover *ts*. Join PD, and draw OE parallel to it and DE parallel to OP. Then OE = half travel and position of the cut-off eccentric which will cut off steam when the crank, OR, reaches the position OP. It need not cut off later than this, for then the main valve has already acted.

The radius O*a* or any other vector of the cut-off valve circle shows the movement of the cut-off valve from its middle position, OE being the half throw of the cut-off eccentric, and $\theta°$ its angular advance. Any radius vector of the auxiliary circle OP will show the relative displacements of the main and cut-off valves from each other. At the crank position OP the cut-off plates are drawn close together. Draw the large circle APDM through P. Then for any position OV of desired cut-off the length of the cut-off valve must be increased by separating the two cut-off plates of the right and left screw to the amount MZ. To cut off at OX of the crank would demand the movement FH, while to cut off at the dead point OB the movement of the plates would be JK. The length over the steam ports in the back of the main valve will be equal to the sum of the length of the cut-off plates plus twice OP. The edges of the cut-off plates when fully extended have a maximum distance from the main valve centre of JK (or MZ if OM is the earliest desired cut-off). But at no moment must the inside edges of the cut-off plates uncover the inside edges of the ports in the main valve. These ports have a width *a*. At the latest cut-off, viz. at OP, the cut-off plates will be drawn together so that x=nil.

To find movement of cut-off plates from centre for any point of cut-off, as with crank at OV, the plates must be screwed out a distance $x = MZ$. If cut-off is at OF position of crank, then $x = FH$.

Taking the position of earliest cut-off at OV, to find the proportions L and l of valves which will be a maximum :—

Calling $FE = y = L - l - x$ for earliest cut-off, the maximum relative movement of the valve centres by the eccentrics is OP. Since at the position OV the valve centres are the opposite way

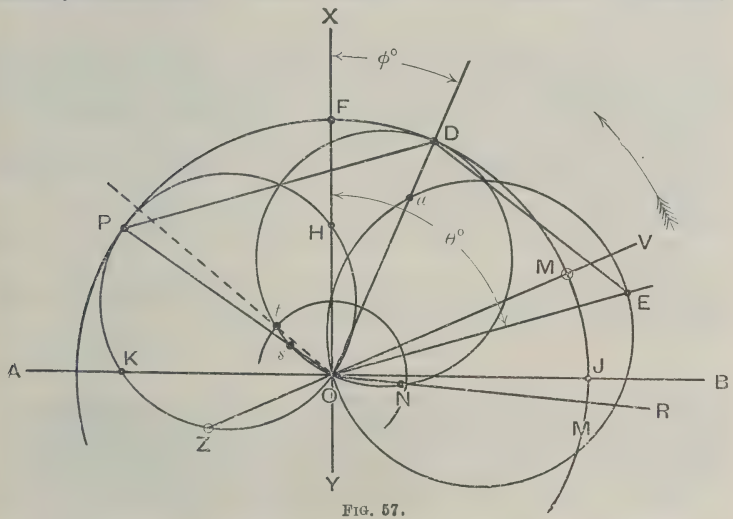


FIG. 57.

removed from cut-off by the amount OZ and that OP is the maximum relative movement, we have $L - l = OP$ when x=nil and x must be $OP+OZ = MZ$ for earliest cut-off. When the crank reaches OP the cut-off valve is moved out already a distance $x = MZ$ and it is moved a further distance OP by the eccentric, so that the outer edge of the cut-off plate is a distance OP+OP+OZ from the middle of the main valve. And when so far the inner edge of

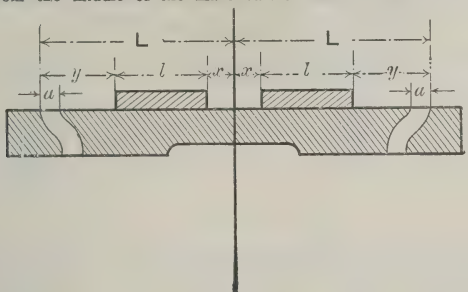


FIG. 58.

the cut-off plate must not overrun the inner edge of the back port a. Therefore $L-a=2OP+OZ$ ∴ $L=OP+OP+OZ+a$ and $L-l=OP$ ∴ $l=OP+OZ+a$. Since the main valve has cut-off at the position Ot there is safety lap of ts on the main port below even, if the inner edge of the cut-off plate then reaches the port a.

In the figure the main valve travels $4\frac{1}{4}$ ins. and the cut-off valve $5\frac{1}{2}$ ins. The cut-off plates are each moved out by screws at earliest cut-off $x=3\frac{1}{4}$ ins., and by eccentric movement $2\frac{1}{8}$ ins.

relative to main valve, or a total of $5\frac{3}{8}$ ins. The main valve has a length $L=5\frac{3}{8}+a$ and the cut-off plate (or $L-OP=l$) a length of $3\frac{1}{4}$ ins. $+ a$.

If $a=1$ in. then $L=6\frac{3}{8}$ ins.; $l=4\frac{1}{4}$; $x=2\frac{1}{8}$ ins. maximum plus $3\frac{1}{4}$ screw-out.

If early cut-off is desired in the general working of the engine with quick action the auxiliary circle OP should be near to 90° from the cut-off position of the crank. Thus at the position OD the two valves are moving at high relative speed. It is usual to give the greater travel to the cut-off valve.

Locomotive Link Motions.

These may be investigated also by the Zeuner method. A link motion is simply built up with two eccentrics, either of which may be made to bear fully and directly on the valve spindle and cause an engine to run forwards or backwards. This is accomplished by connecting the two eccentric rods by a link in which the end of the valve spindle is carried. At the extreme end position the spindle is wholly driven by one eccentric, while at intermediate positions the two eccentrics combine to give a gradually shorter movement to the valve.

There are three general link gears. 'Stephenson's,' in which the link is raised or lowered to give the desired position of the spindle block or link block; and the link is concave to the axle and has a radius equal to the length of the eccentric rods.

'Gooch's,' with link convex to the axle and not movable up and down; movement of the block being effected by a radius rod and the link having this radius of curvature.

'Allen's,' in which the link is straight and both link and radius rod are raised and lowered in opposite directions.

All three forms are made with open or crossed rods, the former giving greater lead of the valve near the middle position of the link block, and the latter giving less lead towards the middle position. In Gooch's motion the lead is of course always constant.

Stephenson Link Motion.

Open Rod Design.

In fig. 59, AE=link. BD=link block in forward and backward gear. OJ=virtual single eccentrics to give requisite travel to B or D in full gear. Join OB, cutting valve circle on OJ at K. Join JM through K. Then the arc struck from c through JMJ will contain the ends of all the eccentric radii or valve circle diameters for all positions of the slider B or D. Dividing this arc in the same ratio as B divides the link AE will give the ends of the valve circles for each point of the link. And the real eccentrics will have the position and half travel OF, OF just as the ends of the link lie beyond B and D. If the link end-pin comes opposite the extreme

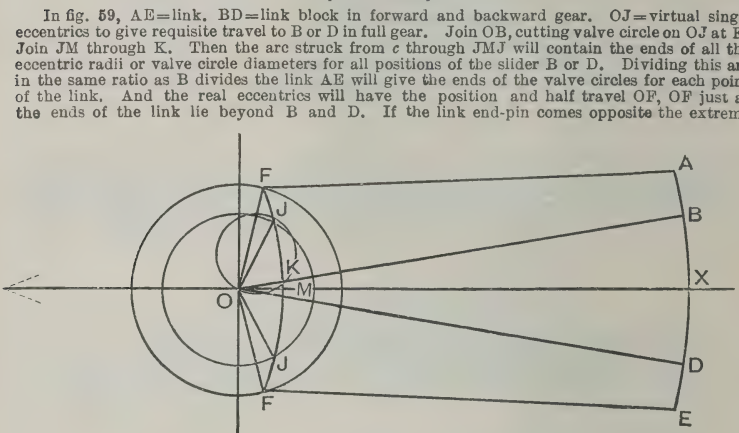


Fig. 59.

positions BD of the block then OJ become the real eccentrics in position and in half throw, i.e. XB : XA : : MJ : MF. Rods are said to be *open* when they are as FA, FE when the eccentric arms point towards the link. With open rods the valve lead increases with the expansion.

Crossed Rod Design.

In fig. 60, AE=link. BD=link block in full forward or backward gear. OJ=virtual single eccentric that would give requisite valve travel in full gear. Join OD, cutting upper valve circle on OJ at L. Join JN through L. N is on the line OX. Then the arc struck from O through JNJ contains the ends of all the valve circles for every position of the slide block, the arc JNJ being divided for each diameter in the same ratio as the slide block divides the link AE.

Thus OG, OG become the actual eccentric arms, and NJ : NG : : XB : XA. Rods are said to be *crossed* when they are as GA, GE, the eccentric arms pointing towards the link. With

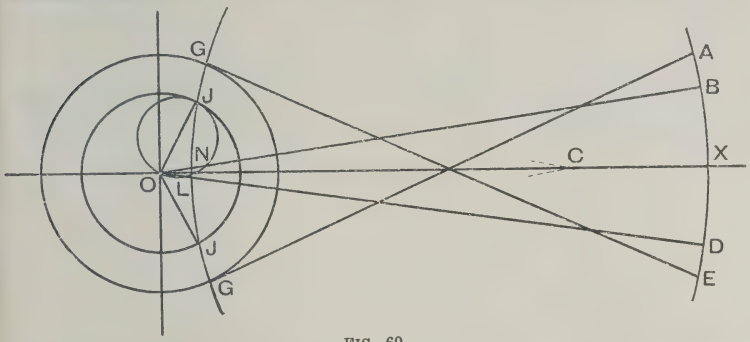


FIG. 60.

crossed rods the lead of the valve decreases with increased expansion. Specially note that OD cutting the valve circle at L is employed in this construction, whereas OB is the line used in the open-rod diagram.

GOOCH'S LINK MOTION.

In this motion, fig. 61, the 'curve' of valve circle centres is a straight line *aa* distant from O an amount $f = \frac{r}{2}\left(\sin\delta^\circ + \frac{c}{l}\cos\delta^\circ\right)$ for the open rod design, and $\frac{r}{2}\left(\sin\delta^\circ - \frac{c}{l}\cos\delta^\circ\right)$ for the crossed rod design, where

r = eccentricity of eccentric or OE;

m is the point from which the link is hung and is a fixed point;

the versed sine of the link arc is $\frac{c^2}{2l_1}$;

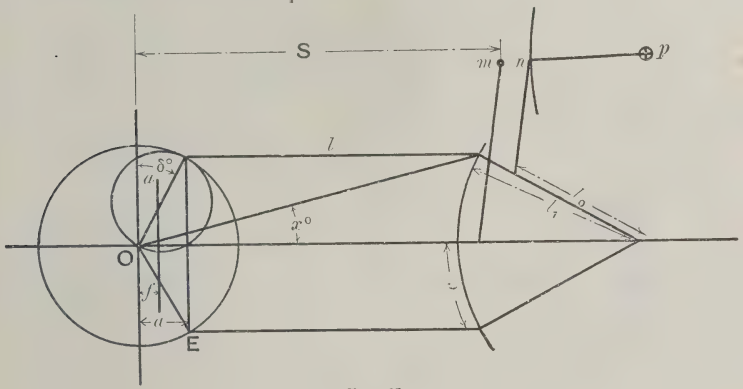


FIG. 61.

m is distant from O a length $S = -\frac{c^2}{2l_1}$;

n is the point from which the radius rod is hung

$On = l + l_1 - l_0 - \frac{c^2(l+l_1)}{2ll_1}$.

The point n should move in a parabola whose parameter $= 2l_1$ or a circular arc of radius l_1.

The co-ordinates of any point on the curve of centres are as per fig. 62, where

for open rods, $OB = \frac{r}{2}\left(\sin\delta^\circ + \frac{c}{l}\cos\delta^\circ\right)$;

for crossed rods, $OB = \frac{r}{2}\left(\sin\delta^\circ - \frac{c}{l}\cos\delta^\circ\right)$.

$BC = \frac{r}{2c}\left(\cos\delta^\circ \pm \frac{c}{l}\cos\delta^\circ\right)$, according as the rods are open for — or crossed for + signs, where c = half length of the link and u = the variable position distance of the slider block from the middle of the link.

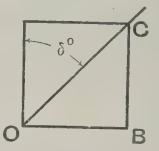


FIG. 62.

The lead is constant at all grades of expansion. The link is so hung that its middle point moves to and fro horizontally by the combined effect of the arc of swing and the cant due to slope of the link, the suspension link being pinned at the centre of the chord of the link.

The radius rod l_o is so hung that its to-and-fro motion produces a minimum of sliding of the link block in the link. The extreme positions of n being known the actual arc in which it moves is arranged in practice with a shorter arm np than theory demands, the best compromise possible being accepted which can be arranged under the conditions. The Gooch gear, for any given distance between axle and cylinder, has very short rods compared with the Stephenson gear. Also the angle δ° in fig. 66 is equal to the angle x° as found in fig. 56, p. 1760, for the simple slide valve, plus the angle δ° in fig. 61, p. 1763.

ALLEN'S STRAIGHT LINK MOTION.

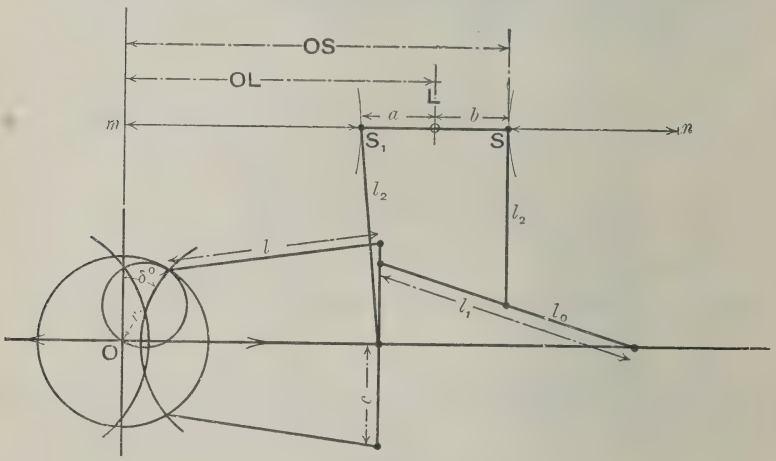


FIG. 63.

$$\frac{b}{a} = \frac{l_o}{l}\left(1 + \sqrt{1 + \frac{l}{l_1}}\right);\quad n = 1 + \left(\frac{l_1}{l_o}\,\frac{b}{a}\right).$$

Curve of centres is a parabola whose vertex is distant from O $= \frac{r}{2}\left(\sin\delta + \frac{c}{l}\cos\delta\right)$ for open

rods, and $\frac{r}{2}\left(\sin\ \delta - \frac{e}{l}\cos\ \delta\right)$ for crossed rods, and is concave towards O for open rods, and convex for crossed rods. Parameter for both open and cross rods $= \frac{nlr}{2c\cos\ \delta}\cos\ \delta \mp \left(\frac{c(n-1)}{nl}\sin\ \delta\right)^2$ for which a circular arc may be substituted of one half the parameter.

S and S_1 should move in parabolas of $2l_1$ and $2l$, or in arcs of l_1 and l radius.

As arranged in practice $a + b = (l_1 - l_0) +$ a little extra l_2 as long as possible.

$$Os_1 = \left(l - \frac{c^2}{2l}\right);\ \mathrm{OL} = l + a - \frac{c^2}{2};\ ms_1 = l;\ ns = l_1;\ Os = l + l_2 - l_0 - \frac{c^2(l + l_1)}{2l\,l_2}.$$

Lead increases with expansion with open rods and decreases with crossed rods.

WALSCHAERTS' GEAR.

(See page 594.)

SECTION XXX

PART IV

STEAM TURBINES.*

FLOW OF STEAM—NOZZLES—BLADE EFFICIENCY—LOSSES—PACKINGS—ROTORS—MATERIALS—TEST DATA—TURBINE TYPES—REDUCTION GEARS.

(Contributed by H. L. Guy, Wh. Exh., Assoc.M.Inst.C.E., M.I.Mech.E., and L. S. Robson, M C., M.Sc.Tech., A.M.I.Mech.E.

In this section the following symbols have been used in the indicated significance, except where otherwise stated :—

c = absolute velocity in ft. per sec.
g = constant of gravitation = 32·2 ft. per sec.^{2}
$H.h.$ = heat drop in B.Th.U.
N = speed in r.p.m.
P = pressure in lbs. per sq. ft.
p = pressure in lbs. per sq. in.
t = temperature °F.
T = absolute temperature °F = $(460+t)$.
U = peripheral velocity in ft. per sec.
v = specific volume in cubic ft. per lb.
W = weight of steam in lbs. per sec.
w = relative velocity in ft. per sec.
λ = ratio of the specific heats of a gas.
ω = angular velocity in radians per sec.
J = Joule's Equivalent = 778 ft. lbs. per B.Th.U.

FLOW OF STEAM.

When steam flows through a simple orifice from a chamber (denoted by suffix 1) into a space (denoted by suffix 2) the velocity with which discharge takes place is always dependent on the steam conditions existing before the orifice, and sometimes dependent upon those existing on the discharge side.

If the pressure, P_1, in the chamber is maintained constant, while that beyond the orifice is gradually reduced from an initial value equal to P_1, the velocity and quantity of discharge gradually increase and depend on the pressure drop over the orifice, until a critical value is reached corresponding to a definite value of the ratio $\frac{P_2}{P_1}$.

Dr. Zeuner † has shown theoretically, and the result is borne out by experiment, that the critical drop is reached when

$$\frac{P_2}{P_1} = \left(\frac{2}{\lambda+1}\right)^{\frac{\lambda}{\lambda-1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

For saturated steam $\lambda = 1{\cdot}135$, and $\frac{P_2}{P_1} = 0{\cdot}577$.

For superheated steam $\lambda = 1{\cdot}3$, and $\frac{P_2}{P_1} = 0{\cdot}546$.

---

* See also Marine Steam Turbines, Section XXXVI.
† *Technical Thermodynamics*, Dr. Zeuner.

After this *Critical Pressure Drop* has been reached, the quantity and velocity of discharge remain constant for smaller values of P_2, the discharge being accompanied by strong acoustic vibrations.

Dr. de Laval was able to obtain—and used in his turbine—much higher velocities than the critical velocity, by adding a conically divergent nozzle to the orifice. In this way the vibrations are eliminated, and although the velocity of exit from the nozzle may be considerably greater than the critical velocity, the velocity at the minimum section or throat of the nozzle and the weight of steam discharged remain unchanged from the values which would exist in a simple orifice.

From this it will be apparent that there is a discontinuity in the phenomena accompanying discharge of steam through an orifice or nozzle, and that the equations existing between the various terms entering into the formulæ of steam flow must be considered in two groups.

(*a*) *When the Pressure Drop is less than the Critical Drop.*

i.e., when $P_2 > 0{\cdot}577\ P_1$, for saturated steam; $P_2 > 0{\cdot}546\ P_1$, for superheated steam.

The theoretical velocity c_1 with which discharge takes place, can be calculated from the formula due to Wantzel and St. Venant,

$$c_1 = \sqrt{2g \frac{\lambda}{\lambda - 1} (P_1 v_1 - P_2 v_2)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The actual velocity of discharge c_2 is less than c_1, on account of the friction and eddy losses in the orifice, and is given by

$$c_2 = \phi\, c_1,$$

$$\text{where } \phi = \text{velocity coefficient} = \frac{\text{Actual velocity of discharge}}{\text{Theoretical velocity of discharge}}.$$

The value of ϕ for various cases is given in Table II., page 1772.

The weight of steam discharged per second is given by the relation

$$W = k\, S \frac{c_2}{v_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where,

v_2 = specific volume of the steam in the lower state, in cubic ft. per lb.
S = the sectional area in sq. ft. of the smallest section or throat of the orifice.
k = coefficient of discharge (see Table II.).

The expansion takes place with very great rapidity and for purposes of calculation it may be assumed to be adiabatic without serious error. The validity of this assumption has been questioned, but Dr. A. Loschge* has proved its approximate correctness by obtaining equal discharge through duplicate nozzles made of metal and of porcelain, and it need only be added that formulæ deduced from this basis agree very closely with experimental results.

For adiabatic expansion—

$$P_1 V_1^{\lambda} = P_2 V_2^{\lambda} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

from which, by combination with (2) and (3), we get

$$W = k S \sqrt{2g \frac{\lambda}{\lambda - 1} \frac{P_1}{v_1} \left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{\lambda}} - \left(\frac{P_2}{P_1}\right)^{\frac{\lambda+1}{\lambda}} \right]} \quad . \quad . \quad . \quad . \quad (5)$$

Alternative Method.

The following method of dealing with such problems, due to Dr. Zeuner, is often more simply manipulated than that already given, especially when used in conjunction with the Mollier or other entropy diagram.

If

H_1 = total heat of 1 lb. of the steam in the higher condition in B.Th.U.;
H_2 = total heat of 1 lb. of steam in the lower condition in B.Th.U.;
h_a = the heat drop available for adiabatic expansion = $H_1 - H_2$.

* *Engineering*, October 17, 1913.

Then

$$c_1 = \sqrt{2g \cdot J \cdot h_a} \quad . \quad . \quad . \quad (6), \quad \text{and } c_2 = \phi \cdot c_1.$$

W can be calculated as before from (3); h_a can either be measured directly off a Mollier diagram or calculated from steam tables in the following manner :—

If $t°$ = degrees of superheat; L_1 = latent heat corresponding to P_1; T_1 = absolute temperature corresponding to P_1; h_1 = sensible heat corresponding to P_1; X_1 = dryness fraction of the steam in the higher state; T_2 = absolute temperature corresponding to P_2; h_2 = sensible heat corresponding to P_2; X_2 = dryness fraction of the steam in lower state; ϕ_1 = entropy of the liquid in the higher state; ϕ_2 = entropy of the liquid in the lower state;

then

$$X_2 = \frac{T_2}{L_2}\left[\frac{X_1 L_1}{T_1} + C_p \log_e\left(\frac{T_1 + t}{T_1}\right) + \phi_1 - \phi_2\right],$$

and

$$h_a = X_1 L_1 + C_p \cdot t \cdot + h_1 - X_2 L_2 - h_2.$$

In these formulæ the mean value of the specific heat of superheated steam at constant pressure C_p for the range of the superheat should be taken.

Nozzles which are designed for a smaller pressure drop than the critical are wholly convergent and must be sized according to the methods given in this section.

Simple orifices, the nozzles of Rateau or Zoelly turbines, and both the rotating and stationary blades of Parsons turbines, fall into this class.

(b) When the Pressure Drop is Greater than the Critical Drop.

i.e., when $P_2 < 0{\cdot}577P_1$, for saturated steam; $P_2 < 0{\cdot}546P_1$, for superheated steam.

In this case let the suffix m denote the conditions in the minimum section or 'throat' of the nozzle.

Then whatever the total pressure difference over the nozzle, that from P_1 to the throat pressure is always equal to the critical value, and the nozzle must be convergent to the throat.

By combining (1) with (2) and (4), the velocity in the throat is obtained.

$$= \sqrt{g\lambda \cdot P_1 V_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

if $\phi = 1{\cdot}0$.

(7) is identical with the expression for the velocity of sound in the conditions present in the throat, so that the velocity in the throat, or the critical velocity, is equal to sound velocity.

By substituting from (1) and (4) in (5)—

$$W = k S_m \sqrt{2g \frac{\lambda}{\lambda + 1}\left(\frac{P_1}{V_1}\right)\left(\frac{2}{\lambda + 1}\right)^{\frac{2}{\lambda - 1}}} \quad . \quad . \quad . \quad . \quad . \quad (8)$$

from which the discharge per second can be obtained, S_m being the area at the throat in square feet.

The nozzle from the throat to the mouth must be divergent, the area at the mouth being calculated from the relation

$$S_2 = S_m \cdot \frac{V_2}{V_m} \cdot \frac{c_m}{c_2} = S \left[\left(\frac{2}{\lambda + 1}\right)^{\frac{1}{\lambda + 1}} \left(\frac{P_1}{P_2}\right)^{\frac{1}{\lambda}} \sqrt{\frac{\frac{\lambda - 1}{\lambda + 1}}{1 - \left(\frac{P_2}{P_1}\right)^{\frac{\lambda - 1}{\lambda}}}}\right] \quad . \quad . \quad (9)$$

Expressions involving λ are strictly applicable only when the expansion takes place wholly in either the superheated or the saturated region, the corresponding value of λ being introduced.

The following expressions will be found to give approximately the same results as (9).

For Superheated Steam.

If $P_1 = (1{\cdot}732 \text{ to } 20)\ P_2$ $\quad S_2 = S_m\left(0{\cdot}7175 + 0{\cdot}163\ \frac{P_1}{P_2}\right)$

$P_1 = (20 \text{ to } 100)\ P_2$ $\quad S_2 = S_m\left(1{\cdot}5 + 0{\cdot}125\ \frac{P_1}{P_2}\right)$

For Saturated Steam.

If $P_1 = (1{\cdot}832 \text{ to } 25)\ P_2$ $\quad S_2 = S_m\left(0{\cdot}777 + 0{\cdot}1216\ \frac{P_1}{P_2}\right)$

$P_1 = (25 \text{ to } 100)\ P_2$ $\quad S_2 = S_m\left(1{\cdot}875 + 0{\cdot}0812\ \frac{P_1}{P_2}\right)$

Introducing the numerical value of λ (1·135 for saturated steam and 1·3 for superheated steam), and with $g = 32{\cdot}2$, formula (7) becomes—

For Saturated Steam. $\quad c_m = 70{\cdot}2\ \phi\ \sqrt{p_1 v_1}$

For Superheated Steam. $\quad c_m = 72{\cdot}38\ \phi\ \sqrt{p_1 v_1}$

In the same way formula (8) would produce two expressions for the discharge depending on the condition of the steam as regards superheat. Researches by Loschge* and Bendemann have however shown that the numerical coefficient of the formula for nozzles having a convergent entrance is the same both for saturated and superheated steam, and that this value is 2 per cent. higher than is obtained by inserting $\lambda = 1{\cdot}135$.

This is in accordance with the previously known fact that for saturated steam the discharge from such a nozzle exceeds the theoretical amount.

Just prior to the publication of these researches Mr. H. M. Martin* propounded the ingenious theory that with steam initially saturated an unstable condition exists in the throat, cooling taking place below the saturation temperature without condensation. The decrease in specific volume which results from this 'Under-Cooling' would account for the measured excess in discharge over that calculated on the assumption of adiabatic expansion.

Prof. Stodola has suggested that this under-cooled steam obeys the same law of expansion as superheated steam, so that the experimental fact that the discharge both for saturated and superheated steam is given by one formula is rational and in line with theory.

The formula for discharge deduced from Loschege's experiments is :—

$$W = 44{\cdot}11\ k\ S_m \sqrt{\frac{p_1}{v}}.$$

Table I., giving the critical velocity and the discharge per square inch of throat area, has been calculated on the assumption that $\phi = 1{\cdot}0$ and $k = 1{\cdot}0$; the necessary correction for other values will be obvious.

The following simple formulæ, the first due to Napier and the second to Rankine, are often useful when an error of a few per cent. is admissible.

For Dry Saturated Steam.

$$P_1 > \frac{5}{3} P_2,\ W = \frac{s \cdot p_1}{70}; \qquad P_1 < \frac{5}{3} P_2,\ W = 0{\cdot}029\ s\ \sqrt{p_2 (p_1 - p_2)},$$

where s = area of orifice in square inches.

Design of Nozzles.

The values of ϕ and k as determined by experiment are given in Table II. for orifices and nozzles of various kinds.

The coefficient of discharge k depends amongst other things upon the friction loss, the coefficient of contraction, and usually for a pressure drop below the critical, upon the ratio $\frac{P_2}{P_1}$.

Professor Rateau found that for nozzle C the value of k was slightly greater than unity when P_2 is less than $0{\cdot}5P_1$.

* *Engineering*, October 17, 1913.

TABLE I.—DISCHARGE OF DRY SATURATED STEAM WHEN $P_2 < 0{\cdot}57\ P_1$.

Pressure before nozzles.—Lbs. per sq. in. g.	260	220	200	180	160	140	120	100	80	60	40	20	10	0
Steam velocity in throat.—Ft. per sec.	1510	1508	1504	1500	1496	1492	1486	1480	1472	1465	1450	1432	1415	1392
Discharge.—Lbs. per sec. per sq. in. throat area	3·92	3·55	3·07	2·8	2·51	2·24	1·94	1·67	1·39	1·09	0·813	0·523	0·376	0·229

TABLE Ia.—DISCHARGE OF SUPERHEATED STEAM WHEN $P_2 < 0{\cdot}54\ P_1$.

Pressure before nozzles.—Lbs. per sq. in. g.	Superheat °F.	260	220	200	180	160	140	120	100	80	60	40	20	10	0
Discharge.—Lbs. per sec. per sq. in. throat area	100	3·61	3·1	2·84	2·59	2·32	2·07	1·81	1·55	1·29	1·02	0·755	0·486	0·35	0·211
Discharge.—Lbs. per sec. per sq. in. throat area	200	3·39	2·91	2·67	2·43	2·19	1·94	1·70	1·45	1·20	0·96	0·712	0·457	0·329	0·198
Discharge.—Lbs. per sec. per sq. in. throat area	300	3·19	2·76	2·53	2·31	2·08	1·85	1·61	1·38	1·15	0·913	0·673	0·434	0·311	0·190

TABLE II.

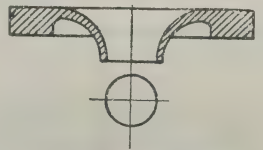
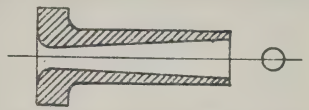
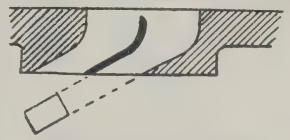
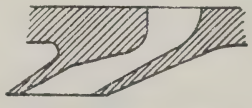
Type of Orifice.	Velocity Coefficient ϕ and Discharge Coefficient k.	Name of Experimenter.	Reference.
A Circular Hole in Thin Plate.	$\phi = 0{\cdot}86$	Rosenhain	Proc. Inst. C.E. vol. cxl.
	$\frac{P_2}{P_1} = 0{\cdot}95 \quad 0{\cdot}8 \quad 0{\cdot}6 \quad 0{\cdot}4$ $k = 0{\cdot}63 \quad 0{\cdot}68 \quad 0{\cdot}77 \quad 0{\cdot}83$	Rateau	Flow of Steam' A. Rateau
B Cylindrical Hole.	$\frac{P_2}{P_1} = 0{\cdot}95 \quad 0{\cdot}8 \quad 0{\cdot}7 \quad 0{\cdot}6$ and smaller values $k = 0{\cdot}79 \quad 0{\cdot}83 \quad 0{\cdot}86 \quad 0{\cdot}88$	Gutermuth	Z. Ver. Deutsch. Ing. January 1904
C  Straight Convergent Nozzle.	$c_e = 500 \quad 900 \quad 1{,}300$ ft. per sec. $\phi = 0{\cdot}936 \quad 0{\cdot}965 \quad 0{\cdot}993$ Saturated steam $\phi = 0{\cdot}954 \quad 0{\cdot}967 \quad 0{\cdot}98$ Superheated steam	Loschge	Z. Ver. Deutsch. Ing. January 11, 1913
	$\frac{P_2}{P_1} = 0{\cdot}95 \quad 0{\cdot}8 \quad 0{\cdot}7 \quad 0{\cdot}6$ smaller values $k = 0{\cdot}94 \quad 0{\cdot}967 \quad 0{\cdot}975 \quad 0{\cdot}99 \quad 1{\cdot}0$	Rateau	'Flow of Steam' A. Rateau
D Rectangular Orifice.	$\frac{P_2}{P_1} = 0{\cdot}95 \quad 0{\cdot}8 \quad 0{\cdot}7 \quad 0{\cdot}6$ smaller values $k = 0{\cdot}8 \quad 0{\cdot}88 \quad 0{\cdot}91 \quad 0{\cdot}93 \quad 0{\cdot}93$	Gutermuth	Z. Ver. Deutsch. Ing. January 1904

TABLE II.—*continued.*

Type of Orifice.	Velocity Coefficient ϕ and Discharge Coefficient k.	Name of Experimenter.	Reference.
E Convergent-cylindrical Nozzle. Circular Throat.	$c_1 =$ 200 500 1,000 1,500 ft. per sec. $\phi =$ 0·923 0·935 0·95 0·97 $\frac{P_2}{P_1} =$ 0·95 0·5 $k =$ 1·0 1·0	Briling	Z. Ver. Deutsch. Ing. February 1910
		Hirn ('Experiments on Air')	'Flow of Steam' A. Rateau
F  Straight Convergent-divergent Nozzle. Circular Throat.	For a nozzle 3 ins. long $\phi = 0·95$ „ „ „ 6 „ „ $\phi = 0·915$ $\frac{P_2}{P_1}$ Less than 0·89 $k = 0·98$	Sibley and Kemble	Journal Amer. Soc. M.E. November 1909
		Gutermuth	Z. Ver. Deutsch. Ing. January 1904
G  Turbine Convergent Nozzle. Rectangular Throat.	$c_1 =$ 1,200 1,400 1,600 ft. per sec. $\phi =$ 0·86 0·89 0·91 $k = 1·0$	Christlein	Z. für G. Turbinenwesen January 1912
H  Turbine Divergent Nozzle. Circular or Rectangular Throat.	$c_1 =$ 1,700 2,000 2,200 2,700 3,300 4,000 ft. per sec. $\phi =$ 0·92 0·92 0·93 4·94 0·95 0·955 $k = 1·0$	Christlein	Z. für G. Turbinenwesen January 1912
		Loschge	Z. Ver. Deutsch. Ing. January 1913.

In reference to nozzles G and H, which are of the types used in impulse turbines, it should be mentioned that although the value $k = 1{\cdot}0$ has been inserted in Table II., Dr. Christlein found that the actual discharge exceeds the theoretical by a few per cent. This effect had already been observed by Mr. Rosenhain for nozzles of form F having well-rounded entrances, and he concluded from his experiments that a loss in efficiency results, and was, therefore, led to recommend that the entrance should be only slightly rounded.

These facts are confirmed by Dr. A. Loschge's researches and explained by Mr. H. M. Martin's theory, which is referred to in the preceding paragraph.

The value of ϕ for nozzle G cannot be very satisfactorily determined from Dr. Christlein's experiments; for a given value of ϵ_1, ϕ appears to increase with the pressure and to decrease with the superheat before the nozzle.

Unfortunately, the results of Mr. Rosenhain's experiments, like those of Messrs. Siberly and Kemble, are vitiated by the fact that the wetness of the steam during the experiments was not measured.

Nozzles of type B are useful for approximate measurements of the amount of steam flowing through a pipe line. It is only necessary to put a thick plate having a cylindrical hole in it between two of the pipe flanges. If the pressure is observed before the plate (P_1) and also in the cylindrical hole (P_2) by means of a hole drilled from the rim through the centre thickness of the plate, the flow can be calculated from formula (5).

Of course, the hole in the plate should be of such a diameter that the ratio $\frac{P_2}{P_1}$ is not less than 0·6.

Several steam meters operate on this principle.

The nozzles of turbines are usually of rectangular cross-section, and the steam in passing through them turns through an angle of about 70°, as indicated in figs. G and H, p. 1773. It is important to bear this in mind when employing data relating to ϕ and k deduced from straight nozzles of circular cross-section—the type usually favoured by experimentalists.

For a given ratio of expansion the length of the nozzle depends on the angle of divergence. If this angle is too great the steam does not completely fill the cone, which results in considerable loss. If the angle is too small the length of the nozzles becomes very great, and many experiments indicate that the friction loss depends on the length. An angle of divergence of 10° is very common.

The experiments of Dr. Rosenhain, and more recently of Messrs. Siberly and Kemble, show that there is no appreciable difference in efficiency for angles varying from 9° to 20°.

If the ratio $\frac{S_2}{S_m}$ is less than that theoretically required, expansion is completed beyond the nozzle. Rateau's experiments demonstrate that when this *under-expansion* is small, little loss results. On the other hand, if $\frac{S_2}{S_m}$ is greater than that theoretically required, the steam is expanded in the nozzle below the condition into which it discharges. Such *over-expansion* results in steam shock with great loss of efficiency.

When the divergent nozzles of a turbine are correctly designed for the normal load, over-expansion takes place at overloads, and under-expansion at partial loads, with nozzle cut-out governing.

The entrance of divergent nozzles should be smoothly rounded off, with a radius about equal to the throat diameter; larger radii are unnecessary and should be avoided.

Since the efficiency of a nozzle depends upon the smoothness of the surfaces with which the steam is in contact, the material of which it is constructed should be capable of withstanding erosion and oxidisation—nickel steel or a bronze highly polished is usually employed for the divergent nozzles of De Laval and Curtis turbines where high steam velocities are encountered, while cast in mild steel plates have given quite satisfactory results with the large nozzle areas and moderate velocities used in the convergent nozzles of Rateau and Zoelly turbines.

The state of knowledge on the efficiency of steam nozzles has been recently advanced by the researches carried out for the Nozzle Research Committee of the Institution of Mechanical Engineers by Dr. Stoney and Dr. Telford Petrie.

The apparatus used is of the impact plate type, largely designed by Mr. H. M. Martin. The first two reports dealt with tests taken on 1-in. wide reaction blading having a nominal outlet angle of 20°, and on impulse nozzles having an outlet angle of 20° with guide blades 0·04 in. thick.

STEAM NOZZLES RESEARCH COMMITTEE—VELOCITY COEFFICIENTS.

Impulse Nozzles : Nominal Angle 20°. Plates—0·04 in. thick.

Reaction Nozzles : Parsons Normal 480 B. Normal Angle, 20°.

Ratio $\frac{\text{Parallel Exit}}{\text{Throat}} = \frac{0}{1}, \frac{1}{1}, \frac{2}{1}, \frac{3}{1}$

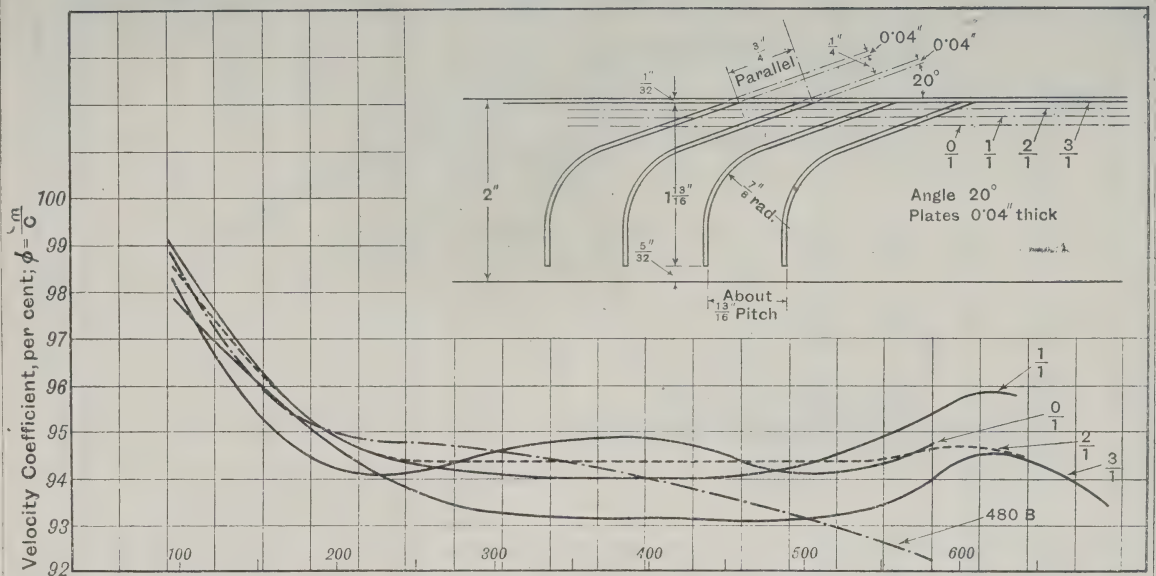


FIG. 1.

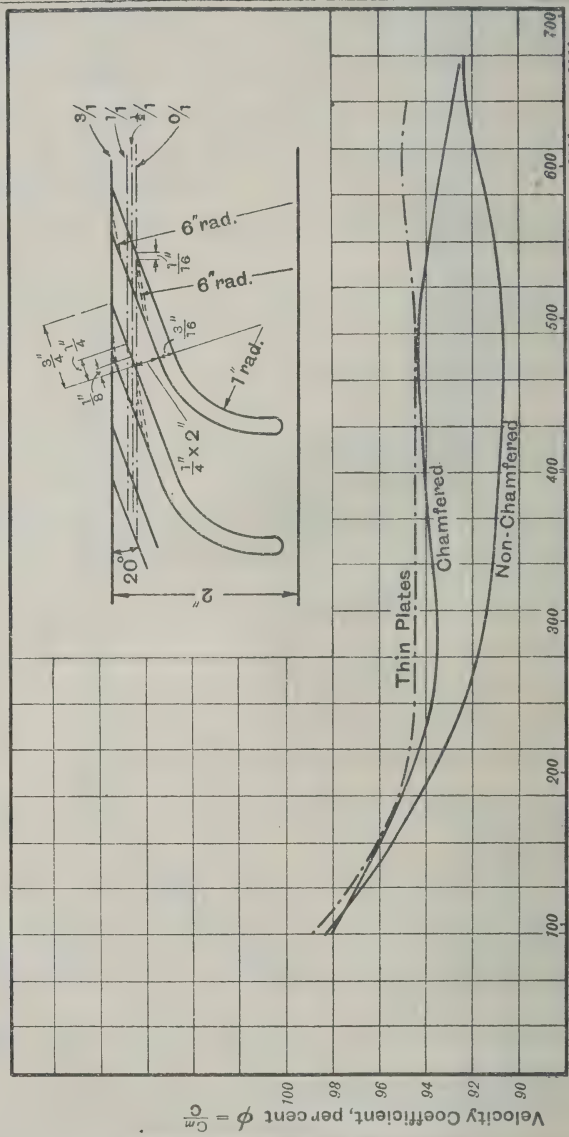

Velocity Coefficient, per cent $\phi = \frac{C_m}{C}$
Theoretical Steam Velocity "C," Ft. & Metres per Sec.
Thin Plates
Chamfered
Non-Chamfered
6" rad.
6" rad.
1" rad.
$\frac{1}{4}$" × 2"
20°
2"
$\frac{3}{4}$"
$\frac{1}{4}$"
$\frac{1}{8}$"
$\frac{3}{16}$"
$\frac{1}{16}$"
9/1
1/1
½/1
0/1

FIG. 2.

TABLE IIIA.—ADIABATIC HEAT DROP IN B.TH.U./LB.

Vacuum. 30 ins. Barm.	Superheat. °F.	Pressure Lbs. per Sq. In. Gauge. 2000	1500	1200	1100	1000	900	800	700	600	550	500	450	400
27·5	0	463·5	449·6	438·0	433·2	427·9	422·1	415·5	407·6	398·6	393·4	387·8	381·4	374·5
	100	504·7	487·1	473·0	467·3	461·2	454·2	446·5	437·6	427·2	421·2	414·9	407·7	399·9
	200	543·2	523·0	506·8	500·6	493·6	485·9	477·3	467·6	456·2	449·7	442·9	434·9	426·5
	300	580·4	557·8	540·2	533·3	525·9	517·5	508·2	497·8	485·7	478·7	471·4	463·1	454·1
28·0	0	472·8	459·1	447·6	442·9	437·7	431·9	425·4	417·6	408·6	403·5	398·0	391·7	384·8
	100	514·5	497·1	483·1	477·5	471·5	464·5	456·9	448·1	437·7	431·9	425·6	418·4	410·7
	200	553·5	533·5	517·4	511·2	504·3	496·7	488·1	478·5	467·2	460·8	454·0	446·1	437·7
	300	591·0	568·6	551·1	544·3	537·0	528·6	519·3	509·0	497·1	490·2	482·9	474·7	465·7
28·5	0	484·5	471·0	459·8	455·2	449·9	444·3	437·9	430·2	421·3	416·2	410·7	404·5	397·6
	100	526·8	509·7	495·9	490·4	484·3	477·5	470·0	461·2	451·1	445·3	439·0	431·9	424·2
	200	566·3	546·5	530·7	524·5	517·7	510·1	501·7	492·1	480·9	474·7	467·9	460·1	451·7
	300	604·6	582·2	564·8	558·1	550·8	542·6	533·5	523·2	511·3	504·5	497·2	489·0	480·2
28·75	0	491·7	478·4	467·3	462·7	457·5	451·9	445·5	438·0	429·0	424·0	418·6	412·4	405·6
	100	534·3	517·4	503·8	498·3	492·2	485·5	478·1	469·3	459·2	453·5	447·3	440·2	432·6
	200	574·2	554·6	538·8	532·7	526·0	518·4	510·1	500·5	489·4	483·2	476·5	478·7	460·4
	300	612·8	590·6	573·2	566·6	559·1	551·2	542·2	531·9	520·0	513·3	506·0	497·9	489·2
29·0	0	500·4	487·3	476·3	471·8	466·6	461·1	454·8	447·3	438·5	433·5	428·2	422·0	415·4
	100	543·7	526·8	513·3	507·9	501·9	495·2	487·8	479·1	469·1	463·4	457·3	450·2	442·7
	200	583·9	564·4	548·9	542·8	536·0	528·5	520·2	510·8	499·7	493·5	486·9	479·2	470·9
	300	622·6	600·7	583·5	576·8	569·7	561·6	552·5	542·4	530·7	524·5	516·8	508·7	500·1
29·25	0	511·5	498·5	487·8	483·3	478·3	472·8	466·5	459·1	450·5	445·6	440·3	434·2	427·5
	100	555·4	538·7	525·4	520·0	514·1	507·5	500·1	491·6	481·7	476·0	469·9	463·0	455·6
	200	596·1	576·8	561·4	555·4	548·7	541·3	533·1	523·7	512·8	506·6	500·1	492·4	484·4
	300	635·2	613·5	596·5	589·9	582·8	574·8	565·8	555·8	544·2	537·5	530·4	522·4	513·7
29·5	0	526·9	514·2	503·7	499·3	494·3	488·9	482·9	475·6	467·0	462·3	457·0	451·0	444·4
	100	571·6	555·2	542·0	536·7	531·0	524·5	517·1	508·8	499·0	493·4	487·4	480·6	473·2
	200	613·0	593·8	578·7	572·7	566·1	558·9	550·7	541·5	530·7	524·6	518·1	510·5	502·5
	300	652·5	631·2	614·3	607·8	600·9	592·8	584·0	574·1	562·6	556·0	549·0	541·1	532·5

TABLE IIIA.—*continued.*

Vacuum. 30 ins. Barm.	Superheat. ° F.	Pressure Lbs. per Sq. In. Gauge. 350	300	275	250	220	200	180	160	140	120	100	80	60
27·5	0	366·3	357·0	351·8	346·0	338·3	332·6	326·3	319·3	311·5	302·5	291·9	279·5	263·9
	100	390·9	380·6	374·8	368·5	360·1	353·9	347·0	339·4	330·9	321·2	310·0	296·6	280·0
	200	416·8	405·7	399·5	392·8	383·8	377·1	369·9	361·8	352·8	342·6	330·8	316·7	299·4
	300	443·9	432·2	425·6	418·6	409·2	402·2	394·6	386·2	376·8	366·2	353·9	339·3	321·3
28·0	0	376·7	367·5	362·3	356·5	349·0	343·3	337·1	330·1	322·4	313·4	303·1	290·7	275·3
	100	401·7	391·5	385·8	379·6	371·2	365·1	358·3	350·7	342·3	332·7	321·5	308·3	291·9
	200	428·1	417·1	410·9	404·2	395·3	388·7	381·5	373·5	364·6	354·5	342·8	328·8	311·6
	300	455·5	443·9	437·4	430·4	421·1	414·2	406·6	398·3	389·0	378·4	366·2	351·7	334·0
28·5	0	389·8	380·7	375·5	369·9	362·4	356·7	350·5	343·7	356·0	327·2	317·0	304·7	289·5
	100	415·4	405·3	399·6	393·5	385·2	379·1	372·4	365·0	356·6	347·1	336·1	323·0	306·8
	200	442·2	431·4	425·2	418·7	409·9	403·4	396·2	388·3	379·5	369·4	357·9	344·0	327·0
	300	470·2	458·7	452·2	445·3	436·0	429·2	421·8	413·5	404·3	393·8	381·7	367·4	349·8
28·75	0	397·8	388·8	383·6	378·1	370·6	365·1	358·9	352·1	344·5	335·8	325·6	313·4	298·3
	100	423·9	413·8	408·1	402·0	393·9	387·8	381·1	373·7	365·4	356·0	345·0	332·1	315·9
	200	451·0	440·2	434·1	427·6	418·8	412·3	405·2	397·4	388·6	378·7	367·1	353·4	336·6
	300	479·2	467·8	461·3	454·5	445·3	438·5	431·1	422·9	413·7	403·3	391·3	377·1	359·6
29·0	0	407·6	398·7	393·6	388·1	380·6	375·1	369·1	362·3	354·8	346·1	336·0	324·0	309·0
	100	434·1	424·2	418·6	412·5	404·4	398·3	391·7	384·4	376·1	366·8	355·9	343·0	327·0
	200	461·6	450·9	444·8	438·3	429·7	423·2	416·2	408·4	399·7	389·8	378·4	364·8	348·1
	300	490·1	478·8	472·4	465·6	456·5	449·7	442·4	434·2	425·1	414·8	402·9	388·8	371·5
29·25	0	419·9	411·2	406·1	400·6	393·3	387·8	382·0	375·3	367·7	359·1	349·1	337·2	322·4
	100	447·1	437·2	431·7	425·7	417·7	411·8	405·2	397·9	389·7	380·5	369·6	356·8	340·9
	200	475·0	464·4	458·4	452·0	443·4	437·1	430·0	422·3	413·8	403·9	392·6	379·2	362·5
	300	504·0	492·6	486·4	479·7	470·7	464·0	456·6	448·6	439·6	429·5	417·6	403·6	386·2
29·5	0	436·8	428·2	423·1	417·8	410·6	405·3	399·5	392·9	385·3	376·9	367·1	355·3	340·8
	100	464·7	455·1	449·6	443·6	435·7	429·8	423·4	416·2	408·2	399·1	388·3	375·7	360·1
	200	493·4	482·9	477·0	470·6	462·1	455·8	449·0	441·3	432·8	423·1	411·9	398·7	382·3
	300	522·9	511·7	505·5	498·8	489·8	483·2	476·0	468·2	459·3	449·1	437·5	423·7	406·9

The figures in the above tables have been abstracted from 'Heat Drop Tables' and 'Extended Heat Drop Tables' prepared by Prof. H. L. Callendar, F.R.S., and published by the British Electrical and Allied Industries Research Association.

The mean velocity coefficient (ϕ) values for the impulse nozzles obtained on those tests are shown on fig. 1, p. 1775, for nozzles with throats of various proportions. The corresponding curves for the reaction blading are shown by the dotted curve.

The notable feature of these tests is that they were carried out as large-scale experiments on commercial blades and nozzles.

The character of the curves, particularly for the impulse nozzles showing a rising value at low velocity, is particularly notable, as the region was not previously explored in published researches.

The third report of the Nozzle Research Committee of the Institution of Mechanical Engineers contains the results obtained with thick plate impulse nozzles summarised in diagram, fig. 2.

It will be observed that although the efficiency of the thick plate nozzles is materially lower than that of thin plate nozzles, the efficiency of the former can be improved to that of the latter by suitably chamfering the outlet.

In this report the experimental results on the efflux angle of the discharged steam suggest that its value is equal to $\sin^{-1}\frac{o}{p}$, and not to the geometrical angle (see fig. 3, and table C).

Experiments were also carried out on straight nozzles well radiused at the inlet and having long parallel throats equal to about three diameters in length.

The efficiency of this nozzle was very high, giving a velocity coefficient θ of about 0·98 for 500 ft./sec., 0·97 for 300 ft./sec., and 0·98 for 1,700 ft./sec.

The experiments also lead to the conclusion that the efficiency of a nozzle is decreased as the superheat is increased.

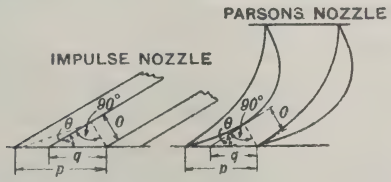


FIG. 3.—Efflux Angles.

TABLE C.

—	$\frac{o}{q}$	θ Degrees.	$\frac{o}{p}$	δ Degrees.	Angle of Exit. Degrees.	Difference Degrees.
20° thin plate	0·34	20	0·308	18·0	17·0	−1·0
20° thick plate	0·34	20	0·190	11·0	14·0	+3·0
Ditto chamfered	0·34	20	0·190	11·0	10·5	−0·5
12° thin plate	0·21	12	0·178	10·3	10·5	+0·2
12° thick plate	0·21	12	0·121	7·0	6·0	−1·0
1 in. Parsons	0·61	37	0·350	20·5	18·5	−2·0
⅜ in. Parsons	0·61	37	0·350	20·5	18·5	−2·0

TURBINE EFFICIENCY AND STEAM CONSUMPTION.

The 'Overall Efficiency' of a turbine is obtained by comparing the work actually done per lb. of steam, with that which would be performed by an ideal engine working on the Rankine cycle and having the same initial and final steam conditions as the turbine.

This latter amount of work is equal to the mechanical value of the change in total heat contents (h_a B.Th.U.) which takes place during adiabatic expansion from the steam conditions before the governor valve, to those obtaining in the turbine exhaust.

h_a is referred to as the 'adiabatic heat drop' or the 'available heat drop,' and its value for the steam conditions usual in practice is given in Table IIIA., p. 1777.

TABLE IIIB.—ADIABATIC HEAT DROP IN B.TH.U./LB. DRY SATURATED STEAM.

Vacuum 30 ins. Barm.	Pressure Lbs. per Sq. In. Absolute.					
	14	15	16	17	18	19
27·0	144·41	148·90	153·11	157·08	160·82	164·35
27·25	149·33	153·79	157·99	161·95	165·68	169·19
27·5	154·67	159·12	163·30	167·24	170·96	174·46
27·75	160·51	164·94	169·10	173·03	176·73	180·21
28·0	166·96	171·37	175·51	179·42	183·10	186·57
28·25	174·20	178·58	182·70	186·59	190·25	193·70
28·5	182·45	186·81	190·90	194·76	198·40	201·83
28·75	191·97	196·30	200·36	204·19	207·81	211·22
29·0	203·51	207·80	211·82	215·62	219·21	222·59
29·25	217·92	222·15	226·17	229·95	233·44	236·73
29·50	237·96	242·10	246·07	249·79	253·23	256·48

If

d = steam consumption in lbs. per S.H.P. hour; η_T = overall efficiency of the turbine, then

$$\eta_T = \frac{2{,}544}{d.\,h_a} \times 100 \text{ in per cent.}$$

If

d_1 = steam consumption in lbs. per kWh.; η_G = generator efficiency in per cent., then

$$\eta_T = \frac{3{,}412}{\eta_G d_1 . h_a} \times 100 \text{ in per cent.}$$

The total steam consumption of the turbine is given by

$$W = \frac{d\ (\text{S.H.P.})}{3{,}600} \text{ lbs. per sec., or, } W = \frac{d_1\ (\text{kW.})}{3{,}600} \text{ lbs. per sec.}$$

While the efficiency calculated with h_a as already defined is the true index of the performance of a turbine when running, it is sometimes the practice—and it is one which has a special value to turbine designers—to calculate the efficiency by taking the heat drop h'_a as that available between the conditions existing after the governor valve, that is, before the nozzles, and the exhaust of the turbine.

This is equivalent to eliminating the consideration of the pressure drop through the governor valve, which is rightly regarded as not being an inherent loss to any given turbine.

Suppose, for example, that two turbines are sold as 1,000 kW. units to operate with certain steam conditions, and that one is actually capable of developing a load of 1,100 kW., while the second can only pull a load of 1,000 kW.

If these two turbines are tested on a load of 1,000 kW., the latter will show the better steam consumption, because with the former the steam will be throttled at the governor to the greater extent. Of course, the first machine is in every way the better from the customer's point of view, provided the excess of power is not too great to be carried by the generator.

The steam consumption which can be used for estimating purposes when using steam at 200 lbs. per square inch, 200° F., and exhausting into a vacuum of 28 ins., is shown by fig. 4 for 1,500

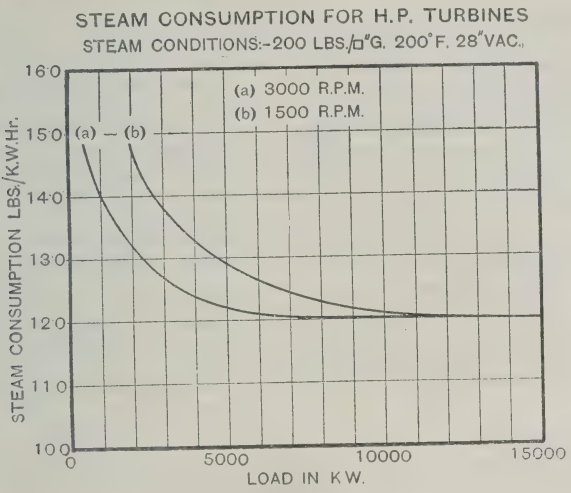


FIG. 4.

and 3,000 r.p.m. The former speed is the highest which can be used for 25-cycle generators, while the latter can be used for either 25 or 50 cycles. These curves illustrate the effect of speed

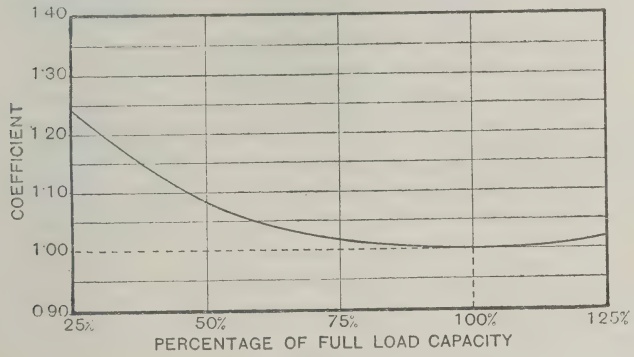
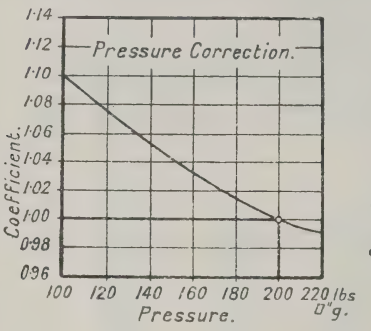


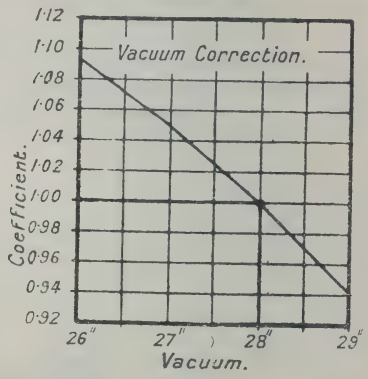
FIG. 5.

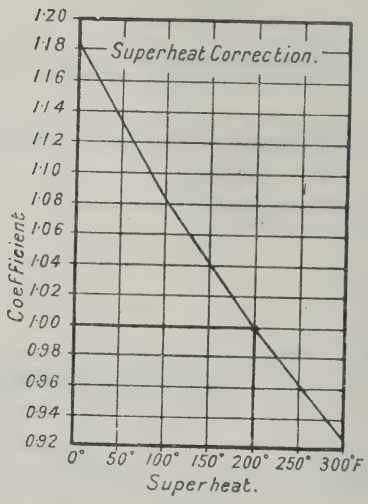
on efficiency. Fig. 5 enables the consumption at fractional loads to be estimated when the consumption at full load is known.

The steam consumption which can be obtained for other steam conditions can be approximately obtained by means of the correction curves shown in fig. 6.

STEAM CONSUMPTION CORRECTIONS FOR TURBINES DESIGNED FOR VARIOUS CONDITIONS.







STANDARD STEAM CONDITIONS:

200 lbs. per sq. in. g.

200° F.

28 in. Vac.

For other steam conditions, multiply the consumptions obtained from fig. 4 (page 1781) by the coefficients obtained from the curves.

FIG. 6.

IMPULSE TURBINES.

A pure impulse turbine is one in which the available heat energy of the steam is completely converted into kinetic energy in stationary nozzles.

The steam jet thus produced performs mechanical work by impinging on a rotating row of blades or successive rows of rotating and guide-blades placed immediately before the nozzles.

An impulse turbine in which the whole pressure drop available is utilised in a single set of nozzles is said to consist of one *pressure stage*. If the pressure drop is subdivided between two, three, or more successive sets of nozzles, it consists of two, three, or more pressure stages.

When the kinetic energy of the steam jet is utilised in one row of rotating blades, the turbine is said to have one *velocity stage* per pressure stage. If, after the first row of rotating blades, a guide blade is placed to reverse the direction of steam so as to impinge on a second row of rotating blades, the turbine is said to have two ' velocity stages,' and so on.

Since a minimum nozzle height will be fixed by practical considerations (not less than $\frac{1}{4}$ in. even in the smallest turbines), the arc over which the nozzles will extend depends upon the total nozzle mouth area required. In the high-pressure stages this may be less than the mean circumference of the blades. Such a turbine is said to be working with partial admission, and the arc over which the nozzles extend is termed the ' arc of admission.'

This arrangement lends itself especially well to conveniently and efficiently dealing with overloads, it being only necessary to increase the ' arc of admission ' by admitting steam to additional sets of nozzles.

In the same way it is possible to divide the total number of nozzles into blocks controlled by separate cutout valves. Thus one block of nozzles can be designed to give all loads from 0 to $\frac{1}{2}$ full load, while these together with a second block give $\frac{1}{2}$ to $\frac{3}{4}$ and so on.

By this means the loss of pressure through throttling at the governor valve is reduced to a minimum, and substantial economies can be effected on partial loads. The nozzle cutout valves may be operated automatically or by hand. Nozzle cutout governing is not possible with reaction turbines.

(a) Single Pressure Stage Turbine with a Single Velocity Stage.

The nozzles of all impulse turbines can be designed by the methods already indicated, while the work done and the efficiency can be obtained by drawing the triangles of velocity a_1 and a'_1 (see fig. 7). All velocities are in feet per second.

Let

W = total weight of steam through nozzles in lbs. per sec.; c_1 = absolute velocity leaving nozzles; U = peripheral velocity at mean diameter of blades; w_1 = relative velocity at entry

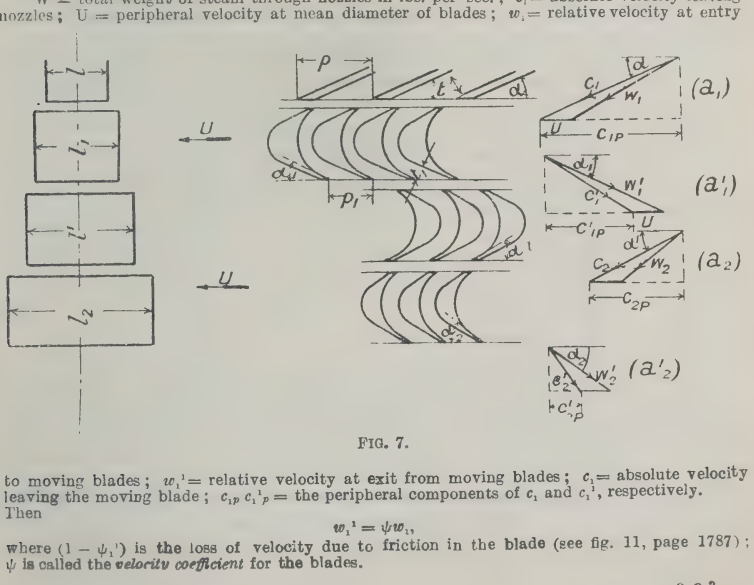


FIG. 7.

to moving blades; w_1' = relative velocity at exit from moving blades; c_1 = absolute velocity leaving the moving blade; c_{1p} $c_1'_p$ = the peripheral components of c_1 and c_1', respectively.

Then

$$w_1' = \psi w_1,$$

where $(1 - \psi_1')$ is the loss of velocity due to friction in the blade (see fig. 11, page 1787); ψ is called the *velocity coefficient* for the blades.

The total work done, or
The indicated work $= \frac{WU}{g}(e_{1p} + e_1{}'_p)$ ft.-lbs. per sec.

The blading efficiency $\eta_B = \frac{\text{(total work in ft.-lbs. per sec.)}}{W.\ J.\ g.\ h_a}$

If

l = the required nozzle height; l_1 = the required blade height;

then

$$l_1 = l \cdot \frac{c_1}{w_1} \cdot \frac{t}{t_1} \cdot \frac{p_1}{p}$$

The de Laval turbine, which is of this type, is made in sizes from 5 to 500 h.p., running at 30,000 r.p.m. in the smallest size, and 10,000 r.p.m. in the largest outputs.

The peripheral speeds are as high as 1,300 ft. per second, and the steam velocity at exit from the nozzles as high as 4,000 ft. per second.

It can be shown that the efficiency varies with $\frac{U}{C_o}$, where Co is the velocity theoretically obtained if $\phi = 1{\cdot}0$.

For a maximum efficiency, $\frac{U}{C_o}$ should be about equal to 0·5 (see fig. 9, page 1786).

(b) Single Pressure Stage. Two Velocity Stages.

In this case the work done in the first row of blades is obtained in exactly the same way as in case (*a*).

The guide-blade returns the steam jet towards the direction of U, the steam leaving it with an absolute velocity $e_2 = \psi_2 e_2{}'$.

The relative velocity of exit from the second rotating blade is $w_2{}' = \psi_2{}' w_2$, knowing which, the velocity triangles a_2, $a_2{}'$ can be drawn. In a_2 and $a_2{}'$ the same symbols are employed as in a_1 and $a_1{}'$, a suffix 2 denoting the guide-blade, while a dash denotes the second moving blade.

The total work per second is now

$$= \frac{WU}{g}\left\{(e_{1p} + e_1{}'_p) + (e_{2p} + e_2{}'_p)\right\} \text{ ft.-lbs. per sec.}$$

Work done in 1st and 2nd Rows of Velocity Wheels. 3,000 R.P.M.

Initial conditions :—200 lbs. per sq. in. g., 200° F. supf.; steam pressure in casing = 34 lbs. per sq. in. g.; temperature in casing = 410° F.

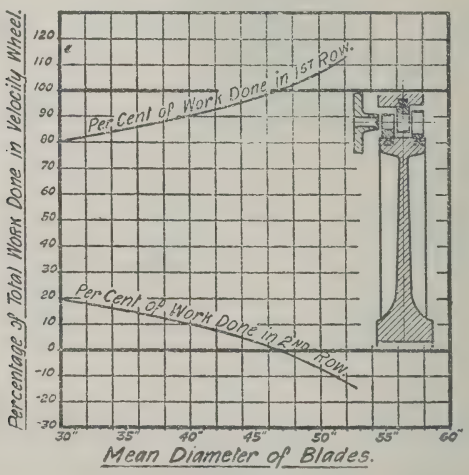


Fig. 8.

and

$$\eta_B = \frac{\text{(Total work in ft.-lbs. per sec.)}}{W \cdot J \cdot g \cdot h_a}$$

The height of the guide-blade $l^1 = l \cdot \frac{c_1}{c_a} \cdot \frac{t}{t^1} \cdot \frac{p^1}{p}$

The height of the second rotating blade $l_2 = l \cdot \frac{c_1}{w_2^1} \cdot \frac{t}{t_2} \cdot \frac{p_2}{p}$

As the mean peripheral velocity of the blades increases the proportion of the work done in the first row increases. This effect is illustrated in fig. 7, from which it will be seen that for the condition stated the second row does no positive work if the mean diameter of the blades exceeds 47½ ins.; it can therefore be dropped with advantage. For this reason two-row wheels are rarely used in modern high-speed machines.

(c) Single Pressure Stage. More than two Velocity Stages.

The solution of this case can be obtained from (*b*) in exactly the same way as (*b*) was obtained from (*a*).

Single-wheel Curtis stages belong to classes (*b*) and (*c*) and are very much employed for small high-speed turbines, where efficiency is not the first consideration.

The peripheral speeds for Curtis wheels are as high as 600 ft. per second, and the steam velocity 2,500 ft. per second.

For a maximum efficiency $\frac{U}{C}$ should be about 0·23 for two rows of moving blades, and 0·15 for three rows (see fig. 9, page 1786).

(d) Multi-Pressure Stage Turbine with Single Velocity Stages.

In order that reasonable efficiencies may be reached with practicable peripheral speeds, it becomes necessary to limit the heat drop utilised in each set of nozzles to some value considerably less than the total available between the conditions before the turbine stop valve and those present in the exhaust.

Up to a certain point, which is rarely reached in the conditions of actual design, the efficiency of each stage increases with the value of the ratio $\frac{U}{C_o}$ (see fig. 9, page 1786).

When the value of $\frac{U}{C_o}$ is known, which corresponds to the desired turbine efficiency, the value of C_o and hence of h_a the heat drop to be utilised per stage can be calculated, since U will be fixed by mechanical considerations.

Then for an equal distribution of energy over the pressure stages,

If H_a = the total heat drop available for the whole turbine,

$$\text{The required number of stages } n = \epsilon \cdot \frac{H_a}{h_a}$$

where ϵ is the '*reheat factor*' which depends on the amount of friction and eddy losses, the number of stages, the stage efficiency, etc., and varies between 1·03 and 1·07.

It is usually the practice to give the first stage a larger heat drop than the remainder, thus reducing the total number of wheels required, and having the considerable advantage of reducing the maximum temperature and pressure to which the turbine casing is subjected.

Having apportioned the heat drops, the steam conditions at each stage can be found—most conveniently by the Mollier diagram—and each stage can be designed as indicated under case (*a*).

The Rateau and Zoelly turbines are of this type.

In practice U varies from 350–600 ft. and the steam velocities from 800–1,600 ft. per second.

(e) Multi-Pressure Stage Turbines with Multiple Velocity Stages.

In precisely the same way as has been explained under (*d*), two- or three-row velocity wheel. may be employed in the individual pressure stages.

The heat drop which can be advantageously used per pressure stage is then greater than in the case where a single row of blades is used, and thus the required number of pressure stages is reduced, but the efficiency is somewhat decreased, as may be deduced from fig. 9.

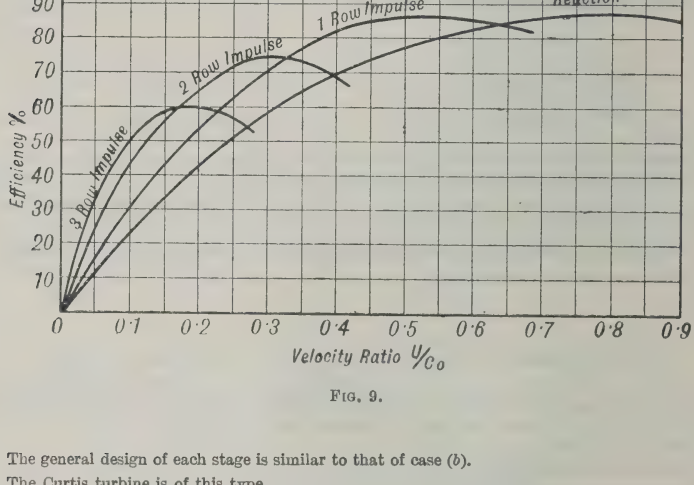


FIG. 9.

The general design of each stage is similar to that of case (*b*).

The Curtis turbine is of this type.

(f) Combination Impulse Turbines.

It has become a common practice to construct pressure-compounded impulse turbines of type (*d*) with a Curtis wheel having two rows of blades in the first stage. By this means a large heat drop can be employed in the first stage with reasonable efficiency and with the consequent advantages already mentioned.

The Curtis-Rateau and Curtis-Zoelly turbines are of this type.

CARRY-OVER ENERGY.

In pressure-compounded impulse turbines, the kinetic energy of the steam leaving the last blade of any pressure stage—except the last—is not entirely lost; part of it at least being recovered in the succeeding stage.

The total energy available in the succeeding stage is then greater than the adiabatic heat drop by an amount equal to $\frac{\alpha c^2}{2g}$, where c is the absolute velocity of exit from the velocity blades and α is a constant.

For full admission, Prof. Rateau has suggested the value $\alpha = \frac{2}{3}$, while for partial admission the value $\alpha = \frac{1}{4}$ may be taken.

THE VALUE OF ψ, THE VELOCITY COEFFICIENT FOR BLADES.

The value of ψ has been the subject of researches by Prof. Rateau* and Dr. N. Briling.†

Prof. Rateau found that ψ was a characteristic of each type of blade depending on their size and spacing and slightly increasing with velocity.

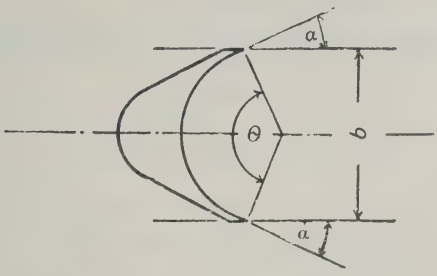
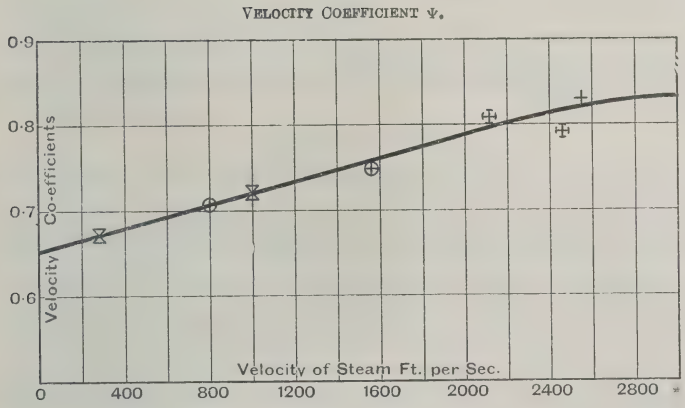


FIG. 10.



⊕ *Rateau. (Engineering 10/12/09) Mean Values.*

⊕ *Values deduced from Rateau's experiments on Bucket Reaction; Congrès International de Mécanique Appliquée 1900.*

⧖ *Briling's Experiments with 20 m/m Buckets (Engineering 29/4/10).*

\+ *Stodola "Steam Turbines" deduced from Delaporte's Test of a de Laval Steam Turbine.*

FIG. 11.

Dr. Briling, whose experiments covered a wider field, found that ψ increased with velocity, decreased with the width of the bucket, and appeared to be principally dependent upon the angle through which the steam was diverted in the blade.

* *Engineering*, December 10, 1909. † *Z. V. d. I.*, January 1906.

Briling deduced the following formula from his experiments :

$$\psi = \psi_0 - 0{\cdot}000432\,\theta^{\frac{4}{3}}$$

where θ = total angle in degrees through which steam is turned in the blade (see fig. 10, p. 1787),

ψ_0 = 0·90–0·97 for blades with rounded inlet edges.
= 0·95–0·99 for blades with sharp edges.

Mr. H. M. Martin* has given a useful diagram combining Rateau and Briling's results with others deduced from various tests, which is reproduced in fig. 10; from this the following expression can be deduced:

$$\psi = 0{\cdot}65 + \frac{w}{15{,}000}$$

w being the velocity of the steam relatively to the blade in ft. per sec.

PITCH OF IMPULSE BLADES.

The best value of the pitch p has been determined experimentally by Dr. Briling † as being connected with the blade angle $\alpha°$—which he takes as being the same at the inlet and outlet—and the blade width by the following relation:

$$p = \frac{r}{2 \sin \alpha°}$$

$$r = \frac{b}{2 \cos \alpha°}$$

Briling is of the opinion that 30° is the most suitable angle for impulse blading.

In practice, the blade angle varies from 20° to 50°, the largest angles being used at the exhaust end of a turbine when it becomes difficult to provide sufficient area. The pitch varies from 0·5 to 0·75 of the blade width.

REACTION TURBINES.

In a reaction turbine the available heat energy of the steam is converted into kinetic energy both in the stationary and the moving blades.

In the most usual case—*considering a stage as consisting of a fixed blade and the following moving blade*—one-half of the available heat drop per stage is utilised in the fixed blade and the remainder in the moving blade. The turbine is then said to have a half degree of reaction.

Thus, both the fixed and moving blades perform the function of a nozzle in precisely the same way, and are identical in form and thermodynamic purpose.

Since there is a heat drop in the moving blades, there must also be a pressure drop across them. This is a characteristic of the reaction turbine as compared with the impulse turbine.

A second characteristic lies in the fact that the arc of admission to the first stage of a reaction turbine must extend around the complete circumference. For this reason nozzle cutout governing is not possible, and overloads can only be dealt with by admitting steam at some intermediate point in the expansion.

The steam enters the moving blades with a relative velocity approximately in an axial direction, and leaves them at a higher velocity in a direction inclined to the plane of rotation, the blades being impelled forward by the reaction of the leaving steam.

Reaction Blading with Half Degree of Reaction.

In this case since the outlet angles of the fixed and moving blades are the same, and the heat drop utilised in both is the same, the velocity triangles at inlet to and exit from the moving blade are identical (see fig. 12, p. 1789).

* *Steam Turbines—their Theory and Construction*, H. M. Martin.
† *Z. V. d. I.*, January 1906.

Hence

$$c_1 = w_1' \; ; \; c_1' = w_1.$$

If h = heat drop in any stage (one fixed and one moving blade) and with the remaining symbols as were used when considering impulse turbines—

$$c_1 = \frac{1}{2\,U\cos\alpha}\,(25{,}050\,\zeta\,h + U^2);$$

where $(1 - \zeta)$ is the energy loss in the conversion of heat to velocity.

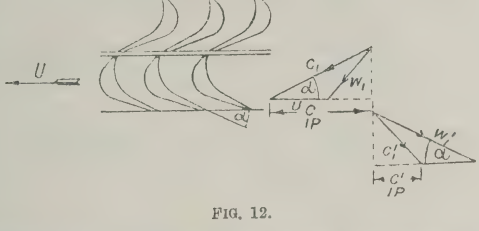


FIG. 12.

Capt. Riall Sankey gave 0·8 as the value of ζ, whereas the statement of Sir Chas. Parsons that the blading efficiency in the low-pressure end of the turbines of the 'Mauretania' reached the value of 85 per cent., shows that ζ may be as high as 0·86 under those conditions, while the efficiencies reported to have been obtained with the Ljungström turbine of quite small capacities could only be reached if ζ was as much as 0·85.

U and α being known, the velocity triangles can be drawn.

The work done per stage

$$= W\,\frac{(c_1^2 - c_1'^2)}{2g}, \text{ or, } = W\,\frac{(2k\cos\alpha - 1)}{g}v^2, \text{ where } k = \frac{c_1}{U}.$$

The blading efficiency

$$\eta_B = \frac{c_1^2 - (c_1'^2)}{2g\,J\,.\,h}, \text{ or, } = U^2\frac{(2k\cos\alpha - 1)}{J\,.\,g\;\;h}.$$

The required blade length can be obtained from the relation

$$\pi\,.\,D_m\,.\,l\,.\,q = \frac{W\,.\,v_2}{W_1'}.$$

where D_m = mean diameter of annulus in feet; l = radial length of annulus in feet = (blade height) + (tip clearance); v_2 = the specific volume after the moving blade in cubic feet per lb.; q = annulus area factor.

At any stage the 'annulus area factor' is the ratio of the available area of the channels at exit from the blades in a direction at right angles to α, to the area of the annulus between the drum and the casing at that stage.

The properties of Parsons blading, with the exception of q and α, were published by M. Lelong,* and will be found in Table IV.

As in the case of impulse blading, the blading efficiency varies with $\frac{U}{c_1}$ and reaches a maximum when $\frac{U}{c_1} = 0{\cdot}945$ if $\alpha = 19°$, the normal angle for all but the last stages of reaction turbine. Under this condition U becomes equal to $c_1 \cos\alpha$. Such a high value is not practicable in ordinary working conditions, nor is it necessary for the realisation of high efficiency, because the curve connecting $\frac{U}{c_1}$ and η_B is very flat in the region of its maximum.

Fig. 13 (p. 1791), which gives this relation for $\zeta = 0{\cdot}8$, is due to Stodola.†

The efficiency for changes of ζ within the limits actually to be met with can be obtained by increasing the efficiency as obtained from fig. 9 by the same percentage as ζ exceeds 0·8.

In electrical work $\frac{U}{c_1}$ varies from 0·5 to 0·8.

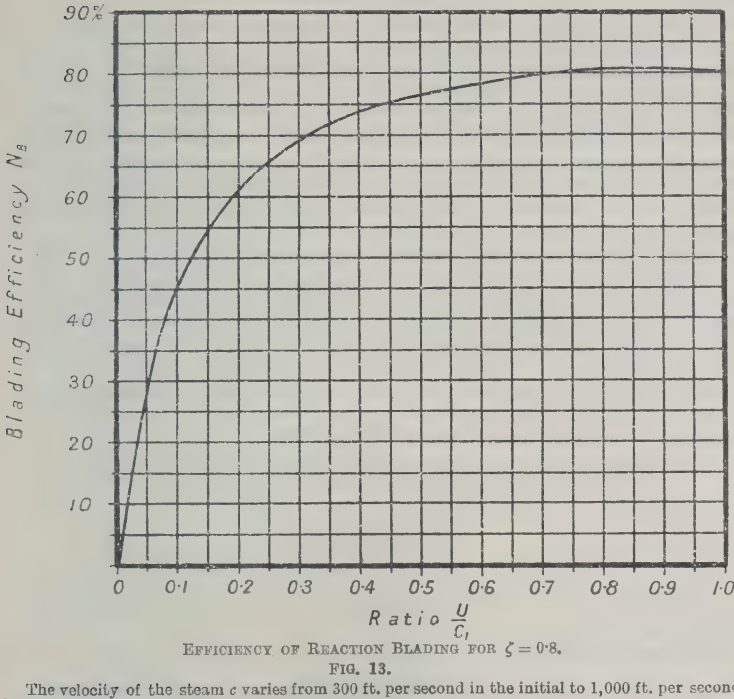
The peripheral velocity of forged steel drums may be as high as 400 ft. per second, and even higher speeds are possible with solid drums, or drums made from special steels of high ultimate strength.

* *Turbines à Vapeur Marines*, R. Lelong; and *Engineering*, March 1, 1912.
† *Die Dampfturbinen*, 4th Ed., A. Stodola.

TABLE IV.—DATA RELATING TO BLADES AND CAULKING PIECES FOR PARSONS TURBINES.

Blades.						Caulking Pieces.								
						Casing.					Drum.			
Section Number.	Class of Blade.	Exit Angle of Blade α. In deg.	Annulus Area Factor q.	Weight per foot run.	Usual Limits of Length.	Section Number.	Number Required per in. run.	Weight per foot run.	Grooves. Width.	Grooves. Depth.	Section Number.	Number Required per in. run.	Weight per foot run.	Grooves. Width.	Grooves. Depth.
									In.	In.				In.	In.
120B	Normal	20	0·3	0·041	Up to $\frac{3}{4}$ ins.	120C	5·8	0·106	0·243	$\frac{1}{4}$	120C	5·8	0·106	0·243	$\frac{1}{4}$
130B	Normal	20	0·3	0·089	$\frac{3}{4}$ in. to 4 ,,	130C	3·7	0·264	0·364	$\frac{1}{4}$	130S	5·2	0·159	$\frac{23}{64}$	$\frac{1}{4}$
130B	Semi-Wing	35	0·53	0·089	$\frac{3}{4}$,, ,, 4 ,,	131C	3·3	0·315	0·415	$\frac{1}{4}$	131S	3·7	0·300	$\frac{13}{32}$	$\frac{1}{4}$
132B	Wing	45	0·68	0·077	$\frac{3}{4}$,, ,, 4 ,,	132C	3·2	0·400	0·406	$\frac{1}{4}$	132S	3·6	0·320	0·400	$\frac{1}{4}$
240B	Normal	20	0·3	0·180	4 ,, ,, 8 ,,	240C	2·75	0·450	0·510	$\frac{29}{64}$	240S	3·4	0·400	0·510	$\frac{3}{8}$
240B	Semi-Wing	35	0·53	0·180	4 ,, ,, 8 ,,	241C	2·54	0·643	0·540	$\frac{9}{32}$	241S	3·2	0·450	0·540	$\frac{3}{8}$
250B	Normal	20	0·3	0·246	8 ,, ,, 12 ,,	250C	2·2	0·783	0·610	$\frac{39}{64}$	250S	2·7	0·600	0·610	$\frac{1}{2}$
250B	Semi-Wing	35	0·53	0·246	8 ,, ,, 12 ,,	251C	2·1	0·917	0·661	$\frac{21}{32}$	251S	2·6	0·670	0·661	$\frac{1}{2}$
252B	Wing	45	0·68	0·237	8 ,, ,, 12 ,,	252C	1·96	1·080	0·716	$\frac{23}{32}$	252S	2·44	0·79	0·716	$\frac{1}{2}$

For Double Wing Blades $\alpha = 60°$; $q = 0·85$ in.



EFFICIENCY OF REACTION BLADING FOR $\zeta = 0{\cdot}8$.

FIG. 13.

The velocity of the steam c varies from 300 ft. per second in the initial to 1,000 ft. per second in the final stages.

In marine work $\frac{U}{c_1}$ varies from 0·3 to 0·45, and for the high-pressure stages the peripheral velocity varies from 80 to 120 ft. per second.

REACTION TURBINE DESIGN.

The blades of Parsons turbines are not usually increased in height from row to row, but are arranged in groups having the same height and the same mean diameter. The angles of the blades of any one group are usually the same, but in some cases they are gradually increased so as to obtain the correct progression of areas for constant steam velocity relatively to the blade.

The blades of such turbines for land purposes are generally placed on three-drum diameters of increasing size, referred to as the H.P., I.P. and L.P. expansions.

The relative distribution of heat drop between the different expansions and their relative geometric properties are given in the following table:—

Expansion.	H.P.	I.P.	L.P.	H.P.	I.P.	L.P.
Adiabatic heat drop used . .	1	1	2	1	1	1
Number of groups of blades .	1	1	2	1	1	1
Mean diameter of annulus . .	1	$\sqrt{2}$	2	1	$\sqrt{2}$	2
No. of rows of blades per group .	1	$\frac{1}{2}$	$\frac{1}{4}$	1	$\frac{1}{2}$	$\frac{1}{4}$
No. of rows of blades per expansion	1	$\frac{1}{2}$	$\frac{1}{2}$	1	$\frac{1}{2}$	$\frac{1}{4}$

The first of these arrangements is that most commonly adopted.

In electrical work there are from two to three groups of blades in the H.P. expansion.

In any given turbine and for groups of blades with the blade angle the same, the ratio $\frac{D^2 z}{v}$

should be constant, where D = mean diameter of the blades, l = their length, and v = the mean specific volume of the steam in the group.

The last three rings on the L.P. expansion were formerly made the same height, but with increasing discharge angle, the latter corresponding respectively to 'normal,' 'semi-wing' and 'wing.' By this means the final ring (*i.e.* wing blades) has double the discharge area of the ring of 'normal' blades of the same height, but without any increase in the centrifugal stresses. The practicable height ratio of $\frac{\text{blade height}}{\text{rotor diameter}}$ was $\frac{1}{4}$ with blades cut from rolled parallel strip. In modern practice a still larger height ratio ($\frac{1}{3}$) is obtained, with a view to obtaining still larger discharge area in the final ring, by the use of twisted blades which are also, if necessary (on account of centrifugal stresses), tapered from root to tip. In these blades the discharge opening ratio $\frac{o}{p}$ is kept constant from root to tip and equal to about 0·71 (*i.e.* wing), but the discharge angle is progressively varied by twisting. The guide ring is made complementary to the final rotating ring, being twisted, but not tapered.

In order to reduce the unsupported length between bearings, and to enable the tip clearance to be kept small, it is becoming the practice in turbines of 5,000 kw. and over to use two cylinders. The first cylinder contains the H.P. expansion only and can be made of cast steel, very fine clearances being possible by eliminating the danger due to the growth of cast iron.

e_1, for any particular stage, can be calculated from the formula already given when the heat drop h is known.

The blading efficiency throughout the turbine can be found from fig. 13 when the ratio $\frac{U}{c}$ is known.

For groups of blades having the same angles and height:

if

n = number of blades in the group considered; R = ratio of expansion over the group;

H_a = adiabatic heat drop for the group,

then the heat drop used in the first stage

$$h_1 = \frac{(1+e)H}{\left[0{\cdot}07 + \left(\frac{R^2-1}{R^{\frac{2}{N}}-1}\right)\right]}$$

The heat drops for the other stages of the groups form with h_1 a geometric progression having $R^{\frac{2}{N}}$ as a common ratio.

In most cases it will be sufficient to calculate the first and last stages of any one group.

Disc and Drum Type.

Just as in the development of the pure impulse turbine combinations of the simple types have been evolved, so the disc and drum type has been introduced into the sphere of the reaction turbine.

In this case the high pressure and less efficient part of the reaction turbine is replaced by a Curtis wheel having as a rule two rows of blades. This design possesses the advantages already mentioned in connection with combination impulse turbines, and in addition it results in a considerably shorter and stiffer rotor than in the pure Parsons machine.

TURBINE LOSSES.

Diaphragm Gland Loss.

The diaphragms of a pressure-compounded impulse turbine must obviously be provided with a clearance between themselves and the shaft, thus providing a leakage area through which steam can pass.

The diaphragm gland is usually made either as a labyrinth packing or as a closely fitting white metal bush.

Although the clearances may originally be very small they are greatly increased by whipping of the shaft, or even by the static deflection.

The percentage leakage can be estimated as being equal to the ratio of the clearance area to the nozzle area in any diaphragm.

In order to reduce this loss, some designers employ carbon packings held together by garter springs in the diaphragms.

The diaphragm clearance is usually from $\frac{10}{1000}$ ins. to $\frac{20}{1000}$ ins. radially.

PACKINGS

Labyrinth Packings.

The glands at the cylinder ends are usually of the labyrinth packing type.

In impulse turbines the labyrinth is invariably of the radial clearance type, while in Parsons turbines the packing at the thrust block end is of the axial clearance type, the other being of the radial clearance type.

Mr. H. M. Martin* has deduced the following formulæ:

$$\text{Weight of leakage steam} = 68 \cdot A \sqrt{\frac{p_o\left(1-\frac{1}{r}\right)}{v_o(N + log_\epsilon r)}} \text{ lbs. per sec.,}$$

where

p_o = pressure before the gland in lbs. per sq. in.; v_o = specific volume of steam in cu. ft. per lb.; r = ratio of expansion through gland; N = number of labyrinths; A = sectional area of leakage path in sq. ft.

If the velocity with which the steam flows out of the last labyrinth is greater than sound velocity, this formula should be amended, but the correction is small, and as it is in the nature of a reduction in the weight of the leakage steam, it can safely be neglected where N is greater than 8, which is the usual case in practice.

N varies between wide limits and may be from 8 in the case of low-pressure glands of impulse turbines to 30 for the high-pressure glands of Parsons turbines.

The clearance usually allowed varies from $\frac{10}{1000}$ in. to $\frac{30}{1000}$ in., but this may increase in actual working in the same manner as the diaphragm clearances.

In order to reduce the loss of turbine efficiency due to gland loss, which might otherwise be considerable, the high-pressure packing is usually connected at one or two points in its length to some intermediate stage in the turbine.

The leakage over each of these sections can be calculated by the formulæ already given, r now being the ratio of expansion over any section, and p_o, v_o the conditions before that section.

Except in those cases where a water gland is used, the gland at the exhaust end is sealed by steam at a higher pressure than atmospheric, taken from a stage in the turbine, and in order to ensure that air is not leaking into the exhaust, the pressure is adjusted until a small quantity of steam is seen escaping into the engine-room.

The cylinder glands of the A.E.G., Zoelly, and Curtis turbines are of the carbon type, for which a considerable reduction in leakage is claimed.

The advantages of these are questionable since they cannot be renewed or adjusted without opening the turbine, and in some designs it is even necessary to strip the rotor.

The carbon rings which are in segments are kept together and caused to bear on the shaft by garter springs.

'BRITIMP' PACKINGS.

(*British Metallic Packings Co.* (1929) *Ltd., 7 Princes Street, Westminster, S.W.* 1.)

Each packing of this brand consists of two complete rings of carbon, X, X (fig. 14). Each ring is in three sections. The outer circumferences of the rings are coned in opposite directions, and the coned surfaces are shrouded by a light metal ring Y.

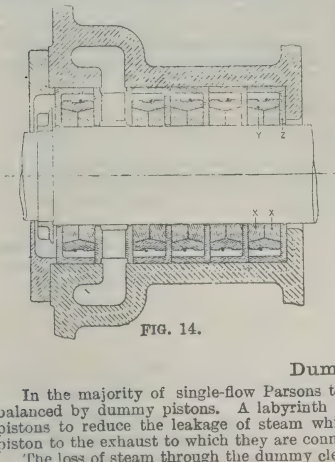


FIG. 14.

This ring is kept in contact with the carbon rings by a coiled wire and spring. The spring is so formed that the possibilities of breakage are very remote.

Each packing unit is enclosed in a chamber ring Z of L section. The relative positions are made definite by stops.

The joints of the sections of all the rings are staggered to ensure steam tightness.

The side pressures serve two purposes—viz. they produce surface friction which supports the weight of the packing units, and they also ensure steam-tight joints between the packings and walls of the chambers. As the shaft is relieved of the weight of the packing it is subjected only to uniform radial contact pressures, with the result that the wear on both the shaft and packings is reduced to a minimum. Means for adjusting the contact pressures are provided on the coiled wire.

Dummy Loss.

In the majority of single-flow Parsons turbines the axial thrust on the blades and drum is balanced by dummy pistons. A labyrinth packing is formed on the periphery of the dummy pistons to reduce the leakage of steam which would otherwise take place across the dummy piston to the exhaust to which they are connected on one side.

The loss of steam through the dummy clearances can be calculated in precisely the same way as the loss through a labyrinth packing.

The double-flow type of turbine obviating the necessity of balance pistons eliminates this loss.

* *Steam Turbines—their Theory and Construction*, H. M. Martin.

Disc Friction.

A considerable amount of work may be absorbed in rotating the discs of impulse turbines in the dense steam present in the first stages of a pressure-compounded (disc) turbine.

Professor Stodola * has given the following formula for the disc friction loss :

$$\text{B.H.P.} = \frac{6 \cdot 1}{10^8} D^2 U^3 \gamma,$$

where,

D = mean diameter of blades in feet ; U = mean peripheral velocity feet per sec. ; γ = density of steam in lbs. per cu. ft.

The experiments upon which this formula is based were performed either in air, superheated steam, or in steam whose exact condition as regards wetness was unknown.

Professor Lewicki has demonstrated that the combined work of disc friction and windage falls off with superheat, some of his experiments showing a reduction of 10 per cent. in the work done for an increase of superheat of 100° C. As the total loss due to disc friction is usually of the order of a few per cent. of the output of the turbine, the reduction with superheat is of little importance.

On the other hand, it seems probable that the disc friction would increase rapidly with the wetness of the steam, but there are at present no data for estimating this increase. For these reasons, the disc friction loss for any given turbine cannot be calculated with certainty.

Windage.

In addition to the disc friction, loss results from the fanning action of the blades as they rotate.

The laws governing this action cannot be regarded as established, the range of experiments has been limited, and results by the same experimenters are discordant. Professor Stodola* gives the following formula :

$$\text{B.H.P.} = \frac{4 \cdot 6}{10^7} \cdot D \cdot l^{1 \cdot 5} \cdot U^3 \cdot \gamma,$$

where,

l = blade length in inches ; D = mean diameter of blades in ft. ; U = mean peripheral velocity in ft. per sec. ; γ = density of steam in lbs. per cubic ft.

Mr. W. Kerr carried out some careful tests on this subject, and by comparing his own results with those of Professor Stodola and Dr. Lasche, suggests that for wheels carrying two rows of blades the work calculated by the above formula must be multiplied by 1·23, for three rows by 1·8, and for four rows by 2·9.†

When mixed pressure turbines are working with low-pressure steam the high-pressure end is rotating idly in an atmosphere of steam, and with disc turbines the work done in disc friction and windage is not negligible. With reaction turbines a windage loss alone is present, and this is mainly due to the fanning action of reaction blading. Sir C. A. Parsons has stated that in the case of a reverse element of a marine turbine rotating in a vacuum the loss is of the order of 1 per cent., so that the loss due to the rotation of the high-pressure element of a mixed-pressure Parsons turbine rotating in steam twelve times as dense as a 28-in. vacuum may be considerable.

Water Glands.

An interesting feature in the development of turbine design consists in the introduction of the water gland.

In this type, an air-seal is produced by a paddle-wheel acting as a centrifugal pump. The pressure difference over the gland is balanced by the centrifugal head produced in the water as it is whirled around covering areas of different internal radii on the two faces of the paddle.

The ribs on the sides of the paddle should be $\frac{1}{4}$ in. deep.

If P_1 (fig. 15) = the pressure on the high-pressure side in lbs. per square ft. ; P_2 = pressure on the low-pressure side in lbs. per square ft. ; R_1 = inner radius in ft. of the whirling water on the H.P. face ; R_2 = inner radius in ft. of the whirling water on the L.P. face ; ρ = density of water = 62·3 ; ω = angular velocity of paddle in radians per second.

Then

$$P_2 - P_1 = \frac{\omega^2 \rho}{240\,A} (R_1^2 - R_2^2) \text{ and } P_3 - P_1 = \frac{\omega^2 \rho}{240\,A} (R_2^2 - R_1^2)$$

The value of A varies from 0·9 to 1·10″.

Although the water gland when first introduced gave rise to troubles, it has proved successful when used in conjunction with a labyrinth packing, for the purpose of preventing air leaking into the turbine casing.

Since a sufficient centrifugal head to form a seal is not developed until the turbine is running at some speed near the normal, it is necessary to steam-seal the glands as the turbine is started up.

The gland must be supplied with a constant stream of water to make up the loss which takes

---

* *Die Dampfturbinen*, 4th Ed., Professor A. Stodola.

† *Engineering*, August 22, 1913.

place by evaporation on the vacuum side of the seal. The water is admitted at the periphery of the gland, and should be supplied from a head of about 15 ft. This will, of course, be equal to $(P_s - P_1)$.

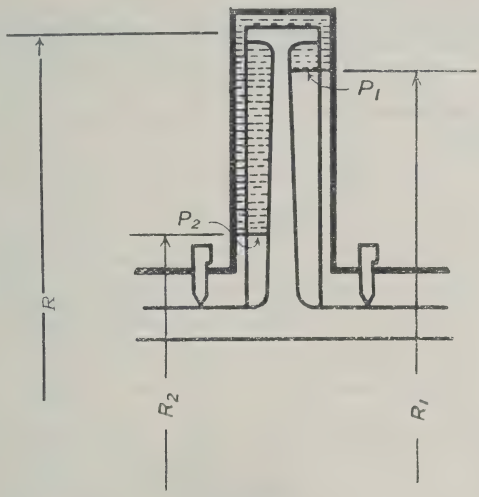


FIG. 15.

A water gland cannot be used to seal against steam at a pressure above atmospheric, because the water boils and the seal is destroyed.

Tip Clearance Loss.

As has already been pointed out, there is always a difference of pressure over both the fixed and rotating blades of a reaction turbine. In consequence of this there is always a leakage of steam

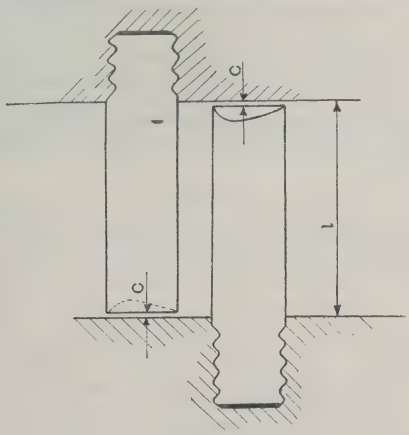


FIG. 16.

at the tips of the blades through the working clearances which have to be provided. In parallel flow reaction turbines there are two principal ways of providing these clearances :

(1) By means of radial clearance reaction blading (see fig. 16).

(2) By means of axial clearance or ' end tightened ' reaction blading (see fig. 17).

The clearance C usually allowed in design varies approximately between the following limits for radial blading, all dimensions being in inches :

$$c = 0{\cdot}0008\,d \text{ to } c = 0{\cdot}0008\,d + 0{\cdot}004\,l + 0{\cdot}015,$$

where,

d = mean diameter of blades

l = blade height.

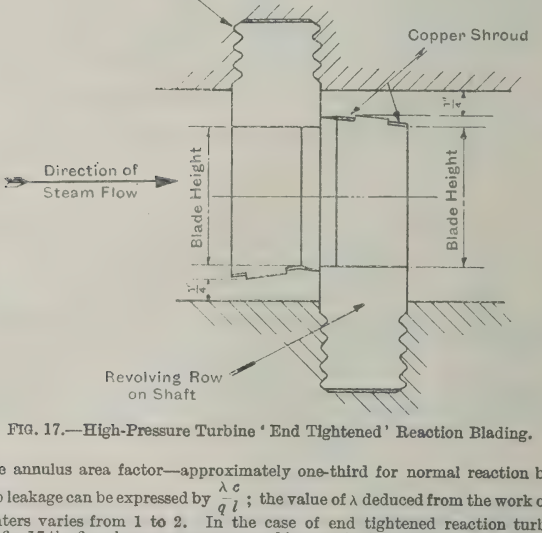


FIG. 17.—High-Pressure Turbine ' End Tightened ' Reaction Blading.

If q is the annulus area factor—approximately one-third for normal reaction blading—loss due to tip leakage can be expressed by $\frac{\lambda c}{q l}$; the value of λ deduced from the work of a number of experimenters varies from 1 to 2. In the case of end tightened reaction turbine blading illustrated in fig. 17 the fine clearances are arranged in an axial instead of a radial direction. This type of blading is usually confined to the high-pressure end of the turbine, and an adjustable thrust block is provided to enable this clearance to be adjusted.

Bearings.

l = the length of a bearing in inches ; d = diameter of journal in inches ; p = pressure on bearing per square inch projected area ; N = speed of turbine in R.P.M. ; μ = coefficient of friction ; v = peripheral speed of journal in feet per second.

Then the work done in overcoming the bearing friction is

$$\text{Friction work} = \frac{\mu \pi l d^2 p \text{N}}{720} \text{ ft.-lbs. per sec.}$$

The heat equivalent of this work must be carried away from each bearing if they are to remain at a constant temperature.

According to Lasche's experiments, if $t°$ F. is the final steady temperature of the bearing, the coefficient of friction is given by the following relation :

$$\mu \,.\, p \,.\, (t° - 32°) = 51{\cdot}2.$$

For turbine bearings $t°$ should be from 120° F. to 150° F.

Assuming a temperature of 140° F., which is about an average value, these expressions can be reduced to:

$$\text{Work done in friction} = 3{\cdot}75\left(\frac{l}{10}\right)\left(\frac{d^2}{100}\right)\left(\frac{N}{1{,}000}\right)\text{b.h.p.}$$

With average bearings bedded in the normal way, it is found that the work done exceeds the amount just given, being usually about twice as great.

One of the striking features of the recent developments in steam turbines has been the successive increases in the specific loadings and rubbing speeds on bearings.

This rating up of bearings has been accompanied by uniform success in all cases where the natural requirements of good lubrication have been safeguarded.

The oil supplied to the bearing has two functions to perform: it must sustain the oil film, and it must flush the bearing copiously to carry away the heat generated in shearing the oil film.

The quantity of oil required by the latter consideration is by far the greater of the two; the necessary amount of oil can readily be estimated from the relation that for ordinary turbine oils—

1·0 b.h.p. in oil friction raises 1·0 gallon of oil per minute by 10° F.

In the average case the temperature of the oil passing through the bearings is such that its temperature rise in passing through the bearing is from 20° to 40° F.; therefore, for 40° F. rise in temperature the oil quantity which should be supplied to a bearing is given by oil which

$$\text{should be supplied} = 2 \times \frac{3{\cdot}75ld^2N}{40 \times 10^6}\text{ gals. per min., or, approximately, } = \frac{2ld^2N}{10^6}\text{ gals. per min.}$$

This represents the minimum quantity required, and may be increased up to double this amount with advantage.

The lengths of turbine bearings vary from $l = 2{\cdot}5d$ to $l = d$, the former value being usual for lightly loaded slow-speed bearings of older design, and the latter being approached with modern high-speed high-rated bearings.

At the present state of the art it is impossible to state the upper limits for permissible loading and rubbing velocity. Bearings are now quite common with loadings up to 110 lbs. per sq. in. and with rubbing velocities up to 130 ft. per sec.

According to modern ideas and practice, the old rules of the form pv = constant are quite unsound—in fact, when p is increased, v can and should be increased.

After leaving the bearings the oil is passed through an oil cooler, where it is cooled to some temperature usually below 110° F., although, of course, the actual temperature depends upon that of the cooling water available.

Turbine bearings give remarkably little trouble, provided the oil is kept clean and there is no tendency to emulsify.

In very small turbines the bearings are sometimes ring-lubricated, and in such cases the peripheral speed should not exceed 40 ft. per sec.; at higher speeds the oil tends to be thrown to the outside of the ring by centrifugal force.

Even in very small turbines there is now a tendency to fit a forced lubrication system provided with an oil cooler.

Centrifugal Stresses in Drums and Discs.

Stresses in thin Ring or Drum.

$$\text{Mean tangential stress} = 2\,(\text{D})^2\left(\frac{\text{N}}{1000}\right)^2\text{lbs./sq. ins.}$$

Solid Parallel-sided Disc.

If U_2 = peripheral velocity at the outside diam. in ft./sec.,

then radial stress at any radius x is $\sigma_r = \frac{4{\cdot}36U_2^2}{100}\left[1 - \frac{x^2}{x_2^2}\right]$ lbs./sq. in.

and the tangential stress $\sigma_t = \frac{U_2^2}{100}\left[4{\cdot}36 - 2{\cdot}51\frac{x^2}{x_2^2}\right]$ lbs./sq. ins.

The maximum stress at the centre is given by

$$\sigma_{r\,max} = \sigma_{t\,max} = \frac{\cdot 36\,U_2^2}{100} \text{ lbs./sq. ins.}$$

If the diameter of the disc = D inches, this can be written

$$\sigma_{t\,max} = 0{\cdot}835\,(D)^2 \left(\frac{N}{1000}\right)^2 \text{ lbs./sq. ins.}$$

Parallel-sided Disc with Hole in Centre.

At any radius x

Radial stress $\sigma_r = \frac{4{\cdot}36}{100} U_2^2 \left[1 + \frac{(x_1^2 - x^2)}{x_2^2} - \frac{x_1^2}{x^2}\right]$ lbs./sq. ins.

Tangential stress $\sigma_t = \frac{U_2^2}{100}\left[4{\cdot}36\left(1 + \frac{x_1^2}{x_2^2} + \frac{x_1^2}{x^2}\right) - 2{\cdot}51\,\frac{x^2}{x_2^2}\right]$ lbs./sq. ins.

Dimensional symbols as shown in fig. 18.

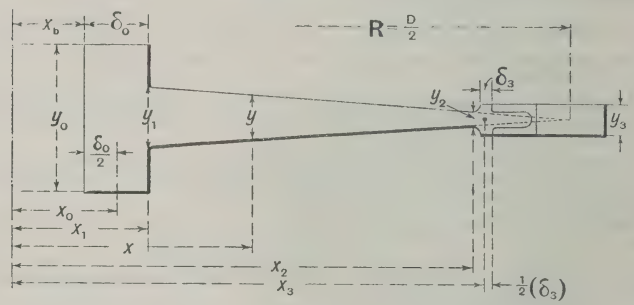


FIG. 18.

Disc with Hyperbolic Profile.

If the profile can be expressed as a hyperbolic function of the form $y = \frac{(\text{constant})}{x^\alpha}$, the following analysis can be used :

Let

ω = angular velocity in radians/sec. = $\frac{N \times 2\pi}{60}$;

E = Young's modulus of elasticity = 30×10^6 lbs./sq. in. for steel :

ν = Poisson's ratio = 0·3 for steel;

σ_2 = load on outer edge of disc due to blade loading in lbs./sq. in. :

σ_0 = stress at bore due to pinch fit;

μ = mass per unit volume ;

then $\alpha = \frac{\log\left(\frac{y_2}{y_1}\right)}{\log\left(\frac{x_1}{x_2}\right)}$.

If $\psi_1 = \frac{\alpha}{2} + \sqrt{\frac{\alpha^2}{4} + \alpha\nu + 1}$, $\psi_2 = \frac{\alpha}{2} - \sqrt{\frac{\alpha^2}{4} + \alpha\nu + 1}$.

and $$a = -\frac{(1-\nu^2)\mu\omega^2}{E[8-(3+\nu)\alpha]}$$

then radial stress at any point

$$\sigma_r = \frac{E}{(1-\nu^2)}\left[(3+\nu)\,a\,x^2 + (\psi_1+\nu)b_1 x^{\psi_1 - 1} + (\psi_2+\nu)b_2 x^{\psi_2 - 1}\right] \quad \ldots.(1)$$

and tangential stress

$$\sigma_t = \frac{E}{(1-\nu^2)}\left[(1+3\nu)\,a\,x^2 + (1+\psi_1\nu)b_1 x^{\psi_1 - 1} + (1+\psi_2\nu)b_2 x^{\psi_2 - 1}\right] \ldots.(2)$$

In order to calculate constants b_1 and b_2, make use of the boundary conditions:—

Extension at outside, where $x = x_3$

$$\xi_3 = \frac{x_3^2}{E\,\delta_3\,y_3}\left(\sigma_3 y_3 + \mu\omega^2\delta_3 y_3 x_3 - \sigma r_2 \frac{x_2 y_2}{x_3}\right) \quad \ldots.(3)$$

Extension at inside, where $x = x_1$

$$\xi_1 = \left(\frac{x_0^2}{E\,\delta_0\,y_0}\sigma_0 y_0 + \mu\omega^2\delta_0 y_0 x_0 + \sigma r_1 \frac{x_1 y_1}{x_0}\right) \quad \ldots.(4)$$

But $$\xi_1 \text{ also} = a x_1^3 + b_1 x_1^{\psi_1} + b_2 x_1^{\psi_2} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(5)$$

and $$\xi_3 = a x_3^3 + b_1 x_3^{\psi_1} + b_2 x_3^{\psi_2} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(6)$$

By equating (3) and (6), also (4) and (5), the two resulting equations can be solved for b_1 and b_2.

When these values are put in (1) and (2) the radial and tangential stresses at any point in the disc can be obtained.

Disc with Straight-sided Taper Profile.

Treating a disc as having a conical profile, Mr. H. M. Martin * has derived a means of arriving at the stresses, which, together with the amendment to incorporate the effect of the hub, as suggested by Mr. B. Hodkinson,† provides a useful method.

The stresses throughout a conical disc are:

$$\text{Radial} = \sigma_r = Tp_c + Ap_1 + Bp_2 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

$$\text{Tangential} = \sigma_t = Tq_c + Aq_1 + Bq_2 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(2)$$

where T = stress in a thin ring

$$= 2D^2\left(\frac{N}{1{,}000}\right)^2 \text{ lbs./sq. in. for steel.}$$

D = diameter of completed cone.

$p_c, p_1, p_2, q_c, q_1, q_2$, are given in the curves, fig. 19, p. 1800, co-ordinated with the ratio $\frac{x}{R}$.

The constants A and B have always to be found by writing 'boundary conditions.' Thus at the inner limit of the conical portion where it joins the hub:

$$\sigma t_1 = \left\{\sigma_4 \frac{2x_1^2}{x_1^2 - x_2^2} + 2\left(\frac{N}{500}\right)^2 (0{\cdot}175x_1^2 + 0{\cdot}825x_b^2)\right\}$$

$$+ \sigma_{r_1}\frac{y_1}{y_0}\left[\frac{x_1^2 + x_b^2}{x_1^2 - x_b^2} - 0{\cdot}3\right] + 0{\cdot}3\Big\} \ldots\ldots\ldots\ldots\ldots\ldots(3)$$

* See *Engineering*, 5th & 26th January 1923.
† See *Engineering*, 31st August 1923 and 7th September 1923.

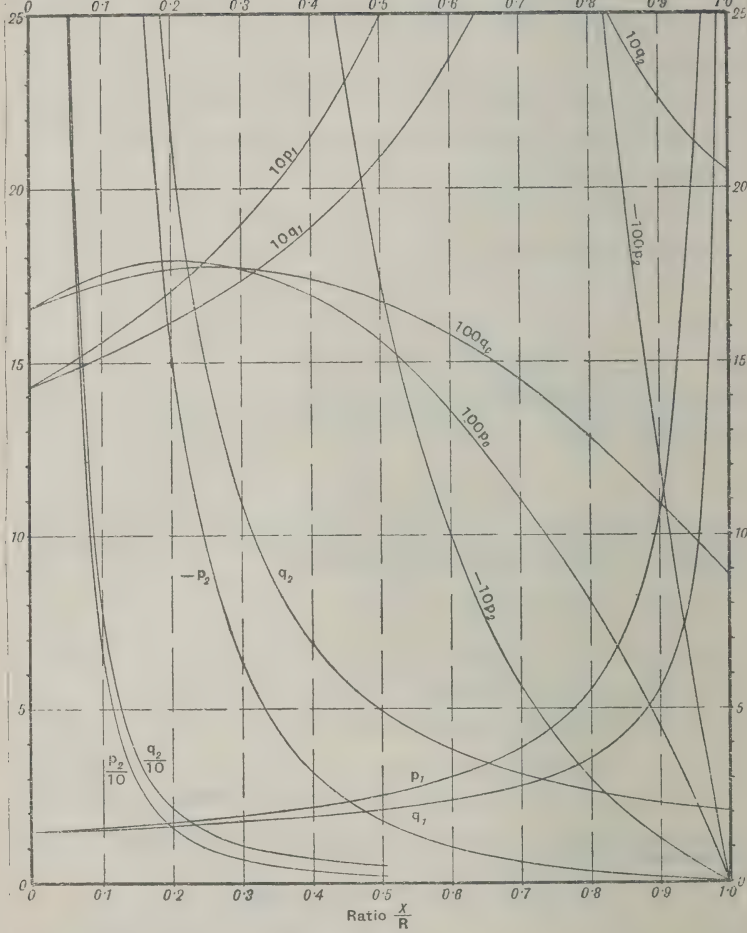

0
0·1
0·2
0·3
0·4
0·5
0·6
0·7
0·8
0·9
1·0
25
20
15
10
5
0
10p1
10q1
100q2
−100p2
100q2
100p2
−10p2
−p2
q2
p2/10
q2/10
p1
q1
Ratio X/R

FIG. 19.

where σ_{t_1} is the ' pinch ' pressure inside the bore in lbs./sq. in. At the rim

$$\sigma_{t_2} = \left\{ \frac{\text{Total centrifugal force of blades}}{2\pi\delta_2 y_2} + 2x_2^2\left(\frac{N}{500}\right)^2 \right\} - \sigma_{r_2}\left\{ \frac{x_2 y_2}{\delta_2 y_2} - 0{\cdot}3 \right\} \quad \ldots (4)$$

Putting in equations (1) and (2) the values of p_0, p_1, p_2, q_0, q_1, q_2, at the inner radius of the cone, and using the resulting expressions as σ_r, and σ_t, in equation (3), an equation involving only A and B is obtained. Writing similarly for the outer radius of the cone a second equation involving A and B, these constants can be calculated, whence the stresses anywhere in the disc are known.

The stresses throughout the hub then are:

$$\sigma_r = \sigma_{r_1}\frac{y_1}{y_0}\frac{(x^2 - x_b^2)}{x_1^2 - x_b^2}\frac{x_1^2}{x^2} - \sigma_4\frac{x_b^2}{x^2}\frac{(x_1^2 - x^2)}{(x_1^2 - x_b^2)} + 0{\cdot}825\left(\frac{N}{500}\right)^2\left(x_1^2 + x_b^2 - x^2 - \frac{x_b^2 x_1^2}{x^2}\right).$$

$$\sigma_t = \sigma_{r_1}\frac{y_1}{y_0}\frac{(x^2 + x_b^2)}{(x_1^2 - x_b^2)}\frac{x_1^2}{x^2} + \sigma_4\frac{x_b^2}{x^2}\frac{(x_1^2 + x^2)}{(x_1^2 - x_b^2)} + 0{\cdot}825\left(\frac{N}{500}\right)^2\left(x_1^2 + x_b^2 - 0{\cdot}575\,x^2 + \frac{x_b^2 x_1^2}{x^2}\right).$$

Pinch fit for Discs and Drums.

In order to ensure that the wheels shall be tight on the shaft at normal speed, they must be pressed on the shafts with an interference fit at least equal to the elastic expansion of the bore due to centrifugal stresses.

This condition can be fulfilled without the maximum stresses in the wheel being greater when the shaft is at rest than the centrifugal stress at normal speed, because in wheels of normal proportions the wheel bore expands during pressing on by only about two-thirds of the pinch allowance, the shaft diameter compressing by the remaining one-third.

Lubrication System.

In all cases except where ring lubrication is employed for small turbines, an oil pump is provided to maintain a pressure of about 10 lbs. per sq. in. in the lubricating system. It should be emphasised that the actual oil pressure is of secondary importance, the prime object being to flush the bearings with a copious flood of oil, and that for this purpose 2 lbs. per sq. in. may be sufficient in a properly designed system. The oil on leaving the pump passes through a cooler of sufficient capacity to maintain the temperature of the oil leaving the bearings at about 60-68° F. above that of the inlet cooling water, bearing temperatures of 150° F. being quite permissible. The main oil pump is usually driven by gearing from the turbine shaft, and the normal pressure is reached when the turbine motor is running at about one-third the normal speed. A hand pump is usually provided to supply oil under pressure during starting up or shutting down, when the speed of the set is less than one-third the normal. An oil strainer should be placed on the suction side of the pumps to prevent grit being circulated. Two to three gallons of oil should be taken out of the tank daily and carefully filtered, the level of the tank being kept constant by making up with freshly filtered oil or, when necessary, new oil. Once in each twelve months the oil tank and all accessible portions of the lubricating system should be emptied and cleaned ; the tank should then be filled sufficiently with paraffin oil to enable the whole system to be flushed by operating the hand pump. The paraffin oil dissolves any gummy matter which may have deposited from the lubricating oil and thoroughly flushes away all dirt to the oil tank ; the paraffin can then be removed and new or freshly filtered oil put in. These simple precautions will result in a real economy in oil consumption and a freedom from troubles and maintenance charges which will amply repay the trouble involved.

There is a large number of lubricating oils supplied by makers of standing, all of which will give satisfactory results in operation. For turbines driving geared installations an oil to the following specification has been found to give satisfactory results :

Heavy turbine oil.

The specific gravity at 60° F. should be from 0·89 minimum to 0·91 maximum.

The flash point taken by a closed test should not be less than 390° F.

The viscosity measured with a Redwood viscometer should be within the following limits :

At 68° F.	Minimum 650 seconds.		Maximum 800 seconds.		
At 140° F.	„ 90	„	„ 100	„	

The acidity expressed as SO_3 should not exceed 0·005 per cent.

One of the most important properties lies in the freedom of a suitable oil from a tendency to form an emulsion. Suitable oils should satisfactorily pass the following test :

When steam is passed into 100 c.c. of oil until the total volume of the condensate and oil measures 200 c.c., the whole of the oil shall separate out within ten minutes, leaving the water clear.

The sludge value of the oil carried out, according to Dr. Michie's test, should not exceed 1·2 per cent.

The following specification would apply to oils suitable for steam turbines driving generators, etc., direct :

Light turbine oil.

The specific gravity at 60° F. shall be 0·87 minimum to 0·89 maximum.
The flash point of the oil measured by a closed test shall not be less than 365° F.
The viscosity measured with a Redwood viscometer shall conform to the following:

At 68° F.	Minimum 300 seconds.			Maximum 400 seconds.	
At 140° F.	,,	60	,,	,,	75 ,,

The acidity expressed as SO_3 shall not exceed 0·005 per cent.
When steam is passed into 100 c.c. of oil until the total volume of condensate and oil measures 200 c.c., the whole of the oil shall separate out within five minutes, leaving the water clear.
The sludge value, measured by Dr. Michie's test, shall not exceed 1·2 per cent.

See also 'Lubrication,' p. 984.

Critical Speeds of Turbine Shafts.

At the critical speed of a loaded shaft, a small disturbance sets up vibrations which may reach a dangerous magnitude. For a uniform shaft, when the obliquity of the loading masses may be neglected, the critical speed coincides with the natural frequency of transverse vibration.

Although turbines have been run for considerable periods at their critical speeds without failure, this is a very dangerous proceeding and should be carefully avoided. It follows, therefore, that the critical speed should always be estimated, and should be at least 30 per cent. removed from the running speed.

This particularly applies to turbines of the disc type, and the calculation can be made with some degree of certainty. In drum turbines, the calculation is much more difficult, but in such cases as usually arise the critical speed is much above the running speed.

Table V. gives the formula for calculating the critical speeds of shafts of uniform diameter in certain standard cases.

Dunkerley has evolved theoretically, and demonstrated experimentally, a law from which can be determined the critical speed of a loaded system which corresponds to any combination of the standard cases.

By means of Table V. in conjunction with Dunkerley's formula, the first critical speeds of sensibly uniform shafts can be approximately calculated for any combination of loading, and any method of fixing the shaft at the bearings.

Thus if a rotor consists of a uniform shaft loaded with weights $W_2 W_3$ etc. as external loads,

n_1 = the critical speed of the shaft without the external loads;

n_2, n_3, etc. = the critical speed of a massless shaft loaded with weights $W_2 W_3$ etc.;

then the critical speed N of the system is given by

$$\frac{1}{N^2} = \frac{1}{n_1^2} + \frac{1}{n_2^2} + \frac{1}{n_3^2} + \cdots\cdots$$

In those cases of rotors having shafts stepped in diameter, as is usually the case in practice, the method already given is only a rough approximation, owing to the difficulty of judging the diameter of the equivalent uniform shaft.

Such cases may be dealt with by obtaining the maximum static deflection y of the shaft under various loads, by the graphical methods due to Mohr.

Then if

N = critical speed in R.P.M.;

$N^2 y = C$, where C is constant.

For impulse turbine shafts supported in bearings which do not fix the direction of the shaft Mr. K. Baumann* has found that the value of C varies from 37,700 to 38,000 if y is in inches.

See also p. 984.

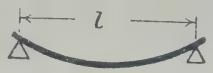
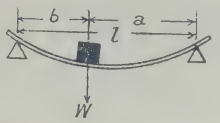
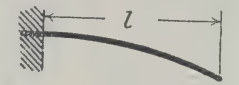
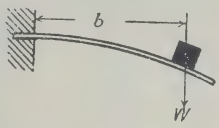
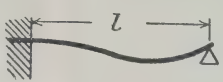
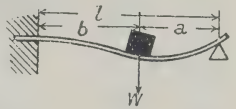
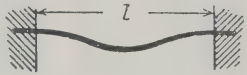
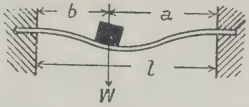
MATERIALS FOR TURBINE CONSTRUCTION.

The use of such high peripheral speeds as are common in turbine practice introduced a new consideration into machine design; because, unlike the conditions existing in reciprocating engine parts, the stresses in the various members are in the main due to centrifugal forces resulting from their own rotation. The stresses in a given turbine will, therefore, be proportional to the square of the numbers of revolutions per minute, and to the density of the material.

With an increase in the possible speed of rotation of a given turbine, the efficiency will in general be improved, since the average ratio $\frac{U}{C_o}$ in the various stages will be increased, or a greater heat drop can be utilised with the original efficiency.

* *Recent Developments in Steam Turbine Practice*, Inst. E.E., April 1912.

TABLE V.

Uniform Shaft with Distributed Load.		Concentrated Load Carried by a Weightless Shaft.	
Method of Support.	Critical Speed in R.P.M.	Method of Support.	Critical Speed in R.P.M.
	$N = \frac{1,853}{l^2}\sqrt{\frac{E \cdot I}{w}}$		$N = 325\sqrt{\frac{E \cdot I \cdot l}{W \cdot a^2 \cdot b^2}}$
	$N = \frac{662}{l^2}\sqrt{\frac{E \cdot I}{w}}$		$N = 325\sqrt{\frac{E \cdot I}{W \cdot b^3}}$
	$N = \frac{2,960}{l^2}\sqrt{\frac{E \cdot I}{w}}$		$N = 246\sqrt{\frac{E \cdot I \cdot l^3}{W \cdot a^3 \cdot b^3}}$ Approximate; holds if load near centre.
	$N = \frac{4,200}{l^2}\sqrt{\frac{E \cdot I}{w}}$		$N = 325\sqrt{\frac{E \cdot I \cdot l^3}{W \cdot a^3 \cdot b^3}}$ Approximate; holds if load near centre.

N = Critical Speed in R.P M.
w = Weight of Shaft or other Distributed Load in lbs. per inch run.
W = Weight of Concentrated Load in lbs.
a, b, or l = Length of Span in ins.

E = Young's Modulus of Elasticity in lbs. per sq. in.
I = Transverse Moment of Inertia in inch units = $\frac{\pi d^4}{64}$ for Round Shafts.
d = Diam. of Shaft in ins.

Note Units employed.

On the other hand, the available heat drop per stage might be increased so that the required number of stages for a given efficiency can be correspondingly reduced, or with the same number of stages the mean diameter of the blades can be reduced inversely as the revolutions per minute, both of which alternatives would result in a cheaper turbine.

Hence, it will be seen that in turbine work it is eminently desirable to use the highest possible peripheral speeds, the limit of which will be determined by the ultimate strength of the material employed.

British Electrical and Allied Manufacturers Association's Specifications for Turbine Forgings, Etc.

The following particulars have been extracted from the Association's specifications for orgings for turbine discs, shafts, and general forgings for use on land turbines.

The following table gives the minimum physical properties acceptable for each of four grades of materials:

Grade *	I.			II.			III.			IV.		
	L.	T.	R.	L.	T.	R.	L.	T	R.	L.	T.	R.
Minimum ult. tensile tons per sq. in.	30	30	30	35	35	35	40	40	40	48	48	48
Yield point	15	15	15	17½	17½	17½	20	20	20	30	30	30
Minimum elongation per cent.	30	23	20	25	20	18	22	18	16	17	14	12
Minimum reduction in area, per cent.	45	40	38	40	35	32	35	32	30	40	35	30
Bending Angle	180°	160°	150°	180°	150°	135°	180°	150°	120°	180°	130°	100°
Test radius	⅜″	⅜″	⅜″	½″	½″	½″	¾″	¾″	¾″	¾″	¾″	¾″

* L.=Longitudinal. T.=Tangential. R.=Radial.

Grades Nos. I., II., and III. are for carbon steels, whereas grade IV. applies to an alloy steel. A margin of 3 per cent. below the reductions in area given in the above table is allowed to the steel maker providing this is compensated by a corresponding increase of 3 per cent. on the elongation.

Bend Tests.

A cold bend test is made upon test pieces having a rectangular section of ¾ in. by ¾ in. These test pieces shall be machined and the edges rounded to a radius of $\frac{1}{16}$nd. The test pieces shall be bent over the thinner section, and the bending shall be done by pressure or by blows. The test pieces must withstand being bent around the radii and through the angles given in the table without fracture.

Additional Test before Rejection.

If either tensile or bending test fail to fulfil the test requirements, and the inspector agrees that the test piece does not clearly represent the quality of the material, two duplicate specimens may, if the maker wishes, be tested, and if the results from both are satisfactory the quality of the material shall be judged therefrom, and not from the original test which failed. If, however, either of the duplicate tests fail the material represented will not be approved, and the manufacturer shall have the option of retreating and again presenting for test.

Table VI. gives the physical properties of steels used for steam turbine blading, the data for which have been supplied by the courtesy of Messrs. Thomas Firth & Sons, of Sheffield.

The blades of Parsons turbines are commonly of brass, the usual mixture being alloy of 63 per cent. copper to 37 per cent. zinc, having an ultimate strength of 20 tons per sq. in., an elastic limit of 12 tons per sq. in., and an elongation of 40–50 per cent.

The blades of impulse turbines are usually made of 5 per cent. nickel steel, although bronze blades are sometimes called for.

The physical properties of a bronze are greatly dependent on the treatment to which it may have been subjected, cold working having the effect of raising the ultimate strength and reducing the ductility.

Such an increase in the ultimate strength is to a large extent artificial, the material returning to its virgin state under the influence of high temperature or vibration.

The importance of this effect will be emphasised by the figures published by M. Breuil † and reproduced in Table VII. respecting the effect of temperature on some materials used for blading.

† *Engineering*, July 5, 1912.

TABLE VI.—PHYSICAL PROPERTIES OF STEELS USED IN STEAM TURBINE CONSTRUCTION.

Nature of Material.	Yield Point, Tons per sq. in.	Ultimate Strength, Tons per sq. in.	Elongation on 2-in. Gauge Points.	Purpose for which Material is Employed.	Remarks.
			Per Cent.		
5 per cent. nickel steel sheets	25	40	20	Turbine blades and shrouding strips supplied mild in sheets for stamping and bending	—
5 per cent. nickel steel sheets	—	35–42	20–24	Turbine blades and shrouding strips	Higher carbon content than above
5 per cent. nickel steel bars	30–35	40–45	25	To be drawn into blade sections	—

For some years so-called rustless irons and steels have been used for turbine blades. These materials can now be regarded as of proved usefulness for turbine blading, owing not only to their comparatively high resistance to erosion and corrosion, but to their attractive mechanical properties.

Rustless steels are a chromium alloy, usually for turbine blading of composition within the following limits: carbon, 0·10 to 0·40 per cent.; chromium, 12 to 15 per cent. Ultimate strength, 40 to 50 tons/sq. in.; yield point, 30 tons/sq. in; elongation, 25 per cent.; izod test, 25 ft./lbs.

Most non-ferrous alloys suffer under high temperatures in a loss of mechanical strength and other desirable physical properties.

The following table illustrates the effect of temperature on some typical materials:

TABLE VII.—NON-FERROUS MATERIALS FOR TURBINE BLADING.

Nature of Material.	Elastic Limit, Tons per Sq. In.	Ultimate Strength, Tons per Sq. In.	Elongation.	Temperature.
			Per Cent.	
70 per cent. copper, 28 per cent. pure zinc	34·9	36·2	11	Ordinary
	34·3	36·2	8·5	100° C.
	32·7	34·3	11	200°
	22·2	24·1	30	300°
	7·6	12·7	75	400°
85 per cent. copper, 15 per cent. manganese	38·7	41·3	10·2	Ordinary
	36·8	40·0	10·5	100° C.
	37·5	41·0	11·0	200°
	33·0	36·8	21·0	300°
	21·0	24·1	68·0	400°

From this it follows that before figures relating to the strengths of alloys can be accepted as working data the effect of temperature and the previous history of the specimens fractured in the test should be known.

Phosphor bronze has been largely employed for impulse turbine blading when conditions were such that abnormal corrosion was expected, and gives satisfactory results with dry steam where the steam velocities are moderate and the stresses induced in the blades relatively small. The following represent the average properties of this material as given by Mr. W. B. Parker.*

Maximum tensile stress . . Not under 24 tons per sq. in.
Elastic limit . . . . . „ 12 „
Elongation on 2 ins. . . 13 per cent.
Modulus of elasticity . . 16,000,000 to 17,800,000 lbs. per sq. in. within a stress range of 0 to 10 tons per sq. in.

The average composition is as follows:

Tin . . . . . From 3 to 5 per cent.
Copper . . . . . From 94·9 to 96·9 per cent.
Phosphorus . . . . . Not less than ·1 per cent.

* *Steam Turbine Blading*, Institute of Metals, September 1915.

Nickel-copper alloys are now being used for the manufacture of turbine blading, the best known being Monel metal (*Messrs. G. & J. Weir, Ltd.*). This metal when drawn into the form of turbine blading has the following properties:

	At 60° F.	At 750° F.
Ultimate stress	41·9 tons per sq. in.	36·4 tons per sq. in.
Yield point	37·7 „	31·2 „
Elongation on 2 ins.	24 per cent.	16·5 per cent.
Reduction of area	60 „	60 „

Modulus of elasticity: 22,500,000 lbs. per sq. in. over a stress range of 1 to 15 tons per sq. in.

The composition (tests by the National Physical Laboratory) is as follows:

Nickel	67%
Copper	28%
Iron	1–2%
Manganese	1–2%
Lead, tin, and antimony	Nil.
Also silicon and carbon	

See also p. 2729.

PROCEDURE IN STARTING UP LARGE TURBINES.

While the general guidance given in the following remarks is, in the main, intended to apply to relatively large units (say turbines of 10,000 kw. capacity and larger), the procedure is an excellent one to follow on smaller sets.

The time intervals indicated can be reduced as the size of the set is reduced.

First operation.—Start up condensing plant and obtain 20 ins. to 24 ins. vacuum.

Second operation.—To warm up turbine, open stop valve sufficiently to start it rotating, then close it down until a steady speed of not more than 10 per cent. of the normal speed is obtained. Maintain this condition for ten to fifteen minutes during which vacuum is increased.

Third operation.—Increase the speed by gradual and steady increments at a rate of from 5 per cent. to 10 per cent. per minute.

Fourth operation.—Synchronise set and switch on to bus bars.

Fifth operation.—Build up load on set at a steady rate of from 5 to 10 per cent. per minute.

The procedure here outlined is a safe one to be recommended in normal practice; in an emergency the time intervals can, of course, be reduced.

TURBINE TEST CORRECTIONS FOR STEAM CONDITIONS.

While it is rarely possible to make tests upon a turbine under the exact steam conditions for which it has been designed, the performance which may be expected under the designed conditions can be estimated from the steam consumption actually obtained during the test.

The usual practice consists in correcting the test consumption for the difference between the steam superheat, pressure, and vacuum during the test, and that which is specified for the turbine.

These 'corrections' are arranged as coefficients by which the observed consumptions must be multiplied, and are determined for any steam conditions from consideration of the changes in available heat-drop and turbine efficiency which take place with any change in that condition.

A very complete list of these corrections has been published by Mr. K. Baumann,* and the values he gives are incorporated in this section.

Superheat Correction.

Where the superheat is different from that specified, the following corrections should be made, both for high-pressure or low-pressure turbines.

Between:—

0–100° F. superheat 1 per cent. improvement of steam consumption for every 10° F. superheat.

100–200° F. superheat 1 per cent. improvement of steam consumption for every 12° F. superheat.

200–300° F. superheat 1 per cent. improvement of steam consumption for every 14° F. superheat.

Correction for Initial Wetness of Steam.

For each 1 per cent. wetness the steam consumption, if measured as condensed water, will be 2 per cent. too high.

---

* *Recent Developments in Steam Turbine Practice*, Inst. E.E., April 1912.

Pressure Correction.

The correction which is to be made for the pressure depends upon whether the change in pressure is before the stop valve or before the nozzles.

Thus, if in the case of a high-pressure turbine working on a constant generator load the pressure before the stop valve is raised, say 10 lbs. per sq. in., the governor valve closes slightly, but the pressure before the nozzle will be very little altered, the only result being that the steam is throttled to a greater extent than formerly.

The effect of such an increase of pressure will be that the steam consumption will be improved by the small amount corresponding to the superheat due to the additional throttling.

In such cases the steam consumption will be improved by 0·5 per cent. for every 10 per cent. increase of pressure.

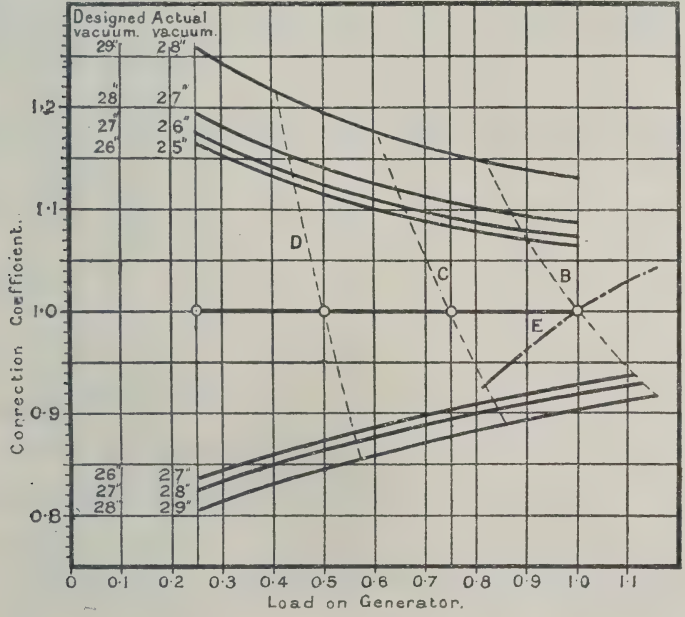


FIG. 20.—Vacuum Correction for Low-pressure Turbines.

When, on the other hand, the pressure before the nozzles is increased, the correction is of quite different magnitude and is:

For high-pressure turbines, an improvement of 1½ per cent. in steam consumption for 10 per cent. increase in pressure.

For low-pressure turbines, an improvement of 4 per cent. in steam consumption for 10 per cent. increase in pressure.

Vacuum Correction.

In any test, the correction for vacuum is usually of greater magnitude than that for pressure or superheat. Although every care has been exercised in estimating these corrections, it is obviously desirable that the total correction should be as small as possible, and if only for this reason every effort should be made to obtain during a test a vacuum as little different as practicable from that for which the turbine was designed.

Table VIII. gives the correction for vacuum which should be employed for various designed conditions with high-pressure turbines or mixed-pressure turbines, working with high-pressure steam.

TABLE VIII.—VACUUM CORRECTION FOR HIGH-PRESSURE AND MIXED-PRESSURE TURBINES.

Partial loads are expressed as fractions of the full load for which turbine is designed.

The correction is to be made when the turbine is run on a vacuum other than that for which it was designed.

Designed Steam Conditions.			Theo. Corr. for 1 in. Vac. Variation.	Actual Correction for 1 in. Vac. Variation.								Range of Vac. Variation. Ins.
Pressure, Lbs. per sq. in. g.	Supt. °F.	Vacuum ins.		H.P. Load $\frac{1}{4}$	M.P. $\frac{1}{4}$	H.P. $\frac{3}{4}$	M.P. $\frac{3}{4}$	H.P. $\frac{1}{2}$	M.P. $\frac{1}{2}$	H.P. Full	M.P. Full	
180	150	26	4·4	3·0		3·8		4·8		6·5		24—26—27
		27	5·5	3·7		4·6		5·6		7·6		25—27—28
		28	7·0	5·1		6·2		7·5		9·7		26—28—28½
		29	10·3	7·5		8·5		10·2		12·8		28—29
140	150	26	4·7	3·2		4·2		5·0		6·9		24—26—27
		27	5·8	3·9		4·8		5·9		8·0		25—27—28
		28	7·3	5·3		6·3		7·8		10·2		26—28—28½
		29	11·0	7·8		8·8		10·7		13·5		28—29
100	150	26	5·0	3·5		4·5		5·5		7·5		24—26—27
		27	6·2	4·2		5·3		6·4		8·7		25—27—28
		28	7·9	5·7		6·6		8·4		10·9		26—28—28½
		29	11·9	8·5		9·5		11·5		14·5		28—29
75	150	26	5·4	3·8		4·8		6·0		8·0		24—26—27
		27	6·6	4·6		5·6		6·9		9·4		25—27—28
		28	8·5	6·1		7·1		9·0		11·6		26—28—28½
		29	12·7	9·1		10·3		12·3		15·5		28—29
50	150	26	5·9	4·3		5·2		6·7		8·7		24—26—27
		27	7·2	5·0		6·0		7·7		10·3		25—27—28
		28	9·2	6·6		7·7		9·9		12·6		26—28—29
		29	13·5	9·9		11·3		13·3		16·8		28—29

The Theoretical Correction is given for 150° F. Superheat.

The actual correction given for various loads can be used for any superheat between saturation and 300° F.

It should be noted that at any load and for the same designed steam conditions and actual vacuum, the correction in the case of a mixed-pressure turbine (M.P.) is greater than for a high-pressure turbine (H.P.).

Thus, for example, the correction for a M.P. turbine at full load and working with high-pressure steam is the same as for a H.P. turbine on half load. The assumption underlying this is that the M.P. turbine is designed to give the same output on H.P. as on L.P. steam, and that, therefore, the conditions in the nozzles and blading of the L.P. end of the M.P. turbine are the same as that of a H.P. turbine designed for twice that output. Changes in vacuum only affect the conditions in the last few stages of a turbine, and these in a M.P. turbine on H.P. steam are operating under the same conditions as those of a H.P. turbine at half load.

Vacuum corrections for low-pressure turbines or for mixed-pressure turbines working with low pressure steam are still larger than those already considered.

In fig. 20 a coefficient will be found for various loads on the generator which is the ratio :—

$$\frac{\text{(Steam consumptions for 1 in. above or below the designed vacuum.)}}{\text{(Steam consumption for the designed vacuum.)}}.$$

While the corrections as given above can be applied to the generality of cases, special conditions may arise where corrections must be separately considered.

Thus, it is possible to obtain little or no benefit by increasing the vacuum with a turbine designed for a certain load at a certain vacuum and modified to operate on a considerably higher load and a higher vacuum; in such a case the velocity in the exhaust would be exceedingly high, and may even approximate to sound velocity.

In back pressure turbines, or the high pressure end of reducing turbines, all corrections will be considerably larger than those given above, and these require separate consideration in each case.

It cannot be too strongly emphasised that every endeavour should be made to arrange the test conditions in such a way that all corrections are as small as possible.

TYPES OF TURBINES.

Turbines may be classified according to the steam conditions under which they operate. It should be noted that the energy available in adiabatic expansion of steam from 180 lbs. per sq. in. g. and 100° F. superheat to 28 ins. vacuum, is about twice that available if the expansion is carried down to atmospheric pressure only. Hence the addition of condensing plant approximately doubles the amount of work which can be done per lb. of coal, without necessitating any increase in boiler capacity.

High-Pressure Turbines.

High-pressure turbines take their steam directly from the boilers, and expansion takes place in the turbine down to the pressure in the condenser. This may be regarded as the fundamental class of turbine. The steam conditions are usually 350–150 lbs. per sq. in. g. pressure, 0°–300° F. superheat, and 27½ ins. to 29 ins. vacuum (30 ins. Barm.). Of course, the range of possible conditions is greater than these: for instance, in collieries and steelworks, steam pressures of 100–50 lbs. per sq. in. are to be met with, while steam temperatures of 800° F. will probably be adopted in the near future.

Low-Pressure Turbines.

Low-pressure turbines take their steam at about atmospheric pressure, the source almost invariably being the exhaust from non-condensing reciprocating engines. The initial steam pressure is usually from 20–15 lbs. per sq. in. abs., and should in all cases be above atmospheric pressure in order to eliminate the risk of air leakage into the steam pipe line. Low-pressure turbines require no governor gear if connected either mechanically or electrically to the recipro-

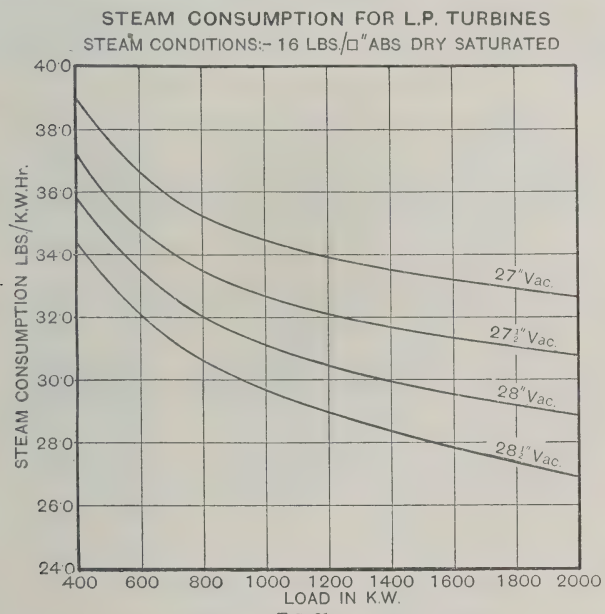


FIG. 21.

cating engine from which they obtain their steam. In the event of the engine being shut down the turbine is supplied with steam from the boilers through a reducing valve.

The efficiency with which low-pressure steam can be used naturally depends not only on the size and speed of the plant, but also on its capacity and costliness. As an average the steam consumption shown in fig. 21 can be used for estimating purposes when dry steam is supplied to a low-pressure turbine at a pressure of 16 lbs./sq. in. absolute.

Back-Pressure Turbines.

Back-pressure turbines take their steam directly from the boilers and exhaust at or above atmospheric pressure. Such turbines may be used in cases where the exhaust steam is required for heating purposes, and the back pressure usually ranges from 1 to 20 lbs. per sq. in. g.

In certain cases the back pressure may be as high as 40 lbs. per sq. in. g.

Back-pressure turbines are used to a considerable extent to drive the auxiliaries, their exhaust steam being utilised for feed heating; by this means the losses in the turbine are recovered in the feed heater, and an overall efficiency for back-pressure turbine and feed heater of over 90 per cent. is obtained.

Mixed-Pressure Turbines.

When L.P. steam is utilised in an L.P. turbine it is obviously desirable that the turbine set may be operated at will independently of the reciprocating engines from which it obtains its steam, so that in the event of the engine being shut down the turbine may operate with reasonable economy. This becomes possible by the addition of what is really a B.P. exhausting into the L.P. turbine, and having a common casing and spindle. This combination forms a mixed-pressure turbine, which can operate not only on either H.P. or L.P. steam, but if sufficient L.P. steam is not available to carry the load, the deficit is automatically supplied by admitting H.P. steam to the H.P. end of the turbine.

The governor gear is automatic and adjusts itself so that all the L.P. steam available is used in preference to H.P. steam.

In mixed-pressure turbines the governor gear is of special importance, as for successful working in parallel with other alternators the changes from L.P. steam to H.P. steam must be effected without any appreciable change in speed or load of turbo-alternator.

Reducing Turbines.

With a B.P. turbine the amount of steam available for heating purposes, and the work done by the turbine, are directly connected. In most installations it is desirable to be able to vary these quantities independently of each other, and to meet this demand 'Reducing' turbines were introduced.

Power Obtained from Heating Steam.

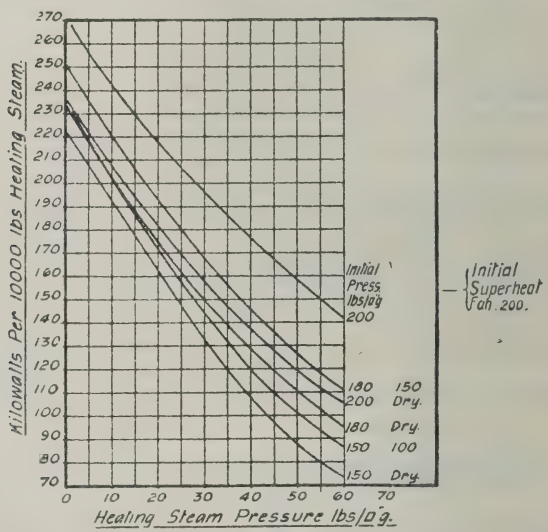


Fig. 22.

An R.P. turbine consists of a B.P. turbine to which an L.P. turbine is added. The B.P. turbine is designed for the maximum amount of steam required by the heaters it supplies.

As the demand for heating steam decreases, an automatic governor gear by-passes the supply to the L.P. end of the turbine, and at the same time the supply of steam to the H.P. end is decreased.

In this way the load the generator can carry is independent of the amount of heating steam required.

The governor gear should be operated in such a way that the pressure of the heating steam remains constant.

An increasing demand for this type of machine has arisen in recent years as a result of the extremely economical results which can be obtained in those manufacturing works requiring both electrical power and heating steam.

In order to obtain the best result the pressure at which steam is taken from the turbine for heating purposes should be as low as possible in view of the requirements of the heating process. The effect of back pressure on the consumption of heating steam can be seen from the following table, while the variation in the power which can be obtained from back pressure on reducing turbines is shown in fig. 22 for various steam conditions.

In recent years marked improvements have been made in the efficiency which can be obtained in H.P. ends of steam turbines. Whereas formerly the first step in expansion was accomplished with an efficiency of from 50 to 60 per cent., efficiencies of from 70 to 80 per cent. can now be realised in turbines of reasonable size and of other than the cheapest construction. It follows that in back pressure or reducing turbines of some considerable size, elaboration and costliness, the kw. obtainable per 10,000 lbs. of steam per hour can, in such cases, exceed those given in fig. 22 by some 25 per cent.

TABLE IX.

PERCENTAGE CHANGE IN STEAM CONSUMPTION OR WORK CAPACITY OF HEATING STEAM FOR A CHANGE IN HEATING STEAM PRESSURE OF 5 LBS. SQ. IN. G.

Initial Pressure. lbs. sq. in.	Initial Superheat. Degrees F.				
	Dry.	50	100	150	200
200	6·85	6·5	6·15	5·8	5·45
190	7·3	6·92	6·52	6·15	5·77
180	7·75	7·35	6·9	6·5	6·1
170	8·22	7·75	7·25	6·82	6·37
160	8·7	8·15	7·6	7·15	6·65
150	9·25	8·65	8·1	7·5	6·95

RECENT DEVELOPMENTS IN STEAM CONDITIONS AND MODIFICATION IN OPERATING CYCLE.

The component elements in a modern steam turbine and boiler plant are indicated in fig. 23.*

The various parts shown of steam turbine, regenerative feed heaters A, B, C, and D, economisers

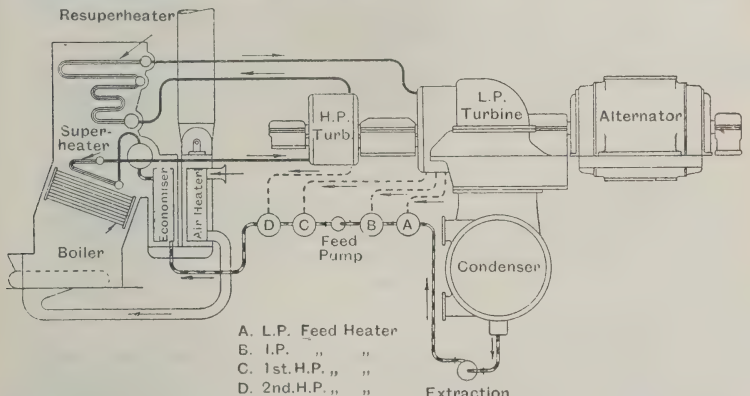


FIG. 23.

* The figures and data used in this section are taken from 'Economic Value of Increased Steam Pressures' of 1927, or from 'Tendencies in Steam Turbine Development' of 1929, both by H. L. Guy, before the Institution of Mechanical Engineers.

boiler, superheater, resuperheater and air preheater, may be used all together, or one or other of the various elements may be omitted. For instance, the resuperheater appears desirable if steam pressures above 500 lbs./sq. in. are used, while air preheating will usually be employed if the feed water is heated above 230° F. by the turbine feed heaters. In order to consider the influence of these elements on plant efficiency, the influence of steam pressure, steam temperature, regenerative feed heating and resuperheating will be considered separately.

Increased Steam Pressure.

The influence of steam pressure on heat consumption of the steam turbine itself is shown in fig. 24 by curve G; while curve M shows the corresponding gain when the effect of the turbine and boiler auxiliaries is taken into account. The meaning of the remaining curves in the figure will be evident.

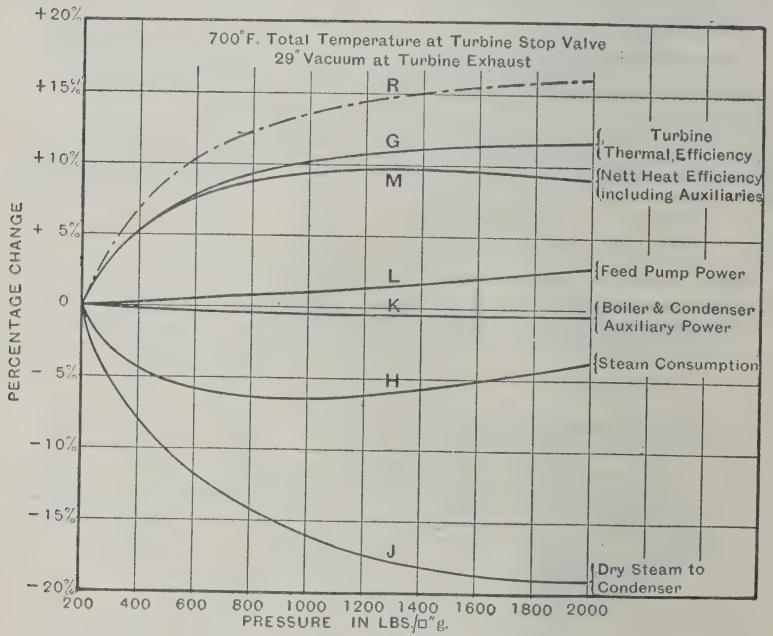


FIG. 24.

Curve R shows the way in which the gain flowing from the use of increased steam pressure is increased by the use of four stages of regenerative feed heating, and therefore represents the relative gain which can be realised in a fairly elaborate modern installation of reasonable capacity. The last reservation arises from the fact that the gain which can be obtained by using increased pressure depends very much on the capacity and speed of the installation. The figures given apply to an installation of about 20,000 kw. capacity if the speed is 3,000 r.p.m., or 40,000 kw. capacity if the speed is 1,500 r.p.m. For much larger installations the returns will be very slightly increased. For sensibly smaller capacities the gains will be materially reduced. From curve R the following table has been deduced, showing the pressure steps and pressures which correspond to definite gains in coal consumption of 3 per cent. each time:

TABLE X.—INCREASE IN STEAM PRESSURE FOR 3 PER CENT. STEPS IN THERMAL EFFICIENCY.

Basis.—700° F. total temperature, 29-in. vacuum, four-stage feed heating. Including boiler and condenser auxiliaries. Capacity not less than 40,000 kw. M.C.R. at 1,500 r.p.m.; 20,000 kw. M.C.R. at 3,000 r.p.m.; 10,000 kw. M.C.R. at 6,000 r.p.m.

Increase in thermal efficiency	3%	6%	9%	12%	15%
Initial Pressure.	Final Pressure.				
From 200 lbs. per sq. in. g to	265	365	505	764	1,360
,, 250 ,, ,, ,, ,,	342	472	703	1,200	—
,, 350 ,, ,, ,, ,,	495	755	1,390	—	—
,, 500 ,, ,, ,, ,,	785	1,530	—	—	—
,, 750 ,, ,, ,, ,,	1,470	—	—	—	—
,,1,400 ,, ,, ,, ,,	—	—	—	—	—
Reduction in coal consumption	2·92%	5·66%	8·26%	10·71%	13·04%

For 900° *F.*—Gain with pressure as above between 200 lbs. per sq. in. and 750 lbs. per sq. in Gain with pressure increased by $\frac{1}{3}$ per cent. per 100 lbs. per sq. in. between 750 lbs. and 1,400 lbs. per sq. in.

For 28-*in. Vacuum.*—Gain with pressure increased by 0·3 per cent. per 100 lbs. per sq. in. between 200 lbs. and 750 lbs. per sq. in. Gain with pressure increased by 0·2 per cent. per 100 lbs. per sq. in. between 750 lbs. and 1,400 lbs. per sq. in.

The absolute values of the most important quantities entering into the calculation of the curves in fig. 24 are given in Table XII. The figures given in lines 21 and 22 under 'Plant Efficiency' are based on a generator efficiency of 96 per cent. and a boiler efficiency of 84 per cent. It should be stated that all the preceding figures are based on an initial temperature of 700° F. and on a vacuum of 29″. The relative position is not sensibly affected by increased steam temperature, but the net gains increase as the vacuum is decreased. For instance, the gain with 28″ vacuum at 2,000 lbs./sq. in. is some 2 per cent. greater than that shown for 29″ vacuum.

Increased Steam Temperature.

The most recent tendency in turbine development has been in the direction of increasing the initial temperature at which steam is supplied to the turbine. The increases in thermal efficiency resulting from increase in initial temperature are given in Table XI.

TABLE XI.—INCREASE IN TOTAL TEMPERATURE FOR 3 PER CENT. STEPS IN THERMAL EFFICIENCY.

Basis.—700° F. total temperature, 29-in. vacuum, four-stage feed heating. Including boiler and condenser auxiliaries. Capacity not less than 4,000 kw. M.C.R. at 1,500 r.p.m.; 2,000 kw. M.C.R. at 3,000 r.p.m.; 1,000 kw. M.C.R. at 6,000 r.p.m.

Increase in thermal efficiency	3%	6%	9%	12%
Initial Pressure.	Increase in Temperature, ° F.			
200 lbs. per sq. in. g	77	151	227	302
250 ,, ,, ,,	77	155	232	309
350 ,, ,, ,,	79	158	237	—
500 ,, ,, ,,	79	159	—	—
750 ,, ,, ,,	77	—	—	—
1,400 ,, ,, ,,	65	—	—	—
Reduction in coal consumption	2·92%	5·66%	8·26%	10·71%

Comparison of Tables X and XI reveals the following interesting characteristics of pressure and temperature increases.

If the total temperature is kept constant, the changes in pressure necessary to secure a given gain in efficiency increase very rapidly, both with the initial pressure and with the magnitude of the gain in efficiency considered. On the other hand, the increase in temperature necessary to secure a given gain in efficiency is sensibly constant at all initial steam pressures whether the step in efficiency be 3 per cent. or 12 per cent.

In contrast with increases in initial pressure, the thermal advantage of increased temperature does not sensibly depend upon the size of the prime mover. A further distinction arises from the fact that a much smaller proportion of the complete boiler and turbine plant is affected by raising the temperature, than by raising the pressure.

Note.—Recently acquired knowledge on the properties of materials at elevated temperature has resulted in recognition of the fact that other things being equal steam pressures must be decreased as the steam temperature is increased. It is probable that the most desirable condition in the immediate future will be obtained by the adoption of a moderate steam pressure coupled with a higher steam temperature than has been normally used in the past.

Regenerative feed heating.

In recent years the practice of heating the feed water progressively by feed heaters supplied with steam tapped from the main turbine at various points in the expansion has developed until it has become standard. From one to four feed heaters are used, depending on the size and

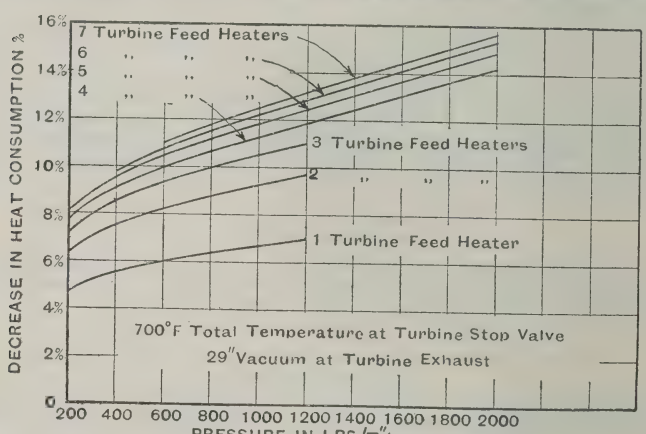


FIG. 25.

elaboration of the plant. The gains in coal consumption which can be expected at various steam pressures with from one to seven feed heaters are shown on fig. 25 for installations including an economiser and air preheater. If no air preheating is done, these gains are reduced to about two-thirds of those shown in this figure.

Resuperheating.

If steam is taken out of the turbine after partial expansion and passed through a superheater, the thermal efficiency is improved and consequently the coal consumption reduced. This gain in efficiency varies with the points in expansion at which resuperheating takes place. The upper

TABLE XII.—VARIATION IN EFFICIENCY WITH STEAM PRESSURE.
(700° F., 29-in. Vacuum

1	Initial pressure (Lbs./sq. in. g.)	200·0	285·3	385·3	485·3	585·3	785·3	985·3	1185·3	1385·3	1585·3	1785·3	1985·3
2	Initial pressure (Lbs./sq. in. Abs.)	214·7	300·0	400·0	500·0	600·0	800·0	1000·0	1200·0	1400·0	1600·0	1800·0	2000·0
3	Superheat (°F.)	321·1	282·2	254·5	231·4	211·7	179·1	152·7	130·3	110·7	93·3	77·6	63·2
4	Total heat from 78·9° F. (B.Th.U/Lb.)	1330·7	1326·2	1321·1	1315·9	1310·7	1300·4	1290·0	1279·2	1269·4	1259·0	1248·7	1238·4
5	Available heat to 29-in. vac. (B.Th.U/Lb.)	453·2	470·2	484·1	494·1	501·6	512·3	519·1	523·6	526·3	527·7	528·4	528·2
6	Theoretical thermal effy. (%)	34·06	35·45	36·64	37·55	38·27	39·39	40·24	40·91	41·46	41·91	42·32	42·65
7	Reheat factor	1·0601	1·0597	1·0594	1·0592	1·0594	1·0604	1·0620	1·0624	1·0666	1·0686	1·0701	1·0711
8	Final dryness	0·9050	0·8875	0·8720	0·8595	0·8490	0·8300	0·8130	0·7980	0·7855	0·7755	0·7675	0·7615
9	Turbine losses factor	1·0000	0·9952	0·9910	0·9878	0·9852	0·9810	0·9778	0·9752	0·9729	0·9711	0·9694	0·9679
10	Comparative turbine effy. (%)	80·00	79·04	78·10	77·33	76·69	75·53	74·60	73·79	73·07	72·44	71·86	71·39
11	Comparative thermal effy. of turbine (%)	27·25	28·02	28·62	29·04	29·34	29·75	30·02	30·19	30·30	30·36	30·41	30·45
12	Power to feed pump (%)	0·35	0·45	0·58	0·71	0·85	1·14	1·42	1·74	2·07	2·41	2·77	3·15
13	Power to boiler auxiliaries (%)	1·50	1·46	1·43	1·41	1·39	1·37	1·36	1·35	1·35	1·35	1·34	1·34
14	Power to condenser auxiliaries (%)	2·00	1·91	1·85	1·80	1·77	1·72	1·68	1·65	1·64	1·63	1·62	1·62
15	Total power to auxiliaries (%)	3·85	3·82	3·86	3·92	4·01	4·23	4·46	4·74	5·06	5·39	5·73	6·11
16	Thermal effy. of turbine corr. for aux. (%)	26·20	26·94	27·51	27·89	28·17	28·50	28·68	28·75	28·76	28·72	28·67	28·58
17	Decrease in heat cons. with 4 F.H. (%)	7·72	8·42	8·99	9·36	9·86	10·60	11·21	11·83	12·43	13·02	13·60	14·20
18	Thermal effy. of turbine with 4 F.H. (%)	29·53	30·59	31·56	32·04	32·55	33·28	33·81	34·23	34·60	34·90	35·19	35·49
19	Total power to aux. with 4 F.H. (%)	3·42	3·40	3·42	3·45	3·51	3·79	4·11	4·52	4·98	5·42	5·96	6·44
20	Thermal effy. of turbine corr. for aux. with 4 F.H. (%)	28·53	29·55	30·38	30·94	31·41	32·02	32·41	32·69	32·88	33·01	33·09	33·20
21	Plant effy. without F.H. (%)	21·12	21·72	22·18	22·49	22·72	22·98	23·14	23·19	23·20	23·17	23·12	23·06
22	Plant effy. with 4 F.H. (%)	23·01	23·83	24·50	24·95	25·33	25·90	26·13	26·36	26·51	26·62	26·68	26·77

part of fig. 26 shows the percentages given of the total expansion which should precede resuperheating. The two curves shown should be treated as giving two limits. Again, in the lower portion, the two curves show the two limits for the reduction in coal consumption. The dotted

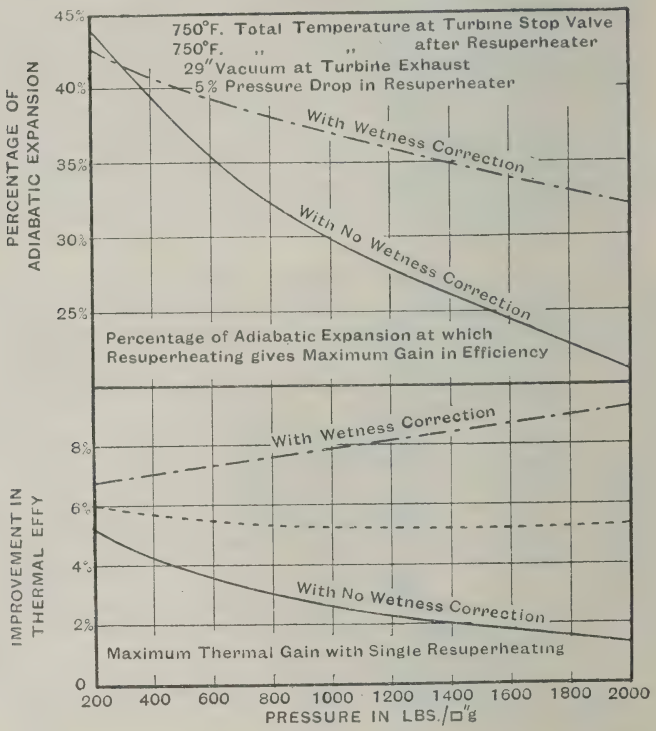


FIG. 26.

curve shows the mean value and represents about the percentage which has so far been obtained by resuperheating.

British Engineering Standards Specification for Steam Turbines.*

No. 132 (Abstract).

Rating.

A steam turbine shall be capable of giving continuously a specified output when running under specified conditions as regards speed, steam pressure, temperature and exhaust pressure.

* By permission of the British Engineering Standards Association.

Output of the Turbine when Working Non-Condensing.

If the specified steam pressure is not less than 165 lbs. per sq. in. absolute, and the rated output of the turbine is not more than 3,000 kw., the turbine shall be capable of developing 30 per cent. of its rated output when exhausting to the atmosphere. For turbines of larger capacity the output must be settled between the purchaser and the manufacturer.

Note.—It is undesirable to work high-pressure steam turbines exhausting to the atmosphere and this should only be done in cases of emergency. Even under this condition, the turbine should not operate in this way for a longer time than is absolutely necessary.

Maximum Speed.

The turbine shall be capable of withstanding for five minutes, without injury, a speed 15 per cent. in excess of the rated speed.

Governing Characteristics.

When the rated load is taken off or on the permanent speed variation shall not exceed 4 per cent.

Speed Adjustment.

Means shall be provided to enable the speed to be adjusted by plus or minus 5 per cent., when working under synchronising conditions.

Emergency Cut-off Speed.

The emergency governor shall be set to shut off steam if the speed exceeds 9 to 11 per cent. of the rated speed. When operating at rated speed and the rated load is thrown off the maximum variation in speed shall not be sufficient to bring the emergency governor into operation.

Economic Rating of the Turbine.

The turbine shall be designed to give a minimum steam consumption at 80 per cent. of its rated output.

Steam (or Heat) Consumptions.

If steam (or heat) consumptions are guaranteed for various outputs, the average steam consumption shall be obtained by

Mutiplying the consumption at the rated output by b_1.
" " 80 per cent. of the rated output by b_2.
" " 60 " " " b_3.
" " 40 " " " b_4.

and dividing the sum by $b_1 + b_2 + b_3 + b_4$, the multiplying factors to be specified by the purchaser or mutually agreed.

The steam (or heat) consumption may be requested and furnished either for some specified load or for a series of loads, but the guarantee shall be confined to the one specified load or to the weighted consumption as above.

Tolerance.

For the purpose of comparing test consumptions with the guarantee performance a tolerance calculated as follows shall be allowed:—

Percentage tolerance (+ or —) =

$$\frac{\text{Arithmetical sum of percentage corrections} - 5}{4} + 2\tfrac{1}{2} \text{ per cent}$$

Hydraulic Test for Parts Exposed to Full Boiler Pressure.

All parts exposed to the rated steam pressure shall be tested with a hydraulic pressure 50 per cent. in excess of the rated pressure. The machine as a whole shall be tested with a hydraulic pressure of 15 lbs. per sq. in. gauge.

Critical Speed of Rotating Element.

The critical speed of the turbine shaft shall not be within per 20 cent. above or 20 per cent. below the rated speed.

TURBINE SPEED-REDUCTION GEAR.

The speed at which it is most desirable to run a turbine is limited only by the strengths of the materials used in its construction. Unfortunately this limit of speed cannot be approached when the turbine is employed for direct driving of propellers, D.C. generators of large output, rolling mills, line shafting, etc., and in such cases where comparatively low speeds are essential not only does the turbine become heavy, costly and inefficient, but direct turbine drive is often quite out of the question.

For this reason several systems of speed reduction have been introduced, and are being successfully applied.

Double Helical Gears.

Speed reduction by means of double helical gears was employed in the earliest days of turbine engineering by Dr. Delaval, whose turbines, running at speeds of the order of 10,000 r.p.m., could not be directly coupled to electric generators. The gearing of these turbines gave remarkably little trouble, and although the outputs were low it is notable that even to-day this is the most successful system of speed reduction.

Double helical gearing for large turbines was introduced by Sir C. A. Parsons, who with characteristic courage first introduced the system in the famous case of the propelling machinery of the s.s. 'Vespasian,' and the drive of a rolling mill at Calderbank Steel Works.

In the earlier installations difficulties were experienced through minute inaccuracies of the gear teeth, which gave rise to very noisy working. This was traced to errors in the gearing of the table of the gear-cutting machine by Sir C. A. Parsons, who has very materially reduced the trouble by arranging a 'Creeping' table giving the gear blank an advance of 1 per cent. relatively to the main table of the machine. The improvements in modern gear-cutting machines have eliminated the inaccuracies referred to, so that the creeping device is no longer essential.

The helix angle adopted varies from 20° to 45°, the usual angle in modern practice being 30°

The teeth of modern double helical gears are of the standard involute type, the pressure angle adopted by different makers varying from $14\frac{1}{2}°$ to 20°.

If p = average pressure in lbs./inch tooth face.
L = total length of tooth face in inches.
D = diam. of pitch circle of pinion in inches.
R = revs. per minute of pinion.

$$V = \text{pitch speed in feet per second} = \frac{\pi D}{12} \times \frac{R}{60} = \frac{\pi D}{720} \times R.$$

Then

$$\text{Horse-power transmitted} = P = \frac{pLV \times 60}{33{,}000} = \frac{pLV}{550}.$$

Dr. Stoney states that V should be kept between the limits 100 and 130 ft./sec. He also gives the maximum permissible pressure $p = 120d$ for pinions smaller than 5 inches diameter, and $p = 175\sqrt{d}$ for larger pinions.

Fig. 27 shows curves co-ordinating p and d, published by Mr. F. H. Hodgkinson (*Engineering*, Jan. 10, 1919).

With pinions having two bearings where E = Young's modulus of elasticity = 30×10^6 lbs./sq. inch,

$$I = \text{the moment of inertia of the pinion} = \frac{\pi D^4}{64}.$$

$$\text{Transverse deflection at the centre} = \delta_1 = \frac{5}{384} \cdot \frac{pL^4}{EI}.$$

This is one of the determining factors in design, and, according to Stoney, should not exceed 0·0002 inch.

For pinions having three bearings, the maximum deflection on each portion of the pinion between two bearings should not exceed 0·0002 inch, and in most cases this is not the condition determining the diameter.

Another condition fixing the design of the pinion arises from the fact that the torsional deflection opens the teeth, thus preventing uniform distribution of pressure on them. This torsional opening, δ_2, according to Stoney, must be limited to less than 0·0001 inch, and is given by the following expressions :

$$\text{For two or three bearing pinions } \delta_2 = \frac{OL}{10 \times 10^4} \text{ inches over one helix.}$$

In the Melville-McAlpine gear, manufactured and developed by the Westinghouse Machine Co., the pinion is supported in a floating frame which permits its axis to take up a slight inclina-

DOUBLE HELICAL GEAR CURVES.

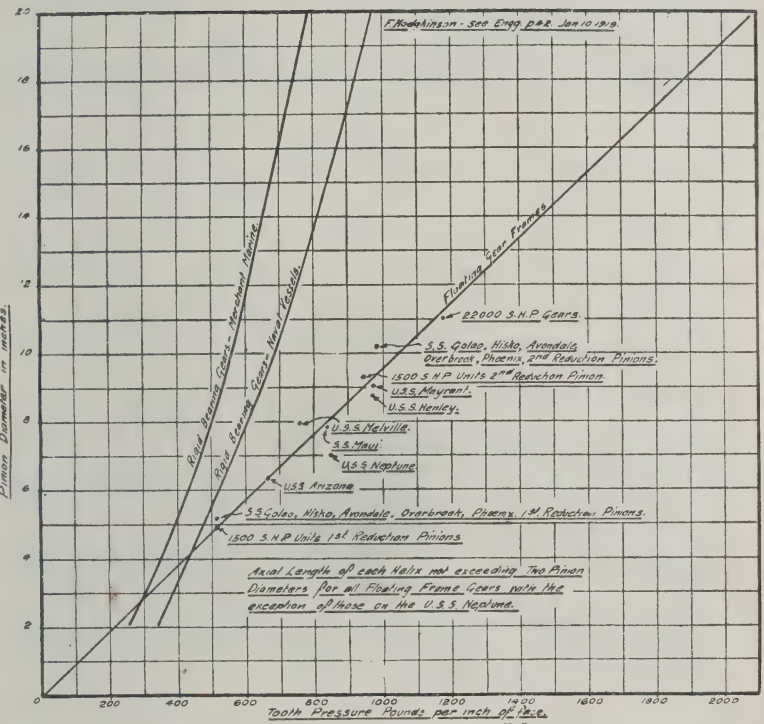


FIG. 27.

tion in the vertical plane relatively to the slow speed shaft. The torque on the pinion tends to increase the helix angle of one half of the pinion and decrease that of the other half, so that if the helix angle of the pinion is the same as that of the wheel, as load comes on, the helix angles become different. If the axis of the pinion is inclined in a vertical plane relatively to the wheel, the change in the helix angle with load can, it is claimed, be very largely compensated, so that with the floating pinion larger torques can be transmitted through a given size pinion than through a pinion supported in rigid bearings. According to the inventors, the floating frame also corrects to a certain extent errors in the teeth and in alignment of the gears.

The action of this gear is well explained, together with some authoritative notes on double helical gearing, in a series of articles by Mr. J. H. McAlpine, which appeared in *Engineering*, commencing May 5, 1916.

In these articles Mr. McAlpine suggests the use of the 'Power Constant' for comparing different gears; these he defines as follows:

If

D = pitch diameter of pinion in ins.; L = length of tooth face in ins.; p = average pressure in lbs. per in. of tooth face; R = r.p.m.; C = power constant; P = horse-power transmitted;

Then

$$P = \frac{\pi D R p L}{33000 \times 12}.$$

For correctly designed gears, p and L vary as D, therefore

$$P = C (D^2 R),$$

and

$$C = \frac{P \times 1000}{D^2 R}.$$

For gears having the pinion supported in two rigid bearings, the power constant 'C' should not exceed 1·5.

'C' is usually from 1·2 to 3·0, for gears having three bearings on the pinion.

Gears having the pinion supported in a floating frame have worked very successfully with a power constant of 4.

The ratio $\frac{L}{D}$ should preferably be equal to 4 in gears having the floating frame.

For best results the pinions for all sizes should have from 26 to 30 teeth; if the number is appreciably reduced interference takes place.

The efficiency in double helical gearing varies from 98·5 per cent. in large gears to 97·5 per cent. in small sets, so that from 1·5 per cent. to 2·5 per cent. of the heat equivalent of the power transmitted has to be dissipated in the lubricating oil cooling system.

The gears are usually lubricated by spraying the oil on the teeth at the point of contact.

The success of a gear depends to a larger extent than is generally supposed on the rigidity and correct design of the gear-box and lubricating system, and where experienced engineers are responsible for these details every confidence can be placed in a turbo-geared proposition.

Rapid advances have been made in the application of steam turbines to the mercantile marine, by the introduction of double reduction gearing. Single reduction gears having ratios larger than 15 to 1 are scarcely practical for ship propulsion, but a reduction of, say, 42 to 1 can be commercially obtained by a double reduction gear having two trains of wheels, the first having a reduction of, say, 6 to 1 and the second 7 to 1.

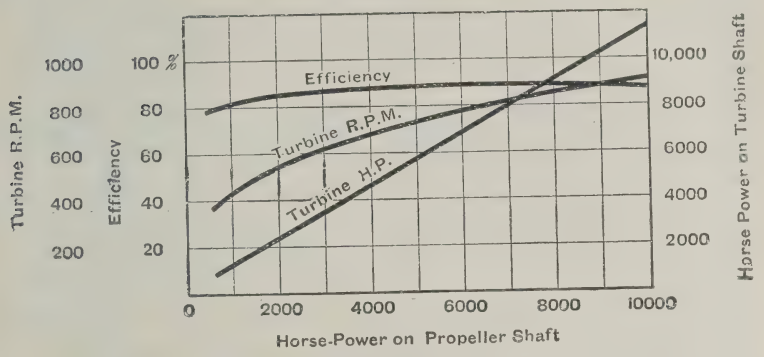
Such double reduction gears are now being used having efficiencies at full load of about 97 per cent.

Now that such large ratios of reduction are commercially practicable, both the turbine and the propeller can be run at the speed most suitable for each respectively.

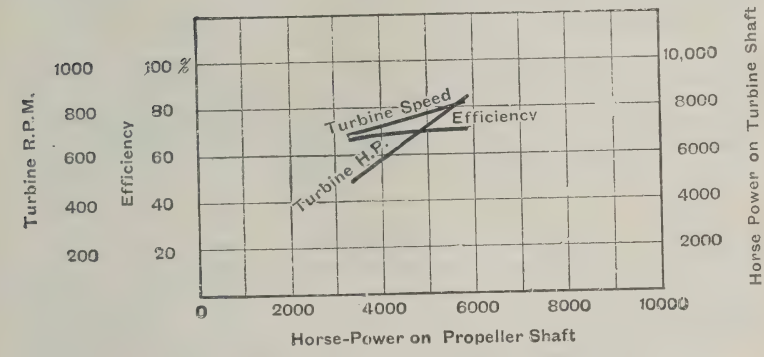
See also p. 930, and 'Reduction Gear for Propulsion,' p. 2381.

The possibilities and limitations of this type of transmission are illustrated by the following figures given by Prof. Fottinger.*

No. of Stages.	Gear Rates.	Estimated Efficiency.
1	5 to 1	90 per cent.
2	7½ to 1	90 „
3	10 or 12 to 1	88 „



FORWARD ELEMENT



REVERSE ELEMENT

FIG. 28.

Unfortunately the ratios usually required for marine purposes are from 10 to 1 up to 20 to 1. and as it is doubtful if more than single stage transmitters could be commercially adopted, it will be seen the field open to this type is limited. A distinct advantage lies in the fact that the discharge from the turbine pump element can be diverted to a reverse water turbine, by which means reversal can be easily and economically effected. The water used in the transmitter is usually the boiler feed, and in this way some of the losses can be recovered in feed water heating, but it is difficult to see how the gain can possibly exceed 1 per cent. measured at the driven shaft.

Fig. 28 shows the results of tests on a 10,000 B.H.P. transmitter.†

* *Engineering*, March 14, 1913.

† *Z. V. d. I.*, May 16, 1913.

TABLE XIII.—PARTICULARS OF SOME NOTABLE INSTALLATIONS OF GEARED TURBINES.

Name of Installation.	Kind, Maker.	Face Width.	Angle of Helix.	Pressure in Lbs. in W.D.	Pitch Line Speed. Ft. per Sec.	Power Constant.	Face Width ÷ Pinion. Diam.	Efficiency of Gear.	Reference.
Hope Natural Gas Company	American Westinghouse Co.	40	30°	715	65·5	2·26	4·0	—	'Engineering,' May 26, 1916.
Cleveland Elec. Illuminating Company	,,	40	30°	925	81·5	2·72	3·85	—	,,
U.S.S. Melville	,,	32	30°	720	47·5	3·02	4·12	—	,,
Cruising Gears, U.S.S. Arizona	,,	24	30°	662	69·2	3·12	3·78	—	,,
S.T.D. 1000 Gear Kw.	,,	20	30°	625	81·5	3·66	3·85	—	,,
U.S.S. Pennsylvania	,,	22	30°	526	37 75	3·99	4·57		,,
Swedish Cruisers T.S.	,,	31	30°	825	118·0	3·61	4·13	—	,,
Prop. for U.S. Torpedo Boat Destroyer W.M. Co.	,,	39	30°	1,255	95·0	3·88	3·94	—	,,
	,,	27	30°	830	104·0	4·03	4·07	—	,,
	Rigid Gear	27	—	710	129·0	1·32 H.P. 0·497 L.P.	2·51 H.P. 1·54 L.P.	—	,,
T.S. Vespasian	C. A. Parsons	24	20°	394	31·8	3·00	4·8	98·5%	'The Engineer,' April 7, 1911.
S.S. Transylvania and Tuscania	,,	About 48	—	442	70·7	1·80	5·05	—	'Engineering,' May 26, 1916.
S.S. Ciudad de Buenos Aires and Ciudad de Monte Video	,,	About 28	—	450	62·0	2·58 H.P. 1·29 L.P.	4·5 3·18	—	,,
Rolling Mill, Calderbank Steel Works	,,	—	23°	270	62·4 24·7	3·75 4·2	—	—	'The Engineer,' April 7, 1911.
Melville-McAlpine	American Westinghouse Co.	—	30°	—	92	—	—	98·99%	'Engineering,' Dec. 3, 1909.
Melville-McAlpine Geared D.C. Gen.	,,	—	30°	—	79	2·85	—	—	'Engineering,' Jan. 31, 1913.
T.S. King Orry	C. A. Parsons	—	44° 22½′	—	—	—	—	—	'Engineering,' June 27, 1913.
	Power Plant Co.	—	—	325	100	1·49	—	—	'The Engineer,' Oct. 1, 1915.
Melville-McAlpine	American Westinghouse Co.	26·5	30°	404	110	1·86	3·93	98·5%	'Steam Turbines for Land Purposes,' Manch. Assoc. of Eng., Jan. 27, 1917.
Messrs. C. & J. Hirst	D. Brown	11	32° 12′	275	65·5	0·96	2·2	98%	
Birmingham Corporation.	Power Plant Co.	12	23°	132	46·3	0·696	2·83	—	

Hydraulic Transmission.

Prof. Fottinger has invented an ingenious system of speed reduction in which the turbine drives a centrifugal pump which delivers water into a water turbine to which the propeller or generator is coupled.

Electrical Transmission.

Various schemes of electrical reduction of speed have been suggested at various times, for which an overall efficiency of from 90 to 92 per cent. is claimed.

The General Electric Company * have installed an electrical transmission of considerable importance in the collier 'Jupiter' for the U.S.A. Navy, the following particulars of which are of interest :—

The ship is of 20,000 tons displacement, and is driven by a six-stage Curtis turbine, in which each wheel has two rows of blades and runs at 2,000 r.p.m., coupled to a bipolar 3 P.H. generator supplying current at 2,200 volts to two motor-driven propellers. The motors have 36 poles, so that the ratio of synchronous speed reduction is 18 to 1. The efficiency of the motors is 95 per cent.

At the designed speed of 14 knots the propeller speed is 110 r.p.m., and the output at the propellers 5,600 s.h.p.

Under these conditions the consumption on test was 12 lbs. per s.h.p., with a steam pressure of 190 lbs. per sq. in., dry saturated steam, and exhausting into vacuum of 28½ ins. Hg. 30 ins. Baro. Thus the thermal efficiency of the turbine generator and motors is 19 per cent., but this does not include any loss in switchgear or cables.

Variations in speed are obtained by varying the speed of the turbine, the gear ratio remaining constant, and in the process of reversal water-cooled resistances are put in circuit with the rotors of the motors.

A system which is virtually an electrical speed reduction has been introduced where direct current is desired with steam turbines as the prime movers, in which the turbine drives an alternator which is electrically coupled to a rotary converter. As compared with direct-current generators driven by double helical gearing, it appears that in plants of over 1,500 k.w. capacity the pure electrical scheme may be the more advantageous. Other considerations peculiar to a particular installation often arise demanding the electrical scheme for smaller outputs than 1,500 kw.

Very large installations of electrically driven ships have been completed for the United States Navy, and this system may possess certain advantages for such ships as operate for a large proportion of their time at speeds very much less than the maximum which they are able to develop. In such cases the turbines can run at their normal speed independently of the speed of the propeller. Further, the main turbines can be disposed in any convenient position in the ship without reference to the position of the propeller. These conditions are not important in the case of the majority of ships in the mercantile marine.

ADDENDUM.

Properties of Steel at High Temperatures.

The use of higher initial pressures and temperatures in steam turbines (and boilers) has been seriously hindered by lack of precise information on the behaviour of materials, particularly steels, at high temperatures. During recent years a good deal of work has been done on the subject, and though much still remains to be investigated, it is now possible to indicate, in general terms, the effect of high temperatures on the more important physical properties.

Ultimate and Yield Strengths.

The curves in figs. 29 and 30, due to Mr. K. Baumann,† show the ultimate tensile stress and yield point, or the limit of proportionality, for a 0·15 per cent. carbon steel and for an alloy steel, at various temperatures. It will be noted that the limit of proportionality of the carbon steel diminishes very rapidly at temperatures beyond 300° F., whilst the yield stress of the alloy steel falls off very little up to about 700° F., indicating that this steel is suitable for use at high temperatures. Mr. Baumann proposes a factor of safety of at least 2 on the proportional limit or 3 on the yield stress.

* *Engineering*, **August 23, 1912.**

† 'Some Considerations Affecting the Future Development of the Steam Cycle,' I.Mech.E., 1930.

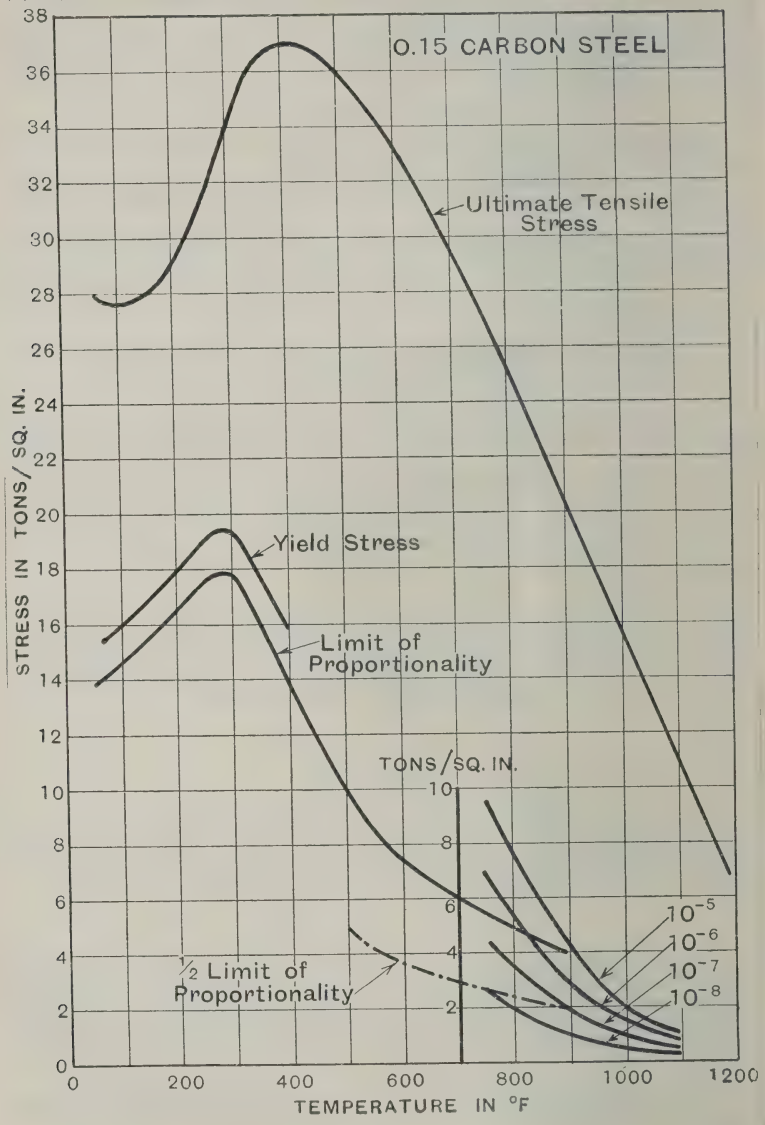

0.15 CARBON STEEL
Ultimate Tensile Stress
Yield Stress
Limit of Proportionality
½ Limit of Proportionality
TONS/SQ. IN.
10
8
6
4
2
10^{-5}
10^{-6}
10^{-7}
10^{-8}
STRESS IN TONS/SQ. IN.
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
32
34
36
38
0
200
400
600
800
1000
1200
TEMPERATURE IN °F

FIG. 29.

CREEP.

It has become recognised that materials sufficiently stressed at elevated temperatures extend continuously or 'creep.' The rate of creep depends on the stress and temperature to which the material is subjected and the character of the material itself. In order to ensure safety it is necessary to keep the rates of creep below certain defined amounts. It seems possible that creep occurs with stress at all temperatures, but increases very rapidly and at an increasing rate, with

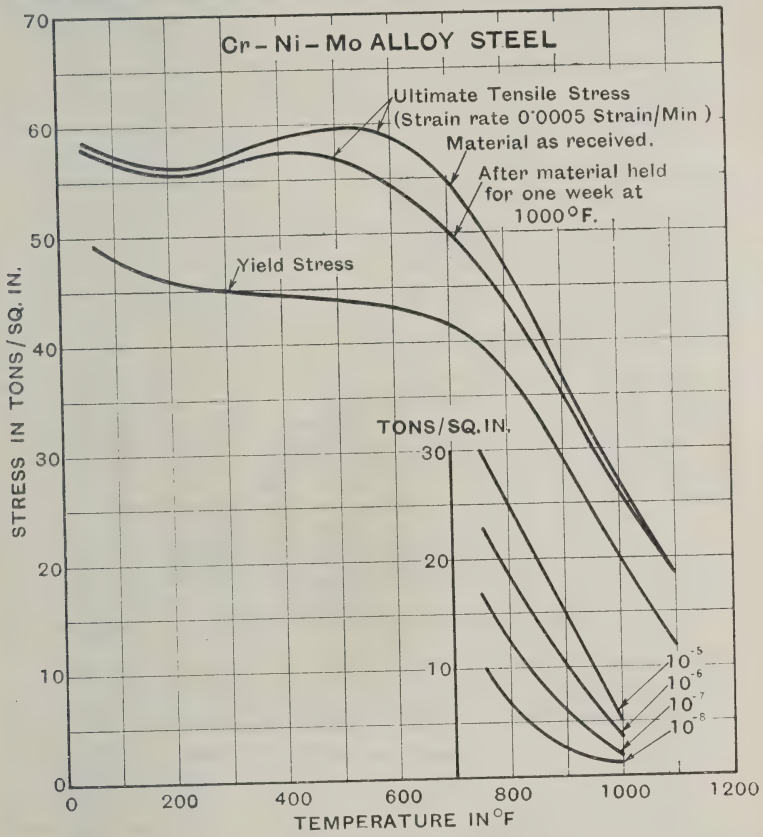


FIG. 30.

temperature and stress. It follows, therefore, that to maintain at higher temperatures the factors of safety on which design experience has been accumulated, it is necessary either to reduce the stresses to which the material is subjected, or to utilise materials known to possess high resistance to creep.

Fig. 31, produced by Mr. R. W. Bailey * from the results of his own tests and those of other investigators, shows the comparative creep resistance of medium carbon steels at temperatures from 750° F. to 1000° F., and indicates the reduction in stress necessary to maintain, for a given steel, the creep rates obtaining at 750° F.

* 'Creep of Steel under Simple and Compound Stresses,' World Power Conference, Tokyo, 1929.

In figs. 29 and 30 are shown also the probable creep strengths of the materials cited, at the higher temperatures, for creep rates of 10^{-8} to 10^{-5}. It should be noted, however, that these curves cannot yet be regarded as established but represent an intelligent anticipation of the creep properties of the materials, from which the actual properties will probably not differ greatly.

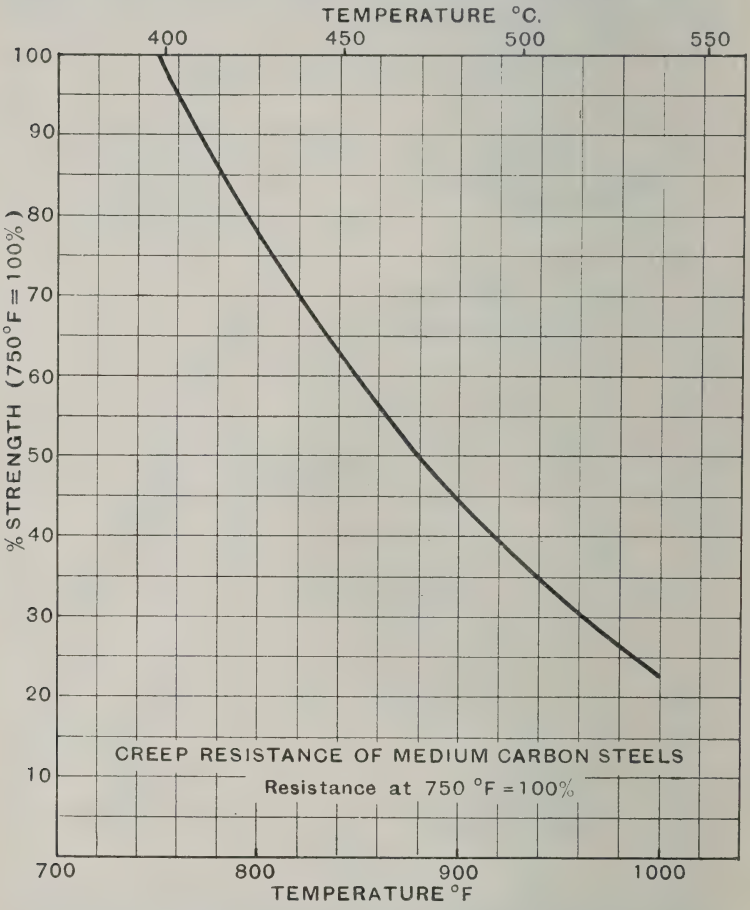


FIG. 31.

The following permissible creep rates are suggested by Baumann * for various parts of steam plant.

Turbine discs pressed on shafts	10^{-9}
Bolted flanges, turbine cylinders	10^{-8}
Steam piping, welded joints, boiler tubes	10^{-7}
Superheater tubes	10^{-6}–10^{-5}

* *Loc cit.*

It will be evident from figs. 29 and 30 that whilst in some cases permissible stresses will be limited by considerations of creep, in other cases the limit will be imposed by the necessary factors of safety on yield stress or proportional limit.

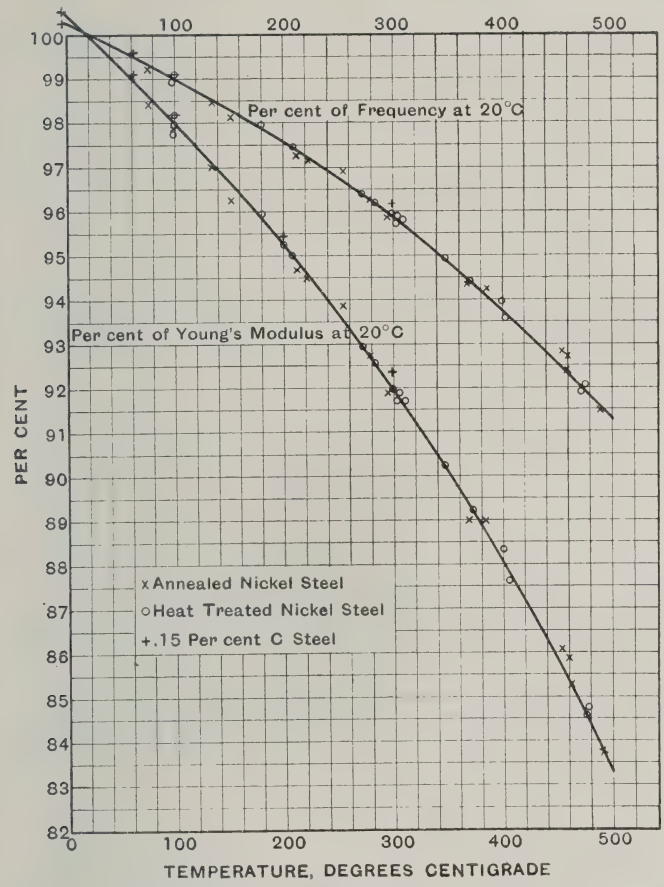


FIG. 32.

YOUNG'S MODULUS.

The variation with temperature of the Modulus of Elasticity E, is shown for various steels in fig. 32 * which gives also the corresponding change in natural frequency of vibration. At 900° F. (480° C.), for example, the value of E is only 85 per cent. of its value at 60° F., so that deflections will be 18 per cent. greater and critical speeds 8 per cent, less than at 60° F.

* Kimball & Lovell. American Physical Society, *Physical Review*, vol. 26, No. 1, July 1925.

SECTION XXX

PART V

STEAM CONDENSERS—HEAT TRANSMISSION IN SURFACE CONDENSERS—AIR PUMPS—ROTARY AIR PUMPS—STEAM JET EJECTOR AIR PUMPS—FEED PUMPS—COOLING TOWERS—STEAM ACCUMULATORS.

(Revised by T. B. Morley, D.Sc., M.I.Mech.E.)

STEAM CONDENSERS.

The principal types of condenser are:—

(1) Jet condensers, in which the condensing water mixes with the steam.

(2 Surface condensers, in which the cooling water and the steam are kept separate.

The jet type involves much less initial cost than surface condensing plant, and requires less attention and cleaning, but requires more power to operate it. The choice of type depends to a very large extent on the nature and cost of the water available for cooling and for boiler feed.

Jet condensers are preferred to surface if the cooling water is fit for boiler feed or if a suitable boiler feed, other than the condensed steam, is available.

If ample water is available, but not suitable for boiler feed, the surface condenser is advisable. In some cases the water may be too bad even for use in surface condensers, causing choking of tubes or serious corrosion; jet condensers may then be used and boiler feed obtained from a limited supply of good water, even though the price of the good water is high.

Jet Condensers.

With low-level jet plants it is necessary that the cooling water should be drawn in by virtue of the vacuum, and an efficient vacuum breaker be fitted whereby, in the event of the pumps connected to the condenser suddenly shutting down, the vacuum will be broken, thus stopping the flow of the injection water, and preventing it from entering the L.P. cylinder.

The mixture of cooling water and condensed steam is preferably withdrawn at the bottom of the condenser, and the air is taken off near the top. This involves separate air and water pumps, but in the case of engines of moderate power having their own jet condenser one pump for both air and water is commonly used. In the barometric or elevated type of jet condenser no water extraction pump is required, as the condenser being fixed at the barometric height is self-draining. Usually, however, an injection pump is required to lift the water partly up to the condenser, as the vacuum can be utilised to a certain extent for drawing the water into the condenser the same as in the low-level plant. These condensers are more expensive than the low-level type, and, due to a long exhaust pipe being necessary, require to maintain a slightly higher vacuum for an equivalent exhaust pressure at the turbine or engine exhaust flange.

The capacity of a contra-flow jet condenser should not, according to *Sim*, be less than one-quarter of the volume of the cylinder or cylinders exhausting to it, and need not be more than one-half. For quick-running engines one-third is generally sufficient. For turbine work or continuous steam flow, 0·008 to 0·011 cub. ft. per lb. of steam per hour is reasonable.

Surface Condensers.

Cooling water should be arranged to pass through the tubes and steam to flow over the tubes. It is usual to divide the tube element in two, three, or four passes according to the design of the plant, the passes being arranged so that the hottest water meets the hottest steam.

Large spaces should be allowed between the tubes at the condenser inlet to obtain a good flow of steam into the heart of the condenser. The steam flow should always be as direct as possible, changing the direction of flow should be avoided, and to ensure an evenly distributed flow over the top of the tubes a dome should be formed on the top of the condenser body.

Internal guide plates are used by some makers, in order to distribute the entering steam uniformly over the tubes.

It is important that the drop of pressure across he groups of tubes should be reduced to a minimum. Baffles in the tube space are not recommended, as their total effect is harmful.

The pitch of the tubes should be such that they will allow sufficient area for the flow of steam without an excessive pressure drop through the condenser.

The number of tubes which can be fixed in a square foot of plate is,

$$N = \frac{166}{p^2};$$

where,

N = number of tubes per sq. ft. of plate; p = pitch of tubes in ins.

Air in the condenser must not be allowed to remain stagnant round the tubes.

Frequently a section of the condenser next the air pump suction branches is partitioned off to act as a cooler for the air passing to the air pump. The first water pass takes place through the tubes in this section. The cooling of the air reduces the volume to be dealt with by the air pump.

Mr. Allen has used a short overflow pipe upstanding in the condenser so as to keep immersed the lower tubes which serve to cool the water going to the air-pump and so improve the vacuum.

Tube Plates.

These are usually of rolled brass alloy of copper 62 per cent., zinc 37 per cent., and tin 1 per cent., and are generally made of a thickness equal to tube diameter + 0·25 in.

Tubes.

These are of solid drawn brass alloy, of copper 67 per cent., zinc 32 per cent., tin 1 per cent., or where salt or corrosive water is used, of Admiralty mixture alloy, of copper 70 per cent., zinc 29 per cent., tin 1 per cent. A still better tube to use where corrosion is liable to occur is copper of 99·5 per cent. purity, but the initial cost of these is often found prohibitive. Tinned tubes should be avoided, since unless the tinning is absolutely perfect, serious pitting is liable to occur. The Admiralty practice is to adopt untinned tubes of the alloy referred to.

The unsupported length of tubes should not exceed 100 times the diameter, and where the length of the condenser exceeds this dimension, tube-supporting plates should be fixed. The tubes should be secured in the tube plates by tape packing and ferrules, the latter having an internal lip to prevent 'creeping' of the tubes.

Ferrules should allow a space for free movement of the tube ends, and should be rounded, so as to reduce to a minimum the loss of head of the water at its entry into the tubes.

According to Stanton, in a condenser of a given surface for maximum efficiency, the tube length should be as great as possible. As regards tube diameter he says that condensers will be equally efficient in which the ratio $\frac{l^{n-1}}{d}$ = constant, where l = length in feet, n = number of tubes, d = external diameter of tube in feet. Tube diameters should be chosen which offer the least aggregate resistance to flow. Resistance is, he finds, proportional to l^{4-2n}, and is therefore least when the tubes are the smallest practical diameter. He considers the best position for tubes is vertical and the water movement downward.

Tube diameters may vary from $\frac{1}{2}$ in. to 1 in., but are usually $\frac{5}{8}$ in. or $\frac{3}{4}$ in., in fact $\frac{3}{4}$ in. diameter and 18 S.W.G. (0·048 in.) thickness are practically standard.

Unannealed hard-drawn condenser tubes tend to become corroded more readily than tubes that have been annealed.

Tube Packing.

(*Asbestos Cement Building Products Ltd.*)

'Compressed Asbestos Fibre Tube Gromets' (*Turner Brothers Asbestos Co., Ltd., Rochdale*) are illustrated in fig. 1. The Gromets are essentially for packing the tube ends of surface condensers,

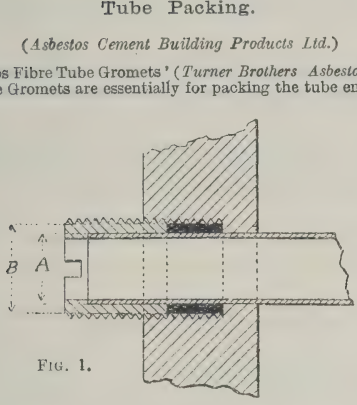


FIG. 1.

feed-water heaters, etc.; they are a sliding fit between the tube and the tube plate, making a perfectly tight joint with a few turns of the screw, after which no 'following up' is necessary.

When ordering, the dimensions at 'A' (the outside diameter of tube) and at 'B' (the apex of thread in socket), together with the length of Gromet required, should be given; the necessary clearance is allowed for in manufacture.

BRITISH STANDARD SPECIFICATION FOR CONDENSER TUBES AND SCREWED GLANDS FOR CONDENSERS FOR MARINE PURPOSES.

(No. 3000—1921).*

(*Abstract.*)

1. The material of the shells from which the tubes for condensers and for screwed glands are drawn shall be an alloy of copper and zinc as follows:—

For Condenser Tubes: 70/30 Alloy.—Containing not less than 70 per cent. of metallic copper and not more than a total of 0·75 per cent. of materials other than copper and zinc (except that, if so specified, it may contain 1 per cent. of tin or 1½ to 2 per cent. of lead).

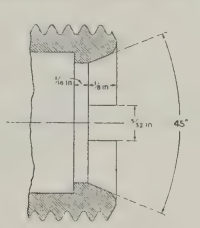


FIG. 2.—View of Outer End of Screwed Glands for ⅝ in. and ¾ in. External Diameter Condenser Tubes.

* By permission of the British Engineering Standards Association.

For Screwed Glands: 60/40 Alloy.—Containing not less than 60 per cent. of metallic copper and not more than a total of 0·75 per cent. of materials other than copper and zinc (except that, if so specified, it may contain 1 per cent. of tin or 1½ to 2 per cent. of lead).

3. The condenser tubes shall be of ⅝ in. or ¾ in. external diameter as may be specified—and they shall not be larger than, nor more than 1½ per cent. smaller than, the specified external diameter. The tubes shall be of uniform diameter throughout and the thickness shall be 18 S.W.G. within the usual manufacturing margins.

5. For ⅝ in. external diameter condenser tubes the average weight per 100 ft. run shall be not more than 32¾ lbs. and not less than 31¼ lbs., and for ¾ in. external diameter condenser tubes the average weight per 100 ft. run shall be not more than 40 lbs. and not less than 38 lbs., *i.e.* 2½ per cent. above or below the theoretical weight.

6. Each condenser tube shall be tested internally by water pressure to 600 lbs. per square inch.

Condenser Tube Corrosion.

See 'Eighth Report to the Corrosion Research Committee of the Institute of Metals.' Abridged Report, '*Engineering*,' Sept. 7, 1928.

Heat Transmission in Surface Condensers.

Let K = coefficient of heat transmission from steam to cooling water, *i.e.* the number of B.Th.U. transmitted per hour per sq. ft. of cooling surface per degree F. of mean temperature difference.

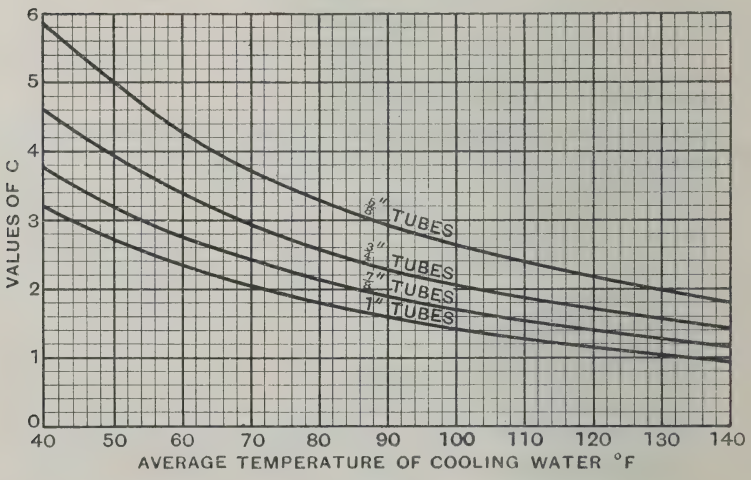


Fig. 3.

$\frac{1}{K}$ represents the resistance to the flow of heat. This resistance is made up of :—

(1) The resistance to the flow of heat from the steam to the external surface of the condenser tube, $\frac{1}{k_1}$ say;

(2) resistance to conduction through the tube $= \frac{t}{k_2}$, where t = thickness of tube in inches and k_2 = conductivity of metal in B.Th.U. per hour per sq. ft. per temperature gradient of 1° F. per inch ; and

(3) the resistance to flow of heat from the tube to the water $= \frac{1}{k_3}$ say.

The total resistance $\frac{1}{K} = \frac{1}{k_1} + \frac{t}{k_2} + \frac{1}{k_3}$.

For brass tubes, 18 S.W.G. thick, the middle term $\frac{t}{k_2}$ is negligible in comparison with the others.

k_1, the coefficient for flow from steam to metal, depends on the velocity with which the steam flows over the tubes, the presence of air in the condenser, and the cleanliness of the tubes.

k_3, the coefficient for transmission from tube to water, is the most important term. It depends upon the water velocity, the diameter of the tube, cleanliness of tube surface, and also on the mean temperature, there being evidence that the heat transmitted is greater at higher temperatures even though the temperature *difference* is the same.

For the usual tube sizes and usual temperatures, it will be seen that the chief variable factor influencing the overall transmission rate is the water velocity.

Data for k_1 and k_3 are not yet thoroughly established, and the data given by various experimenters for the coefficient K also vary widely. This is due not only to experimental error but to the fact that frequently some of the factors influencing the transmission rate are not taken into account at all.

The resistance to heat flow may be expressed as—

$$R = R_a + R + R_t + R_w,$$

where

R_a is due to the effect of air blanketing the surface of the condenser tubes ;

R depends on the viscosity and density of the film of condensate clinging to the tubes ;

R_t is the resistance to conduction through the metal ;

R_w depends on the velocity of the condensing water in the tubes, its density and viscosity, and the tube diameter.

Sim uses units such that $R = \frac{1000}{K}$, where K is defined above.

In these units $R_a + R_f$ together may be taken as 0·6 to 1·2 for turbine plants, and 1·2 to 1·5 for reciprocating engines.

R_t for clean tubes 18 S.W.G. thick may be taken as 0·08. It increases considerably with dirty tubes.

R_w for a particular tube diameter $= \frac{C}{v^{0 \cdot 82}}$, where v is the water velocity in feet per sec.

Values of C are shown by fig. 3.*

The increased transmission rate—and consequently reduced cooling surface necessary—due to high water velocities is not all gain, however, for high velocity means increased power to circulate the water. The final design must be decided with reference to cost of pumping as well as initial cost of plant.

* Redrawn from Sim's *Steam Condensing Plant* (Blackie & Son, Ltd.), which work provides a full exposition of this important subject.

Fig. 4 shows the head lost in friction per 10 foot of tube element for $\frac{5}{8}$ in., $\frac{3}{4}$ in., and 1 in. external diam. by 18 S.W.G. smooth brass tubes. For application it is necessary to take the total length of the tube element (the length of the tubes multiplied by the number of water passes) and add the 'loss at entry (including loss at exit also) per pass,' values for which are given on the bottom curve.

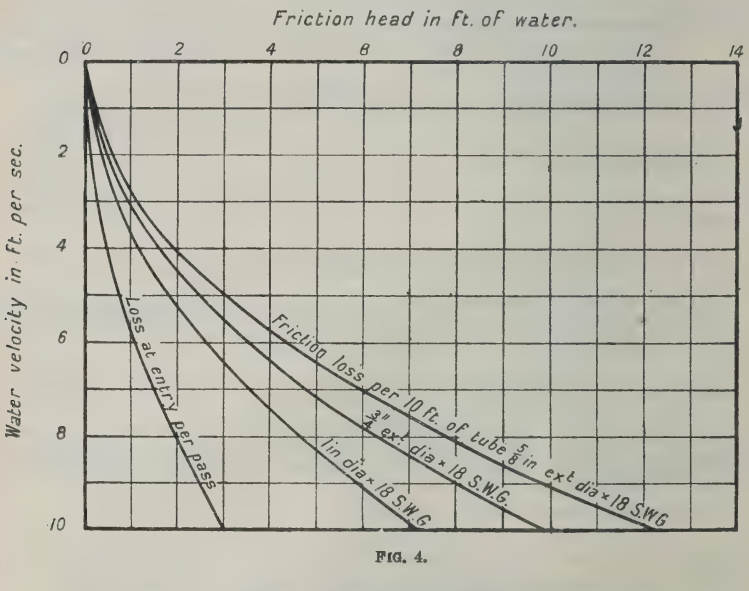


FIG. 4.

MEAN TEMPERATURE DIFFERENCE.

The true mean temperature difference for condensers, heaters, coolers, etc., in which heat is transmitted between two fluids separated by a metallic wall is not the difference of the arithmetic mean temperatures.

If one of the fluids is cooled from T_1 to T_2 and the other heated from t_1 to t_2, then,

$$\text{mean temperature } t_m = \frac{(T_1 - t_1) - (T_2 - t_2)}{\log_e \frac{T_1 - t_1}{T_2 - t_2}}$$

Fig. 5 enables t_m to be readily determined from the maximum and minimum temperature differences $T_1 - t_1$ and $T_2 - t_2$.

COOLING SURFACE.

If

t_m = mean temperature difference between steam and water; t_1 = inlet water temperature; t_2 = outlet water temperature; T = steam temperature in condenser; K = heat transmission B.Th.U. per hour per sq. ft. per F.°; then,

$$\text{Cooling surface per lb. steam per hour} = \frac{1{,}050}{K \times t_m}$$

(1,050 is the number of B.Th.U. given up by 1 lb. steam in the condenser, under usual conditions).

For rough approximations, surface condenser cooling surface may be based on 6 lbs. of steam per square foot of surface; 10 pounds is sometimes adopted, but is too severe unless the temperature of the cooling water is low, say 60° F., or less. For tropical countries the lower rate of condensation should be taken, and even less when very hot condensing water is anticipated.

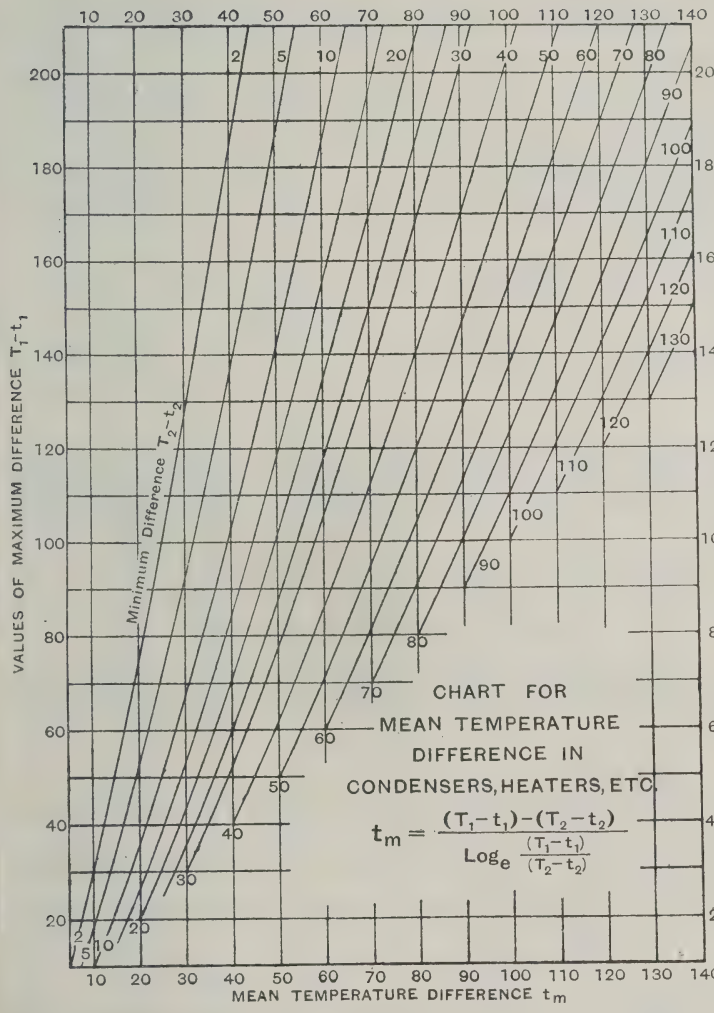

VALUES OF MAXIMUM DIFFERENCE T_1-t_1
MEAN TEMPERATURE DIFFERENCE t_m
Minimum Difference T_2-t_2
CHART FOR
MEAN TEMPERATURE
DIFFERENCE IN
CONDENSERS, HEATERS, ETC.
$t_m = \frac{(T_1-t_1)-(T_2-t_2)}{\text{Log}_e \frac{(T_1-t_1)}{(T_2-t_2)}}$

FIG. 5.

The usual proportions adopted for various services in machinery equipped with air-extracting apparatus of customary type and capacity are shown in the Table below, the figures given being basis of power output at full load.

(Petty, *Power Engineer*, Aug. 1929.)

Type of Machinery.	Cooling Surface in sq. ft.			Steam condensed per sq. ft. per hour. Lbs.
	Per I.H.P.	Per S.H.P.	Per kW.	
Mercantile marine reciprocators	1·0–1·5	1·1–1·7	—	9·5–14
Single-screw combination machinery for cargo vessels	1·3	1·5	—	8·25
Geared-turbine steamships	—	0·845–1·25	—	8·5–10·5
Geared-turbine machinery for destroyers	—	0·4–0·525	—	25–30
Geared-turbine machinery for cruisers and battleships	—	0·45–0·75	—	13·5–22·5
Land power stations (large plants)	—	1·4–2·5 (Alternator efficiency, 96 per cent.)	1·0–1·8	5·375–12·375
Waterworks pumping engines	1·25–2·0	—	—	6·5–13

Quantity of Cooling Water.

If T_1 = temperature of cooling water at inlet to condenser; T_2 = temperature of cooling water at outlet of condenser; T_3 = temperature of steam at inlet, corresponding to the vacuum; T_4 = temperature of the condensed water at outlet; h_3 = sensible heat at inlet; h_4 = sensible heat of the condensed water at outlet; L = latent heat in B.Th.U. per lb. of steam; W = steam weight; Q = water weight; R = ratio of water and steam quantity; x = dryness fraction; then,

Heat (in B.Th.U.) given up by exhaust steam $= W\left\{xL + (h_3 - h_4)\right\}$;

Heat (in B.Th.U.) gained by cooling water $= Q(T_2 - T_1)$.

In practice $h_3 - h_4$ may be neglected, therefore

$$WxL = Q(T_2 - T_1), \quad \text{and,} \quad \frac{Q}{W} = R = \frac{x\,L}{T_2 - T_1}.$$

To allow a small margin, x—which generally varies between 0·8 and 1·0—can be neglected, and as L only varies slightly with the vacuum, it is generally assumed to be a constant for all vacua at 1,050 B.Th.U. per lb. of steam. Therefore the general equation for all vacua can be taken as

$$R = \frac{Q}{W} = \frac{1050}{T_2 - T_1}.$$

For surface plants, T_2—T_1 ranges from 8° to 20° F., and for jet plants, 4° to 8° F.

The ratio of the amount of cooling water to that of the steam to be condensed usually varies from 35 to 40 for jet, and from 60 to 70 for surface condensers.

It is important that the condensed steam should be at as high a temperature as possible, since this water is usually returned to the boilers, therefore it is essential that all the heat possible should be recovered.

Every 8°–10° F. difference in the hot-well temperature means approximately 1 per cent. difference in fuel consumption. The gain obtained with an increase in hot-well temperature can be calculated as follows :—

$$\text{Percentage gain in B.Th.U.} = \left(\frac{100\,T}{H_T + T - T_4 + 32}\right) \times \frac{100}{B_p}$$

where,

T = difference in hot-well temperature (F.°); T_4 = final hot-well temperature (F.°); H_T = total heat of steam at boiler pressure (B.Th.U. per lb.), reckoned from 32° F.; B_p = percentage boiler efficiency.

In surface condensers for turbines, $(T_3 - T_4)$ is about 5° to 8°, and for reciprocating engines about 10° to 15°.

Vacuum Efficiency may be expressed as the ratio of the vacuum at the condenser inlet to vacuum corresponding to the temperature of the condensed steam; it is dependent both upon the design of the condenser and the air pump.

The *Efficiency of a Condenser* is best stated as a 'Coefficient of Performance,' which measures the extent to which the two conditions are fulfilled that (1) the condensate should be cooled as little as possible below the steam temperature, and (2) the quantity of cooling water should be a minimum, and hence its rise of temperature a maximum.

Sim takes as 'Coefficient of Performance' the ratio $(T_2 - T_1) \div (T_3 - T_4 + 10)$, and states that it should be 1·0 to 1·2 for good surface condensers.

Degree of Vacuum.—The maximum vacuum obtainable is dependent upon the inlet circulating water temperature. For engine work 26 ins. to 27 ins. is the maximum vacuum desirable, owing to the limitations in size of the low-pressure cylinder and ports, etc. For turbine work high vacua are necessary. The most economical vacuum depends upon individual installations, and is generally decided by the power required to be expended by pumping the cooling water, price of coal, available water supply, etc.

The following may be taken under average conditions to be the maximum commercial vacua.

Temp. of inlet water, F.°	55	60	65	70	75	80	85
Vacuum in ins. (baro. 30 ins.)	28·8	28·6	28·5	28·3	28·0	27·8	27·5

PROPERTIES OF SATURATED STEAM AT LOW PRESSURES.

Per Increment of Vacuum reduced to 30 ins. Barm.

Inches of Vacuum.	Absolute Pressure Lbs. per Sq. In.	Temp. °F.	Spec. Vol. Cub. Ft. per Lb.	Inches of Vacuum.	Absolute Pressure Lbs. per Sq. In.	Temp. °F.	Spec. Vol. Cub. Ft. per Lb.
0	14·69	212	26·79	25	2·45	133·8	143·5
1	14·20	210·3	27·64	25·2	2·35	132·1	149
2	13·71	208·5	28·54	25·4	2·25	130·5	155
3	13·22	206·7	29·52	25·5	2·21	129·8	158·5
4	12·73	204·9	30·61	25·6	2·16	128·9	161·5
5	12·24	202·9	31·75	25·8	2·06	127·2	169
6	11·75	201	32·95	26	1·96	125·3	177
7	11·26	198·9	34·37	26·1	1·91	124·4	181
8	10·77	196·8	35·80	26·2	1·86	123·4	185·5
9	10·29	194·6	37·35	26·3	1·81	122·4	190
10	9·80	192·3	39·1	26·4	1·76	121·4	195·5
10·5	9·56	191·1	40	26·5	1·71	120·4	200·5
11	9·31	189·9	41	26·6	1·67	119·6	206
11·5	9·07	188·7	42	26·7	1·62	118·4	212
12	8·82	187·4	43·15	26·8	1·57	117·3	218·5
12·5	8·58	186·1	44·25	26·9	1·52	116·2	225
13	8·33	184·8	45·5	27	1·47	115·1	232
13·5	8·09	183·4	46·8	27·1	1·42	113·9	239·5
14	7·84	182	48·2	27·2	1·37	112·6	247
14·5	7·60	180·6	49·6	27·3	1·32	111·3	255
15	7·35	179·1	51·2	27·4	1·27	110	265
15·5	7·11	177·6	52·8	27·5	1·23	108·7	276
16	6·86	176·1	54·6	27·6	1·18	107·3	286
16·5	6·62	174·4	56·4	27·7	1·13	105·8	298
17	6·37	172·8	58·3	27·8	1·08	104·3	310
17·5	6·13	171·1	60·5	27·9	1·03	102·7	324
18	5·88	169·3	63	28	0·98	101·1	339
18·5	5·64	167·4	65·5	28·1	0·93	99·4	356
19	5·39	165·5	68·4	28·2	0·88	97·6	374
19·5	5·15	163·5	71·4	28·3	0·83	95·7	395
20	4·90	161·4	74·7	28·4	0·79	93·8	418
20·5	4·66	159·3	78·4	28·5	0·74	91·7	444
21	4·41	157	82·5	28·6	0·69	89·5	474
21·5	4·17	154·7	86·9	28·7	0·64	87·1	508
22	3·92	152·2	92·1	28·8	0·59	84·6	548
22·5	3·68	149·6	97·7	28·9	0·54	81·9	597
23	3·43	146·8	104·3	29	0·49	79	653
23·5	3·19	143·9	112	29·1	0·44	75·8	721
24	2·94	140·7	121	29·2	0·39	72	806
24·5	2·70	137·4	131	29·3	0·34	68	913
24·6	2·65	136·7	133·5	29·4	0·29	63	1060
24·8	2·55	135·2	138·3	29·5	0·25	59	1230

Air in Condensers

Air enters the condenser in the following manner :—

1. With the steam entering from the boiler having been dissolved in the feed water ; this is usually a small amount in the nature of 1 to 3 per cent. of the total volume of feed water.

2 Air leakage from the atmosphere through the glands of the prime mover or other sources ; this cannot easily be measured, and an allowance has to be made accordingly to cover contingencies.

3. In jet condensers air in solution with the cooling water which is released when in the vacuum space; this varies from 1 to 5 per cent. of the total volume of water.

Due to air contained in the cooling water in the case of a jet condenser, a much larger air pump is required than for a surface plant of similar duty.

For turbine work to cover cases (1) and (2) it is usual to allow for 6 to 8, and, in the case of low-pressure turbines, up to 11 lbs. of air per 10,000 lbs. of steam.

For engines, about two or three times this amount is required according to the type of engine.

These figures are for one unit only ; if two or more turbines or engines exhaust to the one condenser an extra margin of 10 or 15 per cent. per extra turbine or engine, respectively, should be allowed.

(See also page 797.)

The Ejector Condenser.

This condenser is at once an air-pump and a condenser. It acts on the principle that steam will flow into a vacuum at a velocity of many hundred feet per second. The water supply to the ejector should flow to it under a head of several feet. The steam condenses in the jet of water, and the combined jet has a velocity which is a mean of the velocity of the steam and of the water. Thus let the water approach at a velocity 30 feet per second and the steam at 1,200 feet per second. Let the combined jet be assumed to have a temperature of 100° F. and the water to be supplied at 60° F. The water gains 40° F. The steam loses say 1,100 B.Th.U. Then $1{,}100 \div 40 = 27$ = ratio of water weight to weight of steam. The total weight of combined jet is 28. The velocity difference is $1{,}200 - 30 = 1{,}170$, to be divided over a mass of 28, or $\frac{1{,}170}{28} = 42$ feet per second nearly. Add the initial velocity of the water 30, and the final velocity is 72 feet per second.

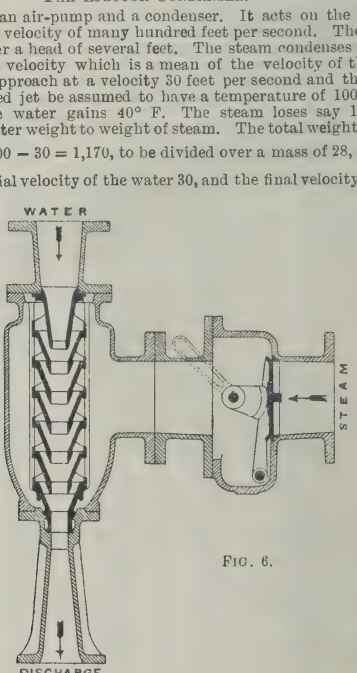


Fig. 6.

This corresponds with a head of $H = (72 \div 8)^2 = 81$ feet ; and since 1 foot of water column equals 0·43 pounds, 83 feet will represent 36 pounds nearly. Since a perfect vacuum is only 14·7 pounds, it follows that with ordinary ratios of water and steam an efficiency of less than 50 per cent. should give an excellent vacuum.

The apparatus (fig. 6) consists of a converging water nozzle into which water is directed from a centrifugal pump or other pressure source, or from an overhead source of 15 to 20 feet head. This jet flows downwards through a series of steam cones to which the exhaust steam has admission at their upper wide portion. The steam flows down each of these short cones and

imparts further velocity to the jet as it condenses, and finally the jet flows into a divergent discharge cone at a velocity as above indicated, carrying with it any air that has come in with the steam, and discharging it. The degree of vacuum is of course that proper to the jet temperature, and the proportion of water can, as seen, be sufficiently great to admit of a moderate final jet temperature and therefore a good vacuum. The ejector condenser occupies very little space and for jet condensing is very simple. The centrifugal water supply pump is conveniently direct driven by an electric motor. While the jet final temperature is amply low for good vacua, the velocity of the combined jet is still more than sufficient to run freely against atmospheric external pressure; but it is always desirable to give the water a fair initial velocity, and there is nothing lost by giving the discharged jet a good fall in a long tail pipe.

The Barometric Condenser is a simpler form of ejector condenser, in which the water enters the head of the apparatus as a spray about the steam entrance, and the vacuum is produced by the fall of the water down a tail pipe of at least 30 feet in vertical drop. By suitable formation of the head the air which enters with the steam is entrained with the water, or, by means of a special surface condenser at the side of the head the air is drawn over cold tubes and carried down to a lower point in the draught tube, whereby it is withdrawn from the head at a minimum temperature, with the object of securing a maximum degree of vacuum.

Mirrlees-Delas-Ginabat Surface Condenser.

(Patented.)

(The Mirrlees Watson Co., Ltd., 45 Scotland St., Glasgow.)

In this condenser it is claimed that every tube does its equal share of work. The tube area opposed to the inlet steam is very large and the steam path through the tubes very short; the tubes are particularly free from air or water blanketing, and the kinetic energy in the steam and

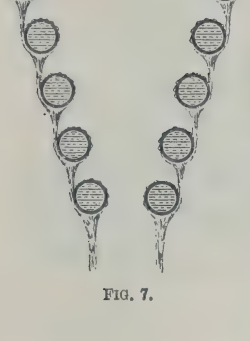


Fig. 7.

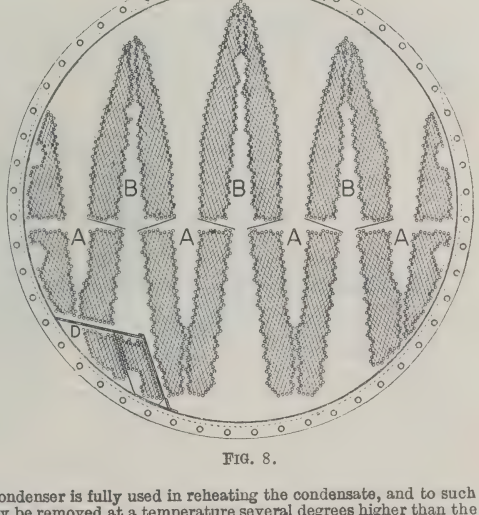


Fig. 8.

condensate passing through the condenser is fully used in reheating the condensate, and to such an extent that the condensate may be removed at a temperature several degrees higher than the inlet steam temperature, and an absolute pressure at the bottom of the condenser slightly higher than the absolute pressure at the condenser inlet; this is a distinctive feature of the arrangement.

Fig. 7 shows the pitching of the tubes. By the arrangement the water of condensation falling from any one tube meets the tube immediately below tangentially to its surface, and thus flows over only $\frac{1}{4}$ of its circumference, leaving the other $\frac{3}{4}$ dry as far as falling water is concerned. Also the water of condensation may be made to travel to the right or left, according as to whether the lower tube is arranged to the right or left of the higher tube.

The method of grouping the tubes is shown in fig. 8. The tubes are arranged in shallow banks, leaving very deep V-shaped openings for entry of steam, thus giving maximum area of contact; the V-shaped openings being carried almost to the bottom of the condenser, all tubes are equally effective. There are also correspondingly deep inverted V openings on the under side.

As stated, the condensate can be made to traverse a tube path into this centre inverted V space, and, doing so, passes over a very small number of tubes and then over only ¼" of their circumference

As there is very little resistance to the steam and incondensible vapours passing through these shallow banks of tubes, the kinetic energy is used in giving a compression effect to the incondensible gases at the bottom of the condenser, and an increment of heat to the condensate: the compression effect is enhanced by the unrestricted fall of condensate through the inverted V-shaped openings.

The heat transmission of the condenser is very much higher than anything previously recorded in practice; in several installations a transmission of 1,000 B.T.U.'s per sq. ft. per deg. temperature difference has been attained.

A recent test gave the following results:

Steam condensed in lbs. per hour	442,000
Steam condensed per sq. ft. cooling surface	16·3 lbs.
Absolute pressure at condenser inlet	0·81 in. Hg.
Temperature of inlet circulating water	45·5° F.
Temperature of outlet circulating water	55·9° F.
Heat transmission B.Th.U./sq. ft./deg. Fah. temp. diff./hr. =	756

The Mirrlees Extraction Pump.

The Mirrlees condensate extraction pump is designed for withdrawing the condensate from a condenser at high vacuum. Formerly centrifugal extraction pumps had to be placed 5 or 6 ft. below the condenser and the prevention of air leakage into the pump at the spindle glands presented difficulties. Both these objections are overcome in the pump to be described which is shown in section in fig. 9.

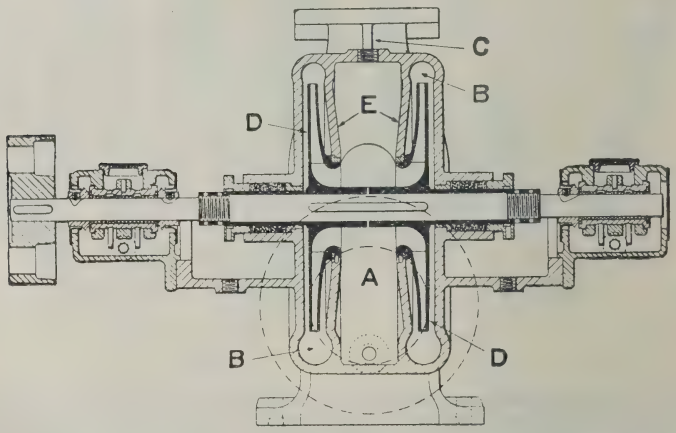


Fig. 9.

There are two impellers with inlets facing each other. Each deals with half the total water and the end-thrusts balance each other.

The impellers discharge into volutes B. The spaces between the impellers and the outer sides of the casing are in communication with the volutes, and therefore the pressure at the inner ends of the stuffing boxes is that of the discharge head.

Instead of air leaking into the pump at the stuffing boxes, water tends to escape outwards. Grease or oil from the packing does not enter the pump and contaminate the condensate. The packing of the stuffing boxes is a simple operation.

The casing is split to facilitate inspection, and the pump can be opened up without any vacuum joints requiring to be broken or the pump alignment being disturbed.

These pumps are designed for speeds of 480 to 1,500 r.p.m.

For higher heads than can be dealt with by a single stage pump, a central diaphragm is introduced and the impellers work in series, the pump thus working as a two-stage pump.

The 'Weir' Regenerative Surface Condenser.

(G. & J. Weir, Ltd., Cathcart, Glasgow.)

This condenser has been introduced to eliminate the temperature loss which exists in the condensate from the vacuum temperature in normal type condensers, and also to give a condenser which delivers its condensate practically air or oxygen free, consequently, although this condenser is suitable for any feed system, it is pre-eminently suitable for fitting to a closed circuit feed water system.

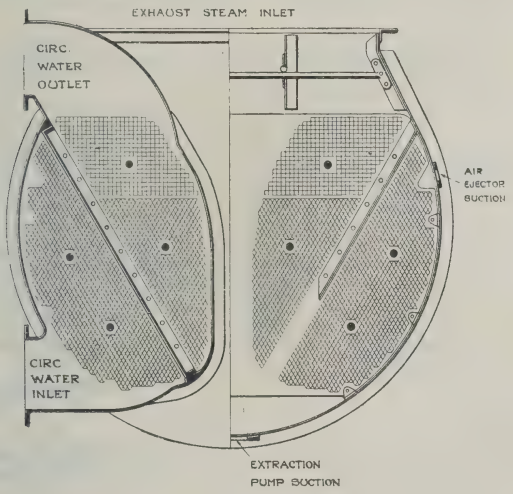


Fig. 10.

The design is the result of a series of researches carried out by Messrs. G. & J. Weir, and in its present form is shown in fig. 10.

It will be seen that a large lane is provided down the middle of the condenser, so that steam can penetrate right down to the bottom; this steam re-heats the condensate dropping from the lowest row of tubes to practically steam temperature. Tests from actual plants show that the difference between the condensate temperature and the vacuum temperature does not vary by more than 2° F. This result is independent of the air leakage present. These tests also show that the oxygen contained in the condensate can reach as low as zero oxygen, 0·015 c.c. per litre being quite a normal figure.

A hood plate is fitted on each side of the condenser, thus giving two reflex flows on the steam side; this arrangement permits a large surface to act as an air cooler if large air quantities are present, while if the latter are small the air cooling surface will be small. This action is automatic, the surface acting as an air cooler, automatically adjusting itself to the air quantity. This property results in the air being drawn off by the air-pump at a minimum temperature for the conditions and resulting in an enhanced condenser performance. The double arrangement shown lends itself to the twin water box design favoured by many present-day land engineers, the idea being that one-half can be shut down for cleaning or tube plugging while the other half is working, so that the main turbine need not be shut down.

The shape of the centre lane, together with the position and length of the hood plate, permit of a large entrance area being presented to the entering steam, with a gradually reducing area to the air suction, so that all the surface is equally effective and a minimum of pressure drop is attained.

The 'Weir' Water Extraction Pump and Air Ejector.

(G. & J. Weir, Ltd., Cathcart, Glasgow.)

To work in conjunction with their 'Regenerative Condenser,' Messrs. G. & J. Weir manufacture a water extraction pump and air ejector. The water extraction pump is of the vertical spindle type, driven by a steam turbine of the velocity stage type having three rows of moving blades mounted in a single wheel and two rows of stationary blades mounted in a guide segment; the pump is shown in fig. 11. To maintain the speed constant under variations of load a speed governor

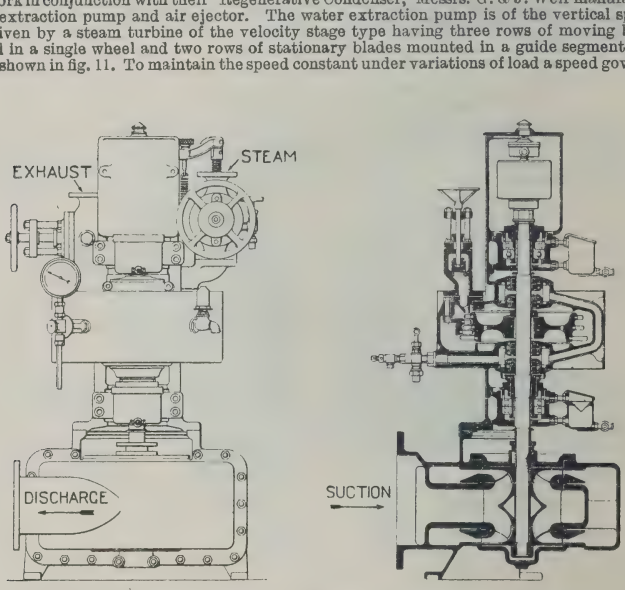


Fig. 11.—Arrangement of Weir Vertical Turbine Extraction Pump Turbine.

is fitted, while in addition a safety governor is fitted which shuts off the steam completely in the event of the pump overspeeding. Auxiliary nozzles under control of the auxiliary stop valves are provided, so that by regulating these the pump may deal economically with large variations in load.

The impellers are located low down at the base of the pump to enable the pump to work satisfactorily under the minimum suction head; the impellers are of the twin type operating in parallel and having the suction placed between the impellers; whirlpool chambers of the diverging

type are employed, and no stuffing gland is fitted at the lower impeller, while a large clearance is provided between the top impeller and the stuffing gland, so that the full discharge pressure reaches the stuffing box, thus preventing air leakage into the pump suction.

The weight of the impellers, turbine wheel, shaft, etc., is carried on a ball thrust bearing, while the shaft is centred by two roller bearings; both thrust and journal bearings run in an oil bath.

The pump end is made of gun-metal, while the framing is of cast iron. The framing and pump end are divided down the centre line, so that the shaft with impellers and turbine wheel can be removed for overhaul without disturbing any of the pipe connections.

The most modern type of ejector as developed by Messrs. Weir is shown by fig. 12. It has three stages and can attain the highest vacua. On dry air tests these ejectors have attained vacua within 0·02 in. of the barometer; such extreme vacua, however, is seldom called for commercially, but this property enables them to reduce the partial air pressure in normal plants down to the very lowest limits. The saving in steam consumption of the three-stage type over

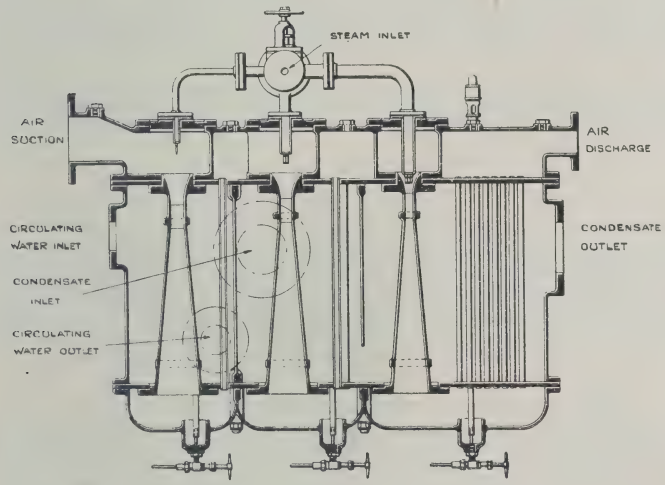


FIG. 12.

the two-stage type is about 30 per cent. at 29 in. vacuum, and increases with higher vacuum. The three stages are generally similar in design, but are proportioned to suit the actual air, steam, and vapour volumes reigning at the respective stages. The steam nozzles are normal convergent, divergent steam ones, wherein the pressure energy of the steam is converted into kinetic energy; the steam issuing from the nozzles at high velocity (over 3,000 ft. per sec.) is delivered into a convergent, divergent diffuser nozzle; entrainment of the air and vapour takes place in the convergent portion of the diffuser, and the pressure rise takes place in the diverging portion; the combined mass of operating steam and entrained air and vapour pass to first intercooler, which is of the surface type and is arranged alongside and partly embraces the diffuser; the operating steam and vapour are condensed out in this cooler, while the air and its associated vapour pass on to the second stage, where the action is repeated, and so on to the third stage, the air being finally discharged to atmosphere.

The steam nozzles are of Monel metal to resist erosion, while the diffusers are of hard gun-metal.

On board ship the first and second intercoolers are circulated with sea-water, while the third or after-cooler is circulated with condensate. On land power stations only the first intercooler is circulated with cold circulating water, the second and third coolers being circulated with condensate. On board ship it is imperative that a full vacuum be maintained during manœuvring,

and to enable the air ejector to do this the first and second stage coolers are circulated with sea-water, while the third stage is arranged that when insufficient condensate is circulating the operating steam simply blows through the after-cooler to a sea-water circulated ejector condenser wherein the steam is condensed. This arrangement is automatic, requires no attention from the ship's engineers, and has been found to work well in practice. On land power stations the above arrangement is not called for, but, owing to the condensate being delivered from regenerative condensers at approximately vacuum temperature, it is imperative to circulate the first inter-cooler with cold circulating water to keep down the surface in the intercooler. With this arrangement the heat in the operating steam of the first stage is lost in the cooling water; but since the steam used in this stage is less than 10 per cent. of the total steam used by the ejector, the actual heat lost is very small indeed.

The vertical design of intercoolers allows of the maximum air cooling and devaporising effect, so that the air is cooled and dried to a maximum for the conditions before it passes on to the next stage.

Between the discharge of the second stage and the inlet to the third stage a light plate non-return valve is fitted. This valve has the effect of stabilising the vacuum produced by the third stage should the steam pressure fall below the designed figure. It also allows of the full vacuum being retained should the ejector steam valve require to be shut for any reason whatever; each intercooler is drained separately to the main condenser, the after-cooler drain passing through a trap.

For land work and mercantile marine work these ejectors are manufactured with cast-iron bodies, steel or brass tube plates, copper tubes, gun-metal diffusers, Monel metal nozzles, while for fast naval craft the bodies can be aluminium alloy, brass tubeplates, Monel metal tubes, gun-metal diffusers, Monel metal nozzles.

Air Pumps. (See also p. 2408.)

The essential feature of a plant having a high vacuum efficiency is an air-pump capable of operating without cooling the condensed steam. Excepting in small plants where the heat loss is not so appreciable, all surface condensers should be arranged with two pumps working on the vacuum side, *i.e.* an air-pump capable of dealing with all the air entering the condenser and a water-pump capable of dealing with all the condensed steam. This is what is known as a 'dry' system. Cases where only one pump is employed for the combined duty are known as a 'wet' system. In order to cut down the size of the air-pump in the latter system to reasonable proportions, the condensed steam is usually cooled by means of the last few rows of tubes in the condenser being arranged so that they are always flooded, or by an intercooler.

Capacity of Air-Pump in order to Remove given Amount of Air.

The diagram (fig. 13, p. 1841) shows the properties of mixtures of air and steam such as exist at the air-pump suction. Thus, if the temperature is 100° F., and the vacuum 26 ins., the volume of 1 lb. of air (together with the associated vapour) is 204 cu. ft. and the weight of air per lb. of steam is 1·75, *i.e.* 204 cu. ft. of the mixture contains 1 lb. of air and 0·57 lb. vapour.

The scale on the right provides a basis for calculating the air-pump capacity if the air leakage and the temperature and vacuum are known. For example, assume a surface condenser installation for a reciprocating engine in which allowance is to be made for 16 lbs. air per 10,000 lbs. steam, vacuum 27 ins., temperature at air-pump suction 100° F. From the right-hand scale the volume of air per lb. of condensate, measured at the conditions of the air-pump suction, is 0·395 cu. ft. for an air proportion of 10 lbs. per 10,000 lbs. steam. Thus in the present case the volume is

$$0{\cdot}395 \times \frac{16}{10} = 0{\cdot}63 \text{ cu. ft. per lb. steam.}$$

If the steam condensed is 15,000 lbs. per hour, the *effective* capacity of the 'dry' air-pump will then be

$$\frac{15{,}000}{60} \times 0{\cdot}63 = 157 \text{ cu. ft. per minute.}$$

For a 'wet' air-pump the volume of the condensate must be added. The actual capacity of the pump must be greater than the capacity thus found, on account of losses of volumetric efficiency due to clearance, slip, leakage, and valve or port resistance.

The diagram shows the reduction in volume due to cooling of the air-steam mixture before it passes to the air-pump.

DIAGRAM SHOWING THE PROPERTIES OF MIXTURES OF AIR AND STEAM, SUCH AS EXIST AT THE AIR PUMP SUCTION.

FIG. 13.

In a jet condensing plant the air introduced by the injection water must also be allowed for.

Leakage of air into the exhaust pipe and condenser should be reduced to a minimum by exercising the greatest care to keep all joints, etc., in good condition.

The amount of air entering the exhaust pipe due to leakage at joints and glands is obviously not proportional to the amount of exhaust steam, nevertheless air-pumps are frequently designed to give an effective displacement proportional to the amount of exhaust steam, and in fact air-pumps for piston engines are frequently proportioned upon the low-pressure cylinder.

Vertical air-pumps are usually single-acting, and low-pressure engine cylinders double-acting. On this basis for ordinary bucket pumps the piston displacement of the low-pressure cylinder is about 6·5 times that of the pump for jet condensing compound engines in this country and 5·5 times for India and hot tropical climates. For surface condensers this air-pump ratio varies from 10 to 12, the lower figure for the smaller and less efficient engines.

Wet-air pumps for jet condensers are frequently proportioned so that their effective displacement is about five times the volume of the water (injection plus condensed steam) which they have to discharge.

Horizontal air-pumps are double-acting, and their volume is therefore one-half that of single-acting pumps.

Air-pumps of the Edwards type usually have about 75 or 80 per cent. as much capacity as the ordinary pumps.

It is tacitly assumed that the vacuum aimed at is about 26 ins., and that considerably greater capacity is necessary for higher vacua.

Air-pump bucket speeds rarely exceed 200 ft. per minute unless the buckets are of the solid plunger type. A usual value is 180 ft. per minute.

The vacuum in a condenser ought to maintain itself an hour with a drop of only 2 lbs.

The smallest air-pump found practicable by Mr. R. W. Allen had 0·5 cu. ft. of displacement per pound of steam condensed.

With circulation water at 65° to 70° F. a vacuum of 28½ ins. at full load with a single-stage air-pump can be maintained without special accessories. It is not advisable to allow less than 0·75 cu. ft. of air-pump volume per 1 lb. of steam condensed, and it ought to be possible to maintain a 98½ per cent. vacuum.

Circulating Pump.

The pump for delivering the required water quantity to the condenser is nearly always of the centrifugal type. The head against which this pump has to deliver is made up of the static difference in water levels between the inlet and discharge points, plus the pipe and condenser friction. In cases where water is drawn from and returned to a river or canal, the discharge pipe end can be sealed, thus forming a closed or syphon system. In arranging syphon systems care should be taken to ensure that the highest point in the system is never above a vertical distance of 25 ft. from the water level, otherwise the air contained in solution with the water is liable to be released and accumulate, thus breaking the syphon and throwing extra head on the pump.

The Edwards Patent Air-Pump.

In the Edwards Patent Air-Pump, fig. 14, foot and bucket valves are entirely dispensed with. The only valves used are those which in the old type of pump are known as 'the head valves,' but which in the Edwards pump are simply called 'the valves,' as there is no need to distinguish them in any way.

The condensed steam flows continuously *by gravity* from the condenser into the base of the pump, and is there *dealt with mechanically* by the conical bucket working in connection with a base of similar shape. Upon the descent of the bucket the water is projected *silently and without shock* at a high velocity through the ports into the working barrel, as shown in fig. 15. The rising water is followed by the rising bucket, which closes the ports, and sweeping the air and water before it discharges them through the valves at the top of the barrel.

The essential feature of the Edwards pump is that the water is dealt with *mechanically*, and is in no way dependent on *pressure* in the condenser to drive it up into the pump; consequently,

increase of speed does not impair its efficiency. The speed of the water must correspond with the speed of the bucket, and the pump thus having a regular quantity of water to deal with at each revolution is particularly well suited to work at high speeds.

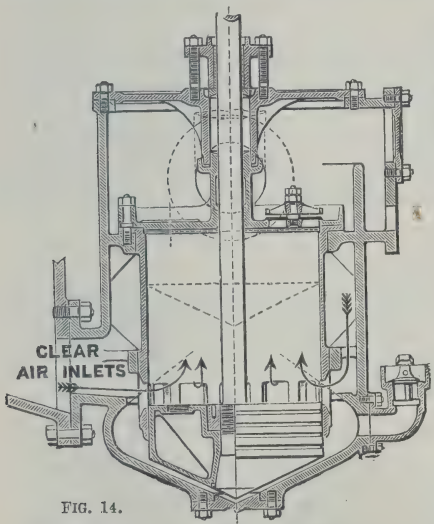


FIG. 14.

Another point of considerable importance in the design of the pump is that *clear air inlets are maintained*. As the bucket descends and the ports open there is absolutely no obstruction between the condenser and the pump. The air has a perfectly free entrance into the barrel, as indicated by the arrow in fig. 15, while immediately afterwards the water is injected at a high velocity; thus, instead of obstructing the entrance of the air, the water tends to compress that already in the barrel, and to carry more air in with it.

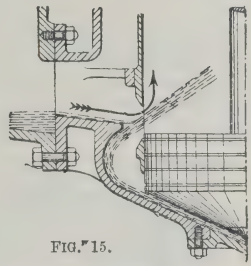


FIG. 15.

Before any air-pump can discharge, all the air in the barrel must be compressed to slightly in excess of the atmospheric pressure, and consequently the greater the amount of air remaining in the clearance water the more inefficient the pump becomes. By reducing the top clearance to a minimum, as is done in the Edwards pump, the efficiency is considerably increased.

SIZE, SPEEDS, AND APPROXIMATE OVER-ALL DIMENSIONS OF THREE-THROW EDWARDS AIR PUMPS WHEN WORKING IN CONNECTION WITH SURFACE CONDENSERS FOR VARIOUS AMOUNTS OF STEAM.

Lbs. of Steam per Hour.	Number of Pump Barrels.	Dia.	Str.	Revs. per Min.	Approximate Dimensions.		
					Height to Centre of Shaft.	Length without Motor.	Width.
10,000 to 12,500	3	8 ins.	8 ins.	160 to 200	5 ft. 1 in.	4 ft. 9 ins.	1 ft. 10 ins.
12,500 to 15,000	3	9 ins.	9 ins.	140 to 170	5 ft. 6 ins.	5 ft. 2 ins.	2 ft. 0 ins.
15,000 to 20,000	3	9½ ins.	9 ins.	150 to 200	5 ft. 6 ins.	5 ft. 2 ins.	2 ft. 0 ins.
20,000 to 25,000	3	10 ins.	10 ins.	160 to 200	6 ft. 0 ins.	5 ft. 4 ins.	2 ft. 1 in.
25,000 to 30,000	3	11 ins.	11 ins.	150 to 180	6 ft. 6 ins.	5 ft. 6 ins.	2 ft. 3 ins.
30,000 to 35,000	3	12 ins.	12 ins.	140 to 165	6 ft. 10 ins.	6 ft. 0 ins.	2 ft. 3 ins.
35,000 to 40,000	3	13 ins.	12 ins.	140 to 160	6 ft. 10 ins.	6 ft. 6 ins.	2 ft. 5 ins.
40,000 to 45,000	3	14 ins.	12 ins.	140 to 160	6 ft. 10 ins.	7 ft. 0 ins.	2 ft. 6 ins.
45,000 to 50,000	3	14½ ins.	12 ins.	145 to 160	6 ft. 10 ins.	7 ft. 0 ins.	2 ft. 6 ins.
50,000 to 55,000	3	15 ins.	12 ins.	150 to 165	7 ft. 0 ins.	7 ft. 6 ins.	2 ft. 8 ins.
55,000 to 60,000	3	16 ins.	12 ins.	145 to 160	7 ft. 0 ins.	7 ft. 6 ins.	2 ft. 8 ins.

By means of the door at the top of the pump the only valves used can be readily examined cleaned, and, if necessary, renewed while the pump is at work.

The door also allows the engineer-in-charge to estimate the air duty of the pump, and after a little experience to tell whether there is any serious leakage into the system.

For marine pumps a weir is arranged between the door and the valves (see fig. 15); this enables the door to be taken off while the pump is at work, without loss of water even when there is a heavy list on the ship.

The table on page 1843 gives the dimensions of 3-throw pumps suitable for electrical and other land installations, and for dealing with the stated quantities of steam, when working in connection with *surface* condensers of the ordinary type.

(*The Edwards Air-Pump Syndicate, Ltd.*)

The Leblanc Air-Pump, Simple Jet Type.

In the Leblanc Simple Jet Condenser and Air-pump, water is jetted across the exhaust pipe from the engine by means of a reversed turbine wheel which rotates in a closed casing alongside the pipe, there being an opening into the pipe at a portion of the turbine periphery. There are thus a series of flat sheets of water traversing down the vertical pipe, and serving both to condense and to draw off the steam and air. The water can be used over again if not too hot, and the pump is thus intended for use as a wet and dry air-pump. As there are quite a number of water vanes acting together and moving with great energy down the eduction pipe, there is only a small pressure difference between adjacent vanes, and there is quite a mass of water always acting. The vanes of water practically act like an extension of the metal vanes of the central fan, so that this fan is in a sense an absolute fit in the casing like a scoop wheel, and air is withdrawn between each such vane and the next following. The fan or turbine is practically a water-packed radial-bladed wheel which scoops out the air as it rotates.

The Leblanc Rotary Air-Pump.

In the Leblanc type of Rotary Air-pump, a section of which is given in fig. 16, the air from the condenser is trapped between thin sheets of water thrown out by a series of turbine vanes, which rotate at a high speed in an enclosed casing. The discharge opening in the casing uncovers a certain portion of the turbine periphery where the sealing water, drawn in at centre and passing through the distributor nozzle, gets shot out in sheets from the revolving blade wheel, down the narrowing part of collecting cone discharge pipe, at a high velocity, trapping on its way a certain volume of air between the water sheets discharged.

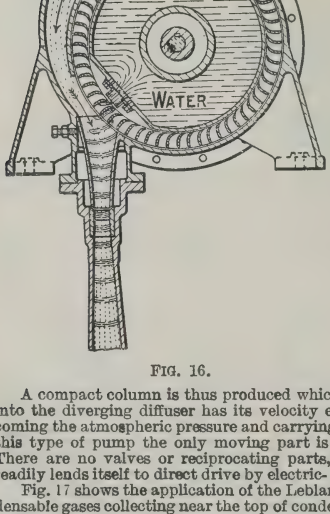


Fig. 16.

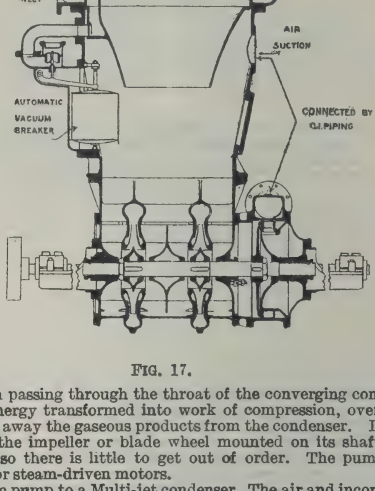


Fig. 17.

A compact column is thus produced which passing through the throat of the converging cone into the diverging diffuser has its velocity energy transformed into work of compression, overcoming the atmospheric pressure and carrying away the gaseous products from the condenser. In this type of pump the only moving part is the impeller or blade wheel mounted on its shaft. There are no valves or reciprocating parts, so there is little to get out of order. The pump readily lends itself to direct drive by electric- or steam-driven motors.

Fig. 17 shows the application of the Leblanc pump to a Multi-jet condenser. The air and incondensable gases collecting near the top of condenser are drawn off by the Leblanc Rotary Air-pump, and the injection and condensation waters are withdrawn from the bottom of condenser by special pump. The pump impellers are mounted on a common shaft and driven by one motor.

The Leblanc Air-pump is not used so widely now that ejector air-pumps are highly developed. It is efficient, however, and more suitable where steam pressure available is not high enough for ejector air-pumps.

Jet or Ejector Air Pumps.

Of recent years great developments have taken place in the design and application of air-pumps in which the vacuum is produced entirely by means of a jet or jets of fluid, which may be water or steam. The air from the condenser mixes with the jet, and the mixture, which just after mixing is at high velocity and low pressure, passes through a ' diffusor ' tube of increasing area in which its kinetic energy is as far as possible transformed into pressure energy, so that the mixture may be finally discharged against atmospheric pressure.

Such types of air-pump have the advantage that there are no moving parts in the complete condensing plant, with the exception of a pump for removing the condensate ; also they occupy very little space, have little weight, can be located in any convenient situation, and require little attention and maintenance.

The types using steam jets appear to be superseding those with water jets.

It is found advantageous not to carry out the compression of the air from vacuum to atmospheric pressure in one stage, but in two. In order to reduce the work in the second stage, and hence to reduce steam consumption, frequently an auxiliary condenser is used between the first and second stage ejectors.

Under certain circumstances, the flow through the diffusor becomes unstable, causing a drop in the vacuum. To overcome this, some makers provide auxiliary air inlets by which atmospheric air can enter at the throat of the diffusor when required ; one firm uses a water jacket round the combining nozzle, and at least one other uses neither of these means, but relies entirely upon the proportions of the nozzles. These forms of the apparatus are described below.

A further recent development is the introduction of an ejector with three stages and two intermediate condensers.

Condensing Plant with Hick-Breguet Ejectair.

(Hick, Hargreaves & Co., Ltd., Bolton.)

The surface condenser is so designed that the whole of the steam comes into intimate contact with the tubes, and the tubes are disposed so as to reduce to a minimum the resistance to the passage of steam.

To ensure the air being withdrawn at maximum density, it is drawn through a ' de-vapourising ' chamber formed in the lower pass of the condenser by steel plates enclosing about 8 per cent. of the total number of condenser tubes. The reduction of volume of the air adds to the efficiency of the air-extracting apparatus.

The condenser inlet is usually connected to the turbine exhaust by means of a galvanised wrought-steel bellows pipe which permits of relative expansion between turbine and condenser. The condenser may be mounted on springs, and the bellows pipe dispensed with, but in this case expansion joints are required for all piping connected to the condenser.

The jet condenser is of the multiple jet type, the number of nozzles being proportionate to the duty of the plant. The upper portion is designed to eliminate, almost entirely, resistance to the passage of steam on to the finely divided water sprays, condensation thus being immediate and complete.

The air and water are extracted at different points entirely independent of one another, the former being extracted by a steam-jet air pump and the latter withdrawn by a pump of the centrifugal type.

The water extraction pump is designed for efficient continuous running under all load variations, and is driven through a flexible coupling by an electric motor or by a steam turbine as required.

The air is extracted from the condenser by the patented ' Hick-Breguet ' Ejectair, combining the advantages of a high degree of air extraction with low steam consumption. To meet varying requirements the apparatus is manufactured in the following types :—

Duplex Ejectair with intermediate condenser of the jet or direct-contact type.
Duplex Ejectair with combined intermediate surface condenser and feed heater.
Three-stage Ejectair with combined surface condenser and heater.

In the first-mentioned type, as shown by fig. 16, the first ejector withdraws the air from the condenser, compressing it slightly and discharging it into the intermediate condenser, where the steam from the first ejector is condensed and the air reduced in volume, thereby increasing the volumetric efficiency of the secondary ejector.

The water from the intermediate condenser along with the condensate of the first ejector is delivered by means of a balance pipe direct to the extraction-pump suction, in this way raising the temperature of the condensate from the main condenser, and conserving the heat in the first jet.

The steam from the second jet, along with the air, is discharged preferably into a surface heater, through which the discharge water from the extraction pump passes ; by this means the heat from the second jet is conserved and the temperature of the condensate further raised.

The apparatus is fitted with a patent stabilising valve which automatically regulates the working of the two ejectors at light loads, and prevents them from becoming unstable. Under ordinary conditions of working the stabilising valve is not in operation, the whole energy of the steam operating the Ejectair being used for extracting the air from the condenser.

The plant is very simple in construction and requires no attention when put into service, the only moving parts being the extraction pump spindle and impeller. It is put into operation by simply opening the steam valve to the two jets, and closing the switches controlling the circulating pump and extraction pump motors.

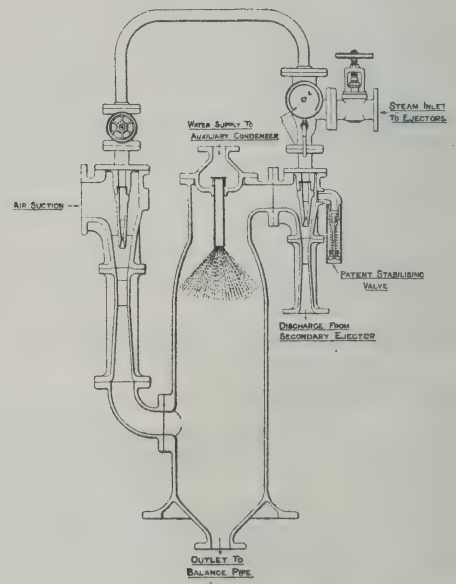


FIG. 18.—Sectional Arrangement of Hick-Breguet Ejectair.

The Mirrlees Ejector Air Pump.

(*The Mirrlees-Watson Co. Ltd.*)

The earlier types of Mirrlees air ejector were provided with groups of small steam nozzles in each stage. Also an auxiliary air inlet was provided at the throat of the second stage ejector. To hasten the production of vacuum when starting, the throat was made larger than necessary for normal working. When the air from the condenser became less than that required to fill the throat, the auxiliary ports admitted atmospheric air and the working of the ejector thus became stable. The latest type of Mirrlees ejector is shown in fig. 19.

In this design there is a single steam nozzle in each stage instead of a group, the auxiliary air inlet is eliminated, and the consumption of operating steam reduced.

Steam enters at G, the air from the condenser at B. The steam from the first ejector is condensed by injection water entering at F, which also cools the partially compressed air delivered by the first ejector. This cooled air and incondensible gases are led to the second ejector, and discharged at L along with the steam used in the second ejector.

The condenser is self-draining through a U-sealing pipe to the condensate pump suction branch.

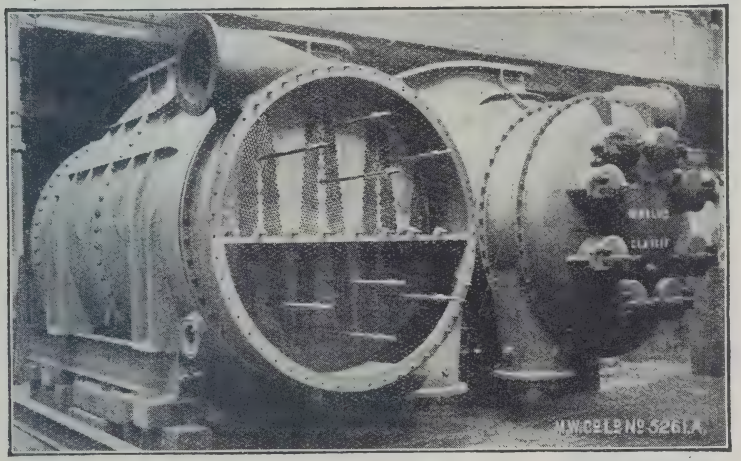
The Mirrlees-Watson Co. state that their experiments with water-jacketed ejectors show that the additional complications involved by the water jacket are unnecessary.

In the Mirrlees ejector the convergent cone is cool, and the discharged air and gases do not attain a temperature higher than 215° F.

A standard works test chart for one size of the latest Mirrlees ejector is shown in fig. 20. The chart shows that there is a great reduction in steam consumption as the air leakage falls below 85 lbs. per hour. In large installations advantage is taken of this by arranging the extractors in batteries working in parallel.

Mirrlees

DELAS - GINABAT CONDENSER



Mirrlees-Delas Surface Condensers as supplied to Carlisle Willow Holme generating Station. Combined steam duty 141,000 lb./hr. Vacuum 29.1" with water at a temperature of 55 deg. F. Surface 16,100 sq. ft.

The MIRRLEES WATSON COMPANY LIMITED

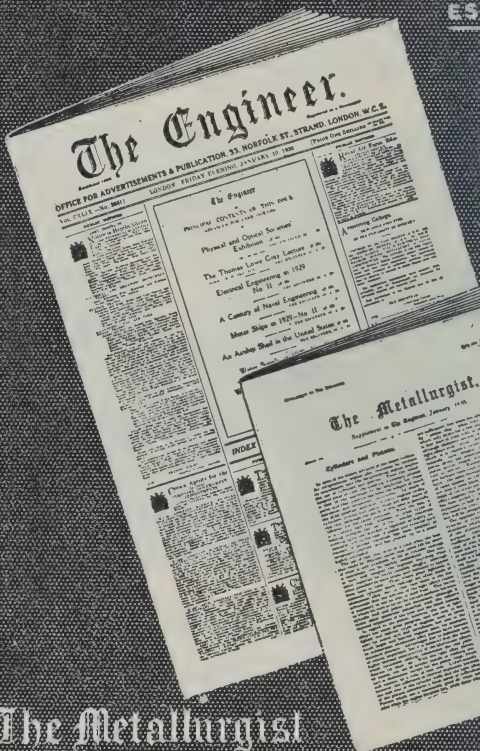
ENGINEERS, SCOTLAND ST., GLASGOW

London Office: Mirrlees House, 7 Grosvenor Gdns. S.W.1

278

The Engineer.

ESTABLISHED
1856



1/-
WEEKLY

The Metallurgist
IS A FREE MONTHLY SUPPLEMENT
TO The Engineer.

EDITORIAL and PUBLICATION OFFICES
28 ESSEX STREET STRAND LONDON W.C.2

ANNUAL SUBSCRIPTION RATE £3 5 0

THE MIRRLEES EJECTOR PUMP.
(The Mirrlees-Watson Co. Ltd.)

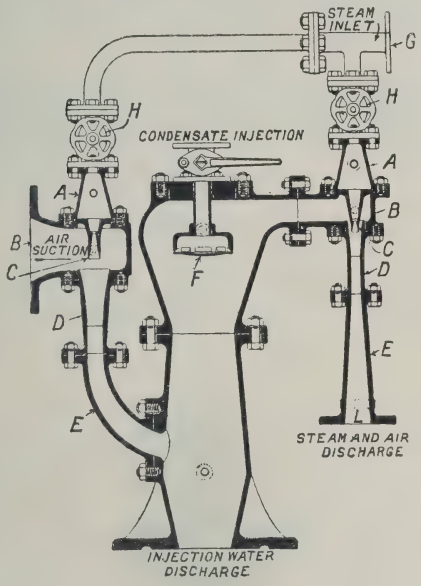


FIG. 19.

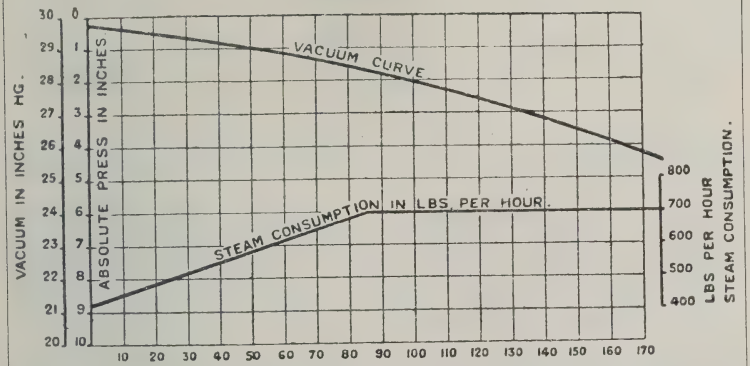


FIG. 20.

Condensing Plant with Radojet Air-Pump.

The construction of the ' straight ' type air-pump is illustrated by fig. 19. The arrangement comprises, essentially, two steam ejectors working in series. The first-stage ejector consists of a number of specially designed and arranged tubular nozzles, whilst the second-stage ejector comprises a specially designed radial nozzle with an adjustable nozzle point, so arranged that the steam passing from the nozzle to the diffuser does so in the form of a radial jet and offers a large surface area of steam for the entrainment of the air.

Steam is supplied to the second stage of the ' Radojet ' through a steam strainer, and a branch pipe with valve from this strainer controls the supply of steam to the first stage. The steam issues from the first stage nozzles at high velocity, entrains the air withdrawn from the condenser, or other vessel in which it is required to maintain a vacuum, and compresses the air and vapour to a pressure several inches above that at the entrance to the ' Radojet.' The mixture of air

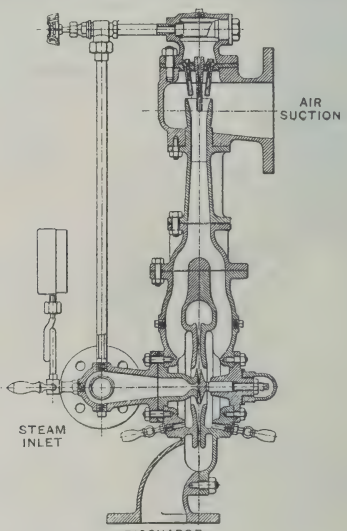


FIG. 21.

and steam then passes through the body of the ejector to the second stage, where it is entrained by the second-stage steam and compressed to atmospheric pressure in an annular diffuser. The whole then passes through the discharge branch to atmosphere, or to some place where the heat of the steam can be utilised.

It will be noted that the second-stage steam compresses not only the air but also the steam coming from the first stage. The air-pump may, however, be arranged with a surface or jet type intercondenser, fitted between the stages, so as to condense the steam used in the first stage, and leave only the air and vapour to be dealt with by the second stage. For equal air handling capacity, this arrangement reduces the steam consumption by about 50 per cent. as compared with an ordinary ' straight ' type ' Radojet.' Where it is desired to recover the heat in the steam discharged from the ejector, the discharge pipe may be led direct into the feed tank, or the steam may be led to a feed heater of the direct-contact or surface type.

The ' Radojet ' air-pump has been adopted in a great many land installations, and has also been widely applied to marine service. For closed feed systems the first stage discharges into a surface intercondenser, and the second stage into a surface type aftercondenser—both units being incorporated, if desired, in one casing. The air-pump has also been fitted in conjunction with the maker's surface and jet type main condensers for turbine installations up to large sizes, and for uniflow, steam extraction, and slow-speed reciprocating steam-engines. For turbine installations the water extraction and circulating pumps are usually driven by electric motors

or small turbines, and this drive may also be adopted in the case of reciprocating engines, or the pumps may be driven through ropes or belts from pulleys on the engine shaft. Where the vacuum obtainable with existing reciprocating or other type air-pumps is unsatisfactory due to age or any other cause, an improvement may be effected by the substitution of a 'Radojet' equipment. Such conversions have frequently been carried out in connection with large reciprocating steam-engines, and have been accompanied by an increase in the power and efficiency of the prime mover. If desired, the air-pump may be arranged in the 'Duplex' form, which provides two complete ejector units with common inter- and after-condensers, and thus affords a spare unit capable of dealing with abnormal loads or air leakages.

The application of the 'Radojet' is not confined to condenser installations, but embraces all processes in which the creation of a vacuum is required, *e.g.* vacuum drying, filtering, and paper making, sugar making and refining, evaporative and chemical processes. In such cases the total absence of moving parts is an important advantage, and the exhaust heat from the air-pump may frequently be used for heating the liquor or other material used in the process. The principle of the radial jet has also been applied to the design of a 'Radojet' Vacuum Augmenter, operated by steam from the auxiliary exhaust range at a pressure of 2 to 5 lbs. gauge. This augmenter is intended for use in conjunction with existing air-pumps where it is desired to increase the vacuum without entailing additional work on the existing pumps.

THE WEIR EJECTOR AIR-PUMP. (See also p. 1838.)

(*G. & J. Weir, Ltd., Cathcart, Glasgow.*)

The most modern Weir Ejector Air-Pump is the three-stage type, totally enclosed, each stage being fitted with after coolers. These three-stage air ejectors have a greater air-handling capacity than the two-stage types, the ratio of increase naturally varying with the quantity of air being handled. At normal duties the ratio is roughly 3 to 2 on the same steam consumption.

THE BAILEY AIR-PUMP.

(See p. 1200).

Measurement of Condensed Steam.

The condensate may be measured by passing over a V-notch in a suitably arranged tank, the height of the water surface above the apex of the notch being measured at a point at such a distance from the notch that the water is nearly still.

If h is the height in inches, Q the flow in cub. ft. per minute, and the angle of the notch is 90°, then $Q = ch^2\sqrt{h}$, where c is approximately 0·305, but, more accurately, varies from 0·31 at $h = 2$ ins., to 0·301 at $h = 9$ ins. or over. Less heads than 2 ins. should not be used, it being better to increase the head by using a notch of narrower angle; the discharge is approximately proportional to the tangent of half the notch angle.

For the 90° notch the cubic feet of flow per minute $= 0{\cdot}305h^2\sqrt{h}$, where h is the depth from the water level in the still pond above the notch measured down to the point of the V.

If one notch will not pass the water, two may be used, and the reading of the instrument doubled.

In the 'close feed' systems, which are now frequently adopted in power stations, the condensate and the make-up feed-water may be measured by means of Venturi meters included in the pipe system.

Measurement of Air-Pump Water.

THE LEA AUTOMATIC WATER RECORDER FOR MEASURING CONDENSED STEAM.

This instrument, made by *The Lea Recorder Co., Ltd.*, of Manchester, measures the condensate from surface condensers by means of 90°, or smaller, 'V' notches.

A float in a chamber, connected to a measuring tank, which is fitted with 'V' notch plate, rises and falls in accordance with the level of water flowing over the 'V' notch.

On the float rod is a toothed rack which turns a pinion on a drum spindle. The pen of the recording mechanism is moved across the record paper (which is clock-driven) by means of the spiral groove cut in the drum. This spiral has an increasing pitch, starting at 0 when the water is down to zero level, or the apex of the 'V' notch, and increasing in the ratio $h^2\sqrt{h}$, so that the pen movement is always proportional to the volume of water flowing. (See p. 1441.)

An important feature of these recorders is that when measuring hot water they possess a compensative action, owing to the fact that when the temperature increases, the density of the water is reduced, and consequently the float sinks slightly, and thus the instrument records a less weight, which is correct. On the other hand, when the temperature falls the float rises, owing to the increased density of the water, and the instrument records a slightly greater weight, which is also correct. The instrument, therefore, remains accurate for changes of water temperature within wide limits.

By means of this recorder the steam consumption of an engine or turbine can be noted at any time, whilst a chart record is secured of the total discharge over a given period.

Any small changes in the running of the plant can be instantly noted on referring to the diagram produced.

TABLE OF FLOWS THROUGH 90° V NOTCHES.

Depth in Notch in Ins.	Flow per Hour.		Depth in Notch in Ins.	Flow per Hour.	
	Galls.	Lbs.		Galls.	Lbs.
2	654	6,540	9	27,300	273,000
3	1,785	17,850	10	35,600	356,000
4	3,640	36,400	11	45,000	450,000
5	6,350	63,500	12	56,050	560,500
6	9,970	99,700	13	68,400	684,000
7	14,650	146,500	14	82,600	826,000
8	20,300	203,000	15	79,900	979,000

(*Barr.*)

The position of the 'V' notch tank varies with the type of plant, *e.g.* sometimes it is placed in the condenser pit, alongside the extraction pump, and the condensed steam flows from it by gravity to a hot-well, or, alternatively, it is inserted between the extraction pump and a lift pump, the latter drawing water from the end compartment of the notch tank, and delivering to an overhead hot-well or storage tank.

In cases where the 'closed feed' system is in operation the recorder tank can be sealed, and the same degree of vacuum maintained as exists in the condenser, in which case the tank is placed immediately underneath the condenser outlet, and between this and the extraction pump.

This apparatus is also extensively used for the measurement of boiler feed-water, in which case the measuring tank is usually inserted on the suction side of the boiler feed pumps, so that water is drawn from it; the supply of water flowing through the recorder tank being controlled by means of an equilibrium float valve.

These recorders can be obtained in two patterns, *i.e.* the 'Indicating and Recording' type, and the combined 'Indicating, Recording, and Integrating' type. With the latter the total quantity can be read off directly on large dials of the Harding type.

Injection Cocks

Are usually of the hollow plug type, in gunmetal. One per condenser is sufficient. Two or three standard sizes are generally available to choose from. The minimum size can be checked, as in the following example:—

Maximum load under normal conditions	1,000 i.h.p.
Steam per i.h.p. hour	15 lbs.
Steam per second	4·17 lbs.
Condensing water per pound of steam	40 lbs.
" " per second	167 lbs.
" " "	2·67 cub. ft.
Vacuum in condenser, worst	24 ins.
" " in feet of water	27 ft.
Vertical height water level to injection cock	11 ft.
Head of water on injection service, 27–11	16 ft.
Assume spent in friction	50 per cent.
Net head at injection cock	8 ft.
Velocity of water at cock $=8\sqrt{8}$	22·6 ft. sec.
Hence area of opening	0·118 sq. ft.
" "	17 sq. ins.

If in doubt add a margin of 25 to 30 per cent. It is, of course, understood that the above level, steam consumption, etc., must be modified to suit the actual conditions. Thus in tropical countries 60 should be substituted for 40 lbs. of condensing water per lb. of steam.

Force or Feed Pumps.

When the boiler feed-pump is driven off the engine it is usual to allow sufficient capacity for other works purposes.

The following are frequently adopted:

Textile spinning mills	25 lbs. water per i.h.p. hour
Textile weaving sheds	40 " " "
Mixed spinning and weaving	30 " " "
Flour mills	35 " " "
Bleach works	50 " " "

It will be seen that an ample allowance is made, and for economical superheated-steam engines a reduction of about 5 lbs. can be made. The above are based on actual pump displacement. The plunger speed should not exceed about 200 ft. per minute.

Pump Openings.

The velocity of water in pump openings should not exceed 500 ft. per minute.

Cooling Ponds.

Where the supply of water for condensing purposes is limited, it is necessary to cool the condensing water and use it again and again. One method of cooling is to pass the hot water into an open pond and draw off water for condensing from the opposite side of the pond, the water being cooled simply by evaporation from the surface.

Spray Ponds.

An improvement upon the simple pond is to lay a system of piping over the pond and let the water issue from nozzles in a fine spray. A large amount of water is, however, carried away by the wind and lost.

A spray pond should have $2\frac{1}{2}$ sq. ft. of area per gallon of water to be cooled per minute. This includes the marginal straps necessary for windage, etc.

Water-Cooling Towers.

Water-cooling towers are of two principal types—chimney type and open type. Both act by bringing the water into intimate contact with air currents, and to expose as large a surface of water to the air as possible the water is either broken up by splash bars into fine drops, or made to run down in thin films. In chimney type towers air is admitted at or near the bottom only, and passes up inside the casing usually by natural draught, though sometimes forced draught is used. Forced-draught towers are smaller than natural-draught for the same duty, but require a large expenditure of power in the fans.

Open type coolers have water-distributing arrangements similar to those in the chimney type, but have no external closed casing, the flow of air across the water is dependent upon the action of the wind. To prevent excessive loss of water, the stack is usually surrounded by louvre boarding, but even then the loss of water is large compared with that in the chimney type.

In chimney type towers the water is usually pumped to distribution troughs at a height of 16 to 25 ft.; the height of the chimney is about 60 to 70 ft. if of timber, 100 to 120 ft. if of steel (which is less usual). The width of the tower should not exceed 25 ft. at most, as the difficulty of getting cool air to the centre increases as the width is increased; nor should the tower be narrower than about 15 ft., in order that it may have an adequate margin of stability.

In all cooling towers the heat is extracted from the water partly by (1) direct heating of the air, partly by (2) evaporation of part of the water. Under typical atmospheric conditions in this country evaporation accounts for about 75 to 85 per cent. of the cooling.

The amount of evaporation possible depends upon the amount of moisture already contained in the air entering the tower and its temperatures at inlet and outlet.

It is important, in specifying or judging the capacity of cooling towers, to take into account the humidity or the wet and dry bulb readings.

The 'relative humidity' of the atmosphere is the ratio of the weight of water vapour contained in a certain volume of air to the weight required to saturate this volume at the same temperature, or it may be expressed as the ratio of the vapour pressure at the dew-point to that at the actual air temperature, the dew-point being the temperature to which the air must be cooled in order that it may just become saturated, so that further cooling would cause condensation to begin. The greater the humidity of the atmosphere, the less, of course, is its capacity for taking up moisture by evaporation. The humidity is ascertained by reading wet and dry bulb thermometers and referring to humidity tables.

The quantity of fresh 'make-up' water required to compensate losses by evaporation, etc. in cooling plants is about 3 to 5 per cent. of that in circulation.

It is stated that the loss by evaporation only in natural draught chimney type cooling-towers is approximately 1 per cent. for each 12° F. by which the water is cooled in the tower.

(*I. V. Robinson*, '*B.E.A.M.A. Journal*,' *March* 1921.)

The amount of air for which the cooling tower should be proportioned may be calculated as in the following example:—

Let the atmospheric temperature and humidity be 65° F. and 60 per cent. respectively, the atmospheric pressure 14·7 lbs. per sq. in., and let the air leave the tower at 85° F. (saturated with vapour).

From steam tables, the vapour pressure at 65° F. is 0·31 lb. per sq. in., thus the air pressure is 14·7—0·31 = 14·39 lb. per sq. in.; 1 lb. air at this pressure and at 65° F. occupies 13·45 cu. ft.; 1 lb. vapour at 85° F. occupies 990 cu. ft.

$$\therefore \text{weight of vapour per lb. air if saturated} = \frac{13 \cdot 45}{990} = 0 \cdot 0136 \text{ lb.}$$

$$\therefore \text{weight of vapour at 60 per cent. humidity} = 0 \cdot 0136 \times 0 \cdot 6 = 0 \cdot 00816 \text{ lb.}$$

At 85° F. the vapour pressure = 0·6 and air pressure 14·1 lb. per sq. in.; 1 lb. air at these conditions occupies 14·3 cu. ft.; 1 lb. vapour at 85° occupies 541 cu. ft.

$$\therefore \text{vapour per lb. air} = \frac{14 \cdot 3}{541} = 0 \cdot 0264 \text{ lb.}$$

Hence, for each lb. of air passing through the cooling tower, 0·0264—0·00816 = 0·0182 lb. water is evaporated.

Heat to evaporate 1 lb. water from water at 65° F. to vapour at 85° F. = 20 + latent heat = 1,061 B.Th.U.

$\therefore$ heat to evaporate 0·0182 lb. = 0·0182 × 1 061 = 19·3 B.Th.U.
Heat absorbed in heating air = 0·237 × 20 = 4·74 B.Th.U.
Total heat taken from water per lb. air = 19·3 + 4·74 = 24·04 B.Th.U.

Ratio of cooling by evaporation to total cooling

$$= \frac{19 \cdot 3}{24 \cdot 04} = 0 \cdot 8 \text{ or } 80 \text{ per cent.}$$

If the water is cooled, say 30°, the weight of water cooled per lb. air $= \frac{24 \cdot 04}{30} = 0 \cdot 801$ lb.

Percentage of water lost by evaporation

$$= \frac{0 \cdot 0182}{0 \cdot 0182 + 0 \cdot 801} \times 100 = 2 \cdot 22 \text{ per cent.}$$

Weight of air to pass through cooler per lb. water

$$= \frac{1}{0 \cdot 801} = 1 \cdot 25.$$

Premier Water-Cooling Towers.

(*The Premier Cooler and Engineering Co. Ltd., Station Works, Shalford, Surrey.*)

'Premier' Cooling Towers, although primarily designed for cooling the water used for condensing purposes in, or in conjunction with, steam engines or turbines, also for cooling the jacket water for internal combustion engines, are equally useful to a degree for other industrial purposes, such as: (1) The liberation of gases in liquids, (2) the evaporation of water in solutions containing certain solids it is desired to concentrate as far as possible by this process.

The Chimney Type Tower, which is almost invariably used for the larger schemes, although usually made in rectangular form with a height ranging from 70 ft. to 100 ft., can, nevertheless, be supplied with chimneys of hexagonal or octagonal construction. The latter type, a considerable number of which have been put into operation in this country, has exceptional strength owing to its shape approaching as near as possible the circular form. The chimneys are designed so as to resist the full wind pressure experienced with a good factor of safety without the assistance of the internal filling or irrigation. The materials used for this class of tower are generally pitch pine and red deal, if of timber construction, but towers can be supplied with steel, concrete, or brick chimneys. In the wooden construction all the bolts, fishplates, grappling irons, etc., are heavily galvanised.

The irrigation system is designed to give the largest area possible for the uprising air, which is brought into contact with the water or other liquid to be cooled or evaporated, in the finest possible form. The towers operate on the splash and film principle, which extensive experience has proved gives the best results. The distributing troughs and gutters are of the open top type, thus allowing the quick removal of solids and foreign matter which invariably accumulate, due to the continuous process of evaporation. The distributing tubes are of 'Venturi' type executed *in acid-resisting earthenware, as also are the splash plates.* The water is thoroughly atomised before falling into the irrigation or lath system. The laths are fixed without the use of nails, being held in place in machined-notched bearers. The design provides for keeping the air in contact as long as possible, whilst allowing the maximum amount of air to be circulated through the plant.

The Open type construction, such as is usually employed for dealing with the jacket water for internal combustion engines, is constructed on somewhat similar lines, but the outside faces are louvred to prevent splashing out. Since this type of tower is not fitted with a chimney, the air

FAMOUS PREMIER PRODUCTS

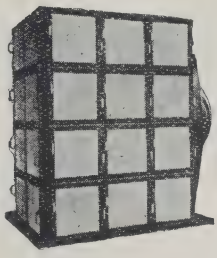
renowned for their efficiency and economy

PREMIER COOLING TOWER

has been erected by us for many of the important engineering firms. Combining strength with efficiency, absolutely weatherproof, Premier plant cools over 2,160,000,000 gallons daily!

PREMIER OIL SEPARATOR

Simple to instal—needs very little attention, no filtering materials required. Guaranteed to separate all oil up to .35 grains per gallon of steam condensed.

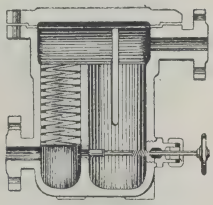


PREMIER ADHESIC AIR FILTER

With its low resistance to air flow, uniform surface and adhesive coating, the "Premier" filtering system is far superior to any other. Guaranteed to remove 98% of all solid impurities

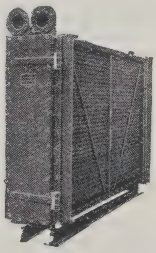
HASTE PATENT VALVE

No springs, hinges, corners angles or pockets; the only working part is the valve itself. For a given size, more water can pass through this valve than any other type.



PREMIER UNIC STEAM TRAP

Simple design ensures discharge of any condensation present. Works satisfactorily with superheated steam.



CLOSED CIRCUIT AIR COOLER

For alternator ventilation, to be found in practically every engineering plant of importance throughout the world. Installations to date exceed 700,000 kw.

The PREMIER

Cooler & Engineering Co. Ltd.

Shalford, Nr. Guildford, Surrey

F. 1852

THREE
CELEBRATED
PAPERS

"THE ENGINEER"
(*Established 1856*)

"THE IRONMONGER"
(*Established 1859*)

"THE CHEMIST & DRUGGIST"
(*Established 1859*)

Proprietors:—

MORGAN BROTHERS (PUBLISHERS) LTD.
28 ESSEX STREET STRAND
LONDON W.C.2

circulation is much slower, and consequently the space occupied for a given duty is much more than with the natural draught Chimney type construction. In cases where the water volume is comparatively small, and it is desired to reduce the temperature to a very low degree, this type of cooler is especially suitable.

In exceptional circumstances, where the space available is limited, Forced Draught Towers can be supplied with considerable benefit, but such plant is uneconomical from the point of view of running cost, as compared with Natural Draught Plant.

STEAM ACCUMULATORS.

Where, as in the case particularly of rolling mill engines and winding engines, large quantities of steam are exhausted intermittently, this steam may be profitably utilised in exhaust steam turbines. In order to give a steady supply of steam to the turbine, the exhaust from the high pressure is first passed into a storage vessel or heat accumulator, which consists essentially of a large container in which a large mass of material is placed which can absorb the heat of the engine exhaust during its working periods and redeliver it to the turbine during the 'bridge' periods, i.e. when the reciprocating engine is not exhausting any steam.

An exhaust steam turbine working with steam at, say, 16 lbs. per sq. in. abs., and a vacuum of 27·5 ins., will require, with allowance for auxiliaries, about 40 lbs. steam per kw. hour, so that for every 1,000 lbs. exhaust steam per hour available, and deducting 16 per cent. for wetness and leakage, $\frac{1,000-160}{40} = 21$ kw. can be obtained.

The proportions of the accumulator depend on the bridge period or time of regeneration, the range of temperature allowable, and the material used to store the heat, as well as on the steam rate. Early accumulators were made up of old boilers packed with old rails, etc., but water is a much more efficient material, for iron has a specific heat of about 0·11 and a specific gravity of about 7, so that its volumetric specific heat is only about 0·77.

The range of temperature to be allowed in the accumulator must be decided upon, and from these figures the mass of water to be employed must be calculated. If the accumulator becomes too hot it will reduce the power of the engines which feed it. If it becomes too cold this proves that the turbine is doing more work than can be supplied by the waste steam. It is considered that usually the stops to be bridged by the accumulator are 60 secs. in colliery work and 30 to 45 in rolling mills. The admission to a steam turbine is assumed to be 16·5 lbs. absolute, the range being 15 to 18 lbs., or 10° F. In some plants, pressure rises more than ½ lb. above atmosphere, and it may fall to 12 lbs. absolute or 5½ ins. of vacuum. A reflux valve is arranged in that case to prevent the vacuum getting back to the engine—an important safeguard for a winding engine.

In calculating accumulators,

$$WTL = wt,$$

where,

W = lb. of steam supply per sec.; T = time of regeneration in secs.;
w = lb. of water in accumulator; L = latent heat of steam per lb.;
t = range of mean water temperature in accumulator.

It should be noted that if the circulation is not perfect, the range of mean temperature (t) of the water will be less than that corresponding to the change of steam pressure. This should be allowed for in using the formula.

It is essential that the steam should mix with the water or this will be locally heated only. Shallow dishes of water piled in tiers with narrow spaces for steam admission between them have been used, or the water may be kept in continuous circulation by a pump which delivers it in a shower through an upper steam space.

In the Rateau-Morison type of steam accumulator, great attention has been paid to efficient circulation. This has been attained by means of a number of vertical open-ended cylinders, each containing a nozzle with upturned star-shaped opening. This is placed 9 ins. below water-level, and sets up a powerful up-flow through the cylinder, which draws from the base of the accumulator and discharges the heated water at the surface.

As many as 30 such circulators are hung on curved branch pipes from each side of the exhaust main which runs along the crown of the accumulator boiler. Below the main is a third row. The accumulator is a plain cylinder laid horizontally with about 6 ft. of water depth. Circulation is so rapid that a difference of only 2° F. occurs between top and bottom of the water. The circulation cylinders have their discharge openings all turned to one end of the accumulator, and all oil is then drifted that way and collected behind two vertical plates and blown off periodically.

Accumulators may be made of old Lancashire boiler shells, or built up of cast-iron segments, or may be of ferro-concrete.

An increase of vacuum from 26 to 28½ ins. will reduce the steam consumption of a turbine by 25 per cent. With this vacuum the condensate will be about 80° F., and if there is any spare steam it should be applied to feed heating. In this case by heating the feed to 200° an economy of 12 per cent. is secured.

THE GALLOWAY STEAM ACCUMULATOR.

This accumulator (figs. 22 and 23) is intended for receiving the exhaust steam at a few lbs. per

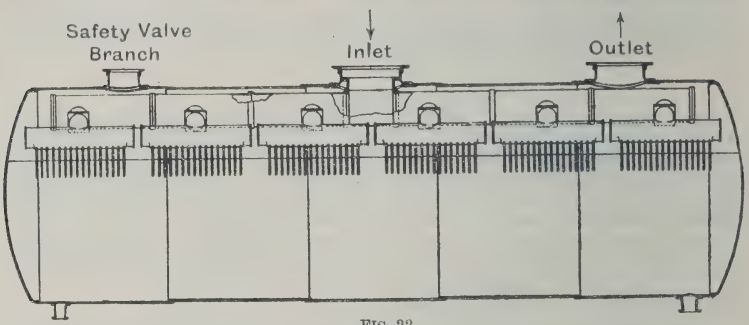


FIG. 22.

sq. in. pressure from engines such as winding engines, which work intermittently, and supplying steam at a steady rate to mixed pressure turbines.

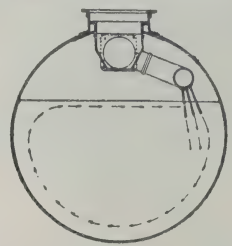


FIG. 23.

As shown in the figs., the steam enters by the central branch through internal distributing boxes, and is led into the water at high velocity through a large number of simple nozzles of small bore, so placed that the steam rapidly sets the whole of the water in the accumulator into circulation. The condensation of the incoming steam is thus carried out very effectively. The nozzles are submerged only a few inches, so that the loss of pressure between incoming and outgoing steam is very slight. The efficiency of these accumulators in operation is evidenced by the uniformity of temperature throughout the whole of the water, the maximum difference being about 1° Fah.

THE RUTHS ACCUMULATOR.

Ordinary steam boilers, especially those built for high pressures, are not well suited to cope with quickly varying loads. Such variations cause a low efficiency in the operation of the boiler plant, which manifests itself in the great discrepancy that nearly always exists between boiler test results obtained at constant load and those obtained under normal practical working conditions. More serious, however, is the detrimental influence which the limited rate of steam supply from the boilers has on the magnitude of the *output of the works* where the boilers are installed. It also happens very often that variations in steam pressure, caused by variable loads, have a serious influence on the quality of the products manufactured.

Investigations made in a great number of different industries show that the actual variations which occur in the steam consumption are usually larger than conceived. Moreover, if the limiting effect on the steam consumption that is exercised by the boilers did not exist, the variations in the steam consumption would be even still larger, as a more intensified work is possible in the factory when steam is always available in sufficient quantities.

By means of the new steam accumulation system of Dr. J. Ruths of Stockholm, all such variations in the steam demand for process and for power purposes, which are detrimental to the work of the manufacturing plant or the boilers, can be completely ' equalised ' and prevented

from reaching the boilers. This system will also prevent inevitable unevenness in the quality of fuel or in the fuel supply to adversely affect the operation of the boilers, the overall efficiency of which can thus be materially increased.

The scope of application of the Ruths Accumulator is thus a much larger one than that of earlier steam accumulators as, *e.g.*, the exhaust steam accumulators of Rateau and others. As in these accumulators, the Ruths Accumulator utilises the well-known ability of hot water to accumulate steam at a rising temperature and pressure and to release it again through re-evaporation by the internal heat of the water at falling temperature and pressure. The pressure ranges applied in Ruths Accumulators are much wider than those used in earlier steam accumulators, and although the Ruths Accumulators co-operate with the boilers, the pressure of the latter is under normal circumstances unaffected by the accumulator pressure. The maximum working pressure in a Ruths Accumulator is usually much greater than the minimum pressure. Ranges like 150–40 50–10 lbs. gauge are common. On account of the use of these wide pressure ranges at comparatively low level on the pressure scale, great steam quantities are stored per unit of volume. The practical heat storing capacity or 'steam reserve' of Ruths Accumulators is as an average 5 to 50 times larger than the maximum 'steam reserve' that can be obtained at a permissible pressure drop in the steam boilers with which they co-operate.

The Ruths Accumulator is usually built in the form of a large cylindrical steel tank, with hemispherical ends. It is constructed like a steam boiler of riveted steel plates to withstand a certain maximum steam pressure. It contains the hot steam-storing water, and is equipped with arrangements for an even distribution of the charging steam into the water, and for re-discharging steam in a safe way. The accumulator is not directly attached to the boilers, but is usually erected out of doors in a factory yard, etc., in order to be out of the way. By means of a single steam pipe and a few control valves of special type and action the accumulator is connected to the steam system, including the boilers, steam engines, or turbines, and other consumers, in such a way that a *complete and positive equalisation* of all the flow of heat energy in the whole works in question is accomplished. All surplus steam that may occur is absorbed by the accumulator, and any shortage of steam that may arise in the supply to the consumers is automatically made up from the accumulator; the boiler plant can therefore be worked under practically constant load and the boiler pressure remains steady at the set maximum level. The following are the advantages obtained as a result of this 'thermo-fly-wheel effect' of the accumulator:

1. Steady supply and practically unlimited reserve of steam at constant pressure for all power or process purposes.
2. Maximum economy in the operation of the whole manufacturing plant, as the same can be intensively utilised.
3. Maximum efficiency in the boiler plant, as ideal load conditions prevail.
4. Reduction in the numbers of boilers in operation, as only so much heating surface is required as corresponds to the average steam requirements of the factory.

The firing of the boilers is guided by a pressure gauge, put up in the boiler-house, which shows the accumulator pressure. As long as this pressure is swinging within the limits of the set range of the accumulator (which is clearly marked on the dial), the firing rate is kept constant.

Fig. 24 shows a section of a Ruths Accumulator. The accumulator is charged through steam entering the check valve BV_1 and passing into the water through the distribution pipe Z, the special nozzles P and the circulation pipes Q. The check valve BV_1 in the steam inlet prevents the water flowing backwards out into the steam pipe. When discharging, the steam is passed out from the steam dome through a 'safety nozzle' F and a check valve BV_2, which prevents the charging steam passing this way. Usual boiler mountings are fitted, including a water-gauge W, made extra long on account of the great variations in level that are a result of the wide pressure range. No regular supply of feed-water is necessary, but only occasional adjustments of the level in order to make it correspond to the pressure. The accumulator does not require daily attendance. The steam passes in and out (according to the requirements of the factory) without any noise or vibrations. The vessel is covered all over by a thick cover I of first-class insulation material, whereby the heat losses are sufficiently reduced as to be negligible. On accumulators erected out of doors the insulation is protected by roofing material E.

The control valves, which govern the proper action of the whole plant, including the co-operation of the accumulator and the boilers according to the variations in the steam demand, are automatic in action and operate on the relay principle with oil as a pressure medium.

The pressure impulses that govern the work of the valves are taken from the various steam pipes, and by suitable combinations of these valves, relays, and impulses, all variations that occur in the steam system are kept away from the boilers and made to fall on the accumulator. This action goes on automatically and accurately without attendance.

Numerous Ruths Accumulators have been working for years with marked success in many different industries and also in purely power stations, especially in Scandinavia and on the Continent. The accumulators are particularly useful in the increasing number of factories where back pressure, pass-out, or mixed pressure steam turbines or engines are employed for producing cheap steam power. In such schemes they work as a very valuable 'clearing department' for handling all the fluctuating requirements of heat and power that are interlinked by means of the steam machinery.

Under the steadying influence which the accumulator exercises on the load and steam pressures it is usually found, in such schemes, that a great deal more power can be produced out of the heating steam than when the variations in the steam and power demand are left unequalised.

THE RUTHS STEAM ACCUMULATOR

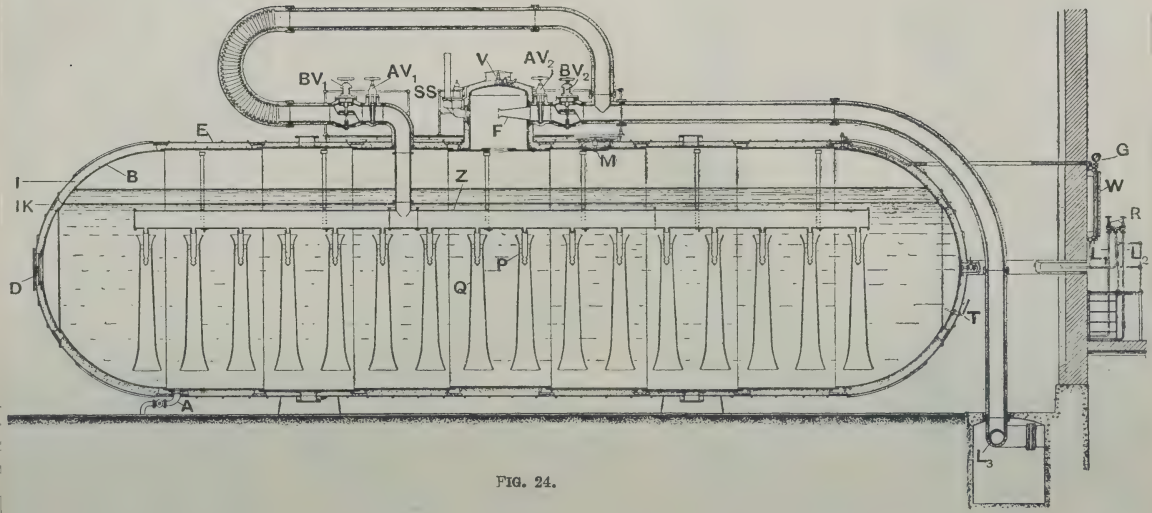


FIG. 24.

The Kiesselbach Thermal Storage System.

In the Kiesselbach thermal storage system (see fig. 25), a water storage tank (or accumulator) is in free communication with the boiler, and water is circulated by a pump from the tank to the boiler and returns to the tank, by a pipe, the open end of which in the boiler drum determines the water level in the drum. The water in the tank is thus at the same temperature and pressure as that in the boiler (hence the system is described as a constant-pressure system).

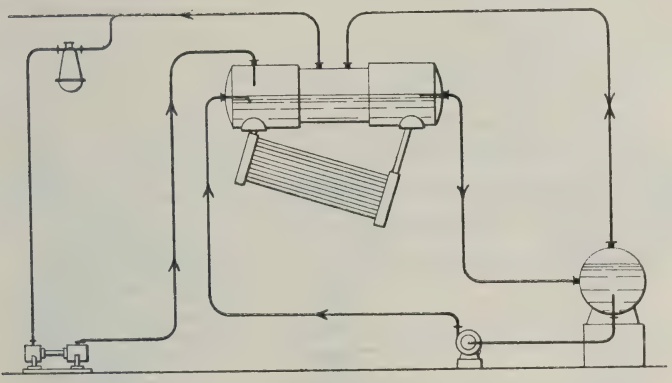


Fig. 25.

The feed-water pump is controlled by a pressure-operated regulator, so that it runs more slowly if the pressure falls and faster if the pressure rises.

If the demand for steam falls the pressure rises, the pump accelerates, and the quantity of water in the accumulator increases. The rate of firing the boiler remains constant, the heat supplied by the combustion of the fuel serving partly to generate the steam, and the balance to heat the extra feed-water which accumulates in the tank.

For an accumulator of a given capacity, the variations in steam demand (duration of demand as well as rate being taken into account) which can be dealt with by this system depend upon the difference between the saturation temperature corresponding to the steam pressure and the temperature at which the feed-water is supplied to the boiler; the thermal storage effect is thus reduced (unless the accumulator is allowed to work also on the variable-pressure system) if the temperature of the feed-water is high.

Under suitable conditions this system, like other storage systems, enables a steadier steam pressure to be maintained and the constant rate of boiler firing improves the efficiency.

Caprotti Valve Gear.

The Caprotti valve gear consists essentially of double-beat poppet valves actuated by means of rocking levers from cams contained in a gear-box. It is intended for reversing engines, such as locomotives and marine engines, and the gear-box contains mechanism for reversing and also for enabling the cut-off to be varied. A feature of the mechanism is that a wide range of cut-offs can be obtained while the lift of the valves and the lead are not affected.

Fig. 26 shows the mechanism as applied to locomotives. Two cams, C and C_1, operate the steam valves, a third cam, U, operates the exhaust valves. To the rocking lever which actuates a steam valve is attached a swinging lever carrying two rollers R and R_1, which make contact with cams C and C_1, respectively. The steam valve is shut when both rollers are on the 'low' parts of the cams, and fully open when both are on the 'high' parts. In forward gear cam C controls the cut-off, cam C_1 the admission—*i.e.* the lead.

Variation of cut-off and reversal are effected by rotating the cams with respect to each other; for this purpose a quick-pitch screw is cut on an enlargement of the camshaft, and two large circular nuts which engage with this screw can be moved axially by links connected to cranks on the reversing shaft L. Rotational movement of the nuts relatively to the camshaft is transmitted to cams C and C_1, respectively, by means of rods, which slide in fitting holes or in slots, according to whether they engage with the part whose rotational motion they are to transmit, or whether they are to pass through a part without sharing in its motion.

In the position for running in a certain direction (see D of fig. 26) the cranks of the control shaft L are in such a position that movement of the shaft gives little motion to one link, that connected to the cam governing the lead, while maximum shift is given to the other, which controls the cut-off. It is thus that the cut-off can be altered without the lead being altered.

The mechanism in the gear-box works in oil, which should not require replenishment for six months.

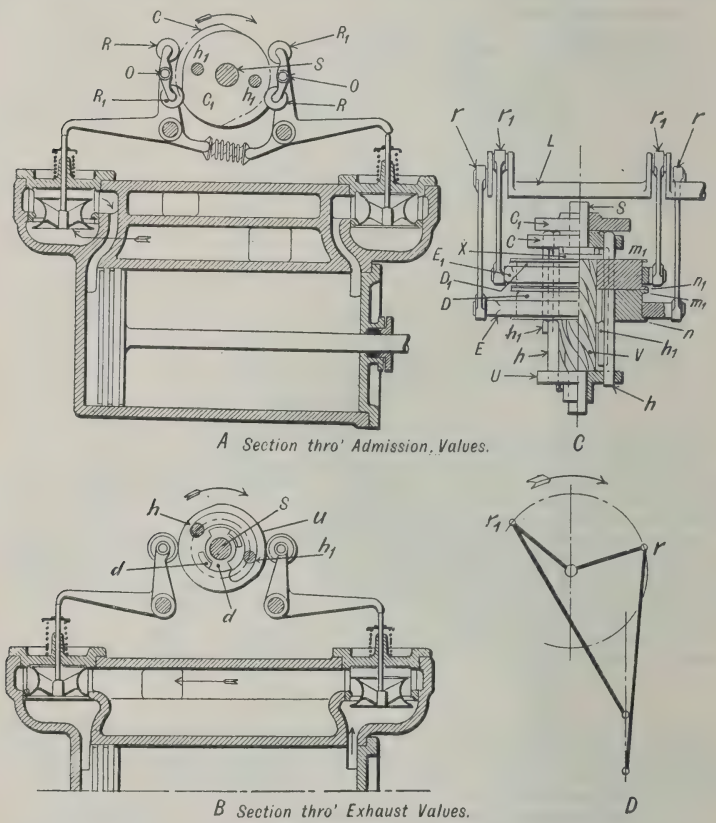


FIG. 26.

For a four-cylinder locomotive there are eight steam valves and eight exhaust valves, and all are operated from two gear-boxes.

The reversing wheel in the engine cab can be easily turned by one hand.

The valve gear has given good results as regards economy of steam and low costs of maintenance. Comparative tests on the L.M. and S. Railway have shown 46·2 lbs. coal per mile for an engine fitted with Caprotti gear, as against 58·3 lbs. coal per mile for a standard engine of the same type on similar service. (Abstracted from *The Engineer*, June 29, 1928.)

SECTION XXXI

INTERNAL COMBUSTION ENGINES

Revised and amplified by A. Vincent Clarke, Assoc.M.Inst.C.E., M.I.Mech.E., A.M.I.E.E.

With contributions by G. A. Burls, M.Inst.C.E., M.Inst.A.E.; T. B. Morley, D.Sc., M.I.Mech.E.; and G. E. Windeler, M.I.Mech.E.

PART I

GENERAL DATA RELATING TO POWER GAS . . pp. 1861-1867

PART II

GAS ENGINES . . . . . pp. 1869-1891

PART III

DIESEL ENGINES — SEMI-DIESEL ENGINES — THE 'STILL' (I.C. + STEAM) ENGINE—HIGH-COMPRESSION OIL ENGINES —LOW-COMPRESSION OIL ENGINES—INTERNAL COMBUSTION TURBINES—LLOYD'S RULES FOR MARINE INTERNAL COMBUSTION ENGINES—HOT-AIR ENGINES . . pp. 1893-1931

SECTION XXXI

Revised and amplified by A. Vincent Clarke, L.C.C., Assoc.M.Inst.C.E., M.I.Mech.E., A.M.I.E.E.

With contributions by G. A. Burls, M.Inst.C.E., M.Inst.A.E.; T. B. Morley, D.Sc., M.I.Mech.E.; and G. E. Windeler, M.I.Mech.E.

PART I

GENERAL DATA RELATING TO POWER GAS.

Interrelations of Units.

In calculations relating to Power Gas, the following physical data and interrelations of units will be found convenient for reference :—

1 metre = 39·37079 ins. = 3·2809 ft. = 3 ft. 3⅜ ins. (app.).
25·4 millimetres = 1 inch.
1 square inch = 6·4516 square centimetres.
1 cubic inch = 16·387 cubic centimetres.
1 gallon = 8 pints = 277·3 cubic inches (app.) = 4·5435 litres.
1 gallon of pure water at 62° F. weighs 10 lbs.
1 cubic foot = 6·23 gallons.
1 cubic foot of pure water at 62° F. weighs 62·3 lbs. = 28·315 litres.
1 cubic foot of average river water weighs 63 lbs.
1 cubic foot of sea water weighs 64 lbs.
1 litre = 1,000 cub. cm. = 61·03 cub. ins. = 1·761 pints.
1 pound (av.) = 7,000 grains = ·45359 kilogramme.
1 gramme = 15·432 grains = ·0022046 lb.
1 gramme per litre = 70·13 grains per gallon = 436·9 grains per cub. foot.
1 kilogramme = 15,432 grains = 2·2046 lbs.
1 British ton = 2,240 lbs.
1 metric ton = 2,205 lbs. = 0·9844 British ton.
1 atmosphere = 14·7 lbs. per sq. in. = 2116·8 lbs. per sq. ft.
" = 33·977 ft. of water.
" = 26,220 ft. of (homogeneous) air.
" = 30·0 ins. of mercury (app.).
" = 760 mm. of mercury (app.).
1 kilogramme per sq. cm. = 14·22 lbs. per sq. in.
" " = 0·9673 atmosphere.
1 inch of mercury = 0·490 lb. per sq. in.
1 inch of water = 0·0360 lb. per sq. in.
1 British thermal unit (B.Th.U.) = heat necessary to raise 1 lb. of water through 1° F.
" " = 778 ft.-lbs. of work.
1 centigrade thermal unit (C.Th.U.) = heat necessary to raise 1 lb. of water through 1° C.
" " = 1·8 B.Th.U.
" " = 1400·4 ft.-lbs. of work.

1 (major) calorie = heat necessary to raise one kilogram of water through 1° C.
" = 2·2046 C.Th.U.
" = 3·9683 B.Th.U.
" = 3087·3 ft.-lbs. of work.
1 British horse-power = 33,000 ft.-lbs. of work per minute.
" " = 42·416 B.Th.U. per minute.
" " = 2,545 B.Th.U. per hour.
" " = 23·564 C.Th.U. per minute.
" " = 1,414 C.Th.U. per hour.
" " = 10·69 calories per minute.
1 metric horse-power = 75 kilogrammetres per second,
" " = 0·9863 British horse-power.
1 kilowatt = 1·34 British horse-power.
Absolute temperature °F. (T) = 460 + ord. temp. °F.
" °C. = 273 + ord. temp. °C.

The volume of 1 lb. of a gas varies with its pressure and temperature and for purposes of comparison a definite pressure and temperature are selected. If normal pressure and temperature be one atmosphere and 32°F., this may conveniently be denoted by N.T.P.

Mass and Bulk of some Gases and Vapours at N.T.P.

Air is by *weight*, composed of 77 per cent. nitrogen and 23 per cent. oxygen.
By *volume* its composition is 79 per cent. nitrogen and 21 per cent. oxygen.
One pound of oxygen is contained in $\frac{100}{23} = 4{\cdot}35$ lbs. of air.
One cubic foot of oxygen is contained in $\frac{100}{21} = 4{\cdot}8$ cub. ft. of air.
One pound of dry air at N.T.P. occupies 12·387 cub. ft.
One cubic foot of dry air at N.T.P. weighs 0·08073 lb.
The following table gives data relating to the heaviness and bulkiness of the more usual gases:—

HEAVINESS AND BULKINESS OF SOME GASES, ETC., AT N.T.P.

Gas.	Symbol.	Specific Gravity; Hydrogen = 1.	Heaviness of 1 Cubic Foot at N.T.P.; Lbs.	Bulkiness of 1 lb. in Cubic Feet at N.T.P.
Column:—	1	2	3	4
Hydrogen	H_2	1	0·00559	178·89
Methane (marsh gas)	CH_4	8	0·04472	22·36
Water (gaseous)	H_2O	9	0·05031	19·88
Acetylene	C_2H_2	13	0·07267	13·76
Carbon monoxide	CO	14	0·07826	12·78
Ethylene (olefiant gas)	C_2H_4	14	0·07826	12·78
Nitrogen	N_2	14	0·07826	12·78
Air	$O_{23}+N_{77}$	14·44	0·08073	12·387
Oxygen	O_2	16	0·08944	11·18
Carbon dioxide	CO_2	22	0·12298	8·13
Normal benzene	C_6H_6	39	0·21801	4·59

p = abs. pressure, in lbs. per sq. foot
= 144 {14·7 + pressure in lbs. per sq. in. above atm.}
v = volume of 1 lb. of gas, in cubic feet;
T = absolute temperature in °F.;
= 460 + ord. temp. °F.

Then for a perfect gas:—

$$pv = c\,T, \qquad (1)$$

where c is constant for any, the same, gas.

For dry air at N.T.P.:—

$p = 14{\cdot}7 \times 144 = 2116{\cdot}8$ lbs. per sq. ft
$v = 12{\cdot}387$ cu. ft. (Table, col. 4.)
T = 460 + 32 = 492° F. (abs.).

Whence, by Eq. (1), we have $c=53{\cdot}29$. Thus the characteristic equation for dry air regarded as a perfect gas, is :—

$$pv = 53{\cdot}29\ \mathrm{T}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Equations (1) and (2) show that in a perfect gas the value of the product pv is proportional to the absolute temperature. If the pressure remains constant, the volume and absolute temperature are mutually proportional. So also, if the volume remain constant, the pressure and absolute temperature are mutually proportional.

The specific heat of dry air at constant pressure, k_p, is 0·2374 B.Th.U. per lb.

The specific heat at constant volume, k_v, is less than k_p by the heat equivalent of the external work of expansion under constant pressure.

If Eq. (1) be differentiated at constant pressure we have :—

$$p\ dv = c\ d\mathrm{T};$$

thus, putting $d\mathrm{T}=1°\mathrm{F}.$, we have $c=p\ dv$; so that c expresses the external work of expansion under constant pressure in foot-lbs. As Joule's equivalent, J, is 778 foot-lbs. per B.Th.U., we have for dry air :—

$$k_v = 0{\cdot}2374 - \frac{53{\cdot}29}{778} = 0{\cdot}1889.$$

The ratio of k_p to k_v is important in the theory of gas engines, and is usually denoted by γ : thus for dry air :—

$$\gamma = \frac{k}{k_v} = 1{\cdot}4056.$$

Isothermal and Adiabatic Expansion.

The two fundamentally important modes of expansion of gases are :—

(1) Isothermal expansion.
(2) Adiabatic expansion.

(1) *Isothermal Expansion.*—When a mass of gas changes volume at constant temperature, the change is described as 'isothermal.' In the case of perfect gases, Eq. (1) shows that in this case the value of the product pv remains constant. If $p_o\ v_o\ \mathrm{T}_o$ be respectively the initial values of pressure, volume, and (abs.) temperature, and pv any subsequent simultaneous values, we have in isothermal expansion :—

$$pv = p_o\ v_o = c\ \mathrm{T}_o \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

The external work W done in expanding from volume v_o to volume v, is expressed by :—

$$\mathrm{W} = \int_{v_o}^{v} p\ dv \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

as, in isothermal expansion, $pv = p_o\ v_o$, we have $p = \frac{p_o\ v_o}{v}$; hence (4) becomes for this case :—

$$\mathrm{W} = p_o\ v_o \int_{v_o}^{v} \frac{dv}{v} = p_o\ v_o\ \mathrm{Log}_e \frac{v}{v_o} \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$\frac{v}{v_o}$ is the volume ratio of expansion, and may be denoted by ρ; thus, from (5), we have in isothermal expansion :—

$$\mathrm{W} = p_o\ v_o\ \mathrm{Log}_e \rho = c\ \mathrm{T}_o\ \mathrm{Log}_e \rho \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

As the internal energy of a perfect gas depends only on its temperature, it follows that in isothermal change the internal energy of the gas remains constant. Hence, in isothermal expansion of a perfect gas the heat given to the expanding gas is the exact equivalent of the

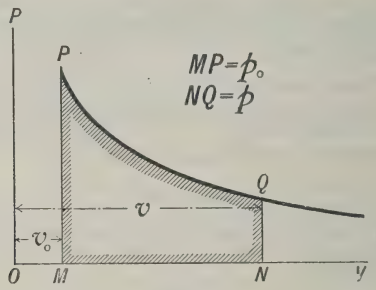


FIG. 1.

external work performed by the gas. And, conversely, in isothermal compression of a perfect gas, the heat that flows from the gas during compression is the exact equivalent of the work done on the gas in compressing it.

Frequently in engineering thermodynamics it is found that actual expansion and compression curves of diagrams can be closely represented by an equation of the form :—

$$pv^n = p_o v_o^n, \text{ a constant} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

where the index n is constant in value.

When $n=1$ we have the particular case of isothermal change, which has just been considered. For all other values of n, the external work done during expansion from volume v_o to volume v—represented by the hatched area MNQP of fig. 1—is given by :—

$$W = \int_{v_o}^{v} p\, dv = p_o v_o \int_{v_o}^{v} v^{-n}\, dv, \text{ by Eq. (7)}$$

That is,

$$W = \frac{1}{n-1}\left\{p_o v_o - pv\right\} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

The expanding substance being assumed a perfect gas, we have $p_o v_o = cT_o$, and $pv = cT$; hence, in expansion of this type Eq. (8) may also be written :—

$$W = \frac{c}{n-1}\left\{T_o - T\right\} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

Thus if n be greater than unity, expansion is accompanied by a falling temperature. If n be equal to unity the temperature remains constant during expansion. While if n be less than unity expansion is accompanied by a rising temperature. Equation (9) shows further that in expansion of this type the external work done is proportional to the change of temperature of the expanding gas.

Again, the heat expended, H, is equal to the sum of the external work and change of internal energy of the gas ; thus we have :—

$$H = \frac{c}{n-1}(T_o - T) + k_v(T - T_o)$$

as $c = k_p - k_v$, this becomes on reduction :—

$$H = \frac{n\,k_v - k_p}{n-1}(T - T_o) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

which shows that the heat expended is also proportional to the change of temperature of the expanding gas ; and that the gas expands with an apparent specific heat which is constant, and expressed by $\frac{nk_v - k_p}{n-1}$ B.Th.U. per lb.

(2) *Adiabatic Expansion.*—When a mass of gas changes its volume without any heat communication from without, the changes are said to be adiabatic.

Thus in adiabatic expansion the external work is done wholly at the expense of the internal energy of the expanding gas ; its temperature, and consequently its pressure, rapidly fall during expansion. And contrariwise, in adiabatic compression, the work done in compressing the gas appears as an increase of the internal energy of the gas ; hence, the temperature and consequently the pressure rapidly rise during adiabatic compression.

The adiabatic equation for a perfect gas is at once obtained from Eq. (10). For as, by supposition, there is in this case no heat communication with outside bodies, we have only to put $H = 0$ in Eq. (10), and then deduce the corresponding value of n. But if $H = 0$ we must have $nk_v - k_p = 0$, so that $n = \frac{k_p}{k_v} = \gamma$ in this case.

Hence, the adiabatic equation of a perfect gas is

$$pv^\gamma = p_o v_o^\gamma, \text{ a constant} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

By aid of Eq. (1) the following two equations are at once obtained from Eq. (11), giving the relation between change of temperature and volume, and change of temperature and pressure respectively, in adiabatic changes :—

$$T = T_o \times \left(\frac{v_o}{v}\right)^{\gamma - 1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

$$T = T_o \times \left(\frac{p}{p_o}\right)^{1 - \frac{1}{\gamma}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

For dry air, as $\gamma = 1{\cdot}4056$, we have accordingly from Eqs. (11) (12) and (13) :—

$$pv^{1{\cdot}4056} = \text{constant} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

$$T = T_o \times \left(\frac{v_o}{v}\right)^{0{\cdot}4056} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

$$T = T_o \times \left(\frac{p}{p_o}\right)^{0{\cdot}2886} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

Thus, in Diesel engines, the ratio is commonly about 13. Assume the initial temperature of the air to be 70°F., and the pressure to be atmospheric. Then $T_o = 460 + 70 = 530$; and $p = 14{\cdot}7$ lbs. per sq. inch : hence from (15) we have : $T = 530 \times 13^{0{\cdot}4056} = 1500°$F. or $1500 - 460 = 1040°$F. by ordinary scale. This is a sufficiently high temperature to at once ignite the charge of fuel oil on injection. The corresponding pressure from Eq. (14) is 540·8 lbs. per sq. in. abs. = 526 lbs. per sq. in. above atmosphere.

Moisture in Air.

Ordinary air always contains water vapour in greater or less degree, and the presence of this vapour modifies its physical properties.

The pressure of saturated steam formed at temperatures below 212°F. is usually described as the 'vapour tension' of water. In table, p. 1866, some data are given relating to the vapour tension of water from 32°F. to 212°F.

DATA RELATING TO AIR SATURATED WITH MOISTURE AT ATMOSPHERIC PRESSURE. ESTIMATED FROM 32°F.

Temperature. t°F.	Per 1,000 cu. ft. of Saturated Air.			Cu. ft. of Saturated Air, per Lb.	Cu. ft. of Dry Air, per Lb.	Density ratio of Saturated to Dry Air.	Heat of 1,000 cu. ft. of Sat. Air from 32°F. at t°F.				B.Th.U. per Lb. of Saturated Air.
	Lbs. of Dry Air.	Lbs. of Water Vapour.	Total Weight in Lbs.				B.Th.U. of Heat of Dry Air.	B.Th.U. of rise of Temp. of Water.	B.Th.U. of Vaporisation of Water.	Total B.Th.U. per 1,000 Cu. Ft.	
Column	1	2	3	4	5	6	7	8	9	10	11
32	80·241	0·3046	80·546	12·415	12·387	0·998	0	0	326	326	4·05
40	78·779	0·4073	79·186	12·628	12·56	0·995	149	3·26	434	586	7·4
50	76·943	0·5855	77·529	12·898	12·84	0·995	329	10·54	621	961	12·4
60	75·068	0·8264	75·894	13·176	13·09	0·993	499	23·14	872	1,395	18·4
70	73·112	1·1494	74·261	13·466	13·34	0·991	660	43·7	1,207	1,911	25·7
80	71·042	1·5625	72·605	13·773	13·60	0·987	810	75·0	1,633	2,518	34·7
90	68·821	2·146	70·967	14·091	13·85	0·983	948	124·5	2,231	3,304	46·6
100	66·396	2·833	69·229	14·445	14·10	0·976	1,072	192·6	2,930	4,195	60·6
110	63·691	3·731	67·422	14·832	14·35	0·968	1,179	291·0	3,839	5,309	78·7
120	60·650	4·926	65·576	15·250	14·60	0·957	1,267	433·5	5,041	6,741	102·8
130	57·204	6·329	63·533	15·740	14·85	0·943	1,331	620·0	6,442	8,393	132·1
140	53·269	8·130	61·399	16·287	15·11	0·928	1,366	878	8,230	10,474	170·6
150	48·729	10·309	59·038	16·938	15·36	0·907	1,365	1,217	10,378	12,960	219·5
160	43·498	12·903	56·401	17·730	15·61	0·880	1,322	1,652	12,917	15,891	281·8
170	37·442	16·000	53·442	18·712	15·86	0·848	1,227	2,208	15,925	19,360	362·2
180	30·431	19·802	50·233	19·917	16·11	0·809	1,069	2,931	19,594	23,594	470·0
190	22·406	24·390	46·796	21·369	16·37	0·766	840	3,854	23,992	28,686	613·0
200	13·117	29·762	42·879	23·322	16·62	0·713	523	5,000	29,101	34,624	807·5
210	2·460	35·971	38·431	26·021	16·87	0·648	104	6,403	34,949	41,456	1078·7
212	0·000	37·383	37·383	26·750	16·92	0·632	0	6,729	36,277	43,006	1150·4

DATA RELATING TO THE VAPOUR TENSION OF WATER FROM 32°F. TO 212°F.

At °F.	Absolute Pressure, in Lbs. per Sq. In. P′.	Cu. Ft. Occupied by 1 Lb. *u*.	Latent Heat, in B.Th.U. per Lb.	At °F.	Absolute Pressure, in Lbs. per Sq. In. P′.	Cu. Ft. Occupied by 1 Lb. *u*.	Latent Heat, in B.Th.U. per Lb.
Col. :	1	2	3	Col. :	1	2	3
32	0·089	3,283	1070·5	130	2·209	158	1017·9
40	0·122	2,455	1066·3	140	2·871	123	1012·3
50	0·177	1,708	1060·9	150	3·699	97·0	1006·7
60	0·253	1,210	1055·7	160	4·719	77·5	1001·1
70	0·359	870	1050·3	170	5·970	62·5	995·3
80	0·502	640	1045·1	180	7·492	50·5	989·5
90	0·691	466	1039·7	190	9·310	41·0	983·7
100	0·939	353	1034·3	200	11·496	33·6	977·8
110	1·264	268	1028·9	210	14·09	27·8	971·6
120	1·681	203	1023·3	212	14·70	26·75	970·4

When air contains a maximum of water vapour it is said to be saturated. The quantity of moisture depends only upon the temperature, and is estimated by aid of :—

DALTON'S LAW.—The pressure, and consequently the mass of vapour which saturates a given volume, is the same at the same temperature, whether the volume already contains a gas or is vacuous.

Consider a volume of V cubic feet of saturated air at a pressure of P lbs. per sq. in., and temperature t°F. By Dalton's Law this consists of (1) V cubic feet of water vapour at the pressure P′ lbs. per sq. in. corresponding to t°F., as in column 1 of above table, together with (2) V cubic feet of dry air at pressure (P − P′) lbs. per sq. in., and temperature t°F.

Thus the weight, S, in lbs. of 1,000 cubic feet of saturated air at temperature t°F. and pressure P lbs. per sq. in. is :—

$$S = \frac{1000}{u} + 80{\cdot}73 \times \frac{492}{460+t} \times \frac{P - P'}{P} \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

the values of u being given in column 2 of table. If the pressure be atmospheric, P = 14·7, and in this case :—

$$S = \frac{1000}{u} + 80{\cdot}73 \times \frac{492}{460+t} \times \frac{14{\cdot}7 - P'}{14{\cdot}7} \quad . \quad . \quad . \quad . \quad . \quad (18)$$

By aid of Eq. (18), the table on p. 1866 has been compiled.

This table may be useful in connection with the heat expended in providing the moisture-laden air supplied to gas producers.

Efficiency.

The most important thermodynamic cycles are the constant volume (C_v) and constant pressure (C_p) cycles, and the general formulae for efficiency are as follows :—

$$\text{Constant volume efficiency} = 1 - \frac{T_1 - t}{T - t_c}.$$

$$\text{Constant pressure efficiency} = 1 - \frac{\frac{1}{\gamma}(T_1 - t_1) + (t_1 - t)}{T - t_c},$$

where,

t = initial temperature of air ;

t_c = temperature of compression ;

T = temperature after explosion ;

T_1 = " " adiabatic expansion ;

γ = ratio of specific heat $\frac{C_p}{C_v}$;

$t_1 = T_1 \frac{\text{atmospheric pressure}}{\text{pressure of explosion}}$.

SECTION XXXI

PART II

GAS ENGINES.

Types of Gas Engines.

Gas engines may be classified under the following types:

Type I.—Engines igniting their charge at constant volume, without compression. This type is obsolete; it included the Lenoir and Hugon.

Type II.—Engines igniting at constant volume without compression, with free piston, and utilisation of vacuum during instroke. This type also is obsolete; it included the Barsanti, Wenham, and Otto and Langen.

Type III.—Engines igniting at constant pressure, with previous compression of the charge. This type includes the early Brayton and Simon engines, and the modern Diesel engine.

Type IV.—Engines igniting at constant volume, with previous compression. This type includes the very great majority of gas engines in everyday use.

The vast majority of present-day gas engines are operated upon the Otto, Beau de Rochas or four-stroke cycle, wherein the sequence of strokes is suction, compression, working, and exhaust; and in most cases, particularly of British engines, the cylinders are single-acting; the large double-acting engines have been perfected mainly by the great Continental firms.

A number, especially of engines of large size, are built upon the two-stroke cycle, both single and double-acting, and much attention has been devoted to this type.

Crossley Gas Engines.

Messrs. Crossley Bros. Ltd., of Openshaw, produce annually a very large number of gas engines of horizontal and vertical design, working on the four-stroke cycle. The range of power is from 2 to 260 b.h.p., and nearly 100,000 engines have been supplied.

Horizontal engines of 30 b.h.p. and upwards can be made with two cylinders, the flywheel being mounted on the crankshaft between the cylinders. These double-cylinder engines are specially suitable for all purposes where very steady driving is required, as they run with great regularity. The vertical engines, which are of the four-cylinder high-speed design, are also most suitable for driving dynamos.

The rated powers of the producer-gas engines are obtained when running on gas made from anthracite, bituminous coal, and wood fuel, but are reduced by 6 per cent. when running on gas made from coke. Working on town gas, the increase in power varies from about 20 per cent. on the small sizes to about 1½ per cent. on the largest size, which develops 130 b.h.p. in one cylinder. The weight of the standard horizontal engine varies from about 2½ to 3 cwts. per b.h.p., and the weight of the double-cylinder engines is about 2 cwts. per b.h.p.

The sizes from 6 b.h.p. and upwards are now all governed on the throttle, giving a smooth drive and low pressures at reduced engine loads. Fig. 1 shows a recent horizontal single-acting four-stroke design; the girder-type bed and balanced crankshaft will be noted. Other features included are the absence of any liner bolts in the water-jackets; adjustable timing to the electric ignition; double ring lubrication to the bearings with forced lubrication to the cylinders and other parts; screw-down producer-gas cock; hardened cams, rollers, and pins; and a combustion chamber so designed that dust and oil cannot collect at the bottom and form troublesome deposits, but are swept out of the exhaust valve with the burnt gases.

The Crossley improved open-hearth suction gas plants are made for anthracite, coke, and charcoal in various sizes from 3 b.h.p. upwards. Also their bituminous suction gas plants are made to use all classes of bituminous fuels containing up to 15 per cent. ash. In addition, their wood and vegetable refuse producer plants can use as fuel, wood blocks, sawdust, bark, sisal refuse, cocoanut shells, coffee husks, mealie cobs, cotton seed, rice husks, sugar-cane refuse, tea prunings, and other waste fuels.

CROSSLEY GAS ENGINE.

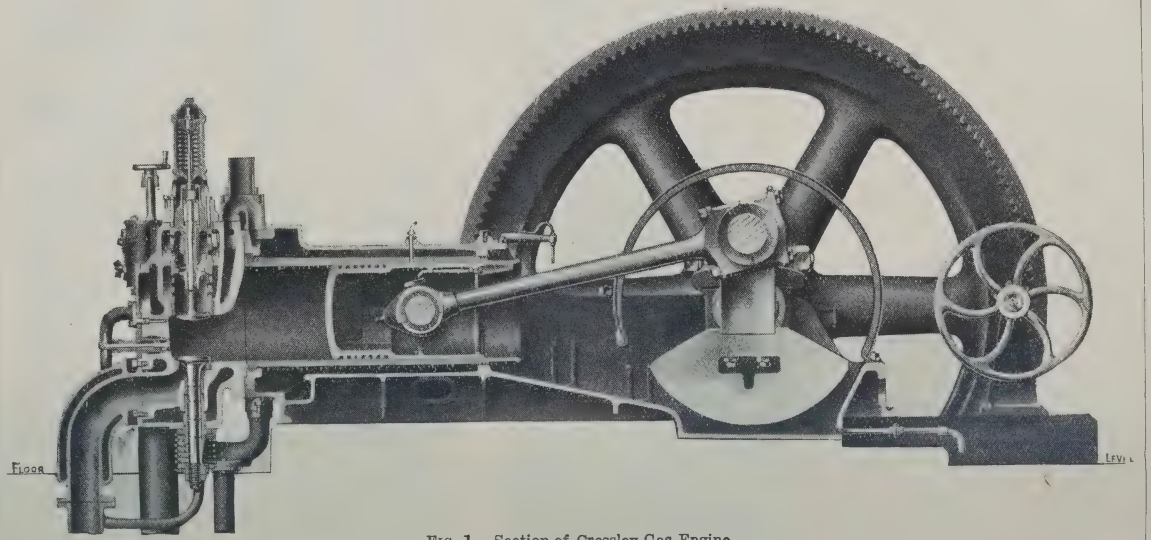


FIG. 1.—Section of Crossley Gas Engine.

Ruston-Hornsby Gas Engines.

The gas engines manufactured by Messrs. Ruston & Hornsby, Ltd., Lincoln, are made in single- and multi-cylinder sizes of both the horizontal and vertical four-stroke cycle types. The range of powers for the horizontal engines is from 13 to 340 b.h.p., and for the multi-cylinder vertical engines from 45 to 330 b.h.p.

These engines have a reputation for strength, reliability, and ease of management, and great attention is paid to mechanical efficiency and steadiness of running. All working parts are designed to allow for ready removal for examination without cumbersome dismantling, and efficient lubrication is provided throughout.

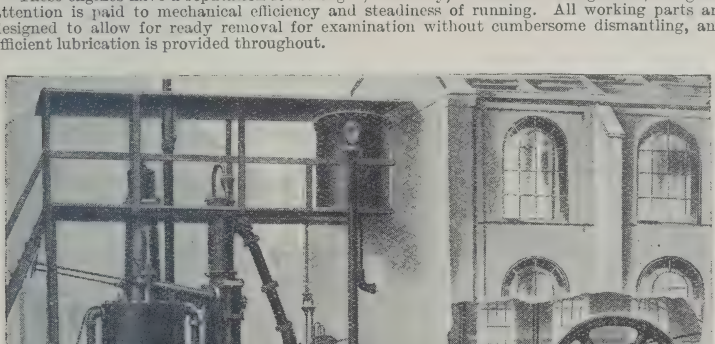


Fig. 2.—Ruston Gas Engine with Suction Producer.

The engines work satisfactorily and with low fuel consumption on anthracite, wood, or other fuel gases. When supplied for working on the oilfields they are arranged for either gas or oil, so that the natural gas found when well sinking can be used, and when this is diminished the engine can be changed over to oil fuel.

Governing is effected by a spring type of governor, extremely sensitive, and arranged so that the speed is kept constant under all loads. The governor acts on the principle of varying the amount of gas and air admitted to the cylinder, but in no case cutting off the charge altogether.

Both the quality and the quantity of the mixture are regulated, the quality by reducing the amount of gas proportionally to the work, and the quantity by reducing the amount of air to the best proportion to give combustion of the gas admitted. This ensures economy of gas, certainty of ignition, and that absolute steadiness and smoothness of running which is essential for certain classes of electrical driving.

National Gas Engines.

Horizontal and vertical four-stroke cycle gas engines are made by the National Gas Engine Co., Ltd., of Ashton-under-Lyne, the power ranging from 2 to 2,000 b.h.p. Up to 10 b.h.p. the engines are of the horizontal overhung cylinder design; for greater powers a special design of girder bed and supported cylinder is adopted.

The horizontal engines are built in various sizes up to 100 b.h.p. per cylinder. Ignition is by low-tension magneto, a sparking-plug being fitted in the combustion chamber. Governing of the smaller sizes is on the hit-and-miss principle, but throttle governing can be supplied if desired, and the larger sizes of engines are fitted with a patent variable admission governing gear.

The National Gas Engine Co.'s vertical engines are of the (*a*) single cylinder per crank design, and (*b*) of the tandem single-acting design. The single cylinder per crank engines are made in three sizes, viz. 14 ins., 18 ins., and 24 ins. stroke, and the powers vary from 90 to 900 b.h.p. The tandem engines are built with 18-in. stroke and 24-in. stroke, and the powers range from 300 to 2,000 b.h.p. The engines are suitable for working on town, producer, blast-furnace, and coke-oven gases, and the cyclic variation is such that there is no difficulty in running electric alternators in parallel.

A recent test of an engine of this type furnished the following results:

Independent test on 600 b.h.p. National Single Cylinder per Crank Vertical Gas Engine, at the works of Newton, Chambers & Co., Ltd., Thorncliffe Collieries, near Sheffield:

Date	May 14, 1926
Duration of test	5 hours
Gas used as measured by meter	53,250 cu. ft.
Barometer	29·42 ins.
Gas pressure	4·95 ins. of water
Temperature of gas	75·7° F.
Gas used per hour at 32° F. and 30 ins. of mercury	9,600 cu. ft.
Calorific value of gas 32° F. and 30 ins. of mercury	491·3 B.Th.U.'s per cu. ft. (higher value 542)
Total kilowatt-hours generated	1,920
Average kilowatts	384
Exciter ,,	4·2
Revolutions per minute	298
Diameter of cylinders	18 ins.
Stroke of pistons	18 ,,
Mean effective pressures, average	64·35 lbs. per sq. in.
Indicated horse-power, total	665·3
B.h.p. total, assuming alternator efficiency 0·93 and excited efficiency 0·8	560·0
Mechanical efficiency of engine	0·84
Efficiency of plant	0·77
Gas at 32° F. and 30 ins. of mercury:	
Used per i.h.p. per hour	14·4 cu. ft.
,, b.h.p. ,,	17·1 ,,
,, kw. hour	25·0 ,,
B.Th.U. per i.h.p. per hour	7,080
,, ,, b.h.p. ,,	8,400
,, ,, kw.-hour	12,300
Indicated thermal efficiency	36 per cent.
Brake thermal efficiency	30·3 ,,
Total thermal efficiency (neglecting power to drive circulating water pump)	27·8 ,,
Load factor	0·96
Circulating water, total	349,400 lbs.
Average inlet temperature of circulating water	92·1° F.
Average temperature of water leaving cylinders	117·3° F.
Average temperature of water leaving exhaust pipe jacket	121·1° F.

Heat Balance.

Dr.		
To calorific value of 1 cu. ft. of gas at N.T.P.	491·3	B.Th.U.
Cr.		
1. By heat given up to circulating water from cylinder walls	183·4	37·4
2. By heat given up to circulating water from exhaust pipe	27·7	5·6
3. Heat converted to electrical energy	136·3	27·8
4. Heat comprising mechanical and electrical losses in engine and alternator	40·5	8·2
5. Heat lost in exhaust gases by radiation, conduction, and otherwise unaccountable for	103·4	21·0
	491·3	100·0

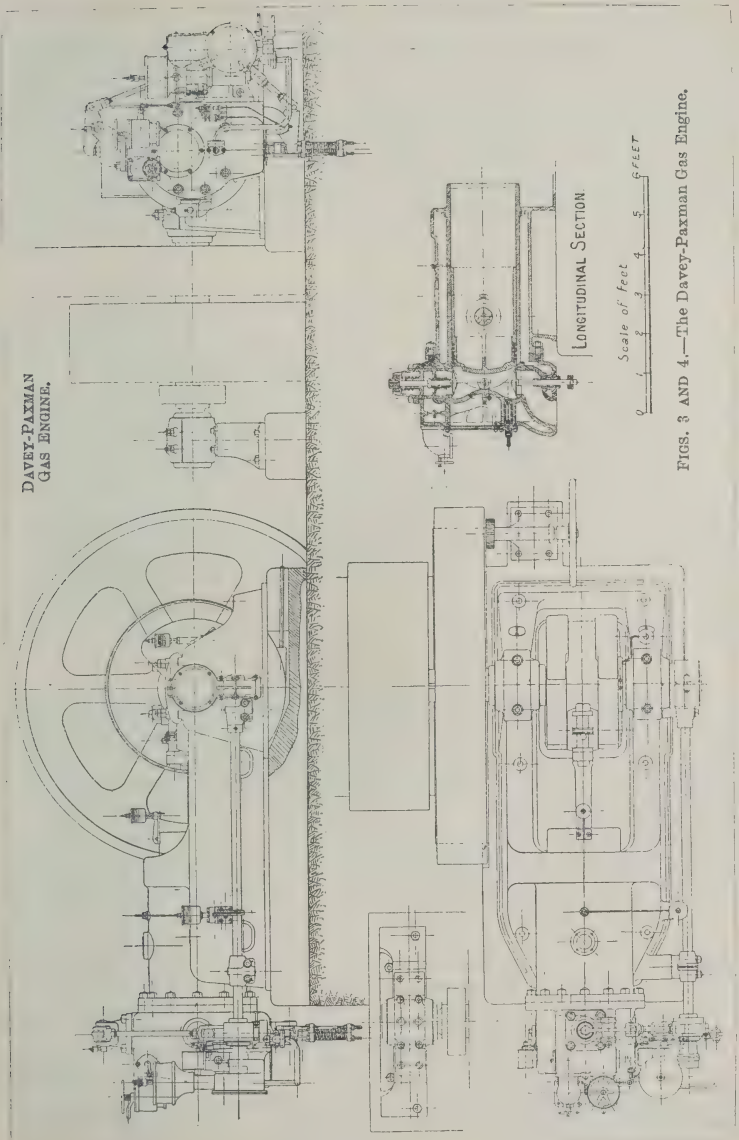
VICKERS GAS ENGINES.

Large gas engines of the horizontal four-stroke cycle type are constructed by Vickers-Armstrong Ltd., Barrow-in-Furness. The engines are double acting, and range from 800 b.h.p. in single line tandem cylinders to 5,000 b.h.p. in twin tandem engines.

A number of these engines have been supplied for driving dynamos, alternators, etc. Eight of these engines for the Appleby Iron Co., of Appleby and Frodingham, have cylinders 800 mm. dia. by 1,200 mm. stroke, and develop 1,200 kw. when running at 100 r.p.m.

Separate cylinder liners are fitted. The inlet valves are located on top, and the exhaust valve at the bottom of the cylinder; they are operated by shaped rolling levers driven by eccentric on the half-speed shaft. The crankshaft is forged in one piece and balance weights are fitted. The main bearings are in four parts, lined with white metal, and have self-aligning wedge adjustment. The main bearings of the 1,500 b.h.p. engine are 21 ins. diam. and the piston rod 10·75 ins. diam.; the latter is hollow to convey the cooling water to the pistons.

Governing is on the 'quantity and quality' system, and it is possible to run direct-driven alternators in parallel. Hand adjustment is provided for regulating the speed while running. Separate hand control for gas and air for each end of both cylinders is provided.



FIGS. 3 AND 4.—The Davey-Paxman Gas Engine.

Davey, Paxman & Co.'s Gas Engines.

The Paxman gas engine of the horizontal type, in sizes from 23 to 270 b.h.p., operates on the four-stroke cycle. For large powers two engines are coupled with a single crankshaft, having the flywheel between the cylinders. The engine is illustrated in figs. 3 and 4. Producers, suitable for anthracite and coke, are also made by Davey, Paxman & Co., Ltd., Colchester.

Special features of the engines are the shape of the combustion chamber, which gives an excellent combustion of the fuel together with freedom from fracture, and the patented timing arrangement for the spark to facilitate starting, the general design following accepted practice and having supported cylinder with removable liner.

Gas engines of the vertical design are also made in various sizes from 25 b.h.p. to 900 b.h.p. in single and multi-cylinder types, and are most suitable for driving factories, electric generators, etc. They occupy little floor space and have a small cyclic variation.

A large number of Davey, Paxman & Co.'s gas engines are in operation, working on a variety of fuels. The utmost attention has been given to securing reliability, and their design of combustion chamber makes the engines very economical.

Fielding & Platt's Gas Engines.

Gas engines of the horizontal four-stroke cycle type are manufactured by Fielding & Platt Ltd., Gloucester, in various sizes from 6 to 150 b.h.p. For larger powers, two engines are coupled together, having a flywheel between the cylinders mounted on the two-throw crankshaft, these engines being made in sizes up to 300 b.h.p. Four-cylinder engines are also made.

The cylinder has a separate liner supported throughout its whole length in the water-jacketed extension of the bed. The variable-admission governor controls the opening of the gas-and-air-mixing valve, thereby varying the volume of charge admitted to the cylinder. The stroke of the mixing-valve is directly under the control of the governor, and, although the volume of charge is varied, the ratio of gas and air in the mixture remains practically constant. By this means an explosion takes place every cycle, and it is claimed that, when working with a suction plant, this steady method of drawing the gas ensures a better quality of gas, and avoids the possibility of trouble through insufficient gas under variable loads.

The cylinder jacket and combustion chamber have separate water circulation, and special care has been taken to reduce the number of parts. Ignition is by means of a low-tension magneto, and the ignition plug can be readily removed from the cylinder for examination. The engines are suitable for running on gas produced from a large variety of fuels, and the ' Fielding ' suction gas plants have several important features, enabling the plant to be quickly started up and to respond instantly to all varying loads.

The Fielding gas engine installations are extremely simple to operate, and the makers state that any man of ordinary intelligence can be entrusted to work it, as they have made a special study of simplicity.

Premier Gas Engines.

The Premier Gas Engine Co., Ltd. (*Crossley Brothers, Ltd.*), of Sandiacre, specialise in multi-cylinder gas engines of the horizontal type, ranging in powers from 120 to 1,000 b.h.p. Large numbers of these engines have been installed for all classes of work, including the driving of alternating current generators in parallel. The speeds of the engines are suitable for coupling to electric generators, and the engines have but a small cyclic irregularity.

From the sectional view, fig. 5, the arrangement of the cylinder, combustion chamber, valves, and cam operating gear can be seen. It will be noticed that the breech end and cylinder form one casting, and the usual joint in the combustion space is avoided. This removes any risk of water leaking into the cylinder.

These engines work on the four-stroke cycle and are made in two, three, and four cylinders, bolted to massive beds which support the crankshafts. The cranks are completely covered by oil splash guards, and the main bearings are supplied with oil by a pump, which is of the valveless oscillating type, operated by the side shaft. The cylinders, piston pin, crank pin, and exhaust valve spindles are also lubricated under pressure automatically.

The valves open directly into the cylinders, the inlet valve being directly over the exhaust valve, a design introduced by the Premier Co. as far back as 1893. A large throttle valve is fitted to control the air supplied to all the cylinders, and can be readily adjusted by hand; in addition, separate throttle valves are fitted on each cylinder. All the valves are operated by a shaft running in self-oiling bearings behind the cylinder, so arranged that the jacket inspection cover can be removed without disturbing it.

Governing is effected by controlling a throttle valve in the gas supply, and the governor is of the high-speed type, powerful and sensitive, driven direct from the crankshaft.

Starting is automatic by means of compressed air, which makes the engines easy to start.

Except in cases where engines are driving alternators in parallel, an isolating valve is fitted in the gas manifold by which the gas supply to one or more cylinders may be shut off when working on light loads with a consequent saving in fuel consumption.

These engines have the advantages of even turning moment, good balancing, forced lubrication, and they occupy little floor space for their power. Very high economy, both in fuel and lubricating oil, is claimed.

As will be seen from the illustration, the Premier gas engine is very accessible, both for cleaning and overhauling, as all of the working parts can be inspected and reached from floor level. This allows of good supervision, and as the piston and connecting-rod can be removed from the crankshaft end of cylinder, this operation can be carried out without disturbing any part of the valve gear or cylinder covers.

The Premier Gas Engine Co., Ltd., also manufacture a special type of cross-head vertical gas-engine, which differs from the usual type in many ways and possesses special features. It is made with 2, 3, or 4 cylinders for moderate powers at moderate speeds. The lower ends of the cylinders are shut off from the crank case by light covers, through which piston rods pass. Cross-heads and guides are therefore used, and the design corresponds in many respects to that of a steam-engine.

The Premier Company were the first to introduce this cross-head type of gas engine, and it is now generally adopted by Diesel engine makers in this country and on the Continent of Europe.

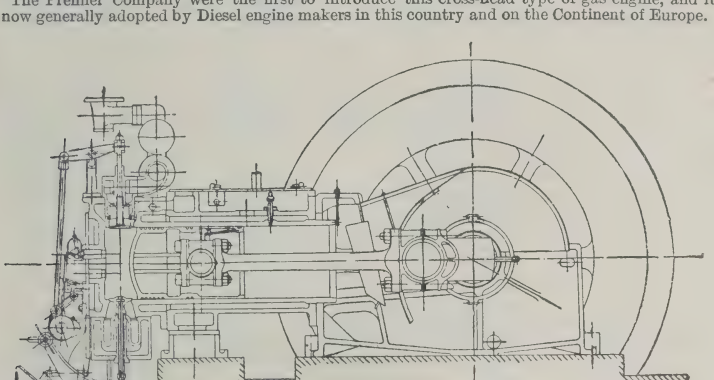


FIG. 5.—Premier Gas Engine.

For powers of 1,000 b.h.p. to 5,000 b.h.p., the Premier Company manufacture a special type of double-acting vertical engine. This engine is of the four-stroke type, and therefore with four cylinders there are four impulses per revolution. As there is a compression on every stroke, smooth running is obtained as a result of this and also because of the forced lubrication system which supplies oil at a pressure to all motion joints.

The design is suitable for powers of 1,000–2,250 b.h.p. continuous load with four cylinders and one crank, and for twice these loads, 2,000–4,500, with two-crank and eight-cylinder design, the speeds being from 125 r.p.m. for the smallest to 94 r.p.m. for the largest. A single crank for every four cylinders is one of the greatest innovations of the engine, and the small space required for this type of engine is a valuable characteristic.

It is specially adapted for the generation of electricity at steelworks and blast furnaces, especially where gases are available as a bye-product, and not produced artificially, and the rough usage of a steelworks is more readily resisted by this moderate speed engine. Fig. 6 shows, in plan and sectional elevation, the D.A.V. type of engine.

TANGYE GAS ENGINES.

Horizontal gas engines of the four-stroke cycle type are made by Tangyes, Ltd., Birmingham, in sizes from 3½ to 490 b.h.p. Two designs, AA and T, are supplied, and a large number of engines have been sold.

The smaller sizes of engines have horizontal cylinders bolted to the beds, and the liners are removable. Governing is on the hit-and-miss principle; the AA design have highly sensitive inertia governors.

Engines of the HT type have the cylinders well supported by the bed, which extends up to the combustion chamber and forms the water-jacket. The cylinder liners are separate and are made of a special hard mixture of cast iron. Governing is on the variable-admission system with Tangye's patent operating gear, which controls the quantity of the charge admitted to the cylinder and an impulse is obtained every cycle.

THE PREMIER GAS ENGINE.
(D.A.V. type.)

Plan and Sectional Elevation.

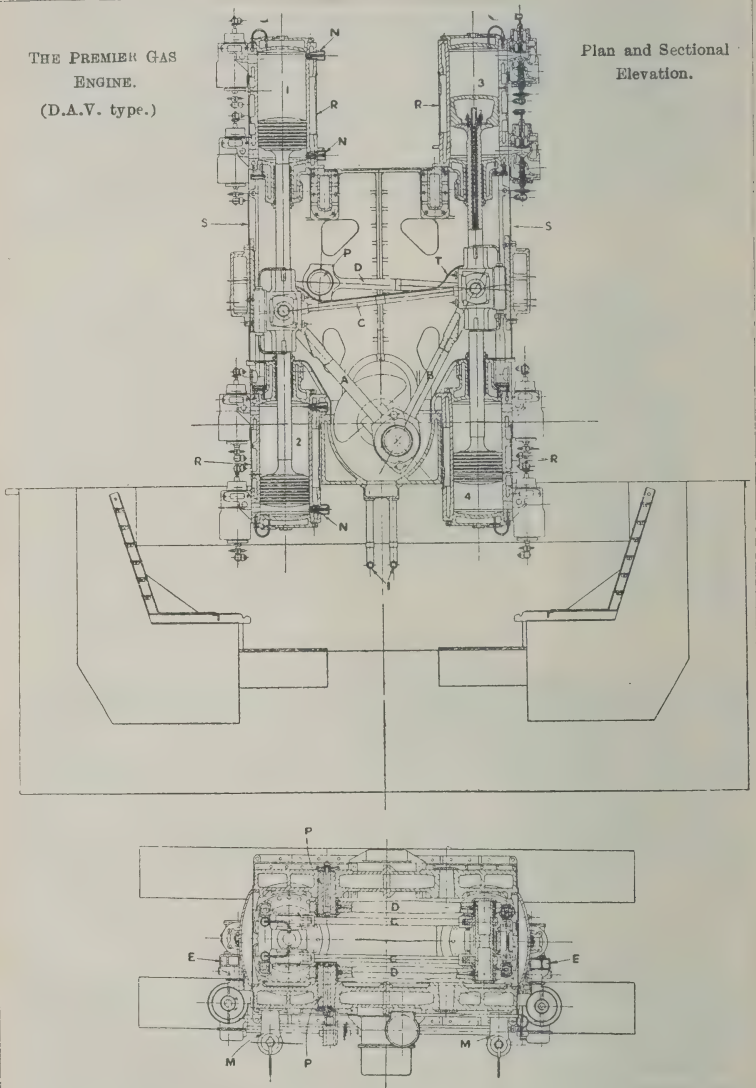


FIG. 6.

Ignition is by means of low-tension magneto, the spark being produced in the combustion chamber by means of a mechanically operated make-and-break arrangement, with variable timing device. On the larger sizes the sparking-plug is arranged so that the internal sparking points can be cleaned while the engine is at work. The engines are designed so that they can make long continuous runs; they are very simple to operate and are easily started, compressed-air apparatus being supplied with the larger sizes.

Tangyes were among the first firms to call attention to the fuel economy to be obtained from the use of gas engines with suitable suction gas producers, especially as engines would work on such inferior fuels as wood refuse, etc. The economy may be gathered from the following table which gives the approximate cost of fuel in a plant developing continuously 50 b.h.p. for sixty hours:

Nature of Fuel.	Rate of Fuel Cost.	Consumption per b.h.p. per hour.	Cost of 50 b.h.p. for 60 hours per week.
'Town' gas	2/- per 1,000 cubic ft.	15 cubic ft.	£4 10 0
Anthracite	40/- per ton	·75 lb.	2 0 0
Gas coke	30/- "	1 lb.	2 0 0
Charcoal	40/- "	1½ lbs.	3 7 0
Bituminous non-caking coal	24/- "	1½ lbs.	2 0 0
Wood refuse	Nil	2½ lbs.	Nil

The figures given above are an average, but the cost of running will depend, of course, upon the size of the engine and the relative prices and qualities of the different fuels.

Gardner Gas Engines.

The Gardner engines are made of both vertical and horizontal design and operate on the four-stroke cycle. The range of sizes is from 1½ to 180 b.h.p. Horizontal engines up to 14 b.h.p. can be supplied to work on either town gas, paraffin, or spirit.

The engines are of very neat design and substantially constructed. They give particularly smooth running, and a large number have been supplied.

The small sizes have been specially designed for gaseous and liquid fuels and are substantially constructed in order to make them suitable for heavy commercial duties and long continuous runs, such as for electric-lighting, etc. The workmanship and design of even the smallest size are exceptionally high class, as the engines are intended for production in large quantities.

The larger horizontal engines are fitted with single fly-wheel and outer pedestal bearings. The special features are the sensitive centrifugal throttle governor, giving low-speed variation, and the silent-running that is obtained. They are easy to start, economical to work, and have a high efficiency.

In addition, a range of Gardner engines made in twelve sizes, from 10 to 180 b.h.p., have been designed for marine and stationary work. They are supplied for running on town and producer gas, paraffin, spirit, and heavy oils, and are most suitable for marine work, driving electric generators, and for industrial purposes.

Peter Brotherhood Gas Engines.

Horizontal engines of the single-cylinder, four-stroke cycle type, developing 10 to 150 b.h.p., are manufactured by Peter Brotherhood, Ltd., Peterborough. The engines are made with two cylinders in sizes from 172 to 300 b.h.p. Producer gas plants capable of working on a wide range of fuels are also made by this firm.

The engines are very simple and neat in appearance, being well proportioned. They are easily operated, and there are no parts liable to get out of order. Several patented features are embodied in the design, which make for efficiency and reliability.

The governor is on the variable-admission principle; the charge of gas and air admitted to the cylinder is varied in quantity to suit the load, but there are no cut-outs and the engine receives an impulse every cycle. Special attention has been given to all the working parts of the valve gear, etc., the cams are of hardened steel and the valves are made from nickel steel, alloy steel being used for the exhaust valve lever.

The bed of the engine is formed with the cylinder jacket and supports the cylinder liner throughout its length. The combustion chamber has specially large water-jacket spaces, which are quite accessible for cleaning, it only being necessary to remove a cover.

Ignition is by means of a low-tension magneto, having powerful double magnets; special timing gear is fitted so that the engine can be easily started. The ignition plug is arranged to

prevent any possibility of the striking spindle bearings becoming coated with rust or deposit, and patented terminals are used on the plug and magneto.

Positive lubrication has been provided, so that when running the engine requires very little attention. A low-pressure self-starter can be fitted for engines of moderate powers, but, for engines of 100 h.p. and over, air starting is recommended, as with this it is claimed that even the largest engines start up as easily as a steam engine.

A number of improved features have been embodied in the producer, making it very simple to operate and capable of working with all grades of anthracite, coke, charcoal, and other inferior fuels.

Lilleshall Gas Engines.

Gas engines ranging in power from 500 to 4,000 b.h.p. are manufactured by the Lilleshall Co. Ltd., of Oakengates, Shropshire. The engines are of the tandem, horizontal, double-acting, four-stroke cycle type, and a working impulse on every stroke is obtained. A section is shown in fig. 7 of the 2,500 b.h.p. engine, from which the general disposition of parts can be clearly seen.

The Lilleshall Co. has recently introduced a number of important improvements, making for additional strength and reliability. The pistons and rods are, of course, water-cooled, the water supply being of a pressure of from 45 to 65 lbs. per square inch on account of the reciprocating motion of these parts; this is supplied by a pump worked from the crankshaft. For the jackets of the cylinders and covers the water is supplied at a pressure of about 15 lbs. per square inch. Each cylinder is provided with an open tank into which all exit water-pipes deliver, the discharge from each being thus always in view; each exit-pipe is fitted with a thermometer and regulating valve by which the temperature of the cooling water can be maintained at the desired degree.

These engines are installed to a very large extent in iron and steel works and collieries, as well as in electrical power stations.

An output up to 1,500 b.h.p. can be developed in one cylinder, the consumption of heat per b.h.p. hour amounting—at normal output—to about 9,900 B.Th.U.

The engines are very strongly constructed, and simple in general arrangement as well as in each individual part. The surface pressures on the moving parts are small, and their lubrication is effected automatically. The frame, cylinder, connecting piece, and rear slide are centrically connected with one another, so that the engine axis is accurately fixed.

The cast-iron cylinders are symmetrically formed, in order to avoid undue strains caused by the heat, and are amply cooled. The cooling-water jackets are easily accessible for purposes of cleaning. In addition to the cylinders, the exhaust-valve seats, the hollow piston rod, and the pistons, are cooled.

The cooling-water consumption amounts per b.h.p. hour to 7·7 gallons at an inlet temperature of 60° F.

The weight of the pistons and piston rod is not carried in the stuffing boxes, but by special crossheads, in the frame, the connecting piece, and the rear slide.

The inlet valve gear consists of a positive action which opens the air valve and gas inlet valve. By means of the governor the valve lift, and consequently the quantity of mixture, is changed according to the engine output. The ignition is effected electrically by a small accumulator battery. The time of ignition can be easily regulated during operation.

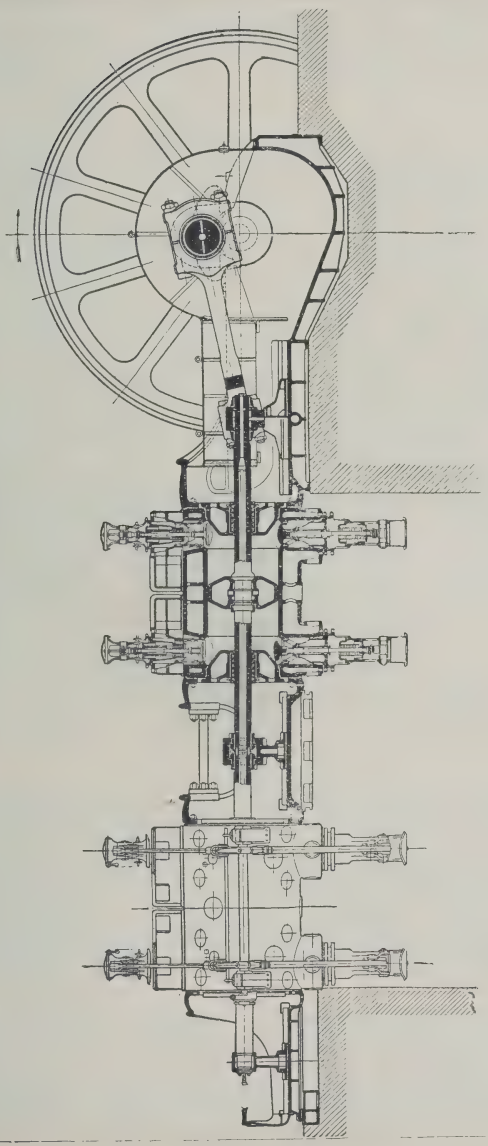
By means of compressed air the engines are very easily started.

Alternate-current dynamos driven by these gas engines can be connected in parallel without any trouble.

From tests of a tandem double-acting horizontal four-stroke engine of this type, developing 1,200 b.h.p. on blast-furnace gas, and having cylinders of 33·46 ins. bore with 43·3 ins. stroke, the following results were obtained :—

Test Results from a 1,200 h.p. Engine using Blast-Furnace Gas.

Item.	Quarter Load.	Half Load.	Three-Quarter Load.	Seven-Eighths Load.	Full Load.
Duration in minutes	33	28	29	26·8	25·33
Revs. per minute	106	105·8	106·3	106·5	105·6
I.H.P.	—	807	1,146	1,312	1,427
B.H.P.	280	557	871·5	1,037	1,186
Mechanical efficiency per cent.	48·5	69	76·2	79	83·1
M.E.P. in lbs. per square inch	—	42·5	60·0	68·7	75·3
Cu. ft. gas per b.h.p. hour	213	145·6	120·5	109·5	101·8
B.Th.U. per cu. ft. of gas	88·5	88·5	89·4	89·6	87·9
B.Th.U. per b.h.p. hour	18,532	12,262	10,794	9,921	8,976
Per cent. brake thermal efficiency	13·8	20·9	23·7	25·8	28·5



Lilleshall Gas Engine.

Fig. 7.—Longitudinal Section of Lilleshall Gas Engine.

Galloway Gas Engines.

Messrs. Galloways, Ltd., Manchester, took up in 1909 the manufacture of the large gas engines of the Ehrhardt and Sehmer type and have since introduced many improvements. The range of powers is given in the following table :—

Electric Generating Engine.

Cylinder Diameter.		Stroke.		R.P.M.	Normal continuous B.H.P.	Max. continuous B.H.P.	Max. Momentary B.H.P.	Blowing Engines.	
Mm.	Ins.	Mm.	Ins.					Max. R.P.M.	Normal B.H.P. at Max. R.P.M.
950	37·4	1,100	43·3	107	1,200	1,300	1,440	95	1,060
1,050	41·4	1,200	47·3	94	1,400	1,540	1,680	85	1,270
1,150	45·3	1,300	51·2	94	1,850	2,000	2,200	85	1,650
1,250	49·2	1,400	55·1	94	2,320	2,550	2,780	80	1,970

The standard Galloway engine is tandem double-acting four-stroke cycle with horizontal cylinders. For sizes over 3,000 h.p. a twin tandem engine is adopted, i.e. two lines of cylinders working on two cranks on a common crankshaft. The engines are designed to run on blast-furnace gas, coke-oven gas, and producer gas.

The special features of the engine are : (*a*) Patented built-up cylinder. The barrel is in two main parts, and the water-jacket is formed by a steel plate casing. Both casting and expansion stresses are eliminated. (*b*) Patent piston with internal diagonal ribs to prevent bending stresses. (*c*) Patented method of fixing piston on rod, allowing of longitudinal adjustment. (*d*) Improved piston rod packing. (*e*) Patented water-cooled exhaust valve box and seat, the exhaust valve itself not being water-cooled. (*f*) Simplified valve gear. (*g*) Powerful servo-motor governor.

The piston and rod are water-cooled and all water discharges are open for inspection purposes. The engine is of simple and strong construction, arranged for ease of accurate alignment, and free to expand when working without distortion or stress. All water-jackets and internal parts are easily accessible for inspection and cleaning.

Experience with a recent installation of these engines showed that the period between overhauls could be safely extended to two years, and even after that lengthy period of running the cylinders and piston rings were found to be in excellent condition, and all the main working parts free from appreciable wear.

Browett-Lindley Gas Engines.

Messrs. Browett-Lindley & Co. Ltd., Patricroft, have developed a vertical gas engine of the ordinary four-stroke cycle type for direct coupling to electric generators and other purposes.

The engines are made with two, three, four, and six cylinders ranging in power from 60 to 1,200 b.h.p. A number of these engines have been supplied for such industrial work as driving mills, etc., and are very satisfactory in operation.

The sectional view of a four-cylinder engine given in fig. 8 shows the accessibility of these vertical engines. The design allows the piston to be easily removed and the cylinder covers are quite free from valves or other gear. The valve gear is quite simple, hardened steel cams are used to operate the valves and the cam roller guides are of the piston type. Both the inlet and exhaust valves can be readily removed for inspection when desired, and the latter are provided with renewable seats.

The governor fitted to these engines is very powerful and the working parts are lubricated by oil under pressure, so that reliable operation can always be expected. Forced lubrication is provided for all of the principal bearings on the engine, the oil being supplied under pressure from a valveless pump.

Features of the Browett-Lindley engines are the simple design, the absolute freedom from back firing, and the equal distribution of power to each cylinder at all loads. A large number of installations are at work on town and producer gas, and very economical running is obtained.

The lowest heat consumption recorded with these engines is 9,540 B.Th.U. per b.h.p., and the highest 10,260. The mechanical efficiency of the engines is about 85 per cent. A cyclical regularity suitable for driving alternators in parallel can be obtained without the need of excessively heavy flywheels. The weights of the standard engines vary from about 2 to 2½ cwt. per horse-power.

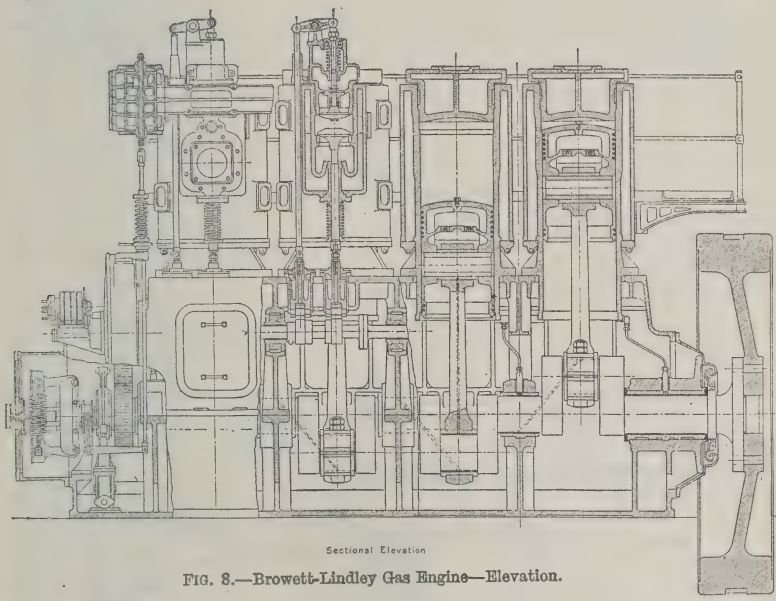


FIG. 8.—Browett-Lindley Gas Engine—Elevation.

FULLAGAR GAS ENGINES.

An important development of the opposed piston type of engine appears in the recent two-stroke design of the late Mr. H. F. Fullagar (fig. 9), and a few of these gas engines are now at work.

Each 'unit comprises *two* open-ended cylinders placed side by side each cylinder contains two pistons as in the Oechelhauser design, but in the Fullagar design opposite pistons of each cylinder are connected by diagonal rods as indicated, thus dispensing altogether with return connecting rods, while a *two*-throw crankshaft suffices for the four pistons of the unit.

The obliquity of the tie-rods is small, and the side thrusts due to this obliquity are taken by the crossheads and guides shown above A and C; it will be seen that pistons A and D are connected, as also are B and C, by long diagonal rods. An explosion between A and B drives B downwards, thus actuating the left-hand crank, and simultaneously drives A upwards, thus actuating the right-hand crank through the tie-rod connecting A and D; the crankshaft accordingly receives as the result of each explosion a practically pure turning couple, or 'torque.' It will also be observed that as the cylinders are open-ended there are no vertical stresses set up in the framing of the engine when running; hence the Fullagar engine may be expected to be light in relation to its power output.

The engine works on the two-stroke cycle, and thus each cylinder furnishes one working impulse per revolution. As a unit includes two cylinders, each *unit* furnishes *two* working impulses per revolution. Stationary engines for land installation usually comprise two or three units, furnishing four or six working impulses per revolution. In the case of great power or where height is limited and the speed must be high, as e.g. in submarines, engines would consist of even four units. The usual inlet and exhaust ports are indicated in fig. 9; the charge is supplied at a pressure of between 2 and 3 lbs. per sq. in. by gas and air pumps, which may be driven by the engine.

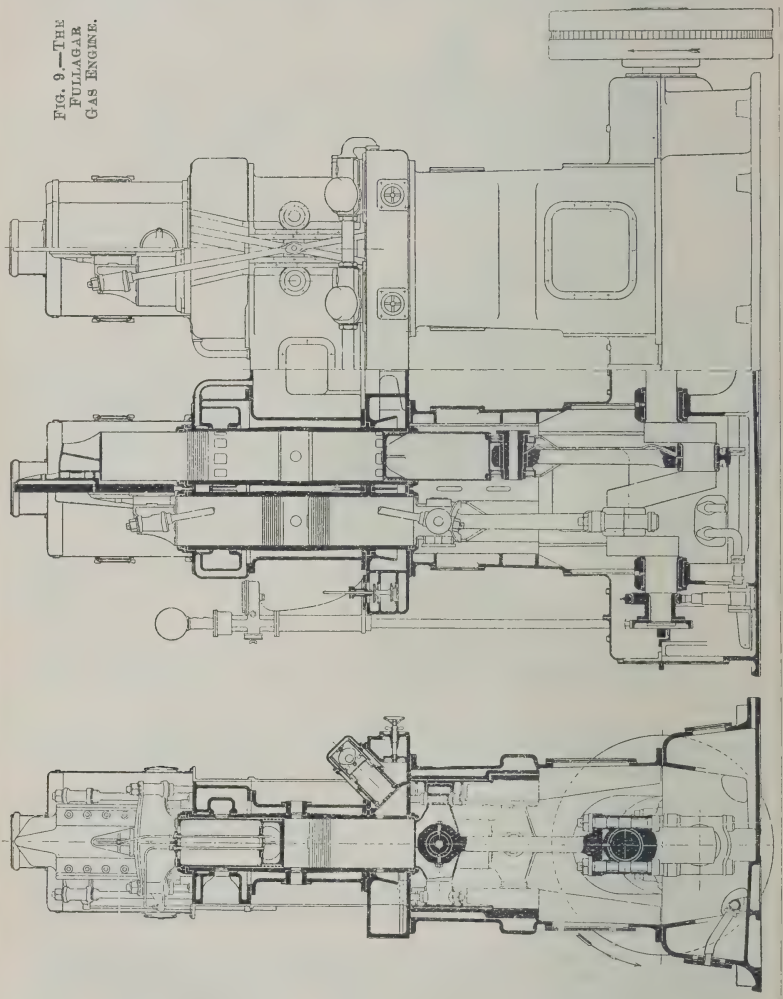


FIG. 9.—THE FULLAGAR GAS ENGINE.

The upper guides are modified to act as scavenge air pumps, the upper crosshead being fitted to operate as pistons, unless separately driven blowers or pumps are required.

The Fullagar engine may be arranged to run in either direction; it is only necessary to alter the timing of the fuel admission valve and the ignition.

An experimental gas engine of 550 b.h.p., constructed by W. H. Allen, Sons & Co., Ltd., of Bedford, to the designs of the Fullagar Engine Ltd., has been installed at the Newcastle Electricity Supply Company's station at Gateshead-on-Tyne; this engine consists of *two* units, the cylinders being of 12 ins. bore, while each piston has a stroke of 18 ins. The normal speed is 250 revolutions per minute, corresponding to a piston speed of $\frac{250 \times 18}{6} = 750$ ft. per minute.

Sectional views of this engine are given in fig. 10.

If p denote the mean effective pressure in lbs. per sq. in. within a cylinder during the working stroke, then the work done upon *each* piston in foot-lbs. is $p \times \frac{\pi}{4} d^2 \times \frac{s}{12}$. Where d and s are respectively the bore and piston stroke, in inches.

As each cylinder contains two pistons, the work done by each cylinder per working stroke is $2 \times p \times \frac{\pi}{4} d^2 \times \frac{s}{12}$ foot-lbs., and this is the work done per cylinder per revolution of the crankshaft. Now a unit comprises two cylinders, and hence the work done by a unit per crankshaft revolution is $2 \times 2 \times p \times \frac{\pi}{4} d^2 \times \frac{s}{12}$ foot-lbs., that is $\frac{\pi}{12} \cdot pd^2s$ foot-lbs. Accordingly if the crankshaft makes n revs. per minute, the indicated horse-power is given by :—

$$\text{I.H.P.} = \frac{1}{33{,}000} \cdot \frac{\pi}{12} \cdot pd^2sn \text{ per unit} \quad . \quad . \quad . \quad (19).$$

As the Gateshead engine includes two units, its i.h.p. will be twice this, or :—

$$\text{I.H.P.} = \frac{\pi}{33{,}000 \times 6} \cdot p.d^2sn \quad . \quad . \quad . \quad . \quad . \quad . \quad (20).$$

Though the mechanical and thermal efficiency of this type of engine is high, the ratio of $\frac{\text{B.H.P.}}{\text{I.H.P.}}$ on the experimental engine was only about 0·8, as for experimental purposes the air blower and gas pumps were of a somewhat temporary character; hence 550 b.h.p. corresponds to $\frac{550}{0{\cdot}8} = 690$ i.h.p. roundly.

Putting 690 for i.h.p., 250 for n, 12 for d, and 18 for s, in Eq. 20 we get as the value of the mean effective pressure :—

$$p = 67 \text{ lbs. per sq. in.}$$

The Fullagar system is being developed for use with liquid fuel by Messrs. Cammell, Laird & Co. Ltd., of Birkenhead, John Brown & Co. Ltd., David Rowan & Co. Ltd., Palmer's Shipbuilding and Iron Co. Ltd., and The English Electric Company, Ltd.

Prices and Weights of Gas Engines.

The prices and weights of gas engines vary considerably, depending on the type and design, but may be taken as from about 6*l.* to over 9*l.* per horse-power at the makers' works. The weight is largely affected by the type and design, engines being single or multi-cylindered, slow speed or high speed, horizontal or vertical, the different designs being necessary to suit the various requirements.

Thermal Efficiency of Gas Engines.

The brake thermal efficiency of four-stroke gas engines has risen from 14 per cent. in 1882 to 32 per cent. in 1908; the following table from Sir Dugald Clerk ('The Gas, Petrol, and Oil Engine,' p. 243) illustrates the rise in efficiency :—

INDICATED AND BRAKE THERMAL EFFICIENCIES OF FOUR-STROKE GAS ENGINES FROM 1882 TO 1908.

(*Sir Dugald Clerk, F.R.S.*)

Year.	Experimenter.	Engine.	Bore & Stroke, inches.	Efficiencies per Cent.		
				Mech.	Indic. Thermal.	Brake Thermal.
1882	Slaby	Deutz	6·75×13·7	87·6	16	14
1884	Thurston	Crossley	8·5 ×14	84·2	17	14·3
1888	Soc. Arts	Crossley	9·5 ×18	86·1	22	18·9
1888	Soc. Arts	Griffin (6 cycle)	9·02×14	80·9	21	17
1888	Kennedy	Beck (6 cycle)	7·5 ×15	87·3	21	18·3
1892	Capper	Crossley	8·5 ×18	82·0	22·8	17·4
1898	Robinson	National	10 ×18	87·0	28·7	25
1900	Humphrey	Crossley	26 ×36	83·0	31	25·7
1900	Witz	Cockerill	51·2 ×55·1	81·7	28	22·9
1905	Inst. C. E.	National	14 ×22	85·5	35	29·9
1907	Burstall	Premier	16 ×24	77·1	41·5	32
1908	Hopkinson	Crossley	11·5 ×21	87·5	36·8	32·2

In tests made on a Crossley-Otto engine in 1882, 1 i.h.p. hour was obtained with an expenditure of 23·7 cubic feet of London gas, while at the Society of Arts trials, held in 1888, a Crossley engine consumed 20·55 cubic feet per i.h.p. hour. Sir Dugald Clerk found, in 1894, when testing a Crossley engine, that the consumption of gas (Openshaw) per i.h.p. hour was as low as 14·5 cubic feet.

At the present time it may be taken that the consumption of town gas per b.h.p. hour at full load varies from 15 cubic feet to 25 cubic feet per hour, according to the size of the engine, and also according to the heating value of the gas.

The increased efficiency that has been obtained in the best known forms of engines is probably due to greater compression of the gas, and to some lesser extent to improved means of displacing the burned gases from the cylinders.

Large stationary gas engines are commonly of the horizontal tandem double-acting four-stroke type, as with this arrangement a working impulse is obtained at every stroke. Pistons and piston-rods are of course always water-cooled in such engines.

Piston speed in large gas engines has reached 800 to 900 feet per minute. Mean pressures of 85 to 95 lbs. per sq. in. can be obtained in the cylinder with gas of 110 B.Th.U., though engines are usually rated on an assumed 70 lbs. only.

BRITISH STANDARD SPECIFICATIONS RELATING TO INTERNAL COMBUSTION ENGINES.

The British Engineering Standards Association have issued three specifications, Nos. 120, 212, and 213 (1925), dealing with gas and oil engines for electrical purposes, copies of which may be obtained from their office, 28 Victoria Street, Westminster, S.W.1, price 2*s.* each.

GENERATION OF ELECTRICITY BY GAS AND OIL ENGINES.

The Electricity Commissioners have published reports* on the output and fuel consumption of electric power stations in Great Britain. The stations are classified in thirteen groups, of

* *Generation of Electricity in Great Britain, year ending March* 31, 1930. Electricity Commission. H.M. Stationery Office, Adastral House, Kingsway, London, W.C. 2. Price 1*s.* 6*d.* net.

which six groups have producer gas engines and eight groups oil engines, according to output. For four or these groups, representing the largest, the smallest, and two intermediate sizes of station, the following figures are abstracted from the report for the year ending on March 31, 1930 :

GAS-PRODUCER STATIONS.

Output of Station Units in Year.	Between 1,000,000 and 2,500,000.	Between 250,000 and 500,000.	Between 100,000 and 250,000.	Under 50,000.
Number of stations in group	3	6	10	9
Total units generated by gas-producer plant	4,843,373	2,239,062	1,768,259	145,565
Total fuel consumed (actual) tons	2,912	2,348	2,196	270
Average fuel consumption per unit generated (converted to basis of coal) lbs.	1·37	2·36	2·82	4·22

	Total units generated.	Total fuel consumed.
Total output of all gas-producer stations (40) for year ending March 31, 1930	12,988,767	Tons. 11,532
Figures for year ending March 31, 1927	20,133,821	22,407
,, ,, ,, ,, 1928	18,920,510	18,610
,, ,, ,, ,, 1930	14,915,912	14,235

GENERATION OF ELECTRICITY BY GAS-ENGINE STATIONS USING TOWN'S GAS.

The total units generated by gas-engine stations using town's gas, during the year ending March 31, 1928, was 2,582,432 (from 23 stations).

For the year ending March 31, 1929, the total units were 3,106,432 (from 19 stations).
,, ,, ,, ,, 1930 ,, ,, 3,107,934 ,, 18 ,,

OIL-ENGINE STATIONS.

Output of Station Units in Year.	Between 1,000,000 and 2,500,000.	Between 250,000 and 500,000.	Between 100,000 and 250,000.	Under 50,000.
Number of stations in group	13	23	39	39
Total units generated by oil engines	17,416,806	7,403,635	5,430,287	685,024
Total fuel consumed (actual) tons	7,236	3,478	2,849	392
Average fuel consumption per unit generated (converted to basis of oil) lbs.	0·77	0·97	1·06	1·29

	Total units generated.	Total fuel consumed.
Total output of all oil-engine stations (160) for year ending March 31, 1930	78,177,202	Tons. 33,549
Figures for year ending March 31, 1927	65,684,561	38,809
,, ,, ,, ,, 1928	62,649,768	27,947
,, ,, ,, ,, 1929	73,714,239	31,455

Gas Engine Starters.

Large gas engines are very usually started by means of compressed air pumped by the engine, when running, into a reservoir. They may be started by means of a small gas engine, after the fashion of the barring engine for steam-engines. This small engine must be able to compress the charge of the large cylinder or one of them sufficiently to ignite; or gas and air may be pumped into the cylinder by a small engine and ignited just as the engine is turned off its dead point.

The Wells Starter.—The Wells Bros. starter consists of a small hand-pump fitted at the rear end of the combustion chamber. The action of this pump is to force gas into the cylinder, the crank of the engine being already fixed in the forward position; the action of a valve then permits the pumping of air into the cylinder, and upon the proper mixture being obtained the charge is ignited by ordinary means.

The Fielding and Platt Starter.—Arranged in close proximity to the engine is an air receiver, into which air is pumped under pressure. After gas has been admitted into the engine cylinder, indicated by a flame at a junction pipe, compressed air is admitted into the cylinder from the receiver, and the mixture is ignited at the moment of combination.

The air pressure self-starter was introduced by Fielding & Platt, Ltd., in 1895, and standardised by practically all other makers, and is in common use to-day. On four-stroke cycle engines it has been brought up to date by an automatic control worked by cam on the countershaft, so arranged that the air impulse is only permitted during the power stroke, and is cut off automatically on the other three strokes of the engine, thus avoiding the risk of attendant leaving the air pressure to oppose the piston on the return stroke. No explosive mixture is used. The air tank is automatically charged by the engine itself.

Cooling Water for Internal Combustion Engines.

With stationary engines a cooling water tank or reservoir is usually provided, and in such cases the loss by evaporation varies from 0·2 to 0·4 gallon per b.h.p. hour; with large gas engines it is usual to employ cooling towers. The water is pumped from a well through the engine jackets, etc., and passes thence to the top of the tower; falling to the bottom, it returns to the well at but little above the atmospheric temperature. With a cooling tower the loss by evaporation is greater than with a cooling tank, and commonly amounts to from 0·4 to 0·5 gallon per b.h.p. hour. From 30 to 35 gallons of cooling tank capacity is required per b.h.p.

The 'Heenan' Patent Water Cooler (p. 1232) is particularly suitable for these installations, and in addition to occupying small space has the advantage that it consumes only approximately 0·1 to 0·2 gallon of water per b.h.p. hour.

The ordinary standard sizes of these machines vary in capacity from 250 gallons of water per hour up to 55,000 gallons per hour, which the machines are capable of cooling from 120° down to 90° F. under temperate climatic conditions.

As an example, a *medium* size cooler of the type in question is capable of dealing with the following problems and providing the corresponding water temperatures:

Application.	Quantity g.p.h.	Water Temperatures.	Heat extracted per Hour.
1. Petrol engines and dynamometers	10,000	170°/100° F.	7 million B.Th.U.
2. Gas, oil, and Diesel engines	,,	120°/90° F.	3 ,, ,,
3. Steam-engine condensers	,,	105°/85° F.	2 ,, ,,
4. Air compressors	,,	85°/75° F.	1 ,, ,,
5. Refrigerating plant	,,	72°/67° F.	500,000 B.Th.U.

DATA RELATING TO COOLING WATER USED IN INTERNAL COMBUSTION ENGINES.

Type of Engine.	Bore and Stroke. Ins.	B.H.P.	Cooling Water Temperatures.			Gallons circulated through Jackets, etc., per B.H.P. hour.	B.Th.U. removed by cooling Water per B.H.P. hour.
			Inlet. °F.	Outlet. °F.	Rise of Temp. °F.		
Normal petrol car engines	—	15 to 75	160	200	40	7·5	3,000
Diesel, 3-cyl. . . .	17·75 × 27	360	54	103	49	5·62	2,750
Diesel, 1-cyl. . . .	11·8×18·1	40	60	100	40	6·25	2,500
Oechelhauser . . .	26·5×37·5	500	72	108	36	5·9	2,125
Nuremberg double-acting tandem	33·5×43·3	1,200	60	100	40	7·5	3,000
National single-cyl. horizontal	22·5×30	140	60	140	80	3·5	2,800
National 6-cyl. tandem vertical	22×24	750	58	100	42	6·5	2,730

The water necessary for cooling the pistons and valves is also included in this table in the cases of engines so provided.

ROSSELINE.

This medium (*Boilerine, Ltd.*) is for the purpose of obviating the incrustation which takes place in water-jackets when salts are present in the cooling water. There are no acids in the preparation, so that joints or attachments cannot sustain any injury. Where scale is already in existence, the action of Rosseline is to cause it to flake and crumble. An average of a quarter pound of Rosseline per brake h.p. per week should be used, the proportion being one pound per 1,000 gallons of water, the approximate quantity of water used to keep a cylinder cool being four gallons per b.h.p. per hour.

Parts of Gas Engines.

In the following formulæ (mainly from Guldner), a maximum explosion pressure of 350 lbs. per square inch is implied.

1. *Cylinders* are of hard, close-grained cast iron; for the inner wall (*i.e.* working barrel) the working stress may be taken in design at 1,500 lbs. per sq. in. to reduce wall thickness and distortion under heat. If d be the cylinder bore, in inches, and t the wall thickness, then :—

$$t = \cdot 047\, d + \cdot 375 \quad . \quad . \quad . \quad . \quad . \quad (21)$$

the ·375 being added as an allowance for reboring, etc. Cylinders up to about 8 ins. bore may have cylinder walls not less than $\frac{1}{2}$ in. thick if cast in one, or $\frac{5}{8}$ in. if with separate working barrels. For bores of 24 ins. and above, the wall thickness may be diminished gradually, or by steps, until at the open end it becomes :—

$$t = \cdot 02\, d = 0 \cdot 5 \quad . \quad . \quad . \quad . \quad . \quad (22)$$

2. *Pistons* are of close-grained cast iron; in large engines cast steel is sometimes used. Piston rings are of medium hard, but not very brittle, cast iron. The rings are usually parallel and are of the Ramsbottom type.

3. *Gudgeon, or piston pins* are usually of steel with a hardened outer surface; the diameter δ may be estimated from the formula :—

$$\delta = 0 \cdot 35 \sqrt[3]{d^2 l_0} \text{ ins.} \quad . \quad . \quad . \quad . \quad (23)$$

where l^0 is the distance from centre to centre of the gudgeon pin lugs, in inches. The intensity of bearing pressure at the moment of explosion should not exceed about 1,800 lbs. per sq. in. of projected area of bearing; this leads to the formula for the length, l, of gudgeon bearing :—

$$l = 0 \cdot 15 \frac{d^2}{\delta} \text{ ins.} \quad . \quad . \quad . \quad . \quad . \quad (24)$$

The gudgeon bush is of phosphor bronze; sometimes white-metal lined in larger sizes.

4. *Connecting Rods* in small engines are usually drop-forgings of mild steel; in large engines steel forgings machined all over are used. Connecting rods are treated in calculations as long

columns supported at the ends; a factor of safety of 12 is usual; Euler's formula then leads to the result that the full diameter, $\triangle$, of a rod of circular section may be :—

$$\triangle = 0{\cdot}128 \sqrt{d\,L} \text{ ins.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

where L is the length of the rod in inches. Towards the gudgeon end, the diameter may be diminished to ·9 or ·8 $\triangle$. The big end of the connecting rod is very commonly of marine type; when of more than about 5 ins. diameter this bearing is usually of cast iron or cast steel, white-metal lined; for smaller sizes bronze is general.

5. *Crankshafts.*—For large engines high-grade steel forgings with an u.t.s. of at least 70,000 lbs. per sq. in. and 20 per cent. ductility in 8 ins.; or open-hearth steel of u.t.s. at least 65,000 lbs. per sq. in. and 18–20 per cent. elongation in 8 ins. may be used. Recently, in special cases, nickel steel has been adopted with an u.t.s. of about 98,000 lbs. per sq. in. and 21 per cent. elongation in 8 ins.

For ordinary steels the allowable working stress is about 12,000 lbs. per sq. in. If s be the stroke, in inches, then H. Allen gives for the crankshaft diameter, D :—

$$D = 0{\cdot}42 \sqrt[3]{d^2\, s} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

With the higher pressures occurring in engines of the Diesel type (see p. 1893) the constant 0·42 must be increased to 0·52 in this equation.

Crank-pin diameter may be about 1·1 times the crankshaft diameter; the length should be about $0{\cdot}18 \frac{d^2}{D}$ ins. The bearing pressure in crankshaft bearings should be about 400 lbs. per sq. in. of projected area of bearing.

The crank web may have a depth of 1·33 to 1·66 D and a thickness of 0·85 to 0·55 D.

Valves of Internal Combustion Engines.

The valves of present-day internal combustion engines of all kinds are practically all of the 'Poppet' or 'Mushroom' type. In small sizes the valves and stems are in one piece, and usually either of medium carbon steel or nickel steel. Large valves, especially when water-cooled, have cast-iron discs and medium carbon steel stems; such valves in some cases attain large dimensions, the Siegener Co., for example, having built Koërting type engines with valves 19·68 ins. in diameter and a lift of 3·14 ins. Valve cages and seats are of hard close-grained cast iron. Inlet valves are sometimes automatic; exhausts must be mechanically operated. It is preferable to place valves with their axes vertical whenever possible. The mean velocity of gas through the valves is 5,000 to 6,000 ft. per minute in good practice.

Packings for Heavy Gas Engines.

See 'Britimp' Brand Metallic Packings, p. 1903.

Flywheels.

6. *Flywheels* when of cast iron in one piece should not have a rim velocity exceeding 100 ft. per second. If b be the breadth, and R and r the outer and inner radii of the rim, all in feet, and if n denote the number of revolutions per minute, then the kinetic energy, or stored work in the rim, is :—

$$\text{Kinetic energy} = 0{\cdot}1205\,(R^4 - r^4)\, b\, n^2. \quad \text{Foot-lbs.} \quad . \quad . \quad . \quad . \quad (27)$$

The 'Cyclic Irregularity' or 'Coefficient of Fluctuation of Speed' is the ratio borne by the difference between the maximum and minimum angular velocity of the crankshaft in any one complete cycle, to its mean angular velocity.

The Cyclic Irregularity is often denoted by $\frac{1}{m}$; its permissible value varies with the nature of the work to be done by the engine; the following are a few typical cases :—

Nature of Work.	m
Ordinary workshops; wood-working, pumping, etc.	30–40
Belt-driven continuous-current dynamos in series	80
,, ,, in parallel	120
Belt-driven alternators in parallel	180
Cotton-spinning machinery	200
Direct-coupled alternators in parallel	250

Mathot has proposed the following formula for rapidly calculating flywheel sizes :—

$$W = C \cdot \frac{m\,P}{R^2 n^3} \cdot 10^6 \text{ tons} \quad . \quad . \quad . \quad . \quad . \quad . \quad (28)$$

where,

W = weight of flywheel rim in tons.
R = mean radius of rim, in feet.
P = b.h.p. of engine.
n = normal speed, in revs. per min.
C = a numerical coefficient.

The following are suitable values for C :—		C
1.	For single-cylinder, single-acting engines	265
2.	For two single-acting cylinders side by side; cranks at 180°	167
3.	" " opposed; cranks at 180°	125
4.	" " side by side; cranks at 0°	125
5.	" " opposed; cranks at 0°	167
6.	" " tandem; cranks at 0°	125
7.	For four single-acting cylinders opposed two and two; cranks at 0°	42
8.	For single-cylinder, double-acting engines	167
9.	For two double-acting cylinders side by side; cranks at 0°	42
10.	" " tandem, on same crank-pin	42

The total weight of the flywheel is about 1·4 W tons.

Flywheels are frequently more massive than the above calculation indicates, especially in cases where the speed fluctuation must be kept small even although one or more explosions be missed due to governor action.

Breakages in Internal Combustion Engines.

The following table is compiled from a wide induction of cases from the records of Insurance Companies, and indicates the relative frequency of failure of the several organs of internal combustion engines :—

ANALYSIS OF FAILURES OF PARTS OF INTERNAL COMBUSTION ENGINES.

Description of parts which are believed to have given way first.	Average before 1904.	1904-1908.	1909.	1910.
	Per cent.	Per cent.	Per cent.	Per cent.
Valves and valve-gear	30·1	33·3	29·6	42·8
Cylinders and cylinder ends	19·3	16·1	15·4	8·5
Pistons	9·7	5·0	2·7	5·1
Connecting-rods and their bolts	11·9	10·3	9·4	9·1
Main shafts	5·1	11·2	13·4	7·9
Governors and governor-gear	5·1	7·0	5·4	3·4
Silencing-boxes and exhaust-pipes	2·8	3·6	1·3	2·8
Flywheels and driving-drums	5·1	5·4	0·7	2·8
Clutches and couplings	1·7	0·0	0·0	0·6
Frames and pedestals	3·4	1·8	1·3	1·7
Bolts *	4·6	1·5	2·0	1·7
Total wrecks, cause not ascertained	0·6	0·4	0·0	0·6
Main driving-ropes and belts	0·6	0·6	0·0	0·0
Miscellaneous	—	2·2	7·4	4·5
Gas-producer plants	—	1·6	11·4	8·5
	100·0	100·0	100·0	100·0

* These do not include bolts in connecting-rods, valve-gear, or other moving parts, the bolts in these parts being tabulated with the parts themselves.

Notes on Fixing a Gas Engine.*

(F. R. Parsons.)

Foundations.

The outline of the foundation of an engine will be determined by the over-all dimensions of its bed or framing. Races for the flywheels must be allowed for if their diameter is such that clearance cannot be obtained otherwise. In a general way a margin of from 3 ins. to 6 ins. all round on engines of from 5 to 25 b.h.p. will meet the case. Above this and up to 100 b.h.p. a margin of from 9 ins. to 12 ins., according to needs and the formation of the bed, will be required. The depth of the concrete foundation depends a great deal upon the character of the subsoil. An engine standing on a good gravel bed would require much less concrete depth than one standing on a clay soil, or made-up ground. Engines up to 10 b.h.p. should in a general way have a bed at least 2 ft. to 2 ft. 6 ins. thick; up to 100 b.h.p. this should be increased to from 3 ft. 6 ins. to

* *Mechanical World*, July 18, 1919.

4 ft. Modern-type horizontal gas engines which have a box-section framing and little cylinder overhang are much less susceptible to vibratory troubles, hence the depth of the bed need not be so excessive.

Opinions vary considerably as to the composition of concrete for engine beds. Always, however, it is much more profitable in the long run to have a fairly strong mixture, thereby diminishing considerably the setting period, and ensuring a good firm job. The following mixture has been found to be a highly satisfactory one for any class of high-speed machinery or engines of any power: broken rubble (stones or bricks), 4; clean river sand, limestone, or granite chippings, 4; cement, 1. Dirt or loose soil adhering to the material detracts from its homogeneity and delays setting. It is a good plan to wash well the stones and larger material from a hosepipe before mixing. Each layer of concrete should be well rammed as it is laid, so as to force the material to the outer extremities of the bed. If the latter cannot be completed right out of hand, the old surface should be well wetted before proceeding with the new material.

Holding-down Bolts.

Always leave ample bolt freedom in the concrete bed so that there shall be no difficulty of getting the engine into position, and also permitting it within certain limits to be moved in either direction if so desired. Use square-necked bolts and washer-plates with square holes, the plates being of ample area in order to resist the pull enforced by the nuts. Grout the bolts in with a mixture of clean fine sand and cement in equal parts when the exact position of the engine has been found, and it has been levelled and set. Holding-down bolts are better not arranged below the concrete foundation, but embedded therein; say, allowing from 6 ins. to 18 ins. of concrete, according to the size and thickness of the bed, below them. Set the bolts and plates on this, then proceed with the filling in. Bolts need not be tied together with a single long plate unless one happens to come so close to the outside of any part of the bed as to have insufficient hold. Always set the bolts in a wood-frame templet having holes in it similarly placed to those in the engine bed.

Size of Gas Meter.

Given the brake horse-power of the engine, the size of a meter required to pass a sufficient quantity of gas per hour can be ascertained by the following rule: Multiply the b.h.p. of the engine by the constant 3·4, then add the constant 5, the result being the size of the meter in 'lights.' For example, suppose the engine we are erecting is 60 b.h.p., then $3{\cdot}4 \times 60 + 5 = 209$. As this rule allows ample margin, we should therefore decide upon a 200-light meter.

Size of Gas Service.

To find the correct size of gas pipe to connect between the meter and the engine, the following rule may be applied: Multiply the meter size, in lights, by the constant 0·008, then add the constant 0·75, the result being the bore of the service pipe in inches. Thus, using the previous example, we have $200 \times 0{\cdot}008 + 0{\cdot}75 = 2{\cdot}35$ ins. In this case, if the connection between the meter and the engine was fairly straight, and not over-long or containing many curves, a pipe 2½ ins. bore would be ample. But if a considerable distance divided the two points indicated, or many bends were introduced, it would be better to use a 2½ in. bore pipe.

Exhaust Pipes.

Generally, engines from 1 to 5 b.h.p. may have an exhaust pipe from 1 in. to 1¾ ins. diameter; 5 to 30 b.h.p., from 1¾ ins. to 2½ ins.; above this and up to 100 b.h.p., from 2½ ins. to 5 ins. in the bore.

Circulating Water.

Almost all gas engines require on an average about 4½ gals. of water per indicated horse-power hour, in order effectively to reduce cylinder temperature, this assuming the temperature of the return water not to exceed 150° F. To effect this by means of circulating tanks would be equivalent to an amount of from 25 to 30 gals. of water per b.h.p. as the contents of the tanks. The effective height of a tank should not be less than 2½ to 3 times its diameter. The higher the tank is situated above the cylinder jacket the more brisk becomes circulation. To keep the bottom of a tank at or about cylinder level causes circulation to be sluggish, and promotes the accumulation of sediment and deposit. For this reason it is generally best to keep the tank bottom at least a foot above the centre of the engine cylinder. A ball tap should be provided to supply cold water to the coldest tank, and if the engine is heavily worked it may also be necessary to arrange for an overflow from the hottest tank. Engines up to 25 b.h.p. can be served from 2 ins. diameter pipes; above this to 50 b.h.p., 2½ ins.; up to 100 b.h.p., 3 ins. to 3½ ins. diameter pipes should be used. The return pipe should be sloped all the way to the tank.

General.

In setting a gas engine on its concrete bed it is advisable to keep the cylinder end slightly the highest, an amount of about ¼ in. in every 3 ft. being ample. This assists in draining the waste oil from the cylinder and other parts of the engine to a point where it may be drawn off and collected.

The exhaust box should not be built solid into the ground or in a pit. It is advisable to mount it on iron rollers placed upon a flat stone or on an iron plate. This provides the necessary means for expansion, without which the joints and connections might suffer.

The gas-inlet pipe to the meter, or from the meter to the engine, should on no account be placed close alongside the hot exhaust pipe. This tends to raise the temperature of the gas, thus making it more rarefied and of less heating value per cubic foot. The cooler the gas admitted the more dense is the charge and the more powerful the explosion.

SECTION XXXI

PART III

OIL ENGINES.

DIESEL ENGINES.

A Diesel engine is a prime mover actuated by the gases resulting from the combustion of a liquid or pulverised fuel injected in a fine state of subdivision into the engine cylinder at or about the conclusion of the compression stroke. The heat generated by the compression of air within the cylinder is the sole means of igniting the charge. The combustion of the charge proceeds at, or approximately at, constant pressure.

The Diesel engine, fig. 1, was patented by Dr. Diesel in 1892, and was brought to a commercial success during the next four or five years by certain engineering firms, with whom he was associated. It differs from the gas engine, or petrol-paraffin engine, where the combustible charge is drawn into the cylinder during suction stroke, and therefore the degree of compression is limited to avoid pre-ignition, the charge being usually ignited electrically. In the case of the Diesel engine, only air is drawn into the cylinder and compressed to about 480 lbs. per sq. inch, and the oil is injected in a fine spray into this compressed air and burns gradually as long as the injection is maintained. During this whole period the pressure in the cylinder does not rise above the compression pressure which therefore is the maximum pressure that the working parts of the engine have normally to withstand.

Diesel engines are made either of the four or two-stroke cycle type, and according to requirements run at various speeds. In the four-stroke cycle type, air at atmospheric pressure is drawn into the cylinder during the first stroke, and compressed during the second stroke to a pressure of about 480 lbs. per sq. in.; this compression of the air causes its temperature to rise to about 950° F., which is more than amply sufficient to ignite the spray of oil injected during part of the third or working stroke. When the injection stops the gases expand and the exhaust valve opens shortly before the piston reaches the end of the third stroke. The products of combustion are expelled during the fourth stroke of the piston.

In engines of the two-stroke cycle type the air, instead of being drawn into the cylinder by the engine piston, is delivered thereto under low pressure and is also used partly to expel the exhaust gases. After the air-inlet ports or valves have closed, this charge of air is compressed to the same pressure (480 lbs. per sq. in.) as in the case of the four-stroke engine. The fuel is injected in the same manner as in the four-stroke engine. The gases expand as soon as combustion is completed, and the exhaust ports are uncovered by the piston shortly before reaching the bottom dead centre, the gases being expelled by the entering charge of air.

The thermal efficiency at full load reckoned on the b.h.p. is about 36 per cent. and the fuel consumption at this load about 0·4 lb. per b.h.p. hour. The products of combustion are invisible and odourless.

Nearly all four-stroke engines are built of the vertical type, with a strong cast-iron 'A' frame, the upper part of which forms the outer cylinder of the water jacket; into this upper part is fitted a liner of special close-grained cast-iron, which is water-jacketed for almost its entire length.

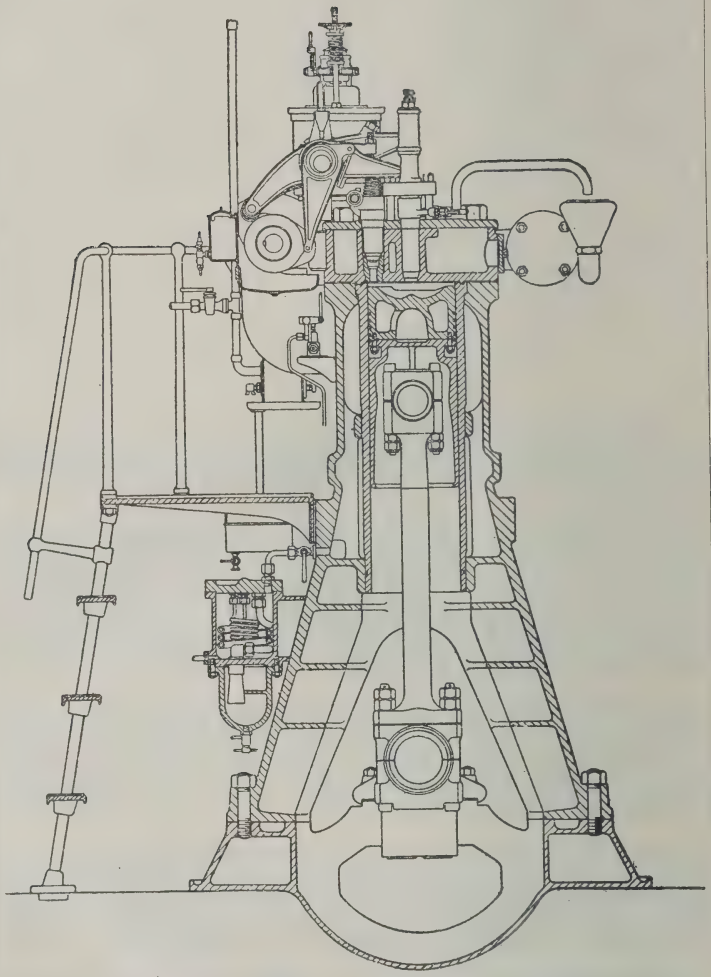


FIG. 1.—The Diesel Engine.

The cylinder cover contains four valves, the two large ones being the exhaust and air-inlet valves. The central valve is a needle valve which gradually admits fuel to the cylinder about 1° or 2° before the completion of the compression stroke, to 15° to 20° of the working stroke. The fourth valve controls the admission of the compressed air which is used in starting the engine. The cover is thoroughly water-jacketed to prevent the valves from overheating.

The exhaust and air-inlet valves are of the mushroom type and are self-contained in cast-iron casings which bed on cone seatings in the cover, so that should it be necessary to replace a valve with the minimum loss of time a complete valve and casing can be substituted.

The starting valve, like the suction and exhaust valves, opens into the cylinder. The fuel admission valve opens outwards, except in the Burmeister & Wain engine, in which it opens inwards. The fuel valve of most Diesel engines works through a packing gland to prevent leakage of injection air, remote cases of valves sticking in this packing are occasionally recorded. To prevent even this possibility, Mirrlees, Bickerton & Day, Ltd., and also Tosi, use fuel valves which require no packing.

All valves are closed by springs and opened by rocking levers actuated from the horizontal camshaft, which is supported by ring-lubricated bearings, bolted to the cylinder frame. The camshaft derives a half-speed motion through spiral gearing from a vertical shaft, which in turn is driven by spiral gearing from the crankshaft. In some engines there is a direct gear drive from the crankshaft to the camshaft. This vertical shaft drives a centrifugal governor acting directly on the suction valve of the fuel pump.

A cast-iron trunk piston is adopted, made sufficiently long to be a guide in the cylinder and to take the side thrust due to the angularity of the connecting rod, and is provided with three to six 'Ramsbottom' rings to maintain a gas-tight cylinder; in recent large engines the crosshead type is, however, becoming usual.

The fuel pump of the engine operates as follows:—

The plunger draws a full stroke of oil from a reservoir into the pump chamber past the suction valve. This quantity of oil is in excess of that required for combustion. The excess is returned past the suction valve on the delivery stroke by the action of the governor, which holds open the suction valve for a sufficient length of time to allow the excess fuel drawn in to escape back into the reservoir. In this way the amount of fuel delivered to the engine is precisely the amount required to do the work taken from the engine at that instant, and steady running is ensured, as there is never any actual cut-out.

A feature of the Diesel engine is the facility and certainty with which it starts, it being quite possible to start the engine within 45 seconds. It was the first type of internal combustion engine to be started from cold on crude oil.

The engine is able to burn any class of oil, from refined to the cheap petroleum residue, as well as tar oil, and provided the calorific value is not less than 18,000 B.Th.U.'s per lb., the makers guarantee the consumption at full load not to exceed 0·4 lb. per b.h.p. hour in the larger sizes and 0·45 lb. per b.h.p. hour in the smaller sizes.

The following working costs of two 'Mirrlees-Diesel' engines of 335 b.h.p. and 500 b.h.p. respectively, installed at the Buxton Electricity Works, Derbyshire, are taken from the published figures of The Diesel Engine Users' Association for 1927–8. It will be observed that as usual in electricity stations the load factor is low. Practically all industrial works have much higher load factors, and therefore still lower working costs are obtained.

Load factor . . . . 26 per cent.

Working Costs.

Fuel	0·325*d.* per unit
Lubricating oil. . . .	0·031*d.* ,, ,,
Wages	0·108*d.* ,, ,,
Water and stores . . .	0·016*d.* ,, ,,
Repairs and maintenance .	0·040*d.* ,, ,,
	0·520*d.* per unit generated.

The amount of cooling water required for Diesel engines varies according to climatic conditions, and ranges between 4 and 8 gals. per b.h.p. per hour, the outlet temperature being from 90° to 120° F.

'Mirrlees-Diesel' Engines.

'Mirrlees-Diesel' engines (figs. 2 & 3) are constructed by Mirrlees, Bickerton & Day, Ltd., of Hazel Grove, Stockport, whose works are specially equipped for the sole manufacture of heavy oil engines.

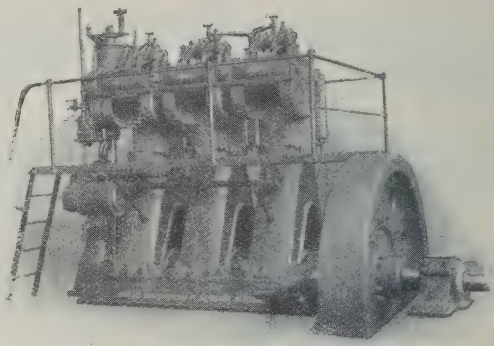


Fig. 2.—165 b.h.p 'Mirrlees-Diesel' Engine.

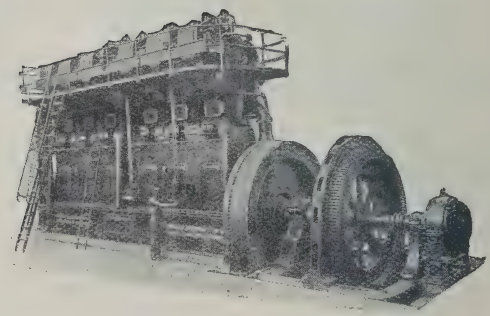


Fig. 3.—1,000 b.h.p. 'Mirrlees-Diesel' Engine.

The first Diesel engine made in Great Britain was a 'Mirrlees-Diesel' of 20 b.h.p., which was made so far back as 1897, and it is interesting evidence of the life of a Diesel engine that this engine is still working daily. 'Mirrlees-Diesel' engines are thus the outcome of thirty-three years' experience, and the total power supplied aggregates over 250,000 b.h.p.

All 'Mirrlees' engines are of the vertical type, and various sizes of cylinders have been standardised as briefly shown below.

Cylinder Output (b.h.p.)	R.P.M.	No. of Cylinders per Engine.	Range of Engine Power (b.h.p.)
Four-stroke Cycle 'Mirrlees' Engines.			
25	450	3 to 6	75 to 150
55	300	1 to 6	55 to 330
100	250	3 to 6	300 to 600
125	200	3 to 6	375 to 750
166	215	3 to 6	500 to 1000
Two-stroke Cycle 'Mirrlees' Engines.			
375	187	3 to 6	1125 to 2250

Engines of 1,125 b.h.p. and upwards are of the two-stroke cycle type, and of 'Mirrlees-Nobel' design. The engines are single acting, and have one piston per cylinder, and a low speed of rotation.

The pistons used in 'Mirrlees' engines are of a greatly improved design; they incorporate a 'floating' piston pin, and special design of top-end bearing, giving larger bearing surfaces than is possible with ordinary practice. Formerly all piston pins have been a driving fit in the piston.

The 'Mirrlees' piston pin is free to rotate both in the piston and connecting rod, thus obviating any piston distortion due to expansion or 'driving in' of the pin, also giving easier dismantling, greater life and reliability.

The fuel consumptions of 'Mirrlees-Diesel' engines are remarkably low, as shown by the following results, and are obtained in regular working over long periods, even with engines of small powers.

At full load	0·4 lbs. per b.h.p. per hour
,, three-quarters . .	0·42 ,, ,, ,,
,, half	0·47 ,, ,, ,,

Messrs. Mirrlees, Bickerton & Day also specialise in enclosed type Diesel engines which have been largely adopted on board ship for lighting and power purposes. Over eighty engines of this type, aggregating over 18,000 h.p., have been supplied to the British Admiralty alone, mainly for use on capital ships of the 'Dreadnought' and later classes. This type of engine has also been adopted by several foreign navies and many of the principal steamship companies.

Another development which has proved a great success is the use of tar oil or creosote as fuel for Diesel engines, and a special device has been introduced by the above firm for this purpose. The modified arrangement of fuel pump, etc., required is shown in fig. 37, p. 2159.

Belliss & Morcom Diesel Engines.

Diesel engines up to 100 b.h.p. per cylinder are standardised by Belliss & Morcom, Ltd., Birmingham, in three, four, five, and six-cylinder units. They are of the vertical design and run up to 375 r.p.m.

Forced lubrication is applied throughout; the overhead camshaft is also lubricated under pressure, and oil escaping from the bearings is led back to a tank under the bedplate for re-use after filtering.

The cylinders are separate castings with liners and detachable heads, water cooled, and each head has one inlet, one exhaust and one fuel valve. The pistons are oil cooled on the higher ratings.

A vertical three-stage air compressor is driven from the crankshaft, and the second stage is below the first to avoid oil-laden air being drawn in from the crankcase. Starting air is admitted to three cylinders, the valves being grouped at the end of the camshaft.

The adoption of the higher speeds and the complete system of forced lubrication and cooling result in a very efficient engine, which runs smoothly, and the freedom from vibration is very noticeable. The loss of power due to altitude is given as 4 per cent. for each 1,000 ft. above sea level, and the loss of power due to air temperature is 2 per cent. for each 10° F. above 80° F.

English Electric Co.'s Diesel Engines.

The Willans-Diesel oil engines are manufactured at the Rugby Works of the English Electric Company Limited, and have been made since 1906. These engines are in use in all parts of the world, driving all kinds of machines, centrifugal pumps, electric generators, etc., both direct coupled and by means of belts or ropes. They are made in standard sizes from 250 to 500 b.h.p.

The engines are of the vertical type, the crank chamber is enclosed and forced lubrication used. The air-compressor for supplying air for fuel injection and starting purposes is fitted on the end of the engine bedplate and driven from the crankshaft. The speed is controlled by a centrifugal governor acting upon the inlet valve of the fuel pump, thus regulating the quantity of fuel exactly to the load. The consumption of residue oil is guaranteed not to exceed 0·41 to 0·45 lb. per b.h.p. hour at full load, depending on the size of the engine, while at half load the consumption per b.h.p. hour is but 15 per cent. higher.

The engines are made to jigs and limit gauges, thus ensuring interchangeability of spare parts, which is of the utmost importance in cases where engines are installed in remote countries. Larger engines in powers from 750 to 3,000 b.h.p. are made to the Fullager designs.

Beardmore-Tosi Diesel Engines.

These engines are made in various sizes up to 2,400 b.h.p. and are of the four-stroke cycle enclosed type. The range of horse-power developed per cylinder is 15 to 400 b.h.p., the dimensions ranging from 180 mm. to 630 mm. Crossheads have been adopted in both the land and marine types. The Tosi design gives the greatest freedom from trouble for the piston and gudgeon pin.

Great attention has been paid to the design of the details of these engines, especially to the lubrication and water circulating systems. The consumption of fuel oil is between ·42 and ·45 lb. per b.h.p. hour. The engines are also very economical in lubricating oil, as only about ·0045 lb. per h.p. is required for the slow-speed engines.

A feature of the Tosi engines is that they will fire regularly at speeds as slow as 40 r.p.m., at which speed compression pressure is 450 lbs., as against 485 lbs. per square inch at normal speed. One reason for the capacity to run at such a reduced speed is the fact that the phase and duration of the fuel injection period can be regulated according to the speed of the engine. At low speeds the quantity of fuel injection air is reduced and the cooling effect of the excessive quantity that would enter the cylinder, were no means of regulation provided, is entirely absent.

William Beardmore & Co., Ltd., also make a range of quick-running oil engines with cylinders 8½ ins. by 12 ins., in four, six, and eight-cylinder units, developing 200 to 400 b.h.p. at 750 r.p.m. Also they make engines with cylinders 12 ins. diameter by 12 ins. stroke in four, six, eight and twelve-cylinder units, developing 420 to 1,260 b.h.p. at 750 r.p.m.

These engines are of the four-cycle high compression airless injection type, with steel monobloc crankcase fitted with steel cylinder liners. The lower part of the crankcase consists of a light tray with an oil sump. The main bearings are phosphor-bronze lined with white metal, forced lubricated, and the crankshaft is of high-grade steel with ground pins and journals.

The pistons are made from forgings of a special aluminium alloy fitted with six cast iron piston rings and a floating piston pin of case-hardened nickel steel.

These engines are used for the Diesel electric cars of the Canadian Railways, and are being taken up by the L.M. & S. Railway because of their extremely light weight, high economy, and reliability.

Allen Diesel Engines.

The Allen Diesel engines are made in various sizes from 50 b.h.p. upwards, and are of the Burmeister & Wain design, which has proved very successful. The engines are of the vertical four-stroke cycle type, totally enclosed, and fitted with forced lubrication.

The construction of these engines has received very careful consideration to enable them to make long, continuous runs, such as are required for work on ships. A low fuel consumption is obtained, and the engines are also economical in lubricating oil.

The Allen Diesel engine is constructed mainly for auxiliary work, such as lighting, etc., on board ship. Further particulars of the engines designed by Burmeister & Wain are given on pp. 2413–16.

The Swiss Locomotive and Machine Works Diesel Engines.

(Bernard Holland & Co., 17 Victoria Street, Westminster, S.W. 1.)

These engines fall into three main classes: (1) airless injection horizontal single- and twin-cylinder engines of from 15 to 250 b.h.p., which command a market where simplicity of design and accessibility are factors of prime importance; (2) vertical airless injection four-stroke engines, ranging from 50 to 800 b.h.p. or, if Büchi exhaust turbo charged,* to over 13,000 b.h.p., which are suitable for electric generating sets, marine auxiliaries, and propelling sets; (3) Blastair and airless injection engines of larger powers up to 2,000 b.h.p., turbo-charged to 3,000 b.h.p., suitable for electricity generating stations, marine propulsion, etc

Schneider Diesel Engines.

Schneider & Co., Le Creusot, have had an extensive experience in the manufacture of large Diesel engines for land and marine purposes. One of the firm's experimental engines, a single cylinder two-stroke cycle type, has developed 1,200 h.p. at 150 r.p.m.

Two-stroke cycle Diesel engines of various powers up to 2,400 h.p. have been made and a number of these have been in regular work for over seven years. Four-stroke cycle engines have also been made and give very satisfactory service.

The Schneider engines are of the vertical type, of both the open and enclosed design, and all of them are fitted with crossheads. The makers have also completed some double-acting stationary Diesel engines of the four-cycle type developing up to 2,600 h.p.

Sulzer Diesel Engines.

Sulzer Bros. of Winterthur, Switzerland, make stationary engines of the four-cycle type from 50 to 1,000 b.h.p. For higher powers they have adopted the two-stroke cycle principle, and all of their marine engines are of this type. A large number of these engines are in operation, and the design of their stationary engines follows that of the marine type shown on p. 2421.

In the larger engines the reciprocating scavenge pumps are replaced by rotary turbo blowers, supplying scavenging air to the cylinder at a pressure of 1½ to 2 lbs. per sq. in. This simplifies the engine, and enables a greater shaft horse-power to be obtained from a given size of engine using the same mean indicated pressure in the cylinder.

Sulzer Bros. now make a new type of airless injection Diesel engine, shown by fig. 4, to meet the demand for oil engines of small powers. The engine operates on the two-stroke cycle, and ignition is obtained by the heat of compression only; the fuel being injected by the fuel pump into a water-cooled space called the pre-ignition chamber, situated in the cylinder head and in communication with the working cylinder. Exhaust ports are arranged in the lower

---

* A description of the Büchi System of Exhaust Turbo-charging is given in the Marine Section, p. 2435.

part of the cylinder liner on the opposite side of which are the scavenge ports, which admit compressed air from the crankcase when the piston is at the bottom of its stroke.

The fuel consumption on a 50 h.p. engine of this type is 0·47 lb. per b.h.p. hour when running on ordinary Diesel fuel oil. The engines start immediately from cold, and are able to run at

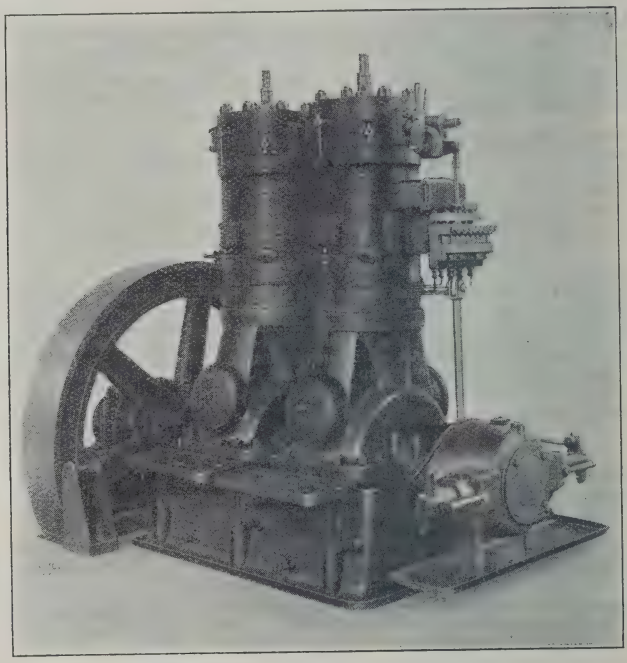


FIG. 4.—Sulzer Airless Injection Diesel Engine.

all loads, even quite light, with regularity. The only valve in regular operation on the cylinder is the fuel injection valve, which is automatic, also the air admission valve on the crankcase is automatic, which greatly simplify the engine, and the robustness of its construction will be apparent from the illustration.

CARELS DIESEL ENGINE.

The Carels Diesel engines are constructed by Messrs. Carels Frères, Ghent, Belgium, in sizes from 50 to 3,000 b.h.p. The first engine was made in 1894, and their standard four-cylinder 600 b.h.p. engine is illustrated by fig. 5.

In 1905 Carels exhibited at Liége a three-cylinder 500 b.h.p. engine; in 1908 they constructed their first 1,000 b.h.p. two-stroke cycle reversible marine engine, and in 1910 they built an engine developing 1,000 b.h.p. in a single cylinder.

Messrs. Carels Diesel & Steam Engines (London), Ltd., which is the British organisation of this firm, have given the following particulars regarding two 320 b.h.p. generating sets installed in 1906 for the Wakefield & District Tramways. The engines are direct coupled to 200 kw. generators at 174 r.p.m., and the average run per day during the sixteen years they have been installed is approximately seventeen hours. Only a small staff is required to run the engines, and the guaranteed fuel oil consumptions which were obtained are: full load, 0·42 lb. per b.h.p. hour; three-quarter load, 0·45 lb.; and half load, 0·476 lb. One class of lubricating oil is

CAREL'S DIESEL ENGINE.

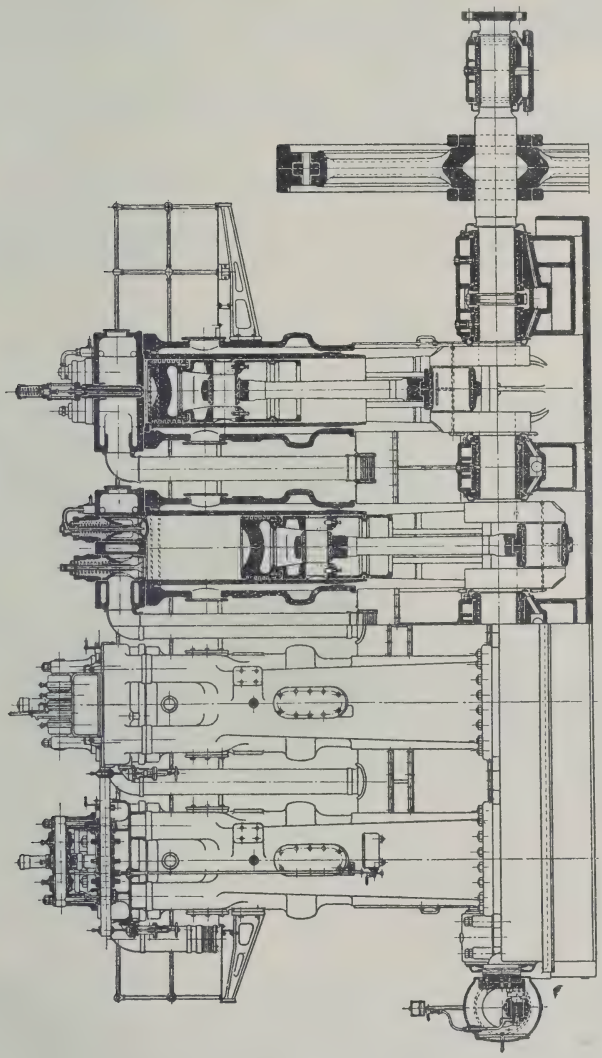


FIG. 5.—Carels 600 h.p. Four-stroke Cycle Diesel Engine.

suitable for all parts of the engine, and the consumption is 4·16 grammes per kw. hour. The average cost for repairs and maintenance over the sixteen hours is approximately 5*s.* per b.h.p. per annum.

These two engines are interesting, as they were among the first Diesel engines to be installed to deal solely with a fluctuating traction load, and their performance has been satisfactory; also the engines are running well at the present time.

Other Makers of Diesel Engines.

Geo. Ansaldo & Co., Rome, manufacture Diesel engines of large powers, and a number are in regular use. One of their 1,100 b.h.p. two-stroke cycle engines running at 100 r.p.m. was recently subjected to severe tests to determine the flexibility and overload capacity. It was found possible to run the engine at as low a speed as 35 r.p.m., and the following are particulars of the overload trials:—

Overload Trial on 1,100 B.H.P. Ansaldo Diesel Motor.

Overload.	Duration of Trial in hours.	R.P.M.	B.H.P.	Fuel Oil consumption, lbs. per b.h.p. hour.	Lubricating Oil consumption, lbs. per b.h.p. hour.
47 per cent.	1	127·7	1,622·5	0·445	—
30 ,,	30	116	1,433	0·440	0·006
12 ,,	5	104·7	1,232·5	0·425	—

These figures dispose of any doubt as to whether the Diesel engine as now constructed can be overloaded for a long period without damage.

Diesel engines are also manufactured by Svenska Aktiebolaget Nobel Diesel, Nynashamn, Sweden, in various sizes up to 3,000 h.p., and are made in both the four-stroke and two-stroke cycles. These engines are of the open vertical type and a large number have been supplied. The Nobel engine is one of the best-known Continental designs, for he first commenced its manufacture in Petrograd in 1897, and it is noteworthy that the Swedish company now intend to specialise on the two-stroke cycle engines.

The Atlas Diesel Co., Ltd., also make these engines in various sizes up to 1,000 b.h.p.

(For further information on Marine types of the Diesel engine, see p. 2411 *et seq.*)

English Electric Co. Fullagar Oil Engines.

The Fullagar oil engine is similar in design to the gas engine described on p. 1881, and is made by the *English Electric Co.*, Rugby, in five sizes, from 750 b.h.p. four-cylinder at 250 r.p.m. to 3,000 b.h.p. The engine is of the vertical enclosed opposed piston type, built up in units of two cylinders placed close together, each cylinder having two opposed pistons, there being one crank and one connecting rod per cylinder. The upper piston of one cylinder is connected to the lower piston of the second cylinder by diagonal rods.

The pistons move in opposite directions in one cylinder: one piston first uncovers the exhaust ports which extend all round the cylinder bore, then the other piston uncovers the scavenge ports, which also extend round the bore, at the other end of the cylinder, and fresh air at a low pressure is supplied by a pump formed in the upper crosshead, flowing from one end of the cylinder to the other, sweeping out the burnt gases and leaving the cylinder filled with fresh air.

The engine works on the Diesel principle, and on the two-stroke cycle, the air in the working cylinders being compressed until the temperature is raised to about 1,000° F., in order to ignite the fuel which is atomised and blown in by compressed air. A three-stage direct-driven compressor supplies the air for starting purposes and fuel injection. Fuel is directed horizontally between the pistons, so that the fuel does not impinge directly against the pistons. Scavenging air, as also that for the compressor, is supplied from chambers in which the upper crossheads work. The crossheads thus act as displacers, the automatic air inlet and delivery valves being arranged behind the chambers in question.

A number of engines are in satisfactory operation driving electric generators in this country and various parts of the world, while several engines of various powers up to 1,500 are now in course of construction at Rugby.

The only mechanically operated valves in regular operation are the fuel admission valves, there being one per cylinder; there are no cylinder covers, which is a very desirable simplification. Starting valves, opened by compressed air, are fitted on two cylinders, the valves being in operation during the first few revolutions only.

All main bearings, connecting rod bearings, and lower crossheads are forced lubricated, whilst the main pistons, upper crossheads, air compressor, and other parts are lubricated by mechanically operated lubricators operated from the cam shaft.

The balance of the engine is excellent, owing to the fact that the centre of gravity of the reciprocating masses working side by side nearly coincide; whilst, owing to its construction and

balance, the engine is suitable for working at higher rotative speeds than the ordinary reciprocating engine. The length of a 6-crank 1,125 b.h.p. engine is no greater than that of an ordinary 500-b.h.p. four-cycle engine; whilst the weight per b.h.p. is about 30 per cent. less than that of the four-cycle engine of equal power, the explanation of this being that the engine is in effect a double-acting two-stroke cycle engine without cylinder covers. Also the height to crane hook for removing pistons is less than that necessary to withdraw a piston and connecting-rod of a four-cycle engine of equal power.

Packings for Diesel Engines.

'BRITIMP' METALLIC PACKINGS.

(*British Metallic Packings Co.* (1929), *Ltd.*, 7 *Princes Street*, *Westminster*, *S.W.* 1.)

Each complete packing (which is equally suitable for heavy gas-engines) is made up of two sets of rings, A and B (fig. 6). The former are the packing rings proper, and they are protected from the heat of the burning gases by the latter.

Each packing ring consists of three or more segments, the number being determined by the diameter of the rod.

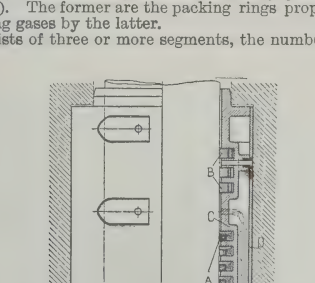


FIG. 6.

Each segment is a combination of gun-metal and cast-iron elements, arranged so as to form steam-tight joints between adjoining segments. The rings are contained in chambers formed in the inner wall, C, of the housing. Overheating of the rings is prevented by circulating water between the inner wall and the outer wall, D, of the housing. The lubricant is introduced by way of the pipe E. The housing is in halves to give easy access to the rings. The outside of the housing is exposed to the cylinder pressure, and it is this radial pressure which makes the joint between the two halves of the housing and eliminates the need of heavy bolting.

Weight of Diesel Engines.

For land installations, considerations of economical production have resulted in cast iron being extensively used in the Diesel engine; due to this, and to the high pressures employed, the weight per b.h.p. is somewhat high; thus, including fly-wheels, Guldner gives as the weight per b.h.p.:

A.	In single-cylinder engines:	600 lbs.
B.	,, two- ,, ,,	520 lbs.
C.	,, three- ,, ,,	350 lbs.

In recent practice, and particularly in cases where weight reduction is of importance, as *e.g.* in Diesel engines for submarines, these figures have been largely reduced; this is effected by using materials of considerably greater tensile strength in place of cast iron, and weights as low as 300, 250, and 200 lbs. per b.h.p. for cases A, B, and C respectively, may be found. In high-speed engines the weight has even been reduced to as little as 60 lbs. per b.h.p. with four-stroke engines, and 45 lbs. with two-stroke designs.

The Diesel engine is more economical in fuel consumption than any other type of internal combustion engine, but in construction it is necessarily a somewhat costly type.

Care of Diesel Engines.

The causes of perhaps 90 per cent. of all the troubles that occur with Diesel engines are either insufficient cooling, improper lubrication, or poor combustion. Good results cannot be expected from the engine unless adequate cooling and lubricating systems are provided and fuel oil of a certain quality used. The owner should not expect his operator to show good results if he has failed to furnish the needed apparatus and materials. The owner should further understand that a Diesel engine requires periodical minor adjustments and repairs, and the engineer should be given an opportunity to do this work.

Following out a schedule of inspection, such as one gives any high-grade machine, will result in a reliability and economy of operation that will justify the slight additional labour involved. There can be no justification for the neglect that the engine so often experiences. (*Power.*)

Cost of Diesel Engines.

The price of Diesel engines varies considerably, depending upon the design and speed, and may be taken as being from about 8*l.* to over 11*l.* per h.p. at the maker's works.

Diesel Engine Crankshafts.*

All shaft failures investigated owe their origin to lack of alignment of the main bearings. This is usually the result of unequal rates of wear of the several bearings supporting the shaft, though in some cases it can be traced to negligence during erection, or, in older engines, to defective foundations.

The following system of assuring perfect alignment of the shaft is recommended, especially to those engineers who have no reserve plant and who have to get repairs done during the week-end.

If the engine is a 4-cylinder, take out bearing C first and micrometer it. Look up the log-book and ascertain how much it has worn since the shaft was last realigned. Then take out bearing A, micrometer it, and ascertain the extent of wear in the same period. If it has worn down $\frac{5}{1000}$ in., and bearing C has worn down $\frac{15}{1000}$ in., then scrape down bearing A $\frac{4}{1000}$ in. more, re-bed it to the shaft, or, preferably, to a special dummy journal, and then replace it. Treat all the bearings in like manner. If the engine is a 2-cylinder, or a single-cylinder unit, then probably the bearing adjacent to the flywheel is most worn. It therefore should be taken out first, measured up, and replaced, scraping only if found 'high' at one end. Then take out the other bearings and scrape them in accordance with the data obtained. In the case of a 3-cylinder engine, the probability is largely in favour of bearing B being the lowest, and this, therefore, should be taken out first.

Thus, where plant is not in excess, the engineer can readjust a bearing each week-end until the whole set is completely put in order, and he runs very little risk of heated bearings if he is careful. The system has its limits. It would not do to apply it in this manner if rate of wear were very variable.

It is necessary to dismantle the whole engine to make a good start with the system. If the engine has been very carefully erected, and is not more than a year or two old, then one may exercise his discretion and assume that any large differences in the thickness of the main bearing shells is due to difference in rate of wear.

As regards the subsequent adjustment, running clearance of main bearings, it is not usually feasible to measure this with feelers. Where possible it is preferable to feel the lift in each bearing, and let the cap closer and closer until the bearing either heats (in which case scrape out the cap) or until the lift just disappears.

Recourse may be had to leads so long as they are considered only as a means of obtaining the *relative* and not the absolute clearance in a bearing. In the ordinary way, if one uses 20 or 21 gauge lead wire, running clearance will be right if the lead squeezes out to $\frac{8}{1000}$ in. or $\frac{9}{1000}$ in. thick. If one uses 16 gauge, then this may squeeze to about $\frac{10}{1000}$ in. or $\frac{11}{1000}$ in. thick. The actual clearance is probably only $\frac{4}{1000}$ in. to $\frac{6}{1000}$ in. The difference between real and apparent clearance depends upon a number of factors. It should be borne in mind that leads do not show the true clearance in a white-metal bearing.

Best Proportion of Shafts.

Tensile strength, not less than 34 tons.
Ductility not less than 25 per cent. in 2 ins.
Diameter of pins and journals, 0·525 to 0·54D.
Length of main bearings, 0·75 to 0·8D.
Length of big end pin, 0·525 to 0·54D.
Thickness of webs not less than 0·32D, but centre to centre of cylinder made minimum possible and any excess over the proposed length of pin and journal to be put into the webs.
Fillets to be not less than one-tenth the diameter of the shaft.

For details of Diesel engine parts see 'Marine Diesel Engines,' Section XXXVI.

* Paper by F. H. Smith read before the Diesel Engine Users' Association, July 12, 1916.

Alignment of Crankshafts and Engine Parts.*

(Geo. E. Windeler, M.I.Mech.E.)

Where small units are built up with separate bedplates and columns or crank cases, it is imperative to check the parallelism of the crankshaft journal from the planed surfaces of the bedplate. The other details which require alignment checking are the connecting rod and the piston, it being essential that the top and bottom bearings of the connecting rod are perfectly parallel and free from twist.

The following is a simple and efficient method for measuring the inaccuracies:

Place into the bottom end of the connecting rod (fig. 7) an accurately ground mandril, say $\frac{5}{1000}$ in. or $\frac{6}{1000}$ in. less in diameter than the bore of the bearing. A similar mandril is placed in the top end bearing. Generally speaking, this mandril is five to six times the length of the bearing. The whole is then placed vertically in two V-blocks, and a scribing block fitted with a micrometer screw is then used to measure the height of the mandril at each end which projects through the rod. The difference between the two dimensions is the exaggerated amount of inaccuracy, and, assuming that the bar is five times the length of the bearing, then the actual amount of error on the connecting-rod bearing itself would be one-fifth of the total error

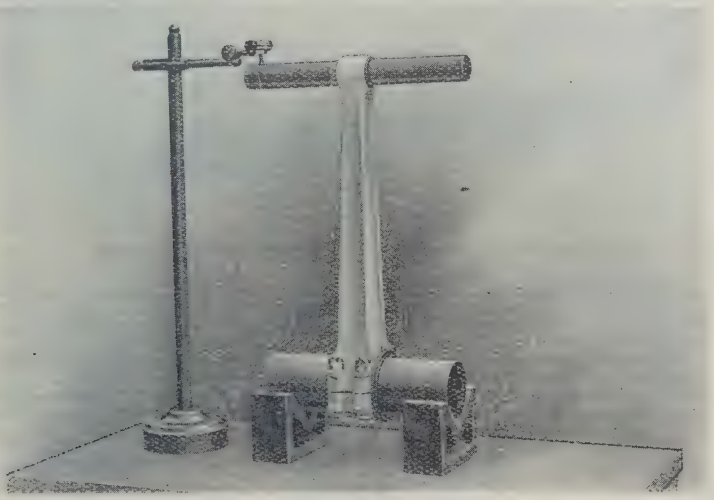


Fig. 7.—Testing the top and bottom bearings of the connecting rod.

shown; so that if the test-bar showed an out of parallel from one end to the other of $\frac{2}{1000}$ in., then the top end bearing of the connecting rod would be out of parallel with its bottom end bearing to the extent of $\frac{1}{1000}$ in. This is an instance of measuring, by exaggeration, very fine inaccuracies.

It is also desirable to test the rod to ensure that the holes lie perfectly parallel with each other in the vertical plane, that is to say, to ensure that the holes are not twisted with each other. To carry this into effect the rod is lowered down into a horizontal position (fig. 8) and a packing piece is placed under the rod immediately below the centre of the bearing. A scribing block with a micrometer screw is then used to test the height from one side to the other, in the same way as the vertical test was carried out. If any twist occurs with small rods, this can be corrected by twisting the rod, but in large rods other means have to be adopted to remove the error.

The same method is adopted for testing whether the piston pinhole is at right angles to the piston (fig. 9).

* Abstract of paper on 'Alignment of Machinery' read before the Manchester Association of Engineers, December 8, 1917.

In moderate-sized units the alignment of the cylinders and parts is often carried out by means of dummy parts which are accurately made, and may be regarded as a form of jig for testing the parts during the course of erection. With larger sized units this is not always practicable and other means have had to be resorted to to obtain the same degree of accuracy.

In large units such as are in general use to-day, and in which the rate of revolution is high and the running clearances very close—as, for instance, in a large vertical engine, say 20 ft. above the crankshaft, with the piston clearance $\frac{1}{100}$ in. diameter when cold—it is imperative that the cylinder and guide should be at right angles to the shaft, and that the

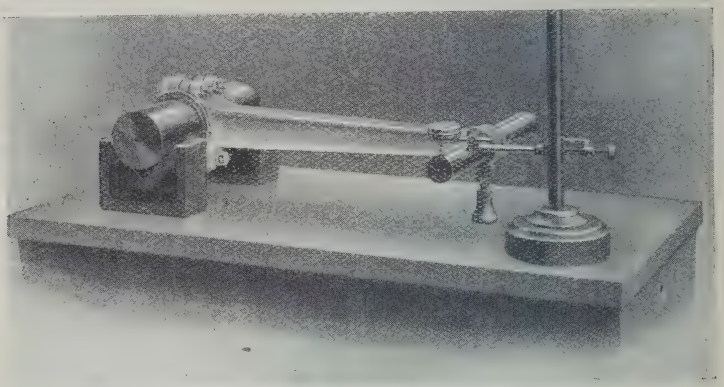


FIG. 8.—Testing the alignment of the holes with the vertical plane.

readings of the alignment should be accurately taken for tabulation. A fine wire is generally used for the alignment of the vertical components, but as it is a flexible member and gives to slightest pressure of contact, really reliable data are not obtained from its use in the ordinary way.

The following describes a simple audible electrical device used in conjunction with the fine-wire system for accurately checking the alignment of engine parts.

It is assumed that the engine bedplate is machined on the under side and parallel with the top side of the bedplate. The bedplate is then accurately levelled on parallel strips, both

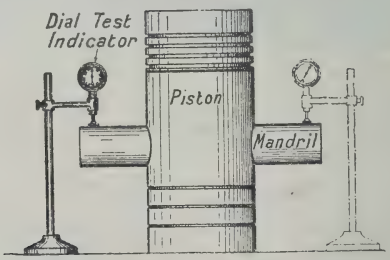


FIG. 9.—Diagram showing method of testing alignment of hole in piston for gudgeon pin.

longitudinally and transversely, and supported by such strips at defined points. The crankshaft is then placed in its bearings and tested for parallelism with the planed surface of the bed, and at the same time the shaft journals are tested for concentricity with each other by similar means to that described under small engines. Assuming this to be satisfactory, the columns are placed accurately in position and the cylinders placed on top of the columns. A fine wire is

then stretched over the whole length of engine, the ends passing over uprights secured to the engine bed and adjacent to the ends of the crankshaft. Suspended at each end is a non-magnetic 'plumb-bob' of sufficient weight to keep the wire taut. The wire is then set so as to intersect the centre lines marked on the end of the crankshaft.

The cylinders are then lined true longitudinally to this wire and correctly pitched one with another. Long wires are then passed through each cylinder, of sufficient length to pass between the crank webs, and a heavy 'plumb-bob' is secured to each wire and immersed in viscous oil contained in a small vessel, to prevent oscillations of the 'plumb-bob' and the wires. These wires are set accurately to the centre of the top end of the cylinders by means of the audible device, and readings are then taken at the top and bottom of cylinders and between the crank webs. The wire is suspended over a rounded knife-edge metal bar at the top end of the cylinder, in such a way as to prevent kinking and cutting of the wire, and this metal bar is supported on hard wood blocks from the top face of the cylinder. The use of wood blocks is necessary to

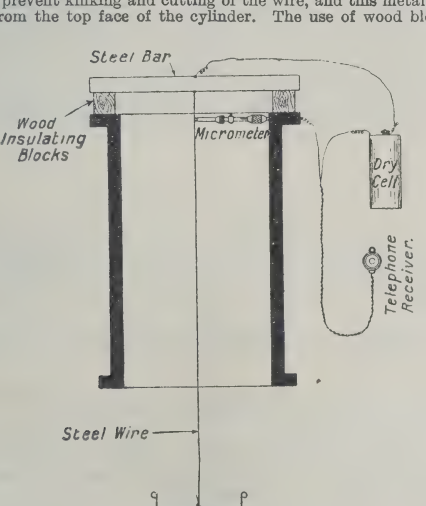


FIG. 10.—Diagram showing method of aligning cylinder using audible warning device.

insulate the metal bar from the engine parts. Connected to the metal bar by one of its terminals is a dry battery (fig. 10), the other terminal of which is earthed to the engine frame with a telephone receiver in circuit; then contact between the suspended wire and the cylinder will cause the electrical circuit to be closed.

The micrometer gauge is then set to half the diameter of the cylinder, less half the diameter of the wire, plus, say, $\frac{1}{2}$/1,000 of an inch. This setting allows $\frac{1}{2}$/1,000 in. pressure against the wire when the wire is set dead central at the top of the cylinder. The micrometer gauge is then inserted between the wire at the points named above, and if the gauge touches the wire the electrical circuit becomes closed by this contact and a distinct noise is transmitted through the telephone receiver. To enable the operator to have the use of both hands, the receiver may be held to his ear by means of an elastic band passed round his head. The micrometer gauge being adjustable, any 'plus' or 'minus' readings can be taken at will with a definite known amount of contact pressure on the wire. The readings obtained are entered on a chart. By reference to this chart the foreman decides what action should be taken to rectify the error shown, and the preliminary and final records are filed for reference.

With this device it is possible to obtain the same readings consistently, and it has the advantage that it is portable and adjustable to any engine, either horizontal or vertical. It

can be sent out to inspectors to check the alignment of engines which have been some time at work. Also the wear of the bore of the liners can be accurately recorded with the exact position at which the wear has taken place.

In multi-cylinder engines it is important that the crankshaft should be properly supported at each journal, otherwise springing of the shaft takes place at each compression and power stroke, which may ultimately result in fracture of the shaft. It is therefore important that the shaft and bearings should be periodically examined. One of the insurance companies has adopted a simple means of measuring the amount, and proceeds in the following manner:

The engine is barred round with each crank on its respective top centre, and the distance between the crank webs is measured with a micrometer gauge at a definite point and recorded. Each crank is then barred down to its bottom centre, the distance between the crank webs again taken, and the difference between the two readings noted. A simple calculation gives a close approximation of the amount that the shaft is clear from its bearings, and a new bearing can be fitted by gauging the thickness of the existing shell from outside diameter to inside diameter and then making the new bearing thickness plus the amount calculated from the figures taken between the crank webs. This method can also be used to detect any want of alignment of outboard bearings of gas and oil engines, and is invaluable for this particular difficulty. It is a very accurate and reliable method, and has saved many cranks from ultimate fracture.

Cyclic Variation Recorders.

A number of arrangements have been made for the purpose of accurately recording the cyclic variations of the rotational speed of an engine; the first attempts were mainly with tuning-forks, and this form of apparatus was well developed by Manlove, Alliott & Co. A variation of this was first used by the writer to test the cyclic regularity of steam engines built by Davey, Paxman & Co. in 1907.

In this apparatus the tuning-fork was replaced by an electric motor running at a constant speed and the records were taken on the peripheries of the flywheels, the flywheels having been suitably prepared.

Mirrlees, Bickerton & Day have taken tests of the cyclic variation of their Diesel engines in a similar manner, and the arrangement of the apparatus is as follows:—

On a base is formed a V, so that an ordinary lathe compound tool rest can be fitted to same, which tool rest is operated across the face of the wheel by a quick-pitch, square-thread screw, actuated by a heavy handwheel, so as to obtain as near as possible uniform movement across the space to be traversed, though this is not essential to the accuracy of the results.

Mounted on the compound rest is a small electric motor arranged in such a manner that it can be worked towards or away from the surface of the wheel by the compound rest. Secured on the end of the motor spindle is mounted a carefully balanced disc into which is inserted a hardened steel, accurately-ground pointer. Immediately below this is fixed a small bracket which holds another adjustable pointer vertically below the point of the rotating pointer on the disc when touching the face of the flywheel of the prime mover.

The object of the fixed pointer is to scribe a line in the form of a helix circumferentially on wax on the rim of the flywheel, whilst the duty of the rotating pointer is to cut short intersecting marks across these lines at constant intervals of time. These objects are obtained by moving the slide rest parallel to the surface of the wheel whilst the prime mover is running at normal revolutions and the electric motor at uniform speed.

The fixed pointer marks a helical line in the wax deposited on the periphery of the flywheel and the pointer marks off distances on this line at equal intervals of time. These distances are a record of the average speed of the flywheel during small intervals of time and from them a speed curve can be plotted.

It is essential that the points are in line vertically, so that the mark cut by the revolving pointer is bisected fairly accurately by the helical line. The electric motor, which is run at a high speed of revolutions, is supplied with current from an ordinary 12-volt motor-car lighting battery, which forms part of the equipment.

Generally speaking, the length of the record should be not less than 4 or 5 revolutions, though the number is only limited by the width of the flywheel surface and the pitch of helical line, which is usually about $\frac{1}{2}$ in. The motor is arranged at such a speed as to give a suitable number of readings on the periphery of the wheel, so that slight variations in the velocity of the rim can be detected. The distances between the marks are then carefully measured off with a steel tape and

vernier, and the figures obtained are plotted in graph form and show clearly the variations that occur in the speed of the prime mover.

This device has been used on different types and sizes of engines with good results.

The theoretical figures and graphs were based on information obtained from diagrams from actual engines several years before the results were taken by the instrument. The graphs shown on fig. 11 indicate these theoretical figures, and the graphs shown on fig. 12 are those produced by the information obtained with the Mirrlees, Bickerton & Day Cyclic Variation Recorder.

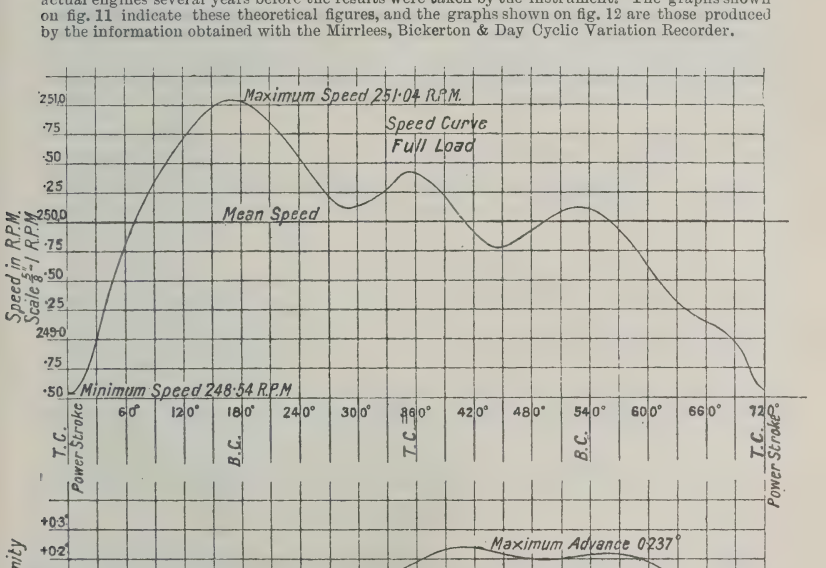


FIG. 11.

Scale.—$\frac{5}{8}$ in. (*i.e.* one square) = 30° of angular movement.

The novelty of the device lies in the use of the periphery of the flywheel of the prime mover as the recording drum, and the application to the flywheel of a delicate recording substance such as wax.

Having plotted a correct speed curve from data given by the trial, the angular deviation from true uniformity of rotation is obtained either (1) fairly accurately by direct calculation from the speed curve, or (2) very accurately by plotting another graph from the speed curve showing the changes in angular position of the flywheel relative to an imaginary wheel rotating uniformly.

The apparatus is inexpensive and of such a character that it can be readily manufactured and easily operated by any careful person with engineering experience. The data obtained can readily be transposed into scaled graphs by a skilled draughtsman

The amount of current used for operating the electric motor is very small, as its duty is to merely maintain at a constant speed the disc carrying the pointer, and to overcome the slight resistance offered by the pointer scratching the marks (one per revolution of the motor) on the surface of the wax.

The method adopted for securing the wax to the wheel is as follows :—The wheel is thoroughly polished and carefully rubbed down, whilst running, with turpentine; the wax, which is pure beeswax, is carefully heated to a liquid state in a glue kettle and then plastered on to the surface of the wheel. The use of the glue kettle was found necessary, as if the wax were overheated during the process of warming it would not adhere to the surface, the overheating apparently reducing its resilience and causing it to become 'short.'

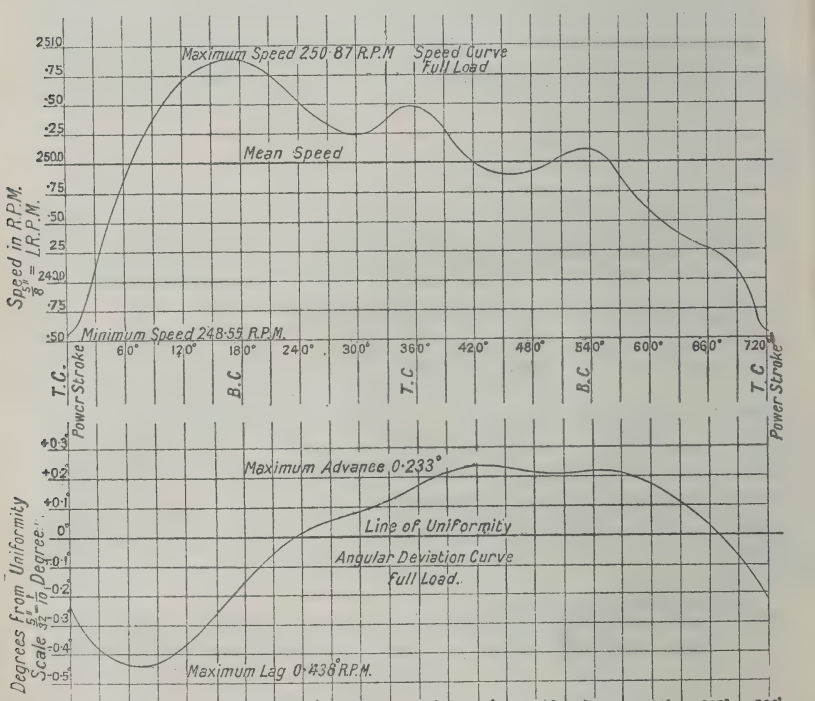


FIG. 12.

Scale.— $\frac{5}{8}$ in. (*i.e.* one square) = 30° of angular movement

The thickness of wax deposited on the wheel should be about $\frac{3}{16}$ in. and turned down till it is about $\frac{1}{16}$ in. thick. It can be readily removed afterwards with a scraper, but it was found undesirable to use the same wax over and over again, as a certain amount of greasy matter (probably oil vapour condensed on the wax) mingled with it and destroyed its nature; so that in carrying out experiments it is desirable to use fresh wax each time.

SEMI-DIESEL ENGINES.*

A semi-Diesel engine is a prime mover actuated by the gases resulting from the combustion of a hydrocarbon oil. A charge of oil is injected in the form of a spray into a combustion space open to the cylinder of the engine at or about the time of maximum compression in the cylinder. The heat derived from an uncooled portion of the combustion chamber, together with the heat generated by the compression of air to a moderate temperature, ignites the charge. The combustion of the charge takes place at, or approximately at, constant volume.

The semi-Diesel engine is the simplest form of oil engine, especially if of the two-stroke cycle type, and can be arranged to work satisfactorily with practically the same range of fuel oil as the Diesel engine.

Moderate compression pressures of 150 to 200 lbs. per sq. in. are adopted, consequently the working pressures are low and a comparatively light and inexpensive engine is obtained. The majority of engines operate on the two-stroke cycle and have solid injection of the fuel.

The weight of the engines varies from about 1½ to 2¼ cwts. per h.p., and the price at the makers' works is from about 7½*l.* to 9*l.* per b.h.p., depending on the size.

Crossley Crude Oil Engines.

The Crossley semi-Diesel engine of the two-stroke vertical type, built in four sizes from 6 to 20 b.h.p., and has a separate scavenging pump. Crankcase compression is eliminated, and this enables forced lubrication to be used for the bearings.

The engines will work on paraffin, gas oil, and the heavier crude and residual fuel oils with good economy, and because of the forced lubrication system the lubricating oil consumption is only about 1 per cent. of the fuel consumption.

The scavenge pump is attached to the side of the engine and delivers the air to the cylinder through water-cooled ports uncovered by the piston. The fuel pump is operated from the crankshaft by a cam, and governing is by allowing the excess oil to pass back to the suction side of the pump, through the control valve in the pump.

These engines can be fitted with a lampless burner, which may be ignited by the heat of compression, so avoiding any external heat at starting.

Marshall Crude Oil Engines.

Marshall, Sons & Co., Ltd., Gainsborough, were one of the earliest makers of the semi-Diesel engine, having commenced their manufacture in 1911, and they make horizontal engines from 8 to 25 b.h.p., and vertical engines from 30 to 450 b.h.p. The horizontal type is largely supplied on portable carriages for agricultural and other work.

The engines operate on the two-stroke cycle, and have an impulse every revolution. The governor is mounted on the crankshaft and controls the stroke of the fuel pump plunger automatically at all loads, so that the amount of fuel injected into the cylinder is proportional to the load and no hand regulation whatever is required. A noteworthy feature of the governor and fuel pump gear is that they are totally enclosed and work in an oil bath.

Marshall engines have a wide range of speed variation and do not cool down when running quite light. Also they will take overloads without trouble and no water injection is required in the cylinder. Separate cylinder liners are fitted on all sizes, and the engines are extremely economical in fuel and lubricating oil. Numbers of these engines have been supplied for Town and other Councils, for lighting, pumping, and sewage disposal. Among the many installations of Marshall engines may be mentioned one of three 200 b.h.p. engines which supply power continuously day and night in a chemical works.

Allen Oil Engines.

Crude oil engines are made by W. H. Allen, Sons & Co. Ltd., Bedford, in single, two and four cylinder sizes ranging in power from 10 to 320 b.h.p. The rating and speeds are moderate and the engines are capable of running continuously at full power.

The engines work on the two-stroke cycle and have an impulse every revolution, the governor being arranged to control the amount of fuel oil injected in proportion to the load. They can be started in a few minutes by preliminary heating of the vaporiser with a blow lamp, which is used at starting only. They will run at any load for an indefinite period without the starting lamp or any other additional device; also no adjustment is required for light running, and the engine will run under no-load without misfiring.

* This illogical name is rapidly falling out of use.—*Ed.*

A noteworthy feature is that the engines will work satisfactorily when running at reduced speeds, down to as low as half the normal speed. Regular running is obtained and the exceptional economy of these engines is still maintained.

The engines will work on almost any good quality crude or residual oil of specific gravity up to ·95, and the fuel consumption is only ·5 pint per b.h.p. per hour at full load. At ¾-load the consumption per b.h.p. is slightly increased and at ½-load the increase is not more than 20 per cent. Thus each engine has a large economical range of power.

Brotherhood Oil Engines.

Single and twin cylinder horizontal crude oil engines from 16 to 130 h.p. per cylinder are manufactured by Peter Brotherhood, Ltd., of Peterborough.

An illustration of an engine of 130 h.p. is shown in fig. 13.

These engines use compressed air for starting only, the injection of the fuel being performed by a pump, the stroke of which is varied according to the load by a centrifugal type governor.

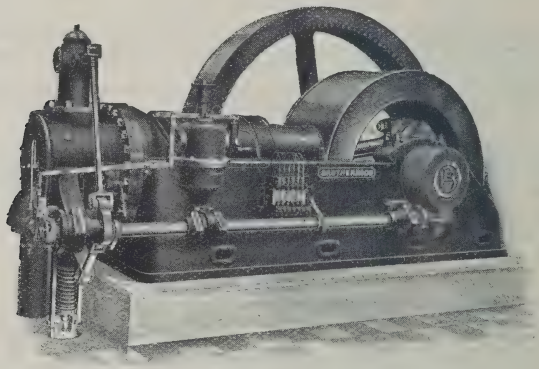


Fig. 13.—Peter Brotherhood Oil Engine.

They start without any preliminary heating when using all kinds of petroleum and its derivatives, such as crude oil, mazout, and tar oil.

The compressor for providing the air for starting is an independent unit, arranged to be driven by belt or separate motor, and, in case of need, by hand.

The consumption of crude oil with a calorific value of about 18,000 B.T.U. per lb. varies from 0·42 to 0·55 lb. per b.h.p. hour at full load, according to size.

Robey Oil Engines.

Semi-Diesel engines for land and marine work are made by Robey & Co., Ltd., Lincoln, in various sizes from 12 to 280 b.h.p. They are of the vertical two-stroke cycle type and are extremely simple.

The fuel is injected into the combustion chamber through a special form of sprayer, which gives good atomisation. The governor is of the centrifugal type mounted direct on the crankshaft and controls the amount of fuel injected in proportion to the load.

Robey engines will run on the usual kinds of refined, crude, or residual oil, and they have worked satisfactorily on tar oil.

Bolinder Engines.

The Bolinder engine (fig. 14), although essentially of the hot-bulb type, contains several features which distinguish it from other motors of this class. It is a high standardised production, and is now made in sizes up to 500 b.h.p. in four cylinders. As in most other motors of this design, the air is drawn in the crank chamber on the compression stroke of the piston, this air being compressed to a relatively low pressure in the crank chamber by the piston on its outward stroke. Just before the piston reaches the end of its stroke the exhaust ports are uncovered in the cylinder, allowing the exit of the exhaust gases, which are finally completely driven out through the exhaust pipe by the admission of the compressed air from the crank chamber through further ports in the cylinder wall uncovered by the piston as it continues on its downward stroke. On the upward stroke the piston compresses the air thus taken into the cylinder, and on reaching the top of its stroke the fuel is sprayed into the hot-bulb, thus causing combustion, vaporisation, and expansion within the cylinder.

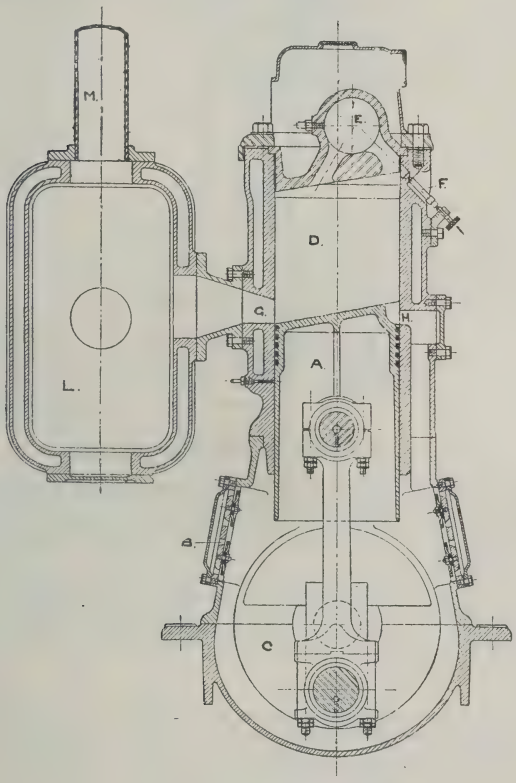


Fig. 14.—The Bolinder Engine.

For marine work, the Bolinder engine is made directly reversible, the principle being, broadly, the injection of fuel to cause an explosion after the engine has slowed down nearly to the point of stopping, and whilst the piston is on the upward stroke, so that reversal of motion follows. This is carried out by a special device regulating the action of the fuel pump.

The Bolinder motor is manufactured in Stockholm, but is handled in Great Britain by James Pollock, Sons & Co. Ltd.

' MIRRLEES-SIMPLEX ' OIL ENGINES.

Messrs. Mirrlees, Bickerton & Day Ltd., of Stockport, are now manufacturing a semi-Diesel type of oil engine known as the ' Mirrlees-Simplex ' engine. This series of engines is an addition to the firm's extensive range of ' Mirrlees-Diesel ' engines, for which they have a world-wide reputation, and the powers are from 35 b.h.p. to 240 b.h.p.

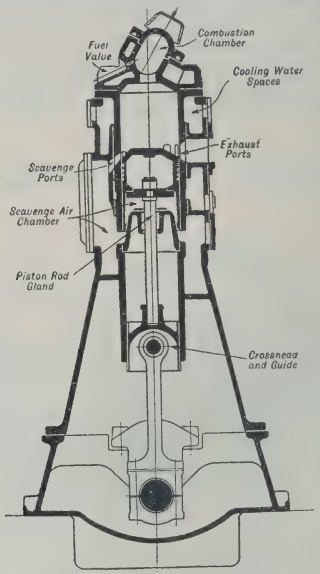


FIG. 15.—Section through ' Mirrlees-Simplex ' Engine.

The ' Mirrlees-Simplex ' engine is a low-compression two-cycle engine employing mechanical fuel injection and forced lubrication for the bearings. The outstanding feature of this engine is the elimination of the use of crankcase compression. This important departure from the common practice has been attained by employing the crosshead type of construction and compressing the scavenging air in the chamber formed between the piston and a gland plate through which the piston rod passes (see fig. 15).

This design completely isolates the crankcase from the cylinder, so a copious supply of lubricating oil can be used for all working parts within the crank-chamber without oil being carried over with the scavenging air into the cylinder, and the consumption is reduced. Also the crank-chamber oil is unaffected by piston and cylinder temperature, no grit is drawn into the crank-chamber with the scavenging air, the crosshead pin is not affected by the piston temperatures, and the side thrust on cylinder walls and piston is eliminated.

FIELDING AND PLATT OIL ENGINES.

The well-known ' Fielding ' Horizontal Four-stroke Semi-Diesel Oil Engines, of which there are many hundreds in operation, have special features, and are of the low compression hot bulb type. They are made in sizes from 12½ b.h.p. to 400 b.h.p., and the compression is 250 lbs. per sq. in. In practice this compression enables engines to be started with 150 lbs. air pressure, and it is important to note that for factory work, if ever the air receiver becomes exhausted these engines can easily be started by hand with the half compression cam in action.

The engines will run without attention from the time they are started, and, being fitted with the ' Fielding ' patent pump and governor control, work with exceptional smoothness. All patterns are of the single flywheel and third bearing type with separate water chamber at back end, and the cylinder is fully supported with oil tray round bed. Up to 100 b.h.p. single-cylinder engines are standardised, and above this power twin cylinder. The consumption of fuel varies from 0·45 lb. per b.h.p. per hour on the larger sizes down to 0·55 lb. per b.h.p. per hour on the smaller engines.

THE STILL (I.C. + STEAM) ENGINE.

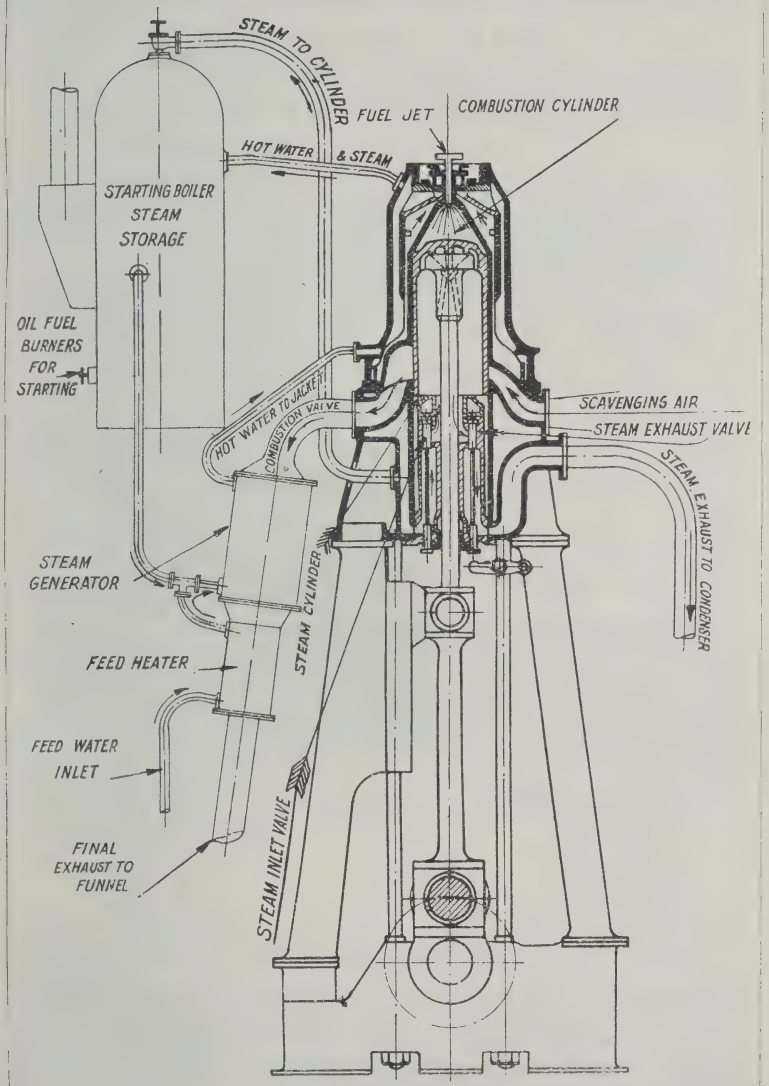


FIG. 16.

The fuel-injection system consists of a plunger pump fitted with a sleeve valve which is under control of the governor, and acts as a cut-off valve. The injection pump works very smoothly and without noise.

Other Makes of Semi-Diesel Engines.

Semi-Diesel engines of the horizontal four-stroke cycle type are manufactured by Tangyes, Ltd., Birmingham, and Bates & Scholes, Ltd., Denton, Manchester.

Engines of the vertical design operating on the two-stroke cycle are also made by Plenty & Son, Newbury; Norris, Henty & Gardners Ltd.; Widdop & Co., Keighley; and the Hamworthy Engineering Co., Ltd., Poole.

Plenty-Still (I.C. + Steam) Engines.

An engine of high thermal efficiency has been developed by the Plenty- Still Engine Co. and is manufactured by several licensees. The main working cylinder uses liquid or gaseous fuel, and in the same cylinder, on the opposite side of the piston, steam is employed. The steam is produced by the recoverable heat from the internal combustion side, which passes through the walls of the cylinder as well as into the exhaust gases.

Heat, wasted in other internal combustion engines, is usefully employed in the Still engine to increase the power of the engine and reduce the consumption of fuel per horse-power developed.

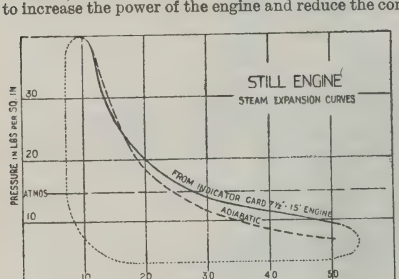


Fig. 17.

As will be seen from fig. 16, the jacket and cooling water form part of the circulating system of a steam generator, which may be an integral part of the engine. The cylinder liner is specially constructed and is about one-third of the usual thickness; it is ribbed externally to add to its conducting surface. The quantity of steam capable of being generated from the rejected heat of an internal combustion engine depends upon the efficiency of the combustion cycle and on the load. At full load the quantity may be about 7 lbs. per b.h.p. of the engine.

In the indicator diagram (fig. 17) taken from the under, or steam, side of the piston, the adiabatic expansion line as plotted shows there is no loss from condensation. Heat is taken from the piston and cylinder walls and the steam exhausts at a slight superheat.

An experimental engine tested by Professor Vernon Boys generated 3·28 lbs. of steam from the jacket alone, and from the jacket and exhaust together 6 lbs. of steam per b.h.p. hour. The average m.e.p. from the combustion stroke was 90 lbs. per sq. in. The steam evaporated by the heat rejected gave 14 lbs. m.e.p. per sq. in. on every return stroke of the piston. Thus, the four-stroke cycle engine had a total m.e.p. of 118 lbs. per cycle.

The possibilities of this design are very great; by admitting additional steam generated by fuel under the boiler, the total m.e.p. can be increased considerably. This enables the engine to take big overloads, and by the use of steam easy starting and manœuvring are obtained.

The thermal efficiency of the Still engine compared with other prime movers is shown in the following table:

Marine steam engine, superheated	16% to 17%
Parsons steam turbines	15 ,, 20
Gas engines	22 ,, 30
Petrol engines	23 ,, 27
Semi-Diesel engines	27 ,, 30
Still gas engine	31
Still petrol engine	33
Diesel engines	30 ,, 36
Still oil engines	38 ,, 44

HIGH-COMPRESSION OIL ENGINES.

The high-compression oil engine is the outcome of experiments to simplify and reduce the cost of Diesel engines. Many attempts have been made to produce engines capable of burning cheap heavy oils with an economy approaching that of the Diesel, but in which the high-pressure air blast is avoided without sacrificing the capability of the engines to start immediately from cold.

Among the first of the successful engines was the Vickers, which has been largely developed for propelling submarines. Several other makes of high-compression engines are now on the market, and the elimination of the air compressor has produced a marked improvement as regards simplicity compared with Diesel engines.

What is known as 'solid injection' is adopted for spraying the fuel into the cylinder, the oil being raised to a high pressure by the fuel pump momentarily at the period of injection, or permanently in a small receiver and admitted to the cylinder by timing the opening of the fuel admission valve.

The weight and price per horse-power are lower on these engines than Diesel engines, but up to the present the economy is not quite so good.

Babcock Oil Engines.

Babcock cold-starting high-compression oil engines are made in six sizes, 30 to 150 b.h.p., and are of the horizontal type working on the two-stroke cycle.

The piston is of the differential type and as the larger diameter acts as an air pump a greater amount of air than the usual displacement of the main piston is supplied to the cylinder, with the result that a better scavenging effect is obtained as well as high mean pressures and increased economy. The fuel consumption is only 0·44 lb. per b.h.p. per hour at full load.

Governing is effected by varying the amount of oil delivered to the cylinder every revolution through a by-pass valve on the suction side of the pump, the fuel pump being operated by an adjustable cam on the crankshaft and the by-pass valve is controlled by a spring-loaded governor working in an oil bath. The sprayer is of the differential type, spring loaded, so that the fuel is injected under heavy pressure.

The important feature of their patent removable crankcase cover is embodied on these engines, thus making the piston and connecting rod very accessible and at the same time dust-proof.

Vickers Oil Engines.

Messrs. Vickers Works at Barrow (now part of Vickers-Armstrongs, Ltd.) were the pioneers of the solid or airless injection type of Diesel engine, and have built these engines in a very wide range of powers. High brake mean pressures are obtainable, especially with engines of the submarine type. A typical small unit is a six-cylinder non-reversing engine with a bore of 14 ins. and a stroke of 14·8 ins., the pistons being of the trunk type and uncooled. When running in submarine conditions at 450 r.p.m. the output is 750 b.h.p.

The engines are of the four-stroke cycle single acting type, the commercial engines being of the crosshead type with water-cooled pistons, and the crank case being separated from the cylinders by an oil-tight diaphragm.

The fuel pumps have four or more units discharging to a common main from which the spray valves derive their fuel supply. The spray valves are mechanically operated, the duration of opening being variable according to the load.

Reversing of Vickers engines is usually arranged by means of a sliding shaft with sets of cams, one set for 'ahead' and one for 'astern' running. The reversing gear first lifts the valve actuating levers clear of the cams, then slides the camshaft along, and finally lowers the levers to contact with the second set of cams.

Messrs. Vickers-Armstrongs, Ltd., also build the Vickers-M.A.N. type two-stroke cycle engine, both single and double acting. The outstanding feature of these engines is the patented scavenge system. The exhaust and scavenge ports are above one another. The scavenge ports are so shaped and directed that the air current is directed towards the piston and then flows along the opposite side of the cylinder to the cover. It is there deflected and follows the exhaust gases to the exhaust ports.

The efficiency of this system is demonstrated by the fact that the scavenge pressure required is very low, about 1 lb./sq. in. These engines are supplied either with air blast or solid injection. A pair of engines of this kind, of a total power of 8,000 s.h.p., have recently been fitted in H.M.S. *Medway*, now in commission as a submarine depot ship.

Crossley Oil Engines.

Crossley Bros. Ltd., of Openshaw, Manchester, make high-compression oil engines in sizes from 20 to 300 b.h.p. The engines are of the vertical and horizontal four-stroke cycle type, with airless injection of fuel, and they are totally enclosed in all the vertical and some of the horizontal sizes.

All of the engines start up from cold without preheating, and will work on crude, refined, or residual and vegetable oils. The patent fuel pump and governor gear are particularly interesting, as the commencement of the fuel injection is constant at all loads. Governing is effected by reducing the amount of the charge delivered by the fuel oil pump at reduced engine loads; the full consumption is thus, as far as possible, proportioned to the power output. Messrs. Crossley state that in a recent test the consumption of (crude) oil for b.h.p. hour at full load was 0·39 lb., and at about half load rose to only 0·44 lb. with their governing arrangement. They point out that this performance compares favourably with that of the more costly Diesel type of engine.

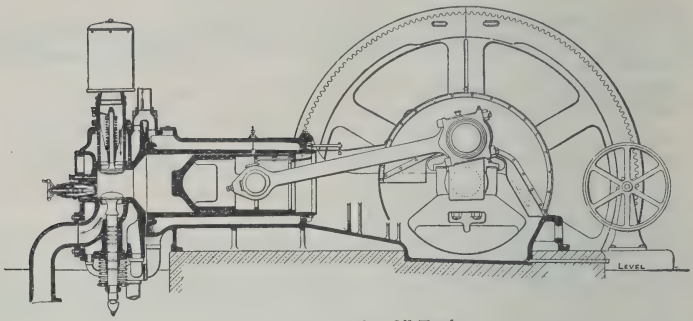


FIG. 18.—The Crossley Oil Engine.

These engines can be supplied as convertible engines to work with either fuel oil or producer gas, whichever may be the cheaper fuel at the time.

CROSSLEY PREMIER OIL ENGINES.

The Premier Gas Engine Co., Ltd., of Nottingham, in combination with Messrs. Crossley Bros., Ltd., now build cold-starting multi-cylinder horizontal heavy-oil engines with solid injection of fuel. The engine is made in units up to 1,200 b.h.p. and is specially adapted for direct coupling to electric generators.

Up to 500 b.h.p. there are three and four cylinders mounted on one bedplate, above 500 b.h.p. there are two bedplates, each having mounted thereon three or four cylinders, with central drive.

The engine has the advantage of low head room and accessibility to all working parts. It is very economical in fuel and lubricating oil, and is specially suitable for working on heavy residual fuel oils; also will start from cold without preheating.

RUSTON-HORNSBY OIL ENGINES.

This engine of Messrs. Ruston & Hornsby, Ltd., fig. 19, is of the solid injection type. The makers have succeeded in dispensing with the high-pressure air blast for fuel injection altogether, a quick-acting fuel force-pump being employed to inject the fuel oil through a specially designed mechanical atomiser.

Instantaneous starting is effected with certainty from cold under all conditions, without the use of a blow lamp or application of external heat.

These engines are of the horizontal and vertical type. The horizontal engines being built in single- and twin-cylinder units with powers ranging from 18 to 280 b.h.p., whilst the vertical engines are of the multi-cylinder type, being suitable for either land or marine purposes, the range of powers being from 27 to 840 b.h.p. The weights and price per horse power are lower on these engines than Diesel engines, and the brake thermal efficiency is higher.

The engines will work from a wide range of fuels, and particular attention is called to the convertible oil/gas engine which, by being convertible to gas, renders the owner independent of an oil shortage, or of a shortage of solid fuels such as coals, coke, etc., by a change over to oil. Thus this engine carries with it the important advantage that it can be altered to run on the most economical fuel available. To change from oil to gas fuel, change the compression plates fitted under foot of the connecting rod, remove the atomiser and substitute the sparking plug.

Low lubricating oil consumption is an important feature of these engines, only 0·0003 to 0·0004 gallon (0·0013 to 0·0018 litres) per b.h.p. hour rated load being used, and one class of lubricating oil only is required. Lubrication is of the pressure type and is entirely automatic.

The oil-fuel pump is driven through a lever by a quick-acting cam on the layshaft, which is completely enclosed and runs in oil. A lever is provided by which each pump can be worked by hand, or thrown out of action altogether to stop the engine. The fuel pumps and side shafts of the horizontal engines are carried by the bedplate, thus relieving the cylinder head.

The mechanical atomiser, which is fixed in the combustion head, consists of a duplex valve loaded by a powerful spring, which is lifted at each stroke of the oil-fuel pump. The valve has two seats of different diameters, the smaller of which closes the opening of the combustion chamber, and the larger a leak pipe to the oil tank. When the smaller valve lifts, the oil is forced through channels converging to a central outlet, which so split up the stream that it emerges on either side of the nozzle in a cone-shaped cloud of oil particles. The movement of the valve is about $\frac{1}{50}$th inch, and is entirely automatic in action. There is no packing or oil-tight diaphragm, or any complication liable to get out of order, and the action, though delicate, is found in practice to be absolutely reliable.

The regulation of the speed is effected by the opening of a by-pass valve by the governor, the toe of the valve resting upon a lever carried upon a cam. The fulcrum of the lever is adjust-

able by a screw, so that although the lift of the valve is minute, there is always a means at hand for easily keeping it in perfect adjustment. The consumption of oil, even the lowest grade of crude, residue, or fuel oils, is complete, while the economy is remarkable.

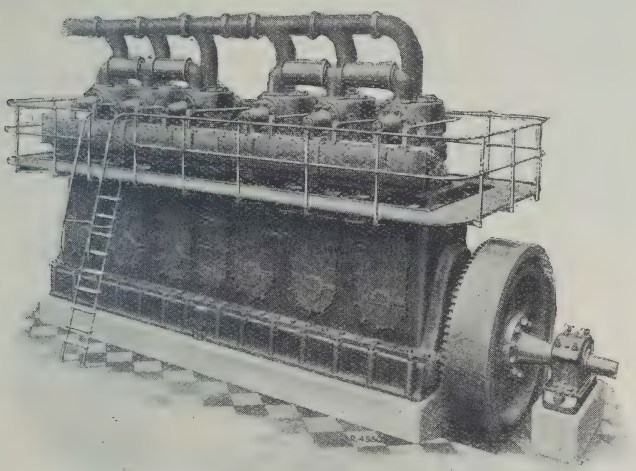


FIG. 19.—Ruston Four-cylinder Vertical Engine.

Tests which have been made show that a high thermal efficiency is obtained with the cheapest fuel oils on the market, the consumption ranging from 0·40 to 0·47 lb. per b.h.p. hour according to the size of the engine.

NATIONAL HIGH-COMPRESSION OIL ENGINES.

The National Gas Engine Co., Ltd., Ashton-under-Lyne, make high-compression oil engines in sizes from 18 to 130 h.p. single-cylinder type, and up to 260 h.p. in double-cylinder engines. The engines are of the horizontal four-stroke cycle type. These engines start up from cold without pre-heating and will work from crude and residual oils. Vertical engines are also made in sizes from 25 to 900 b.h.p.

A special fuel pump and governor gear are fitted, which give efficient and silent running, and the engines fire at all loads. The fuel consumption of these engines is as low as 0·42 lb., and does not rise above 0·5 lb. at half load.

CAMPBELL OIL ENGINES.

Campbell heavy oil engines are now made by Messrs. Kryn & Lahy, Letchworth, and work on the four-stroke cycle with a compression pressure of about 300 lbs. per sq. in. The sizes are from 10 to 140 b.h.p.

The fuel is injected by a cam-operated ram pump through a mechanical atomiser which delivers the fuel into the combustion chamber in the form of an extremely fine spray or mist. The speed of the engine is governed by increasing or diminishing the stroke of the fuel pump and thus the quantity of fuel injected in each cycle; there is no by-pass on the fuel pump. This system provides a working impulse in every cycle at all loads, thus ensuring minimum speed fluctuation and approximate equalisation of the working stresses in consecutive cycles.

PETTER HIGH-COMPRESSION HEAVY-OIL ENGINES.

The Petter heavy-oil engine has recently been modified to dispense with the usual hot bulb, and is built in sizes from 25 b.h.p. in one cylinder to 400 b.h.p. in four cylinders.

The vaporiser is entirely water-cooled and ignition is effected spontaneously by the internal heat of the compressed air, after the initial heat at starting. The piston and under-side of the cylinder head are entirely symmetrical, so that they are free for expansion. The atomiser is a simple

automatic spraying device arranged to be adjustable to suit different grades of fuel oil. The improved design enables a very low fuel consumption to be obtained, this being only 0·43 lb. per b.h.p. per hour in the larger sizes.

The engine is shown in fig. 21, p. 1921. Crankcase compression is adopted and is maintained dry from lubricating oil, with the result that the consumption is extremely low. This, together with the improved efficiency, renders the exhaust practically invisible at all loads.

Smaller engines of the semi-Diesel type from 5 to 42 b.h.p. are also manufactured by Petters, Ltd., and start from cold by means of their special cartridge.

Petter marine engines are made in units of one, two, three, four, and six cylinders, ranging in power from 5 to 400 b.h.p.

The single and two-cylinder engines are fitted with a clutch and reverse gear; the three, four, and six-cylinder engines are directly reversible by compressed air, and these engines do not require a clutch, but are supplied with an independent air compressor and electric generator oil engine driven set. Also these engines have a thrust block of the Michell type.

All of the engines are fitted with bilge and circulating pumps, and the main bearings are water-cooled, and the marine engines comply with Lloyd's, Bureau Véritas, and Board of Trade requirements.

Blackstone Oil Engines.

Messrs. Blackstone & Co., Ltd., of Stamford, make cold-starting heavy fuel oil engines in sizes from 6 to 1,200 b.h.p., and their patent system of spring injection enables the engines to start from cold without preheating and work with great economy.

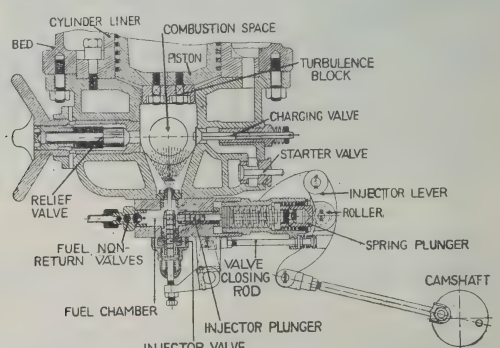


Fig. 20.

Fig. 20 shows a sectional view of the combustion chamber and the spring injection system. The fuel is injected at the same point at all loads and the combustion period is prolonged so that there is no great rise in initial pressure at the moment of injection. The fuel consumption on the larger engines is only 0·4 lb. per b.h.p. per hour.

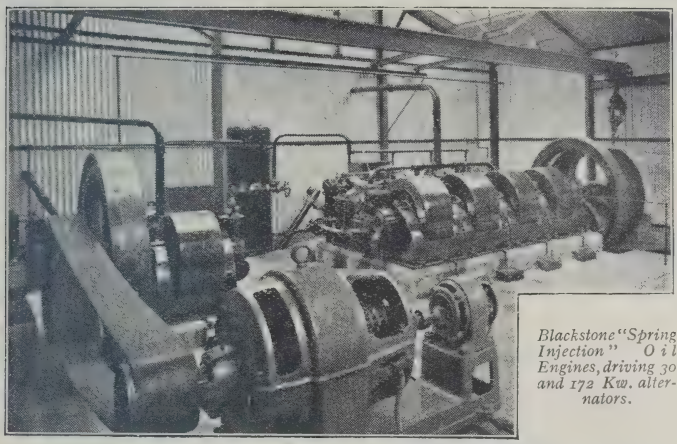
The fuel-injection arrangement consists of a low-pressure fuel pump, a moderate-pressure fuel injector, consisting of two members with a spiral spring between, and a spring-loaded fuel injection valve. The correct quantity of fuel is delivered by the pump and displaces the injector plunger outwards, there being no pressure on this at the moment. The injector lever next compresses the spring, and the injector plunger is held by abutting upon the fuel retained by the injector valve, which is held on its seating by the compression of an auxiliary spring. This spring is released by the timed operation of the injector lever, so allowing the fuel which is under pressure of 1,000 lbs. per sq. in. to open the injector valve. The injector plunger is then forced inwards and injects the fuel through the valve jet orifice into the combustion chamber.

Stationary and portable horizontal hopper-cooled engines in five sizes from 5 to 13 b.h.p., working on the same principle, are also made and on these starting from cold is obtained by inserting a smouldering fuse into the combustion chamber.

Paxman Oil Engines.

Cold-starting high-compression heavy fuel oil engines are made by Davey, Paxman & Co., Ltd., Colchester, in single, two, three, four, five and six cylinder units, ranging from 25 to 1,200 b.h.p. at speeds of 360 to 250 r.p.m.

The engines work on the four-stroke cycle with 380 lbs. compression and a maximum pressure in the cylinder of about 550 lbs. Each cylinder stand is a separate casting bolted to the bed, and has a removable liner. The cylinder head is fitted with inlet valve, exhaust valve, and spring



Blackstone "Spring Injection" Oil Engines, driving 30 and 172 Kw. alternators.

FOR 1/- AN HOUR

in fuel and lubricating oils, a Blackstone engine will develop 55 b.h.p., or 36 units of electricity continuously.

The method of injecting fuel into the combustion chamber—"Spring Injection"—produces a constant burning rather than explosion of the fuel, giving a high mean effective pressure but retaining a comparatively low initial pressure.

A guaranteed minimum fuel consumption is maintained, which, in engines with cylinders of 55 or more b.h.p. does not exceed 0·4 lbs. per b.h.p. hour at full and three-quarter loads.

BLACKSTONE

"Spring Injection" Oil Engines

can be started instantly from cold on their normal fuel and are accurately governed. Independent low pressure fuel pumps are fitted to each cylinder and all working parts of the engine are completely accessible. These units are built in single and multi-cylinder types from 6 to 1200 b.h.p.

Full particulars will be sent on request for publication YB.

BLACKSTONE & CO., LTD. (Est. 1837)

STAMFORD, LINCOLNSHIRE, ENGLAND.

Associated with A.G.E., Ltd.

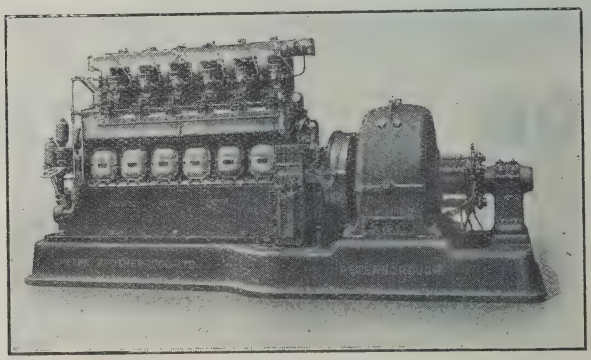
F. 1920

The BROTHERHOOD RICARDO

HIGH SPEED DIESEL ENGINE.

900/1000 R.p.m.

Four cycle, 100 to 400 B.H.P. in 2 to 8 cylinders.



No inlet or exhaust valves ; a single sleeve.

Single hole unchokeable fuel injector.

Fuel consumption ·4 lbs. per B.H.P. hour.

Weight for 400 B.H.P. under 40 lbs. per B.H.P.

PETER BROTHERHOOD, LTD.

ENGINEERS - - PETERBOROUGH

The first builders of engines of this type

Write for Catalogue R.D. 330

F 1921

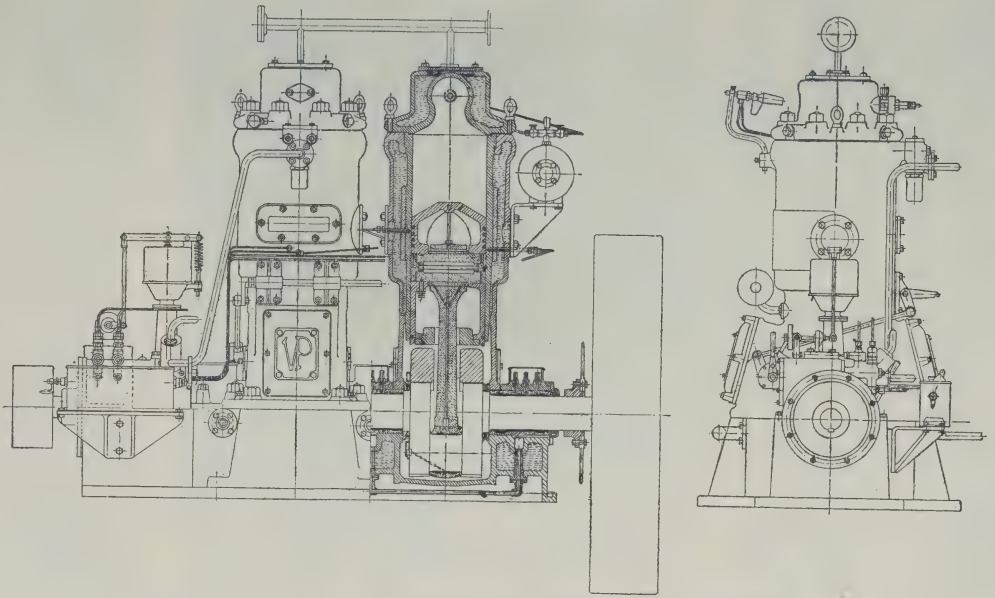


FIG. 21.—**Petter High-Compression Heavy-Oil Engine.** Sectional Arrangement.

injection fuel valve. The crankshaft bearings are ring lubricated, and a mechanically operated lubricator is fitted for the piston and connecting-rod bearings.

The valve gear is their patented eccentric-driven type, which gives a very smooth action to the valves and in a large measure is responsible for the absence of shock and the quiet running of the Paxman engine.

The engines can be converted to work on town, producer, or natural gas, simply by an alteration to the turbulence plate, and the same cylinder heads and valve gear are used. This is of great advantage for oilfields, where natural gas is often obtained before oil is struck.

McLaren-Benz Oil Engine.

High-speed lightweight high-compression oil engines are made by J. & H. McLaren, Ltd., Leeds, in sizes up to 450 b.h.p. for stationary and marine work. They are of the two, four, and six cylinder type, running at 250 to 800 r.p.m., and the weight of the 90 b.h.p. engine is only about 4,800 lbs.

The engines work on the four-stroke cycle, with compressive pressure of 450 lbs., and the fuel is injected into an ignition chamber, which is connected by a funnel-shaped liner with the cylinder. The fuel is completely vaporised, and the vapour passes through three holes in the base of the liner to the larger volume of compressed air in the cylinder.

Messrs. McLaren use these engines on their motor cable ploughing tackles and traction engines, and supply them for rail locomotives, etc., where powerful engines of light weight are necessary.

LOW-COMPRESSION OIL ENGINES.

The early oil engines were all low-compression engines, and the desire to use the heavier grades of petroleum, which are cheaper than the more refined grades, caused the development or higher pressures, so that the increased temperature could completely vaporise the heavier fuels.

The low-compression oil engines still have, and will always continue to have, a wide application, owing to the low working pressures and small cost.

In the normal 'oil engine' a heavy kerosene is usually employed as fuel; Russian oil is, in general, rather preferred, though the ordinary American brands are also used with quite satisfactory results. A standardised oil described as R.V.O. (Russian vaporising oil) of specific gravity 0·825 and flash point 86° F. (close test) is regarded as the normal fuel for these engines; other descriptions commonly used, however, include Rocklight, Russolene, Homelight, Gladiator, Phœbus, Silver Spray, Crown Diamond, Royal Daylight, White Rose, Tea Rose, and Bear Greek. Among the heavier fuels used are refined shale-oils as Broxburn and Scotch Lighthouse, and also the refined heavy oils of Burma, Assam, Roumania, etc.

The average composition of the petroleum oils used is 85½ per cent. carbon, 13½ per cent. hydrogen, and 1 per cent. oxygen, and the mean calorific value, by Berthelot calorimeter, is roundly 19,600 B.Th.U. per lb. When arranged to use either petrol, kerosene, or heavier oils, engines usually develop powers in the proportion of about 100 : 90 : 80, respectively.

It should be carefully noted that the nominal power of internal combustion engines in general is for location at sea-level; when required to work at an altitude a deduction from the nominal power should be made of 3 per cent. for each 1,000 ft. of height, due to the diminished density of the air.

In the ordinary oil engine the heavy kerosene fuel is vaporised in a heated chamber maintained at the necessary temperature either by heat derived from the successive explosions (hot-bulb vaporisers), or from the exhaust of the engine; or, lastly, in a few cases, by means of a separate blow-lamp. In most cases a preliminary heating of the vaporiser by a blow-lamp is necessary for 10 to 15 minutes before the engine can be started. These engines are practically all of the single-acting four-stroke type, and compress a mixture of kerosene vapour and air into the combustion chamber.

Ignition of the charge is sometimes automatic; high and low tension magneto ignition is also used, while in cases where lamp-heated vaporisers are adopted tube ignition is general. On account of the care and attention necessary where lamps are used, and also in some cases due to increased fire risks, makers are in general favouring one or other of the methods of lampless ignition.

Blackstone Oil Engines.

When starting these engines the vaporiser and the igniter are heated by a lamp in from four to six minutes. The flywheel is then turned forward by hand (unless a self-starter is used) to obtain the first explosion, after which the engine works automatically, the necessary heat for the vaporiser and for ignition being maintained by the successive explosions; a continuous burning lamp is therefore dispensed with. A timing or ignition valve is used to regulate the period of ignition. During compression this valve cuts off all heated parts, and the utmost regularity of the explosions under all loads is secured. A patented water drip valve is fitted to enable a small quantity of water to be admitted to the vaporiser to regulate its heat on long, continuous, heavy loads. As long as there is oil in the tank the engine continues to work.

The engine is governed by means of our patent direct-acting shaft governor, which controls the vapour valve—this system having been found to give the most economical and efficient results. It regulates the oil consumption to the power given off. The speed may easily and instantly be regulated whilst the engine is running.

Any ordinary lamp oil may be used, but the efficiency of the engine depends upon the quality.

National Oil Engines.

Oil engines of the horizontal four-stroke cycle type are made by the National Gas Engine Co., Ltd., Ashton-under-Lyne, in sizes from 2½ to 47½ b.h.p. These engines are made in two types: (*a*) the petrol-paraffin engines, which start on petrol and then run on paraffin, and are fitted with electric ignition, and (*b*) the hot-bulb engines, a special feature of which is the vaporiser and ignition timing device, which secures effective vaporisation of the oil and the proper timing of the ignition at all loads. A lamp must be used for starting on the hot-bulb engines, and the vaporiser needs to be heated for about ten minutes to make it hot enough to start the engine.

The engines are supplied both stationary and portable, and are suitable for agricultural and industrial work.

Hamworthy Oil Engines.

Hamworthy oil engines are made in nine sizes—2½, 4, 6, 9, 11, 15, 20, 25, and 30 b.h.p.—and are manufactured on strictly repetition lines by the Hamworthy Engineering Co., Poole.

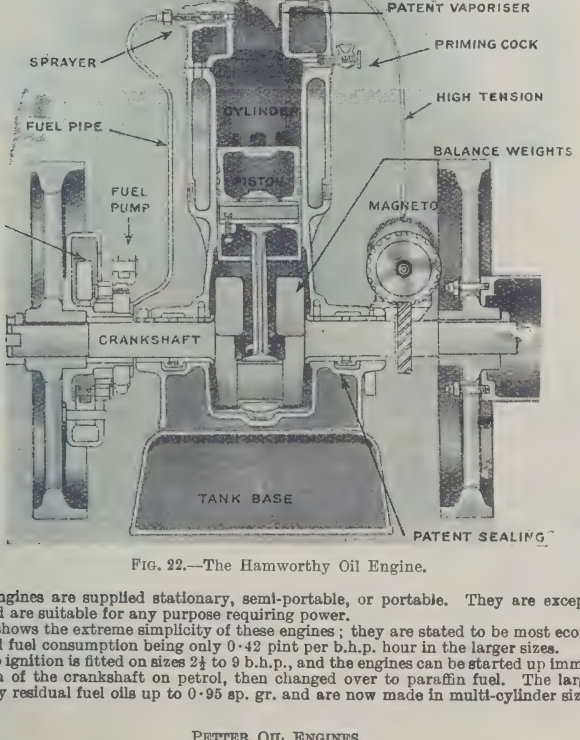


FIG. 22.—The Hamworthy Oil Engine.

These engines are supplied stationary, semi-portable, or portable. They are exceptionally reliable, and are suitable for any purpose requiring power.

Fig. 22 shows the extreme simplicity of these engines; they are stated to be most economical, the full load fuel consumption being only 0·42 pint per b.h.p. hour in the larger sizes.

Magneto ignition is fitted on sizes 2½ to 9 b.h.p., and the engines can be started up immediately by one turn of the crankshaft on petrol, then changed over to paraffin fuel. The larger sizes work on any residual fuel oils up to 0·95 sp. gr. and are now made in multi-cylinder sizes up to 180 b.h.p.

Petter Oil Engines.

Engines of the two-stroke vertical type have been manufactured by Petters, Ltd., of Yeovil, for many years. The small sizes are the well-known Petter 'M' type engines, suitable for farm, estate, and factory machinery, electric lighting, pumping, and other purposes.

The Petter 'M' type engines, fig. 23, are made in three sizes : 1½, 3, and 5 b.h.p. ; and in three patterns : stationary, for fixing on concrete (as illustrated), semi-portable on wood skids, and portable on four-wheeled carriage.

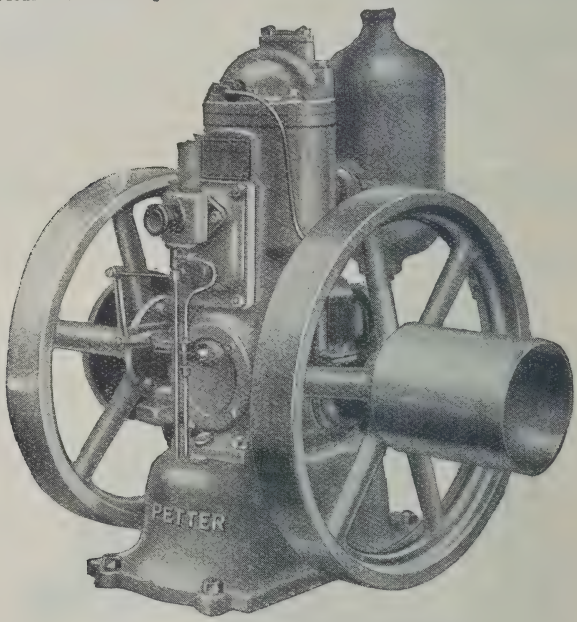


FIG. 23.—The Petter 'M' type Oil Engine.

These engines work on paraffin and have magneto ignition. They are started on a small quantity of petrol, and change over automatically to paraffin after working for a few minutes. By means of the patent fuel injector, the paraffin is vaporised and burnt with the same completeness as in a petrol engine, and at half the cost of fuel. They can also be supplied for running on petrol only.

THE DORMAN ENGINE.

W. H. Dorman & Co., Ltd., Stafford, manufacture two, four, and six-cylinder engines in various sizes developing 8 to 100 b.h.p. at 1,000 r.p.m., as shown in the following table:

No. of Cylinders.	Bore m/m.	Stroke m/m.	Volume c.c.	B.H.P. 1000 r.p.m.	Treasury Rating.
4	66	100	1368	8	10·5
4	69	100	1496	10	11·8
4	80	130	2613	19	15·9
2	120	140	3166	20	17·8
4	95	140	3970	25	22·4
4	100	140	4398	30	24·8
6	95	140	5954	44	33·6
6	100	140	6597	48	37·2
4	120	140	6333	45	35·7
4	120	185	8369	72	35·7
6	120	185	12553	100	53·6

The cylinders are monobloc except on the 100 h.p. six-cylinder engine, where they are cast in two pairs of three each, and on the 45 h.p. engine, where the cylinders are cast in two pairs. The larger sizes are fitted with detachable heads of the Ricardo patent turbulent type. Side-by-side valves have been adopted, owing to their silence in operation, and the inlet valves are 3 per cent. nickel steel, while the exhaust valves are of stainless steel.

These engines are specially designed to meet the demand for increased passenger and weight carrying capacities of vehicles, and the increasing use of internal combustion engines for locomotives and rail cars.

Other Makers of Oil Engines.

Engines of the horizontal four-stroke type are also made by Crossley Bros., Ltd., Manchester; Fielding & Platt, Ltd., Gloucester; and Tangyes, Ltd., Birmingham.

Vertical engines are made by Gardner & Sons, Patricroft; the Parsons Oil Engine Co., Ltd., Southampton; J. I. Thornycroft & Co., Ltd., Basingstoke; the New Pelapone Co., Leeds; and Lister & Co. Ltd., Dursley.

INTERNAL COMBUSTION TURBINES.*

(Contributed by T. B. Morley, D.Sc., M.I.Mech.E.)

The heat engine which has the highest thermal efficiency is an internal combustion engine; the one which can be made in units of maximum power and which, moreover, has many attractive features, such as almost perfect balance due to its purely rotary motion, and small bulk in relation to output, is the steam turbine. It is not to be wondered at, therefore, that engineers have reflected upon the desirability of a prime mover which should combine the merits of both these types. The internal combustion turbine has been the dream of many speculative and inventive minds, but, though many hundreds of patents have been taken out for internal combustion turbines, few have yet been constructed, and none has so far passed the experimental stage.

It is evident, therefore, that there are serious difficulties to be overcome in order to produce a successful internal combustion turbine.

Speaking generally, in all internal combustion turbines, fuel, gaseous or liquid, is burned, and the products of combustion, issuing from the combustion chamber through nozzles at high velocity, impinge upon the blades of a turbine rotor. In comparison with the cycle of a reciprocating engine, the chief difference is that, whereas in that engine the expanding gases do work upon a piston, in the turbine the energy of the gases takes the form of kinetic energy, which is represented in the acquisition of velocity by the jets of gases which are led to the turbine blading.

The combustion, as in the case of the reciprocating engine, may be at constant pressure or at constant volume, or at a combination of the two conditions. If it is at constant pressure, in order to obtain a cycle of reasonable ideal efficiency, the working substance must be compressed before ignition or else the products of combustion must be expanded below atmospheric pressure by an exhauster. In either case, a pump or an exhauster is necessary, and it is the large amount of work necessarily absorbed in this stage of the cycle—relatively much greater than in the corresponding stage of a steam turbine cycle, *viz.* the compression in the air pump and feed pump—which is one of the serious obstacles to the success of this type of internal combustion turbine.

If, as is desirable for a large unit, the compression or exhauster is of the turbo type, its relatively low efficiency compared with a piston compressor greatly accentuates this drawback.

This particular difficulty may be almost entirely avoided by the adoption of a cycle in which the combustion is at constant volume. In this case, however, it is not possible to produce a steady continuous jet; the action of each combustion chamber must necessarily be intermittent, the jets of gas issue at varying velocity, and the blading angles and blade speeds cannot satisfy the conditions for maximum efficiency as they cannot vary in accordance with the varying jet velocities. The loss of efficiency due to this is not, however, of very great magnitude.

The most serious difficulty in the production of a reliable internal combustion turbine is a practical one. In all cases the temperature of the jet of gas striking the blading must be very high in order that the cycle may be efficient; the blading and the rotor must, therefore, be made of materials capable of withstanding temperatures say up to 800° C., at least, without scaling or loss of form, and without undue reduction of strength, as the blades and rotor must run at high speed and are, therefore, subject to considerable centrifugal stresses as well as to the tangential driving effort.

The combustion chamber and nozzles are exposed to still higher temperatures, but as they are stationary and not so severely stressed, nor so delicate in form, the problem of suitable material for them is not so acute.

* See also 'The Internal Combustion Turbine,' by T. B. Morley, *Proc. Manchester Association of Engineers*, February 1925.

In order to reduce the temperature of the jet the gases resulting from the combustion of the fuel may be diluted, with steam for example, before they enter the expansion nozzles leading them to the blading. Interesting types of mixed-fluid internal combustion turbines, *i.e.* turbines in which the working substance is a mixture of gas and steam, are described in Davey's 'The Gas Turbine' (*Constable & Co.*, 1914), but though certain of the turbines proposed, in which the steam is generated partly from the heat of the turbine exhaust and partly from the heat passing through the walls of the combustion chamber, and in which the cycle is carried out at pressures below atmospheric, offer good prospects of success, no actual turbine on these principles has been made.

For a description of experimental machines made before 1914, Davey's book may be referred to.

The type of internal combustion turbine which has most nearly approached practical success is that designed by Herr Holzwarth, and constructed by Messrs. Thyssen, in Germany. This turbine is of the constant volume type. A series of combustion chambers are arranged symmetrically round the shaft, which in the later constructions is horizontal. Gas (or oil) and air at about 20 lbs. per sq. inch abs. pressure are admitted through separate inlet valves; the passage leading from the combustion chamber to the nozzle is closed during the charging operation by a nozzle valve. After the introduction of the charge, all valves are closed, and the charge is ignited. When the pressure has in consequence reached a certain amount it overcomes the spring which holds the nozzle valve shut, the nozzle valve opens and is then held open. The gases pass to the nozzle, expand, and acquire velocity in the nozzle and impinge on the blades. When the pressure in the chamber has fallen sufficiently, scavenging air is passed through it, and then the cycle is repeated. Similar cycles take place in the other combustion chambers. Several Holzwarth turbines have been built and run on gas or on oil fuel, and test results on units in sizes up to 1,000 h.p. have been published: machines up to 5,000 kw. capacity have been built. Unfortunately the earlier published results were incomplete and open to severe criticism. Tests made in 1923 on a 1,700 k.w. unit showed an overall thermal efficiency (based on output at switchboard) of 14·2 per cent. Though Holzwarth turbines have been built from 1908 onwards, and orders were placed in 1919 by the Prussian State Railways, none seems as yet to have been in regular service.

Several other forms of internal combustion turbines are being experimented upon in Germany, U.S.A., and England.

Combinations of reciprocating engine and turbine, the turbine being driven by the exhaust gases from the reciprocating engine, offer good prospects of success. The Rateau exhaust turbine for supercharging aero-engines is one example of this combination.

As regards the internal combustion turbine, considered as a prime mover by itself and not as an adjunct to a reciprocating engine, the prospects of success in the near future are very doubtful, unless the metallurgist can provide a material which will withstand the very high temperatures called for in the blading.

Lloyd's Rules for Marine Internal-Combustion Engines.

The following rules for the survey of internal combustion engines for marine purposes have been drawn up by Lloyd's Register of British and Foreign Shipping :—

General.

Section 1.—In vessels propelled by internal combustion engines the rules as regards machinery will be the same as those relating to steam-engines so far as regards the testing of material used in their construction and the fitting of sea connections, discharge pipes, shafting, stern tubes, and propellers.

Construction.

Section 2.—1. The following points should be observed in connection with the design of the engines :—

2. The shaft bearings, connecting-rod brasses, the valve gear, the inlet and exhaust valves must be easily accessible.

3. The reversing gear and clutch must be strongly constructed, and easily accessible for examination and adjustment.

4. In engines of above 60 b.h.p. which are not reversible, and which are manœuvred by a clutch, a governor or other arrangement must be fitted to prevent racing of the engine when declutched.

5. In vessels intended for open sea service and fitted with engines in which the lubricating oil is circulated under pressure, a spare oil pump is to be supplied with all connections ready for immediate use, and two independent means are to be arranged for circulating water through the oil cooler, where this is fitted.

6. When the engines are of the closed-in type they must be so fitted that the contained lubricating oil can be drained when necessary, and in wood vessels an easily drained metal or metal-lined tray must be fitted to prevent leakage of either fuel oil or of lubricating oil from saturating the woodwork.

7. Carburettors, where petrol is used, and vaporisers, where paraffin is used, should be so designed that when the engine is stopped the fuel supply is automatically shut off. If an overflow is provided in the carburettor or vaporiser, a gauze-covered tray with means of draining it must be fitted to prevent the fuel from flowing into the bilges.

Strong metallic gauze diaphragms should be fitted either between the carburettor (or vaporiser) and cylinders or at the air inlets.

8. If the ignition is electric, either by magneto or by coil and accumulator, all electric leads must be well insulated and suitably protected from mechanical injury. The leads should be kept remote from petrol pipes, and should not be placed where they may be brought into contact with oil.

The commutator must be enclosed, and the sparking coils must not be placed where they can be exposed to explosive vapour.

9. In paraffin and heavy-oil engines, where lamps are used for ignition or for vaporising, these lamps should be fixed by some suitable bracket, and the flame enclosed when in use.

10. The circulating pump sea suction is to have a cock or valve on the vessel's skin placed on the turn of the bilge in an easily accessible position, and the circulating pipe is to be provided with an efficient strainer which can be cleared inside the vessel. The discharge overboard is to be fitted with a cock or valve on the vessel's skin if it is situated under or near the load line of the vessel.

11. The pumping arrangements are to be the same as would be required in the case of a steam vessel of the same size and power, with the exception that no bilge suction need be fitted to the circulating pump. In the cases of vessels fitted with water ballast, the water ballast pump must have, in addition, one direct suction from the engine-room bilges.

In open launches, and in small sailing vessels in which the engines are auxiliary only, a suitable additional hand pump, fitted to draw from the engine-room bilges, may be accepted in lieu of a power-driven pump.

12. Where the cylinder liners are made of hard, close-grained cast iron of plain cylindrical form accurately turned on the outside as well as bored on the inside, so that their soundness can be ascertained by inspection, they need not be hydraulically tested by internal pressure. If, however, they are made of complicated form, the question of testing must be submitted.

13. The water jackets of the cylinders, and the water passages of the cylinder covers and pistons must be tested by hydraulic pressure to 30 lbs. per sq. in. and must be perfectly tight at that pressure.

14. The exhaust pipes and silencer are to be efficiently water-cooled or lagged to prevent damage by heat, and if the exhaust is led overboard near the water-line, means must be arranged to prevent water from being syphoned back to the engine.

15. The machinery must be tried under full working conditions, the report stating the approximate speed of vessel, the number of revolutions of the engines at full power, both ahead and astern, and the lowest number of revolutions of the engines which can be maintained for manœuvring purposes.

Rules for Determining Sizes of Shafts.

Section 3.—1. The diameter of the crank shaft must not be less than that given by the following formula:—

$$\text{Diameter of crank shaft} = \sqrt[3]{D^2 \times (A\,S + B\,L)}$$

where D = diameter of cylinder in inches or millimetres,
S = length of stroke in inches or millimetres,
L = span of bearing adjacent to a crank measured from centre to centre in inches or millimetres.

The values of A S + B L are as given in the following tables:—

TABLE I.—FOR PETROL OR PARAFFIN OIL ENGINES.

4-Cycle Single-Acting Engines.	2-Cycle Single-Acting Engines.	For Smooth Water Service Values of A S + B L	For Open Sea Service Values of A S + B L
1, 2, 3, or 4 Cylinders	1 or 2 Cylinders	0·0308 S + 0·0143 L	0·0342 S + 0·0159 L
6 ,,	3 ,,	0·0310 S + 0·0143 L	0·0344 S + 0·0159 L
8 ,,	4 ,,	0·0351 S + 0·0136 L	0·0389 S + 0·0151 L
12 ,,	6 ,,	0·0474 S + 0·0120 L	0·0526 S + 0·0133 L

The above table applies to engines in which there are bearings adjacent to each side of each crank. If there are two cranks between consecutive bearings the same rules will apply, but L will be the span between the centres of the bearings adjacent to the two cranks.

TABLE II.—FOR HEAVY OIL ENGINES IN WHICH THE MAXIMUM PRESSURE IN THE CYLINDERS IS 200 LBS. PER SQUARE INCH.

4-Cycle Single-Acting Engines.	2-Cycle Single-Acting Engines.	Values of A S + B L
1 up to 8 Cylinders	1 up to 4 Cylinders	0·0298 S + 0·0144 L
12 ,,	6 ,,	0·0376 S + 0·0118 L

TABLE III.—FOR HEAVY OIL ENGINES IN WHICH THE MAXIMUM PRESSURE IN THE CYLINDERS IS 550 LBS. PER SQUARE INCH.

4-Cycle Single-Acting Engines.	2-Cycle Single-Acting Engines.	Values of A S + B L
1, 2, 3, or 4 Cylinders	1 or 2 Cylinders	0·0643 S + 0·0340 L
6 ,,	3 ,,	0·0673 S + 0·0347 L
8 ,,	4 ,,	0·0643 S + 0·0340 L
12 ,,	6 ,,	0·0673 S + 0·0347 L

2. For heavy oil engines in which the maximum pressure in the cylinders is intermediate between 200 and 300 lbs. per sq. in., or between 300 and 450 lbs. per sq. in., or between 450 and 550 lbs. per sq. in., the coefficients A S + B L are to be interpolated between those given in the above tables.

3. For auxiliary engines of the above-mentioned types the diameters of the crank shafts may be 5 per cent. less than given by the foregoing formulæ applying to main engines for open sea service.

4. In solid forged shafts the breadth of the webs should be not less than 1·33, and the thickness not less than 0·56 times the diameter of the shaft as found above, or if these proportions are departed from, the webs must be of equivalent strength.

5. The diameter of the intermediate shaft must not be less than that given by the following formula:—

$$\text{Diameter of intermediate shaft} = C \times \sqrt[3]{D^2 \times S}$$

where D = diameter of cylinder;

S = stroke of piston;

the values of C being those given in the tables, p. 1929.

TABLE VI.—FOR PETROL OR PARAFFIN OIL ENGINES.

FOR SMOOTH WATER SERVICE.

4-Cycle Single-Acting Engines.	2-Cycle Single-Acting Engines.	Values of the Coefficient C where A = 0·0025.	A = 0·0050.	A = 0·0100.
2 Cylinders	1 Cylinder	0·240	0·250	0·256
3 ,,	—	0·255	0·264	0·270
4 ,,	2 ,,	0·272	0·277	0·282
6 ,,	3 ,,	0·268	0·301	0·304
8 ,,	4 ,,	0·322	0·324	0·325
12 ,,	6 ,,	0·363	0·364	0·365

FOR OPEN SEA SERVICE.

4-Cycle Single-Acting Engines.	2-Cycle Single-Acting Engines.	Values of the Coefficient C where A = 0·0025	A = 0·0050	A = 0·0100
2 Cylinders	1 Cylinder	0·252	0·262	0·269
3 ,,	—	0·269	0·276	0·283
4 ,,	2 ,,	0·286	0·291	0·296
6 ,,	3 ,,	0·313	0·316	0·319
8 ,,	4 ,,	0·338	0·340	0·341
12 ,,	6 ,,	0·382	0·382	0·383

TABLE VII.—FOR HEAVY OIL ENGINES IN WHICH THE MAXIMUM PRESSURE IS 200 LBS. PER SQUARE INCH.

4-Cycle Single-Acting Engines.	2-Cycle Single-Acting Engines.	Values of the Coefficient C where A = 0·0025.	A = 0·0050.	A = 0·0100.
2 Cylinders	1 Cylinder	0·210	0·220	0·240
3 ,,	—	0·225	0·234	0·252
4 ,,	2 ,,	0·239	0·248	0·263
6 ,,	3 ,,	0·268	0·272	0·284
8 ,,	4 ,,	0·289	0·290	0·298
12 ,,	6 ,,	0·334	0·335	0·338

TABLE X.—FOR HEAVY OIL ENGINES IN WHICH THE MAXIMUM PRESSURE IS 550 LBS. PER SQUARE INCH.

4-Cycle Single-Acting Engines.	2-Cycle Single-Acting Engines.	Values of the Coefficient C where A = 0·0025	A = 0·0050	A = 0·0100
2 Cylinders	1 Cylinder	0·230	0·255	0·305
3 ,,	—	0·245	0·268	0·309
4 ,,	2 ,,	0·260	0·280	0·312
6 ,,	3 ,,	0·283	0·296	0·318
8 ,,	4 ,,	0·310	0·316	0·323
12 ,,	6 ,,	0·360	0·364	0·368

In using the foregoing tables the appropriate value of A is found from

$$A \times W \times d^2 \times R^2 = D^3 \times S$$

where D = diameter of cylinder in inches
S = stroke of piston in inches;
d = diameter of flywheel in feet;
R = revolutions of engines per minute;
W = total weight of flywheel in tons.

6. If a separate flywheel shaft is fitted its diameter must be at least equal to that of the crank shaft.

7. Where ordinary deep collars are used the diameter of the thrust shaft measured under the collars must be at least $\frac{21}{20}$ths of that of the intermediate shaft. The diameter may be tapered off at each end to the same size as that of the intermediate shaft.

8. The diameter of the screw shaft must not be less than the diameter of the intermediate shaft (found as above) multiplied by $\left(0{\cdot}63 + \frac{0{\cdot}03\ P}{T}\right)$, but in no case must it be less than $1{\cdot}07\ T$

where

P = the diameter of the propeller in inches or millimetres,
T = the diameter of the intermediate shaft in inches or millimetres.

This size of screw shaft is intended to apply to shafts fitted with continuous liners the whole length of the stern tube, as provided for in Section 11, Clause 3 (Rules for Engines and Boilers of Steam Vessels). If no liners are used, or if two separate liners are used, the diameter of the shaft should be $\frac{21}{20}$ths that given above.

The diameter of the screw shaft is to be tapered off at the forward end to the size of the thrust shaft.

Fuel Tanks and Connections.

Section 4.—1. All fuel tanks and their fittings are to be in accordance with Section 49 (rules for steel ships).

2. Strong and readily removable metallic gauze diaphragms should be fitted at all openings on petrol tanks.

3. No glass gauges are to be fitted to fuel tanks containing either petrol or paraffin oil.

4. Filling pipes are to be carried through the deck so that the gas displaced from the tanks has free escape to the atmosphere.

5. Separate fuel tanks should be provided with metal-lined trays to prevent any possible leakage from them flowing into the bilges or saturating woodwork. Arrangements are to be provided for emptying the tanks and draining the trays beneath them. For petrol tanks the trays must have drains leading overboard where possible, or they should be gauze-covered trays with means for draining them.

6. All petrol or paraffin oil pipes are to be of annealed seamless copper with flexible bends. Their joints are to be conical, metal to metal. A cock or valve is to be fitted at each end of the pipe conveying the fuel from the tank to the carburettor or vaporiser. The fuel pipes should be led in positions where they are protected from mechanical injury and can be exposed to view throughout their whole length.

7. The engine-room and the compartment in which the fuel tanks are situated are to be efficiently ventilated.

8. An approved fire-extinguishing apparatus must be supplied.

Periodical Surveys.

Section 5.—1. The machinery is to be submitted to survey annually. At these surveys the cylinders, pistons, connecting-rods, crank and other shafts, inlet and exhaust valves and gear, clutches, reversing gear, propeller, sea connections, and pumps are to be examined. The electric ignition is to be examined and the electric leads tested. The fuel tanks and all connections are to be examined and, if deemed necessary by the surveyor, to be tested to the same pressure as required when new. If practicable, the engines should be tested under working conditions.

2. The screw shaft is to be examined annually and drawn at intervals of not more than three years in the cases of shafts fitted with continuous liners, and of shafts fitted with approved glands, or other approved appliances, at their after ends to permit of them being efficiently lubricated. In the cases of other shafts, they should be drawn at intervals of not more than two years. (On the application of owners the Committee will be prepared to give consideration to the circumstances of any special case.)

High Compression Oil Engines.

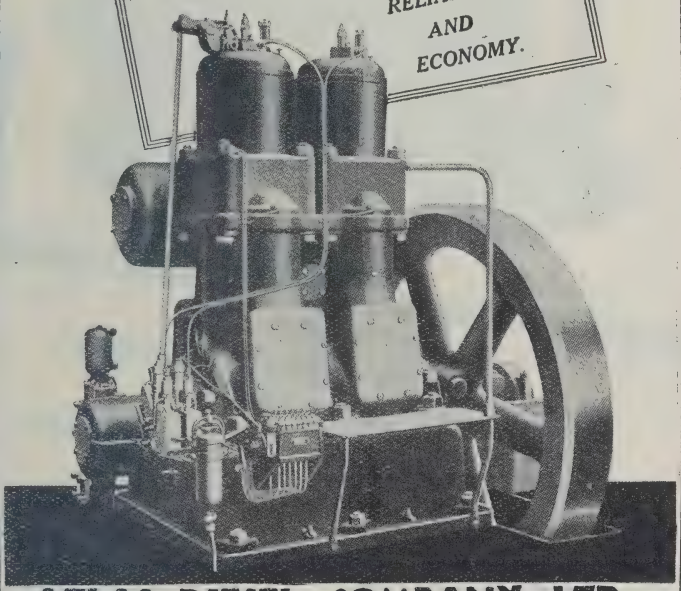
Rules for the Construction and Survey of Diesel engines and their auxiliaries have been adopted by the General Committee of Lloyd's Register, and can be obtained on application to Registrary, 71 Fenchurch Street, London, E.C. 3.

T.L. C.761

Polar

2-Cycle Airless Injection DIESEL ENGINES

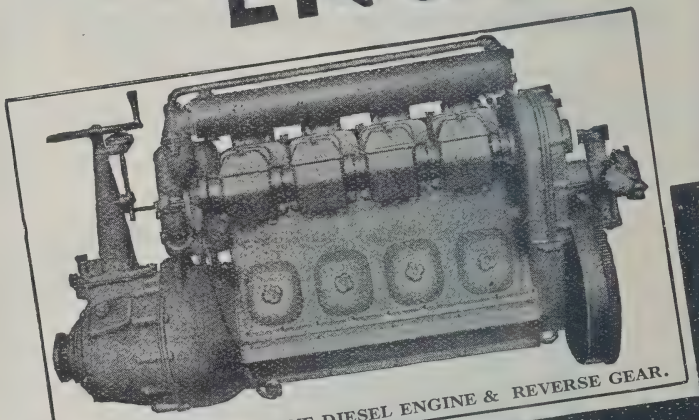
SIMPLE IN DESIGN,
SIMPLE TO OPERATE,
FUEL CONSUMPTION = .375—.365 LBS.
PER B.H.P. HOUR.
UNEQUALLED FOR
RELIABILITY
AND
ECONOMY.



ATLAS DIESEL COMPANY LTD.,

NEW OXFORD HOUSE, HART STREET, LONDON, W.C.1.

GLENIFFER ENGINES



60/80 H.P. 4 cyl. MARINE DIESEL ENGINE & REVERSE GEAR.

For Stationary and Traction Duties and Marine Propulsion

HIGH SPEED DIESEL ENGINES
45 TO 120 H.P.

PETROL - PARAFFIN ENGINES
7 TO 105 H.P.

Full particulars from the manufacturers

GLENIFFER ENGINES LIMITED,
TEMPLE WORKS, ANNIESLAND, GLASGOW, W.3.

Telephone: Western 5500 (2 lines). Telegrams: "Glengine, Glasgow."

Hot-Air Engines.

Hot-Air engines are now but very rarely used; in the best construction a consumption of 1·8 lb. of gas coke has yielded 1 indicated horse-power, or 2·54 lbs. of coke per b.h.p., the h.p. being calculated after deducting the work required to compress the air. The absolute efficiency is therefore only about 7·9 per cent. referred to b.h.p. In the smaller sizes of hot-air engine the consumption of fuel is usually very much higher than this, and is often 20 lbs. or more per b.h.p. hour.

It is practically impossible to construct an air-engine of any great power, as air, like all other gases, expands but one volume for each 493° of temperature through which it is raised; and, in order to double its volume, we must raise it 493° more, that is, to 986° F., which is much too high for practical purposes. Even if air could be worked at this temperature, it would be necessary to have a feed-pump of nearly the capacity of the engine, and the engine itself would have to be more than twice the size of a steam-engine of the same power.

The Gleniffer Cold Starting Oil Engine.

(*Gleniffer Engines Ltd., Anniesland, Glasgow.*)

The Gleniffer heavy oil engine is of the relatively fast-running type, and is made in units having three, four, and six cylinders. As will be seen from the cross-section (fig. 24), the valves

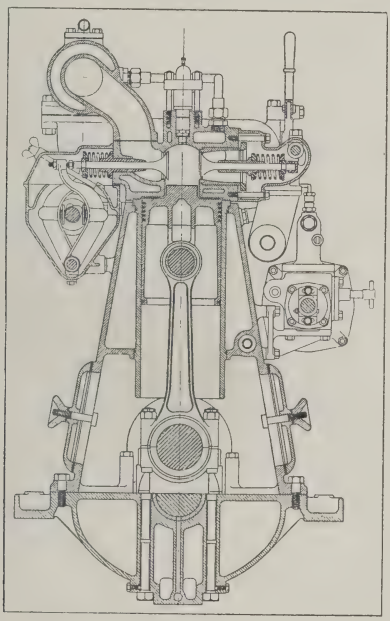


FIG. 24.

are placed horizontally facing each other in the cylinder head, and are operated by the camshaft through levers and pull rods for the admission valves. The engine is of the solid injection type, having the sprayer central over the top of the combustion chamber proper. A projection on the top of the piston increases the turbulence at the top of the stroke by entering a restricted passage in the head. The makers guarantee a fuel consumption of Diesel oil not exceeding 0·44 lb. per b.h.p. per hour. Starting is effected by a compressed air motor, which is mounted over the flywheel, and engages, through a self-disengaging pinion, a gear ring on the flywheel. As soon as the engine starts the air motor is stopped. Compressed air is obtained by employing one of the main engine cylinders as a compressor whilst its fuel pump is cut out of action.

Though solidly constructed the weight is moderate for the power developed, being, in the case of the 4-cylinder 80-h.p. unit, 27 cwts. or 38 lbs. per h.p., and in the case of the same engine arranged for marine duty with reducing gear air receivers, &c., 34½ cwts. or 48 lbs. per h.p.

SECTION XXXII

AUTOMOBILES

(Contributed by Ernest W. Birch, A.M.I.A.E.)

SECTION XXXII

AUTOMOBILES.

(Contributed by Ernest W. Birch, A.M.I.A.E.)

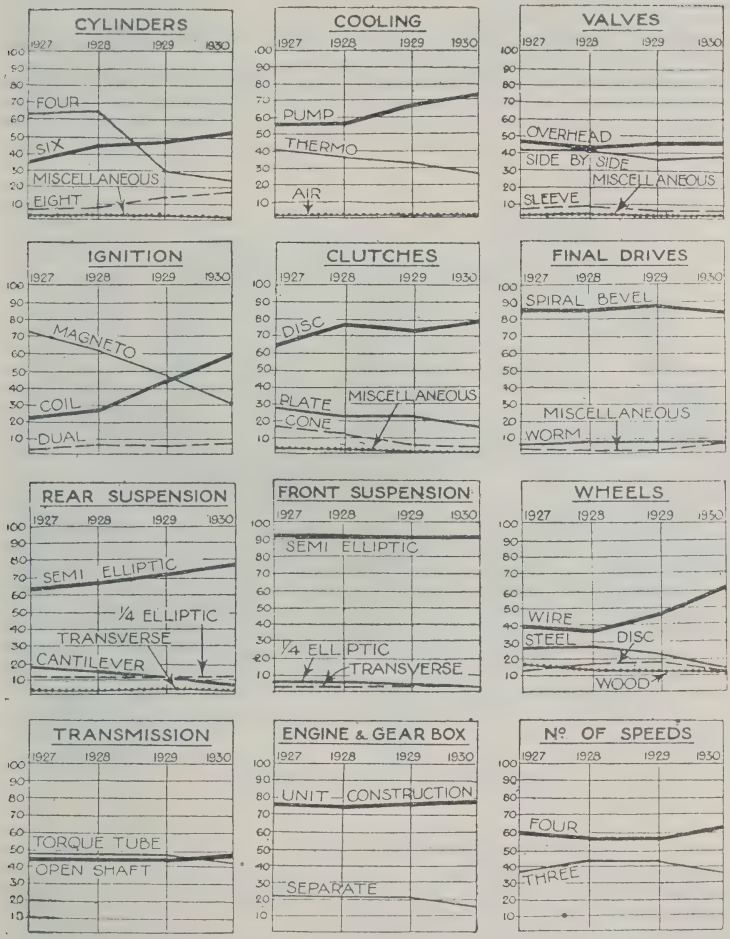


FIG. 1.—Progress in Design, 1927–1930. (*The Autocar.*)

AUTOMOBILES.

Progress in Design.—The four-stroke single-acting, inverted type of engine is in general use. Latest developments show a large increase in the number of six-cylinder engines.

Owing to the horse-power tax it is necessary to make engines of small size, but in America where there are no such restrictions, the eight-cylinder engine is being developed. The progress in design since 1927 is illustrated by the graphs shown in fig. 1. Overhead valves are still the most popular, and bi-metal pistons are coming into use, aluminium alloy being used for the head and gudgeon-pin bosses, and steel or cast-iron sleeves for the skirt. Coil ignition has made great advances, and this is to a great measure due to the increased use of the six-cylinder engine. Further improvements are in the increased size of oil sumps, supply of extra oil to the cylinder walls, and provision for oil cooling.

The single plate (disc) clutch is practically universal, and four-speed gear boxes are increasing in use, and the central gear change is general. Half-elliptic suspension springs are universal in front and are used in the majority of cases for the rear.

Front-wheel drive is receiving considerable attention, and is now standard on several makes.

Other important developments are supercharging, evaporative cooling, thermostatic control of the cooling system, air cleaners, etc., and the improved technique in the manufacture and treatment of the special steels and alloys necessary to meet the exacting requirements of the modern automobile.

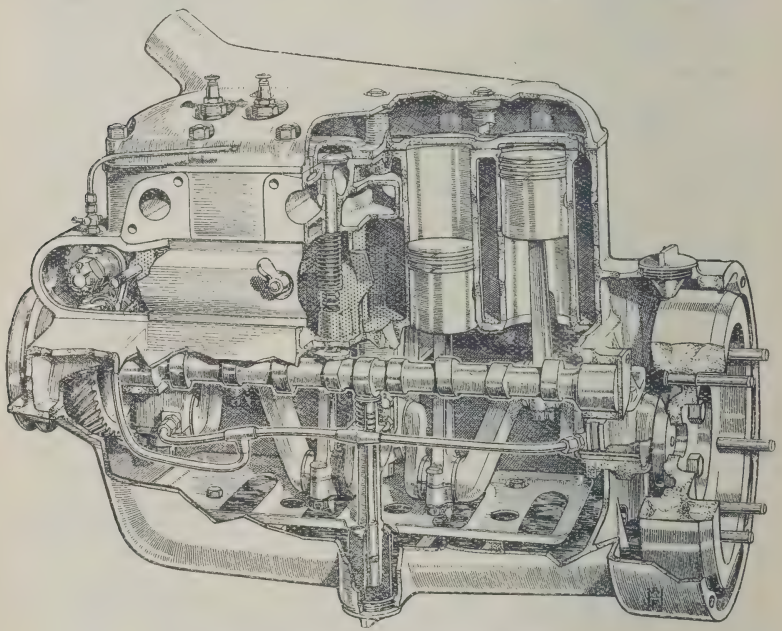


FIG. 2.—Sectional View of the Morris Engine.

THE MORRIS ENGINE.

(Oxford and Cowley Models).

This make represents one of the best examples of 'continuous flow' production in this country. Details of 1929 models are as follows (fig. 2):

Four cylinders cast *en bloc* with the upper half of the crankcase, and fitted with detachable cylinder head. The crankshaft has three white-metal bearings carried upon special bearing blocks attached to the cylinder block.

'*Cowley*' connecting rods 'I' section steel with split small end in which gudgeon pin is clamped by set screw, the gudgeon pin taking its bearing in the piston bosses. '*Oxford*' connecting rods of Duralumin, gudgeon pin fitting the same as for '*Cowley*' model. Piston in both models of aluminium alloy with three rings, the upper one being fitted with special spring centring-ring behind it to reduce liability of piston slap. Side by side valves are used. Camshaft and magneto driven by ekew-gears at forward end of crankshaft. Thermo-syphon cooling assisted by belt-driven fan. Lubrication is by plunger type oil pump driven from an eccentric in the centre of camshaft.

The '*Cowley*' engine develops 22·3 h.p. at 2,800 r.p.m. and the '*Oxford*' 30 h.p. at 2,250 r.p.m.

The six-cylinder model is rated at 17·7 h.p. with a bore of 69 mm. and stroke of 110 mm.

The engine is fitted with an overhead camshaft which operates the overhead valves. The crankshaft is carried in four main bearings, and special attention has been directed in the design of the engine to accessibility.

The 'Morris Minor' has a bore 57 mm., stroke 83 mm., rated at 7·9 h.p. Overhead camshaft and valves. Cylinders cast *en bloc* with detachable head.

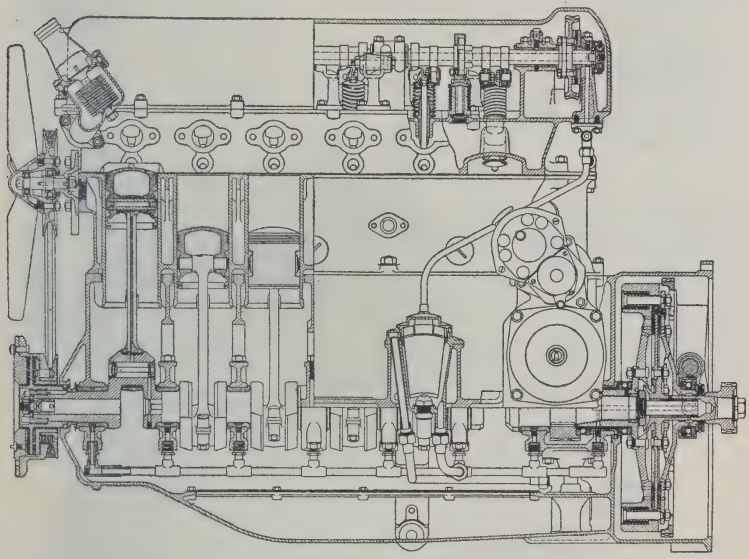


FIG. 3.—Lanchester Engine.

THE LANCHESTER ENGINE.]

Six cylinders 3·1 ins. bore by 4·5 ins. stroke. Treasury rating 23·1 h.p. Swept vloume 203·8 cub. ins. (3,327 c.c.). Power developed at 2,800 r.p.m. is 57 h.p., the brake m.e.p. being 79 lbs. sq. in.

Cylinders cast *en bloc* with detachable head carrying the valves. *Domed* aluminium piston fitted with three light pressure rings and a scraper ring. Hollow floating gudgeon pins with end buttons to prevent scoring of cylinder.

Connecting rods 'I' section with bronze small end bushes and white metal liners for big ends. Crank pin bearings $1\frac{7}{8}$ in. diam. by $1\frac{5}{16}$ in. long. Overhead hollow central camshaft with inclined valves. Camshaft carried on five plain bearings. Damping device provided for damping torsional irregularities. Hollow crankshaft $1\frac{7}{8}$ in. diam. (except for end bearing, which is $2\frac{1}{4}$ in.) carried on seven main bearings of die-cast white metal. Front, middle, and rear bearings $1\frac{7}{8}$ in. long, intermediate journals, $1\frac{3}{4}$ in. long.

Beyond the crankcase proper there is an extension enclosing a vertical shaft and a cross shaft with the gears for driving the overhead camshaft and auxiliaries. An eighth bearing $2\frac{1}{4}$ ins. diam. by $1\frac{5}{8}$ in. long, with white-metal-lined bronze shell, takes the weight of the flywheel.

Daimler Double Sleeve Valve Engine.

This engine operates on the four-stroke, single-acting cycle, but the poppet valves are replaced by cylindrical sliding valves. The piston is enveloped by two open-ended shells which are in turn

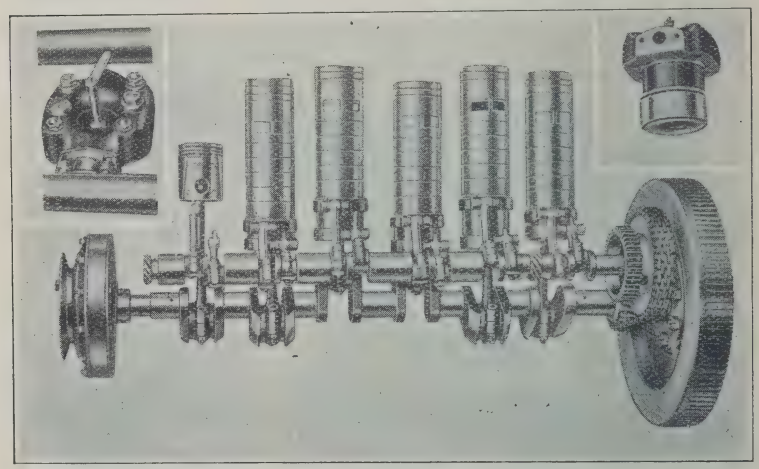


FIG. 4.—Daimler Engine. Sleeve-valve assembly.

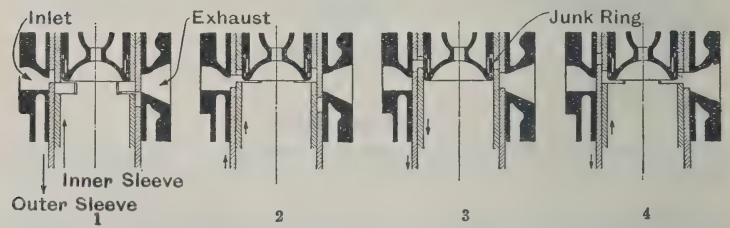


FIG. 5.—Daimler Engine. Valve cycle.

contained within the cylinder casting. These shells are reciprocated in a vertical direction by short links or connecting rods mounted on an independent shaft (fig. 4).

Cycle of Operations.

Diagram 1 (fig. 5) shows the inlet port opening and the exhaust port closing at the beginning of the suction stroke. The inner sleeve is rising and the outer sleeve is descending.

Diagram 2 shows the exhaust closed and the inlet closing at the beginning of the compression stroke. The ports in the inner sleeve are passing up into the cylinder head, where they are sealed by the junk ring during the period of maximum pressure and temperature at the beginning of the firing stroke (*see diagram* 3).

Diagram 4 shows the exhaust already well opened at the beginning of the exhaust stroke. The sleeves are now again moving to the position shown in *diagram* 1.

The complete cycle of operations—suction, compression, firing, and exhaust—occupies two revolutions of the crankshaft.

The latest development is the use of steel sleeves made of seamless drawn tubing in place of cast iron, which, owing to their lightness and great port area, enable the engine to sustain high power with perfect balance at high speeds.

Franklin 26 H.P. Air-cooled Engine.

(American.)

Six cylinders $3\frac{1}{2}$-in. bore by $4\frac{3}{4}$-in. stroke. A special feature of this engine is the system of air cooling. A Sirocco fan directly connected to the front end of the crankshaft supplies air through

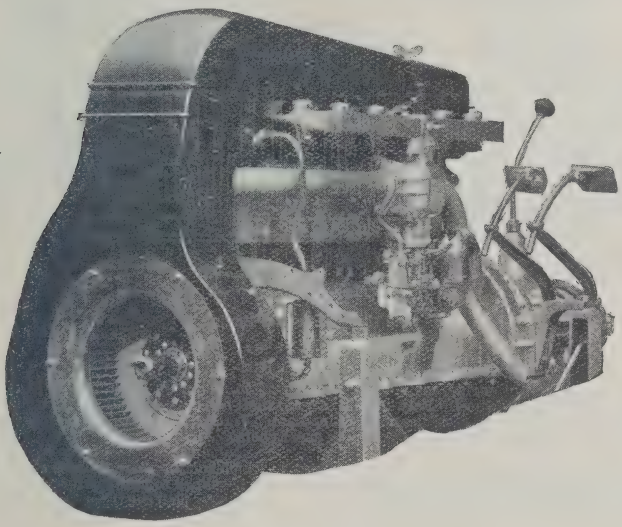


Fig. 6.—Franklin Air-cooled Engine.

a duct over the top of the cylinders, thence through self-enclosing jackets surrounding the cylinders. Deflectors are used to direct the air current in order to secure uniform cooling. At a car speed of 20 m.p.h. the fan absorbs 0·32 h.p. and delivers 1,350 cu. ft. of air.

Crankshaft case-hardened at the seven main bearings. Camshaft has four plain bearings, with cams forged integral with shaft. Timing gear, oil pumps, and ignition driven off camshaft by silent chain. Connecting rods 'I' section of Duralumin, drilled for lubrication of cylinder walls and piston. Aluminium alloy piston with four rings. Gudgeon pin clamped in small end of rod and chromium plated. Cylinders, unit castings of nickel cast iron with copper-cooling flanges cast into cylinder walls.

Packard Engine.

(American.)

Eight cylinders cast *en bloc* with detachable head, 3½-in. bore by 5-in. stroke, swept volume 385 cu. ins. or 6,300 c.c. 106 b.h.p. at 3,200 r.p.m. (*normal speed*), brake m.e.p. 62 lbs. sq. in. Compression ratio, 4·5 : 1.

Side valves of silicon-chrome steel. Aluminium alloy pistons with four rings, weight 24 ozs. Gudgeon pins, hand push fit-in piston boss. Connecting rods of 'I' section, bronze bushed at small end, white-metalled at big end (2 3/16 in. diam. by 1⅜ in. long). Firing order: 1, 3, 2, 5, 8, 6, 7,

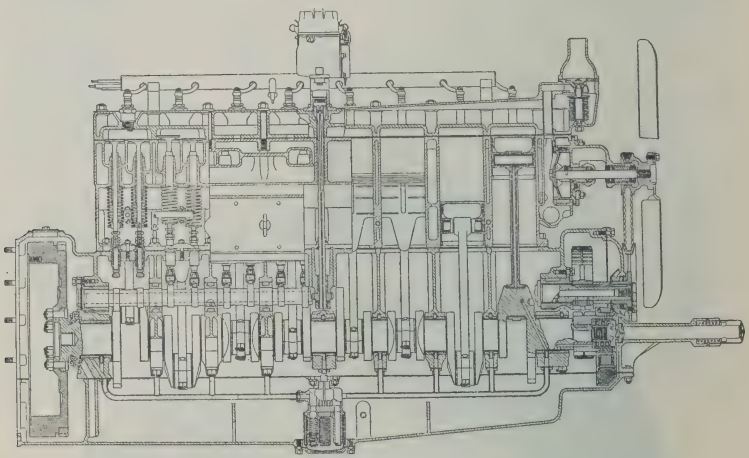


Fig. 7.—Packard Engine.

4, from forward end of engine, main crank bearings 2⅜ in. diam. by 1⅜ in. long. Rear bearing 2½ in. long to support flywheel. Forward end of crankshaft fitted with torsional vibration damper. Hollow camshaft with eight plain bearings, 1⅜ in. diam. Vertical oil pump driven off camshaft by spiral gears. Camshaft and dynamo driven by silent chain.

Water-pump mounted on end of cylinder block in which suction passages are cored. Pump impeller fixed by screw to spindle, the other end of which is splined to take fan boss.

Thermostatic control is fitted to cooling system. Ignition by coil, the timer being driven by a vertical extension of oil-pump shaft.

Analysis of English, American, and Continental Passenger Automobile Practice, 1929–1930.

The figures are the result of an analysis of 100 English cars, 100 American, and 260 Continental. (Figures are in percentages unless otherwise mentioned.)

Main Features of Representative British and Foreign Cars.

Abbreviations used: O.H. = overhead. S.S. = side by side. S.B. = spiral bevel. HYP. = Hypoid. W. = worm. S.P. = single plate. M.P. = multi-plate. C. = cone. I.C. = inverted cone.

Make.	Makers Rating.	Treasury Rating.	No. of Cylinders.	Bore and Stroke, mm.	Cubic Capacity, c.c.	Position of Valves.	Type of Ignition.	Clutch.	No. of Speeds (Forward).	Final Drive.	Wheel Base. Ft.	Wheel Base. Ins.	Track. Ft.	Track. Ins.	Weight of Complete Chassis. Cwt.
British															
Alvis	12/50	11·8	4	68 × 102	1,482	O.H.	Mag.	S.P.	4	S.B.	10	0	4	6	14
Armstrong-Siddeley	30	29·5	6	88·9 × 133·4	4,964	O.H.	Mag.	S.P.	3	S.B.	11	3	4	8	—
Austin	20	23·4	6	80 × 114	3,410	S.S.	Mag.	S.P.	4	S.B.	10	10	4	8	24
Austin	7	7·8	4	56 × 76	748	S.S.	Coil	S.P.	3	S.B.	6	3	3	4	6
Bentley	37·2	38·0	6	100 × 140	6,597	O.H.	Dual	S.P.	4	S.B.	12	0	4	8	36
Crossley	15·7	15·8	6	65 × 100	1,991	O.H.	Mag.	S.P.	4	S.B.	10	3	4	8	18
Daimler	50	49·4	12	81·5 × 114	7,136	Sleeve.	Coil	S.P.	4	W.	13	7	5	0	43
Ford	14·9	14·9	4	78 × 108	2,033	S.S.	Coil	M.D.	3	S.B.	8	7½	4	8	14
Humber	16/50	15·9	6	65 × 106	2,110	*	Coil	M.P.	4	S.B.	10	0	4	8	23
Lanchester	40	38·4	6	101·6 × 127	6,180	O.H.	Dual	S.P.	3	W.	12	6	4	10	37
Morris-Cowley	11·9	11·9	4	69·5 × 102	1,550	S.S.	Mag.	M.P.	3	S.B.	8	9	4	0	14
Morris-Isis	18	17·9	6	69 × 110	2,468	O.H.	Coil	S.P.	3	S.B.	9	6	4	8	20
Morris-Oxford	15	14·9	6	63·5 × 102	1,938	S.S.	Coil	S.P.	3	S.B.	9	6	4	8	17
Rolls-Royce	40/50	43·3	6	108 × 140	7,679	O.H.	Dual	S.P.	4	HYP.	12	6	4	10	38½
Rolls-Royce	20	21·6	6	76·2 × 114·3	3,130	O.H.	Mag.	S.P.	4	S.B.	10	9	4	8	24
Standard	15	14·9	6	63·5 × 101·6	2,062	S.S.	Coil	S.P.	4	S.B.	9	10	4	8	17
Vauxhall	20/60	20·7	6	75 × 110	2,916	O.H.	Coil	S.P.	4	S.B.	10	3	4	9	26
Wolseley	32/80	32·0	8	80 × 100	4,022	O.H.	Coil	S.P.	4	S.B.	11	9	4	8	27

* *Side exhaust and overhead inlet.*

AUTOMOBILE PRACTICE, 1929–1930.—*cont.*

Make.	Makers Rating	Treasury Rating.	No of Cylinders.	Bore and Stoke, mm.	Cubic Capacity, c.c.	Position of Valves.	Type of Ignition.	Clutch.	No. of Speeds (Forward).	Final Drive.	Wheel Base. Ft. Ins.	Track. Ft. Ins.	Weight of Complete Chassis. Cwt. Approx.
FRENCH													
Ballot	13/45	13·0	4	70 × 130	1,995	O.H.	Mag.	S.P.	4	S.B.	10 2	4 5	29
Berliet	23/70	22·4	4	95 × 140	3,969	O.H.	Mag.	M.P.	4	S.B.	12 3	4 9	30
Citroën	13/30	12·9	4	72 × 100	1,629	S.S.	Coil	S.P.	3	S.B.	9 4	4 4	14
Darracq	16	16·9	6	67 × 95	2,000	O.H.	Coil	S.P.	4	S.B.	10 7	4 7	18
Delage	30	30	8	77 × 109	4,050	O.H.	Coil	S.P.	4	S.B.	11 11	4 8	25
Hispano-Suiza	37·2	38·0	6	100 × 140	6,597	O.H.	Coil	S.P.	3	S.B.	12 0	4 9	29
Voisin	20/400	31·0	12	64 × 100	3,860	Sleeve	Coil	M.P.	4	S.B.	11 9	4 8	—
GERMAN													
Benz	16/40	15·9	4	80 × 130	2,620	S.S.	Batt.	C.	4	S.B.	10 5	4 8	—
Mercedes	38/250	38	6	100 × 150	7,069	O.H.	Dual	M.P.	4	S.B.	13 4	4 10	30
ITALIAN													
Alfa-Romeo	17/75	16	6	65 × 88	1,752	O.H.	Coil	M.P.	4	S.B.	10 2	4 7½	—
Bianchi	15/50	15·0	4	78 × 120	2,294	O.H.	Mag.	S.P.	4	S.B.	10 4	4 6	21
Fiat	9	8	4	57 × 97	990	O H.	Mag.	S.P.	3	S.B.	8 5	4 0	10
Fiat	20/70	25	6	82 × 118	3,739	S.S.	Coil	S.P.	4	S.B.	9 10	4 8	20
AMERICAN													
Cadillac	35	37	8	86 × 125	5,786	S.S.	Coil	S.P.	3	S.B.	11 8	4 11	—
Kissel	32·6	33	8	81 × 114·3	4,711	S.S.	Coil and Dis.	S.P.	3	S.B.	11 5	4 8½	40
Marmon	27	28	8	76 × 101	3,580	O.H.	Coil and Batt.	S.P.	3	HYP.	10 0	4 9	—
Moon	27	27	6	86 × 101	3,520	S.S.	Coil and Batt.	S.P.	3	S.B.	10 0	4 8	—
Packard	39·2	40·0	8	88·9 × 127	6,304	S.S.	Batt.	S.P.	3	S.B.	11 11	4 10	39
Studebaker	27·3	28	6	86 × 105	3,624	S.S.	Coil	S.P.	3	S.B.	9 7	4 9	—
Willys-Knight	27·2	28	6	86 × 120	4,170	Sleeve	Coil	S.P.	3	S.B.	10 0	4 8	25

Number of Cylinders	2	4	6	8
English	1	26	55	18
American	—	4	55	41
Continental	1	40	42	17

Position of Valves :	Side by side.	Overhead.	Sleeve.
English	38	55	7
American	78	14	6
Continental	40	53	6

Stroke Bore Ratio (actual).

English—Max. : 1·77 (4·5 ins. stroke by 2·54 ins. bore). Min. : 1·3 (5·5 ins. stroke by 4¼ ins. bore). Average, 1·55.

American—Max. : 1·65 (4⅜ ins. stroke by 2⅞ ins. bore). Min. : 1·05 (4⅜ ins. stroke by 4⅛ ins. bore). Average, 1·41.

Continental—Average, 1·48

Stroke (actual).

English—Max. : 5·5 ins. Min. : 3 ins.
American—Max. : 5·5 ins. Min. : 3⅞ ins.
Continental—Max. : 6·29 ins. Min. : 2·55 ins.

Bore (actual).

English—Max. : 4·25 ins. Min. : 2·2 ins.
American—Max. : 4½ ins. Min. : 2⅜ ins.
Continental—Max. : 4·33 ins. Min. : 2·0 ins.

Type of Clutch :	Multi Disc.	Single Plate.	Cone.
English	18	80	2
American	18	81	1
Continental	29	66	5

Fuel Supply :	Gravity.	Pressure.	Vacuum.
English	39	—	61
American	1	39 *	60
Continental	32	3	65

Final Drive :	Bevel. Etc.	Spiral Bevel.	Worm.
English	7	84	9
American	5	89	6
Continental	5	93	2

Rear Springs :	½ Elliptic.	½ Elliptic.	Cantilever.	Transverse.
English	11	78	7	4
American	1	94	2	1
Continental	8	77	11	4

Cooling System :	Thermo-Syphon.	Pump.	Air-cooled.
English	27	72	1
American	2	96 †	2
Continental	38	59	3

The following data refer to American engines and cars.

Engine :	Valve Location.
'L'	78
'T'	1
'I'	14
'F'	1
Sleeve	6

Camshaft Drive.—Helical gear	15
Chain	84
Spiral bevel	1
Oiling System.—Force feed	59
Splash	1
Pressure	22
Force feed and splash	18

Gear-Box.—Combined with engine (*unit power plant*), 95. Separate, 5.

Rear Axle.—Full floating	6
¾-floating	7
Semi-floating ...	87

Torque taken by springs ...	82
,, ,, torque tube ...	11
,, ,, arm	7

Rear Axle Gear Ratio.—Max., 5·6; min., 3·5; average, 4·71.

* Mechanical pumps.
† 64 per cent. fitted with thermostatic control.

Supercharging.

The object of supercharging is to inject a greater quantity of fuel mixture in the period allotted by the timing diagram, thus maintaining the volumetric efficiency at higher engine speeds and increasing the power output.

To supply the required quantity of mixture at the necessary pressure, a pump is provided. There are four types in use : (*a*) Roots blower, (*b*) piston pump, (*c*) vane pump, (*d*) turbo-compressor.

The various methods of supercharging can be classified as (*a*) forced induction, (*b*) supplemented induction, (*c*) scavenging and use of two-stroke cycle.

In system (*a*) the whole of the charge is supplied to the cylinder at a pressure higher than atmospheric, the normal suction of the engine not being utilised.

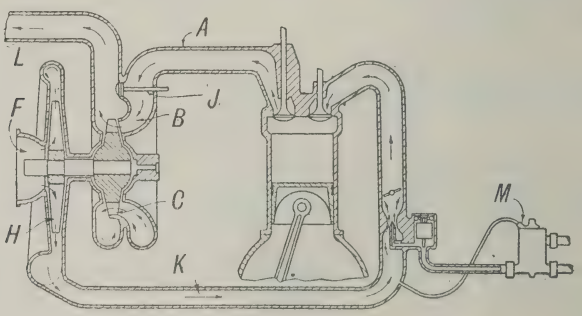


FIG. 8.—Exhaust-turbo Supercharger.

A. Manifold.
B. Nozzle.
C. Turbine wheel.
F. Air intake.
H. Impellor.
K. Air-discharge.

(*b*) The normal suction of the engine is utilised to furnish the greater quantity of the mixture the charge being supplemented by special means.

(*c*) A pump is used that will furnish a volume of air slightly above atmospheric, which is used to free the cylinder and combustion chamber of residual exhaust gases.

Superchargers. Exhaust-driven Turbo-Compressors.—Fig. 8 shows the application of an exhaust-driven supercharger applied to an aero engine. The exhaust gases are collected in manifold A and pass through nozzles B to the turbine wheel C, which is directly coupled to the centrifugal compressor D. In another form the carburettor is connected direct to the suction side of the compressor, and the fuel-air mixture delivered *via* a cooler to the engine.

Positive Displacement Compressors or Eccentric Vane Blowers.—Two general types (*a*) in which the vanes pass through fixed slots in the rotor and the blades are free to fly out under centrifugal action till they make contact with a drum or ring (see fig. 9*a*) ; (*b*) in which the vanes pass through oscillating slots in the rotor and the blades are attached to a lay shaft which coincides with the axis of the casing. The blades do not run out under centrifugal action, but run with a fixed tip clearance inside the casing (see fig. 9*b*).

In the fixed slot type, friction and wear of the blades have been experienced, also the lubrication and friction in the blade guides have proved troublesome.

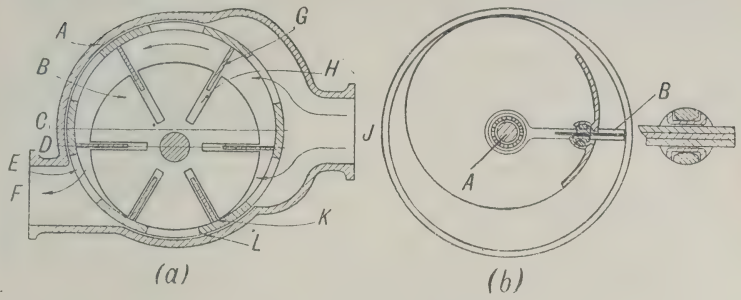


FIG. 9.—Eccentric Vane Blowers.

A. Casing.
B. Rotor.
E. Rolling drum.
F. Outlet.
G. Blade.
J. Inlet.
A. Layshaft central to casing, eccentric to rotor.
B. Blade.

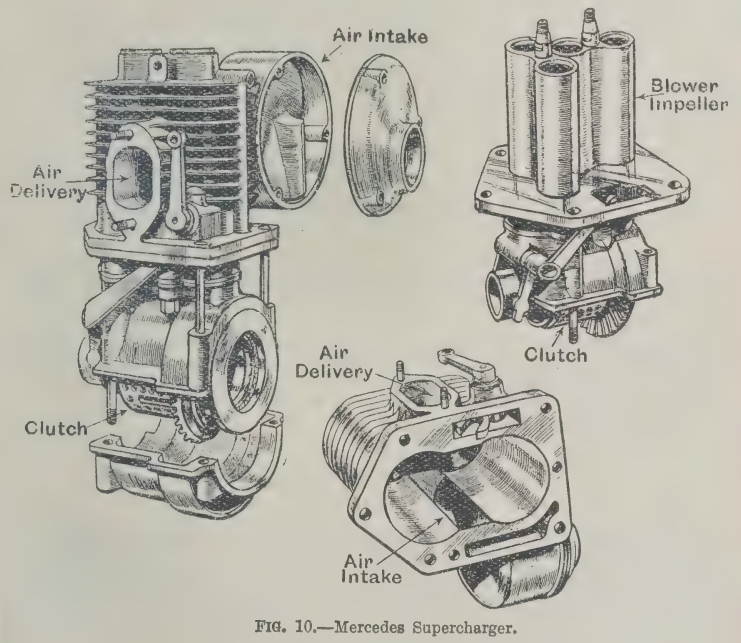


FIG. 10.—Mercedes Supercharger.

Fig. 10 gives details of a Roots' blower type supercharger used on the Mercedes engine. It only becomes operative when the accelerator pedal is depressed past the full open throttle position. This brings into operation a multi-disc clutch that engages a set of bevel gears, and automatically connects the supercharger and closes the normal carburettor intake.

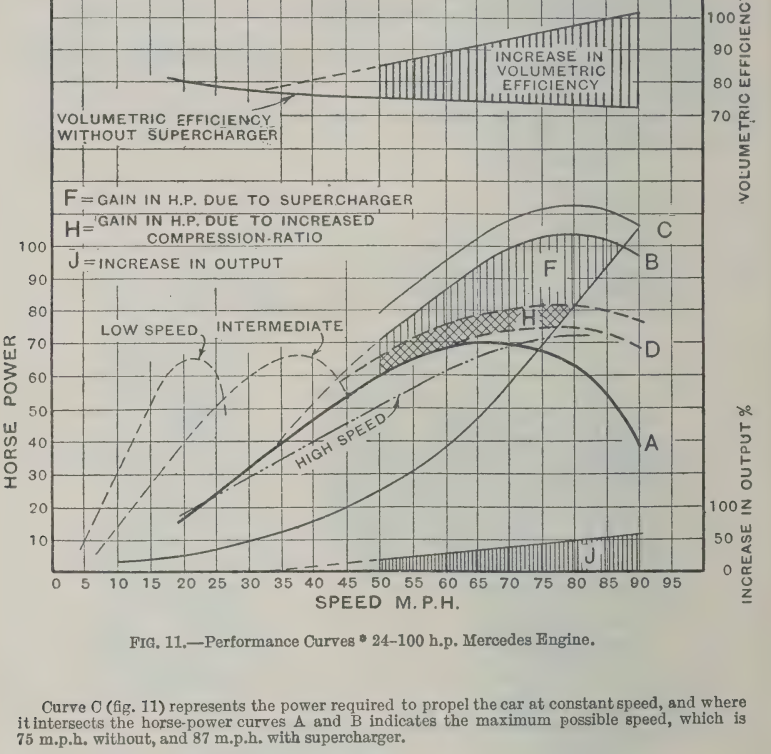


Fig. 11.—Performance Curves * 24–100 h.p. Mercedes Engine.

Curve C (fig. 11) represents the power required to propel the car at constant speed, and where it intersects the horse-power curves A and B indicates the maximum possible speed, which is 75 m.p.h. without, and 87 m.p.h. with supercharger.

Curve A = power output without supercharger.

B = power output with supercharger.

Supercharger Design (Roots' Type).

(*Schwitzer.*)

The mathematically correct cycloidal section is shown at A, fig. 12, in which the diameter of the generating circle is one quarter of the pitch circle. The area included between the housing bore and the sides of the wing section represents the 'bucket' in which the air is carried from the low-pressure to the high-pressure side. The larger this area in relation to the area of the wing section the greater the volumetric efficiency. This area can be increased by using a larger generating circle as at B. The resulting epicycloid will mate with the corresponding hypocycloid in the other

* C. W Iseler, *Proc. S.A.E.*

wing up to the point *a*, which is reached by the generating point when the generating circle has rolled one-eighth of the way round the pitch circle. Beyond this point the curves will not mate at tip and waist, hence the relief shown. The gain in volumetric efficiency over type A is about 7 per cent.

At C the increasing of the size of the bucket is carried to the limit making the diameter of the generating circle one-half that of the pitch circle, the hypocycloid being a straight radial line. These proportions would cut the wing in two if the whole line were used. As at B the curves will not mate beyond *a*, and the gaps can be filled in with circular arcs having their centres on the pitch circle. The gain in efficiency is about 18 per cent. over that at A. The section at D is built up of

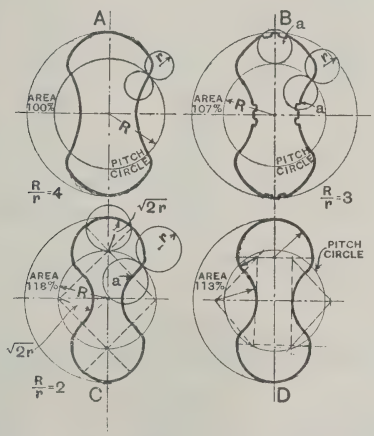


FIG. 12.—Wing Sections for Roots' type Supercharger.

an approximation of circular arcs. It is more efficient by about 13 per cent. than section at A. It is substantially the section used in the latest 'Mercedes' supercharger. This section gives adequate waist strength and the clearance between a pair of wings gradually increases. With this design more backlash can be allowed in the gears than in a mathematically perfect section running with uniform clearance.

The following table shows the effect of the addition of a supercharger to a standard engine:—

R.P.M.	B.H.P.		Power required by Supercharger.	Friction H.P.	
	A.	B.		A.	B.
400	6	7	—	1·0	1·3
800	16	20	0·6	3·0	3·6
1,200	24	32	1·7	4·5	6·2
1,600	31	45	2·7	7·3	10·0
2,000	37	56	4·0	10·0	14·0
2,400	41·5	64	5·8	13·5	19·3
2,800	43·5	68·5	8·3	17·7	26·0
3,000	43	70·0	10·4	20·5	31·0

A. Without Supercharger. B. With Supercharger.

Fuel Distribution to Cylinders.

Tests made on a standard four cylinder 22·5 h.p. car, showed the following values, the latter being determined from gas analysis.

Cylinder No.		CO_2 in Exhaust. Per Cent.	Air Fuel Ratio.	Distribution. Per Cent.	Excess or Deficiency. Per Cent.
A.	1	8·1	10·9	28·5	+ 14·0
	2	8·4	11·1	27·4	+ 9·6
	3	12·0	13·5	21·8	— 12·8
	4	11·6	13·3	22·3	— 10·8
B.	1	9·0	11·5	27·2	+ 8·8
	2	9·5	11·9	26·6	+ 6·4
	3	11·4	13·1	23·2	— 7·2
	4	11·9	13·4	23·0	— 8·0

A = tested at 15 m.p.h. up a 4 per cent. gradient. B = tested at 30 m.p.h. up a 4 per cent. gradient.

The tests show that the two front cylinders (Nos. 1 and 2) are subject to 'loading-up,' particularly at the lower speed.

Air Filters.

Measurements of engine wear of two identical 6-cylinder engines after running under similar conditions:—

		With Air-filter.	Without Air-filter.
Cylinder walls		0·0011 in.	0·0090 in.
Pistons		0·0005 ,,	0·0036 ,,
Ring groove clearance	Upper	0·0110 ,,	0·0650 ,,
	Middle	0·0040 ,,	0·0660 ,,
	Lower	0·0014 ,,	0·0230 ,,
Percentage of incombustible matter in carbon		6·5	31·0

Carburettors.

Sections of typical carburettors are shown in fig. 13, p. 1950.

Location of Carburettor Intake.

As the result of road tests, the best position appears to be (*a*) on the side of the engine on which the fan blades have a descending motion; (*b*) about mid-way between the radiator and the dash; (*c*) about mid-way between the side of the bonnet and the centre line of engine; (*d*) about one-third of the distance down from the top of the bonnet towards the top of the cylinder head.

Engine Starting.

Within certain limits the richness of the fuel determines the number of revolutions the engine must make before an explosion is obtained. Richness of mixture in the cylinder rather than that of the mixture leaving the carburettor determines whether or not an explosion will occur, and it is suggested that this is a function of delayed vaporisation of fuel left in liquid form in the manifold and cylinder during previous cycles. (*T. O. Eisinger.*)

Detonation and Anti-Knock Substances.*

The phenomenon of detonation appears to be substantially as follows:—When a mixture of hydrocarbon vapour and air is compressed to a high pressure and to a temperature approaching that of its self-ignition temperature, and is then ignited from any one point, the flame at first spreads by the normal process of combustion, compressing before it the unburnt portion of the charge. When the rise in temperature of the unburnt portion due to compression by the burning gases exceeds the rate of heat dissipation by a certain margin, spontaneous ignition of the former takes place, and an explosion wave is set up which strikes the cylinder walls with a hammer-like blow, thus causing the metallic ring, known as 'pinking' on detonating.

A number substances will suppress detonation (*anti-knock*), whilst others will induce it.

Element.	Form in which Used.	Effect on Detonation.
Iodine	Element.	Suppresses.
Bromine	Element.	Induces.
Oxygen	Element.	Induces.
Oxygen	Ethyl nitrite or nitrate	Induces.
Nitrogen	Aniline.	Suppresses.
Selenium Tellurium Tin	Di-ethyl compounds	Suppresses.
Lead	Tetra-ethyl compounds.	Suppresses.

'*Anti-knock* substances are efficient when present in minute proportions, and in the case of lead-tetra-ethyl, one molecule in 200,000 molecules of air-fuel mixture is sufficient.

Element.	Compound.	Corresponding Per Cent. by Volume.	Corresponding Number of Molecules of Theoretical Air-Fuel Mixture to one Molecule of Anti-Knock.
—	Benzene	25·0	150
Iodine	Ethyl-iodide	1·6	2,150
Nitrogen	Xylidine	2·0	2,600
Tin	Tin-tetra-ethyl.	1·2	7,100
Selenium	Di-ethyl-selenide.	0·4	11,750
Tellurium	Di-ethyl-telluride	0·1	50,000
Lead	Lead-tetra-ethyl.	0·04	215,000

In using lead-tetra-ethyl it is necessary to add some halogen-containing compound to counteract the harmful action on the sparkling plug electrodes. Carbon tetra-chloride, or ethylene dibromide can be used, the function of these compounds being to convert the lead into lead halides immediately after combustion. Lead-tetra-ethyl is a colourless liquid. S.G. 1·62, boiling point 152° C., miscible with petrol in all proportions and poisonous.

Approximate composition of ethyl fluid: Lead ethyl, 54 per cent.; ethylene dibromide, 36 per cent.; monochlornaphthalene, 9 per cent.†

A usual proportion is 5 c.c. of above mixture to 1 gal. of fuel.

* Report of the Empire Motor Fuel Committee, *Proc. I.A.E.*, vol. xviii.

† Malowar oil.

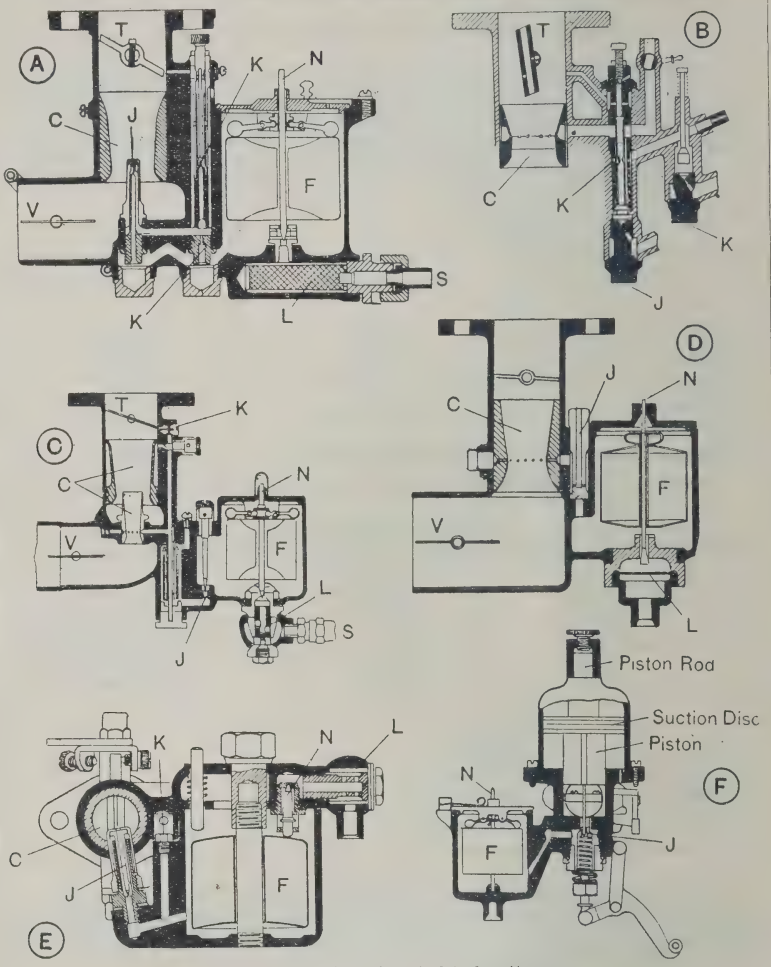


FIG. 13.—Sections of Typical Carburettors.

A. Zenith (Vertical Type).
B. Claudel-Hobson (Power Jet Model). (Float chamber not shown.)
C. Stromberg (Model M).
D. Brown & Barlow. (Auxiliary jets not shown.)
E. Solex (Horizontal Type MH).
F. H.V. Type. Controllable-Jet S.U. Carburettor.

C = Choke Tube. F = Float. J = Main Jet. K = Auxiliary Jet.
L = Filter. N = Needle. S = Fuel supply. T = Throttle Valve.

CHARACTERISTICS OF BRITISH MOTOR FUELS.

(*Ormandy.*)

Fuel.	Highest Useful Compression Ratio.	Specific Gravity. A.P.I. Deg.	Distillation Temperature °F.				
			Initial Boiling Point.	20%	50%	90%	Dry
Petrols No. 1							
Red Line Super . .	5·05	61·7	102	176	234	329	358
Shell	4·76	62·5	100	171	239	351	379
Power . . .	4·73	56·8	127	214	252	325	383
Pratt's Perfection .	4·61	61·2	102	178	234	325	361
Glico	4·54	60·6	109	194	252	336	374
Russian Oil Products (R.O.P.) . . .	4·52	55·0	162	225	259	331	396
British Petroleum (B.P.)	4·46	63·1	117	181	230	315	356
Blue Bird . . .	4·39	61·9	111	192	244	331	378
Dominion . . .	4·32	62·6	108	185	237	329	367
Standard Anti-knock Petrols							
Borneo, old straight run	5·08	48·4	158	217	246	304	347
Borneo, new straight run	5·06	49·0	151	217	252	311	349
Anti-knock Mixtures							
Pratt's Ethyl Petrol .	5·25 to 5·34	62·2	115	183	241	325	367
Red Line Benzol Mixture	5·14	54·7	108	189	239	351	383
Pratt's Benzol Mixture .	4·98	52·8	115	181	219	322	367
National Benzol Mixture	4·88	51·3	127	185	223	338	390
Commercial Petrols							
Red Liné Commercial .	5·03	57·8	95	199	262	363	383
Pratt's Gasoline . .	4·66	61·0	106	187	246	329	372
Mex No. 3 . . .	4·63	59·2	104	189	248	342	388
Blue Bird No. 3 . .	4·59	56·6	138	217	253	327	383
British Petroleum No. 3	4·46	59·6	117	189	241	333	379

Carburation.

Specific Gravity of various fuels at 15° C.:

Bowley's Special	... 0·684	Shell 'Borneo'	... 0·753
Carless	... 0·704	Refined Peruvian	... 0·729
Express	... 0·707	Benzol	... 0·846
Pratt's	... 0·719	Rocklight Paraffin	... 0·825
Carburine ...	... 0·720	White Rose Paraffin	... 0·800
Shell	... 0·721		

Mechanical Losses in Engines.*

The mechanical losses remain practically constant, irrespective of the torque, for though the friction increases slightly with an increase of load, the work done in filling the cylinder increases as the throttle is closed, and the two practically balance. An 80 per cent. mechanical efficiency at full load becomes only 55 per cent. at 30 per cent. of full torque, while a 90 per cent. efficiency becomes 73 per cent., so that the difference that has to be taken into account is not that between 80 per cent. and 90 per cent., but between 55 per cent. and 73 per cent.

Of the gross mechanical losses, piston friction in a well-designed engine accounts for about 60 per cent. of the total. Pumping losses account for about 25 per cent., while the friction of the bearings and all the auxiliary gear amounts to about 15 per cent. Piston friction can best be reduced by keeping both the weight and bearing surface of the piston down to the minimum. Pumping losses are not susceptible of much reduction. A little can be done by the use of large valves and the most suitable timing, but on reduced loads the losses are mainly due to the restriction through partially closed throttle. These losses are therefore more or less inevitable unless means can be found to work with a weakened instead of a throttling mixture. Unless means be adopted to augment turbulence during the compression the valves must be kept fairly

* H. R. Ricardo, *Proc. I.A.E.*

small and the velocity through them high, or slow and incomplete combustion, with loss of power, efficiency, and excessive heat flow will result. The friction of the bearings is relatively small.

It is almost impossible to assess the losses due to general vibration. However, an engine cannot vibrate without absorbing a considerable amount of energy. Every effort should be made to ensure the best possible balance.

Excessive turbulence gives rise to excessive heat losses and a narrowing of the mixture range.

It gives rise also to excessive roughness of running. The optimum degree of turbulence from the point of view of efficiency corresponds to a rate of pressure rise of 30 to 40 lbs. per sq. in. per degree of crankshaft travel. It is the rate of pressure rise, and not the maximum pressure which gives rise to harshness. A high compression engine, so long as it is not detonating, will be as sweet running as a low compression one, provided its rate of pressure rise is the same or less.

Heat Losses in Engines.

Heat Flow through Cylinder Walls.

Average Heat Flow. B.Th.U.'s per sq. in. per minute.

Water-Jacketed Engines.	6-Cylinder Passenger Car.	6-Cylinder Motor Coach.	12-Cylinder Aircraft.
Valve position	In head.	L-head.	In head.
Brake horse-power	51	92	415
Heat loss. B.Th.U.'s per min.	2,500	7,400	10,700
Surface, sq. ins.	540	1,150	2,520
Average heat flow B.Th.U.'s per sq. in. per min.	4·6	6·4	4·3

Heat Losses to the Cylinder Jackets.

Type of Cylinder.	Heat Loss. B.Th.U.'s./min/B.H.P.	Water evaporated. Lbs./min./B.H.P.
Aviation	30	0·03
Valve in head	50	0·05
L-Head	70	0·07
T-Head	80	0·08

To guarantee that the cylinder block will always be full of water, it is necessary to pump more water than can ever be evaporated. In practice it has been found desirable to pump $2\frac{1}{2}$ times the maximum evaporative requirements. The water pipes should be large enough to carry this quantity of water without undue pressure drop, or at a velocity exceeding 6 ft. per sec.

$$\text{Minimum inside diameter, } d'' = \sqrt{\frac{(4 \times 28{\cdot}8 W_p) \div (60 \times 60 \times 12\pi)}{}} = \sqrt{0{\cdot}0085\, W_p}.$$

W_p = weight of water pumped in pounds per min.;

28·8 = volume in cubic inches of 1 lb. of water at 212° F.

For average conditions in a cylinder jacket, the temperature drop through the water film is approximately 20–25° F. The temperature of cylinder walls will increase approximately with an increase in water temperature until it has reached 185–190° F., when no further rise in the cylinder wall temperature will occur so long as the heated surface is wet with water or foam. The effect of a carbon or lime deposit $\frac{1}{32}$ in. thick may cause an increase of inner wall temperature of some 200° F.

Analysis of Heat Losses.

(*H. R. Ricardo.*)

In the case of a well-designed engine, with a compression ratio of 5 : 1, about 32 per cent. of the total heat of the fuel is converted into useful work on the piston, 28 per cent. is carried away by the cooling water, and the remaining 40 per cent. is lost to radiation, exhaust, etc. Of the total heat carried away by the cooling water, approximately 6 per cent. will be lost to cylinder walls during the combustion period, 7 per cent. will be yielded up during expansion, and 15 per cent. during exhaust stroke.

Of the 6 per cent. lost during the combustion period about 40 per cent. appears as useful work, or 2·4 per cent. of the total heat of the fuel. Of the 7 per cent. lost during expansion, about 20 per cent. will be utilised, or 1·4 per cent. of the total heat of the fuel. Of the 15 per cent. lost during exhaust stroke, no part could be utilised.

Although 28 per cent. of the total heat of the fuel is carried away by the cooling water, only 5·8 per cent. could be converted directly into useful work on the piston, and the efficiency of the engine would be increased from 30 per cent. to 35·8 per cent., a gain of barely 12 per cent.

If all the loss of heat to the cylinder walls be suppressed, the temperature of the working fluid would be corespondingly higher, with a result that the losses due to the increase of specific heat and of dissociation at the higher temperature would be substantially increased, and a net gain from 32 per cent. to about 34·5 per cent.

These figures indicate the relatively small part the loss of heat to the cylinder walls plays in an internal combustion engine, and how misleading it may be to assess that loss of heat by the heat carried away by the cooling water. As a first approximation it is fairly correct to assume that, of all the heat carried away by the cooling water of a cylinder, little more than 10 per cent. would actually be converted directly into useful work.

Water Pump.

(Gear Type.)

Capacity of gear pumps can be calculated from the volume of the space between the teeth of the gears. This space can be assumed to be equal to one-half the volume between the outside cylinder and the root cylinder of the gears. As two gears are working, the theoretical delivery per revolution will be equal to the difference in area between the outside and the root circle of one gear times its width. Tests of pumps running at 1500–2000 r.p.m. with boiling water show that the actual delivery is 20–30 per cent. of this amount.

$$W_p = 0{\cdot}25 \times 0{\cdot}0346 \frac{\pi}{4} (D^2 - d^2)FN = 0{\cdot}0068\ (D^2 - d^2)FN$$

or,
$$\frac{147W_p}{N} = (D^2 - d^2)F$$

where,
d = root diam. of gear-ins.; D = outside diam. of gear-ins.; F = width of gear-ins.; N = r.p.m. of pump gears; W_p = weight of water pumped; lbs. per min.; 0·0346 = weight of 1 cu. in. of water at 212° F.

(*A. G. Herreshoff*, '*Proc. S.A.E.*')

Boilerine Radiator Tablets.

These tablets, invented and manufactured only by Boilerine Limited, are for the purpose of obviating the fur which forms in radiators and water-jackets. There is nothing in the preparation detrimental to the joints or metal parts. Where scale is already in existence 25 tablets per gallon of water take the lime into solution, dissolving the particles, thus leaving the walls of the cylinder-jackets and radiator tubes perfectly clean.

Following this cleaning process all water put in the radiator should be ' Boilerined ' to the extent of 2 tablets per gallon, up to 14 degrees of hardness, and one extra tablet for every 7 degrees beyond; one tablet per gallon for rain water to prevent rust.

When water is poured into the radiator, it amounts to putting in a quantity of lime or chalk, or if soft water, a means of producing rust.

If this is done without ' Boilerine ' tablets, the lime adheres to the walls of the cylinders and radiator tubes until efficiency is choked out of the cooling system.

Boilerine Limited have proved from tests that thirty hours heating of water at 190° F. gives nearly the same amount of deposit as half an hour's boiling, thus, surely, though unseen, the motor is, at the workin gtemperatures, storing up in its cooling system this cause of trouble, consequently the flow of water is reduced, producing overheating, or allowing the impurities in the water to cause rust and corrosion, hence leakages.

These claims are confirmed by the reports given on tests of twelve and six months' duration respectively, made by Thomas Tilling, Ltd., and W. H. Dorman & Co., Ltd., Stafford.

Crankcase Corrosion.

The presence of water in the crankcase is responsible for the corrosion. The quantity of water present in the products of combustion is dependent upon the hydrogen content of the fuel, the mixture ratio, and the humidity of the air that enters the engine. The amount of water that may be condensed on the cylinder walls or in the crankcase depends upon the effectiveness of the piston and piston rings in preventing gas leakage, the temperature of the cylinder walls and crankcase, and the extent of the breather action. Condensation of water from the gases of combustion will start at approximately 130° F. when petrol is used, and at 123° F. when a 50 per cent. mixture of petrol and benzol is used. The relative freedom of some engines from water accumulation is due to their higher operating temperatures, or to the better interchange of air by breather action, which results in dilution of the gases in the crankcase and consequent reduction of the saturation temperature of the gases.

Water alone will cause corrosion, but the action may be accelerated by the formation of weak sulphurous or sulphuric acid.

In order to reduce water accumulation, pistons must reduce blow-by, and in addition means should be provided to reduce the warming-up period and ventilate the crankcase. Either air- or steam-cooling, with provision for controlled heating of the crankcase, is favourable for the solution of the problem.

(*M. A. Thorne*, '*Proc. S.A.E.*')

Chromium plating of the crankcase has proved an effective measure against corrosion (see p. 2704).

Engine Efficiency Constants.*

The following equation for efficiency is recommended by Tizard and Pye, and is based upon a 20 per cent. weak mixture, which corresponds very closely to the conditions under which the maximum observed efficiencies are now obtained.

$$E = 1 - \left(\frac{1}{r}\right)^{0 \cdot 296}.$$

r.	E.	*r.*	E.	*r.*	E.	*r.*	E.	*r.*	E.	*r.*	E.
3·8	0·3261	4·4	0·3552	5·0	0·3790	5·6	0·3995	6·2	0·4175	6·8	0·4329
3·9	0·3316	4·5	0·3595	5·1	0·3826	5·7	0·4026	6·3	0·4200	6·9	0·4354
4·0	0·3366	4·6	0·3637	5·2	0·3861	5·8	0·4057	6·4	0·4229	7·0	0·4376
4·1	0·3410	4·7	0·3674	5·3	0·3896	5·9	0·4087	6·5	0·4251	7·5	0·4495
4·2	0·3461	4·8	0·3717	5·4	0·3930	6·0	0·4116	6·6	0·4280	8·0	0·4596
4·3	0·3506	4·9	0·3753	5·5	0·3963	6·1	0·4145	6·7	0·4307	—	—

Firing Order.

From considerations of balance it is necessary to get as uniform a turning-moment as possible. Four-cylinder engines are so arranged that there is a power impulse every stroke or two power impulses every revolution, the one beginning as the other ends. The crank throws are placed all in one plane, the two centre throws being together and at 180° to the two outer throws. With this arrangement two orders of firing are possible, viz. 1–2–4–3 and 1–3–4–2; both are used, but the latter is the more general.

With a six-cylinder engine the throws are divided into three positions or 120° apart. 1 and 6 are always in line, 3 and 4 are always in line, and 2 and 5 are always in line. For a right-hand crank the firing order is 1–5–3–6–2–4 or 1–2–4–6–5–3, and for a left-hand crank 1–4–2–6–3–5 or 1–3–5–6–4–2.

For an eight-cylinder 'V' engine:

$$\left.\begin{matrix}8\text{–}7\text{–}5\text{–}2\\1\text{–}4\text{–}6\text{–}3\end{matrix}\right\} \text{ or } \left\{\begin{matrix}1\text{–}3\text{–}4\text{–}2\\8\text{–}6\text{–}5\text{–}7\end{matrix}\right.$$

For an eight-cylinder 'in-line':

(*a*) 1–5–3–7–4–8–2–6
(*b*) 1–7–3–5–4–2–6–8
(*c*) 1–3–2–5–8–6–7–4
(*d*) 1–5–2–6–4–8–3–7

* I.A.E. Data Sheet No. 205.

Effect of Changes in the Compression Ratio upon Engine Performance.*

With an increase in the compression ratio there is usually an increase in b.h.p. and a decrease in consumption up to a ratio of 14 : 1, provided there is no serious pre-ignition or detonation. The changes in indicated thermal efficiency with changes in compression ratio are in close agreement with those based on the air-cycle efficiencies at the various ratios.

Change in the compression ratio does not produce a consistent change in the friction h.p. volumetric efficiency, or in the range of fuel-air ratios over which the engine can operate. The ratio between the heat loss to the jacket-water and the heat converted into b.h.p. or i.h.p. decreases with an increase in the compression-ratio.

Explosion Pressures in Petrol Engines.

(Research Association of British Motor and Allied Manufacturers.)

The general expression for the explosion pressure is :—

$$P_3 = \frac{0{\cdot}3\,(r-1)}{1-r^{\cdot 3}}\left\{P_i + P_0 \cdot \frac{r}{r-1}\cdot\frac{r^{\gamma'-1}-1}{\gamma'-1}\right\}$$

where

r = compression ratio.
P_3 = explosion pressure, lbs. per sq. in. absolute.
P_0 = pressure at the end of the suction stroke, which is assumed to lie between 12 and 14 lbs. per sq. in. absolute.
P_e = brake mean effective pressure, lbs. per sq. in.
γ = ratio of specific heats; this for the explosion curve is assumed to be 1·3, and for the compression curve, where the ratio is denoted by γ' = 0·64 + 0·05P_0.
P_i = theoretical mean indicated effective pressure, and may be written $P_i = \frac{P_e}{\eta\, K}$ where η is the mechanical efficiency, and K the diagram factor representing the proportion of the actual indicator diagram to the sharp-cornered constant-volume cycle diagram assumed in the calculations.

The above equation may be reduced to the equivalent form :—

$$P_3 = (0{\cdot}72 + 0{\cdot}482r)\left\{\frac{P_e}{\eta\, K} + 5{\cdot}427 + 0{\cdot}780\,P_0 + r\,(0{\cdot}642\,P_0 - 4{\cdot}801)\right\}$$

which agrees with above exact equation to within one part in 400.

Piston Clearance and Compression Ratio.

Analysis of 100 different makes of engines show the average clearance for cast-iron pistons to be 0·013 in. at the top and 0·003 in. at the bottom.

The extreme values were :—Top, max. 0·026 in., min. 0·002 in. Bottom, max. 0·005 in., min. 0·001 in.

Average compression ratio 4·42 : max. value 5·76 ; min. value 3·33. (Compression ratio = Ratio of total cylinder volume to combustion chamber volume.) In general 4·5 may be considered as high compression, 4·0 as medium, and 3·5, or below, as low.

Compression Pressure for Various Percentages of Combustion Space Volume.

Combustion chamber, percentage of piston displacement.	Compression Pressure. lbs. per sq. in. gauge.			Combustion chamber, percentage of piston displacement.	Compression Pressure. lbs. per sq. in. gauge.		
	High value.	Average value.	Low value.		High value.	Average value.	Low value.
25	125	112	97	33	89	80	68
26	118	107	92	34	86	77	66
27	113	102	88	35	83	74	64
28	108	97	84	36	80	72	62
29	103	93	80	37	77	70	60
30	99	89	77	38	74	68	58
31	95	86	74	39	72	66	56
32	92	83	71	40	70	64	55

* Report No. 205, National Advisory Committee for Aeronautics, U.S.A.

$$\text{Compression Ratio, } r = \frac{\text{Combustion space + swept volume}}{\text{combustion space}}.$$

$$\text{Air Standard Efficiency, } E = 1 - \left(\frac{1}{r}\right)^{1\cdot4} - 1.$$

Brake M.E.P.

d = diameter of piston.; s = stroke; n = r.p.m. of crankshaft; P = total m.e.p. on piston, lbs. per sq. in.; p = brake m.e.p., lbs. per sq. in.; c = power strokes per revolution of crank.

If d and s are in inches,

$$\text{B.h.p.} = \frac{s.d.^2p.n.c.}{504300}; \quad \text{P} = \frac{396000 \text{ b.h.p.}}{s \times c \times n}; \quad p = \frac{504300 \text{ b.h.p.}}{s.d.^2n.c.}.$$

If d and s are in millimetres,

$$\text{B.h.p.} = \frac{s.d.^2p.n.c.}{8262 \times 10^6}; \quad \text{P} = \frac{\text{b.p.h.} \times 1006 \times 10^6}{s.n.c.}; \quad p = \frac{\text{b.h.p.} \times 8262 \times 10^6}{s.d.^2n.c.}.$$

Pistons.

The material most generally used at present is Cast Iron, but great advances have been made in the use of non-ferrous alloys, and the following, which is an extract from a paper by L. Aitchison, indicates their advantages.

Light alloy pistons are in general use for aero engines and racing car engines. The main advantages in their use are :—

(*a*) Owing to their light weight, friction is reduced, and also the loading on the connecting-rod bearings, due to the reduction of inertia of the reciprocating parts.

(*b*) Owing to the lower temperature of the crown (due to the greater thermal conductivity) little or no carbonisation of the lubricating oil takes place on the under-side.

(*c*) Owing to the lower mean temperature of the oil film between the piston and cylinder walls, less oil may be used without risk of piston seizure.

A disadvantage in their use is due to larger coefficient of expansion, which necessitates increased clearance and is the cause of 'piston-slap.'

The Ricardo Piston.

This piston, fig. 14, is a 'slipper' type and aims at securing increased power by the reduction of friction and weight. The contamination of the lubricating oil due to gas leaking past the rings, which takes place in the ordinary form of piston, is reduced by allowing the leaking gas to escape before reaching the piston skirt. Also the lower ring groove is bevelled and drilled to enable the ring to act as a scraper.

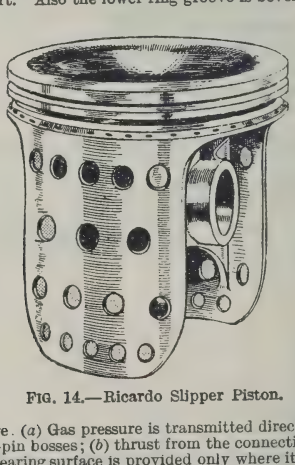


Fig. 14.—Ricardo Slipper Piston.

Features of this piston are. (*a*) Gas pressure is transmitted direct from the crown of the piston to the centre of the gudgeon-pin bosses; (*b*) thrust from the connecting rod is transmitted directly to the bearing surfaces; (*c*) bearing surface is provided only where it is needed; (*d*) gases that pass the piston-rings may escape freely into the crank case; (*e*) no wad is transmitted through the side-

walls, or past the piston-ring grooves; (*f*) the lower piston-ring acts as a scraper, so that oil that has worked up past the slippers can be returned through the small holes beneath. Tests made with this piston showed a reduction in oil consumption from 0·06 lbs. per h.p. hour, to 0·035 lbs. At the same time fuel consumption dropped 4 per cent. with about 4 per cent. increase in b.h.p. For cylinder bores ranging from 3½ ins. to 5 ins. the weight of the piston, complete with gudgeon pin and piston rings, can be reduced to 0·1 lb. per sq. in. of piston area.

Tolerances on Weights of Pistons and Connecting Rods.

(S.M.M. & T. Provisional Standard No. 53, 1928.)

TOLERANCES ON WEIGHTS OF PISTONS.

Diameter of Piston.	Maximum tolerance. Aluminium Pistons. oz.	Cast-iron Pistons. oz.
Up to 70 mm. inclusive	±¼	±½
Over 70 mm. and up to 100 mm. inclusive	±½	±¾
Over 100 mm. and up to 130 mm. inclusive	±¾	±1

TOLERANCES ON WEIGHTS OF CONNECTING RODS.

Weight of Connecting Rod.	Max. Tolerance. oz.
Up to 50 oz. inclusive	±¼
Over 50 oz. and up to 100 oz. inclusive	±⅜
Over 100 oz. and up to 150 oz. inclusive	±½

The weight of a connecting rod shall be that of an assembled rod, the tolerance being on the gross weight. Not more than half the tolerance shall be taken on the small end of a connecting rod.

DESIGN OF PISTON RINGS.

The practice of making eccentric piston rings with the idea of obtaining equal distribution of radial pressure is now being replaced by the use of concentric rings which have been subjected to hammering or peening. The effect of peening is to reduce the nominal modulus of elasticity, E_n, by about 10-15 per cent. (There is no definite elastic modulus for cast iron, as no part of the stress-strain diagram is straight.) The minimum value for the modulus is given in B.E.S.A. specification, No. 5003/1927, as $13 \cdot 5 \times 10^6$ lbs. per sq. in., and the maximum value as $15 \cdot 5 \times 10^6$ lbs. per sq. in. According to the British Piston Ring Co., Ltd., the value of E_n for good piston ring iron varies from approximately $15–17 \times 10^6$ lbs. per sq. in., but the effect of hammering is to reduce this by about $1 \cdot 5 \times 10^6$, so that the value of E_n for the finished rings should be between $13 \cdot 5–15 \cdot 5 \times 10^6$ lbs. per sq. in.

If

d = diameter of cylinder bore, ins.;

t = radial thickness of ring, ins.;

b = width of ring measured parallel to cylinder axis, ins.;

δ = Free gap in ins. (measured in a straight line tangential to the circumference of ring and parallel to the plane of the ring);

p = pressure between ring and cylinder wall per sq. in. of contact area;

f = calculated maximum stress in tons per sq. in., assuming the material to be truly elastic, and of modulus E_n;

E_n = nominal Young's Modulus;

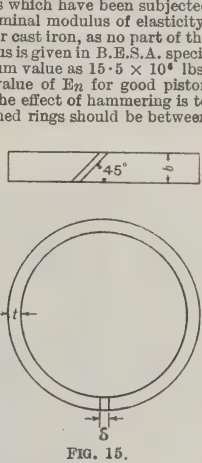


FIG. 15.

Q = diametral load in lbs. applied along diameter of ring corresponding to a change of gap of δ_1 ins.;

then

$$E_n = \frac{5 \cdot 37 \left(\frac{d}{t} - 1\right)^3 Q_1}{b \cdot \delta_1}$$

Minimum thickness of ring $= \frac{d}{30}$.

Maximum thickness of ring $= \frac{d}{30} + 0{\cdot}008$ in.

Width of ring varies in steps from $\frac{d}{16}$ for 2-in. size to $\frac{d}{24}$ for the 6-in. size.

Tolerance $= \begin{matrix} -0{\cdot}001 \text{ in.} \\ +0{\cdot}000 \text{ in.} \end{matrix}$

Maximum free gap $= \frac{d}{7{\cdot}5}$. Such a gap produces approximately the same stress in the ring during the operation of placing it over the piston as it has when in its closed position in the cylinder, but since the ring is continuously closed in the cylinder it is desirable to use a smaller free gap so as to reduce the stress in the ring under working conditions. A free gap of $3\frac{1}{2}$ times the minimum thickness is more satisfactory in practice.

A ring with a free gap of $3{\cdot}5 \times \frac{d}{30}$ gives a minimum pressure of 9·15 lbs. per sq. in. (taking $E_n = 13{\cdot}5 \times 10^6$ lbs. per sq. in.).

$$\text{Cylinder wall pressure } p = \frac{\left(\frac{\delta_1}{t}\right) E_n}{7{\cdot}08 \left(\frac{d}{t}\right) \left(\frac{d}{t} - 1\right)^3},$$

where δ_1 is the difference in inches between the free gap and the gap when in the cylinder measured in the same manner as δ.

$$\text{Maximum stress } f = \frac{\left(\frac{\delta_1}{t}\right) E_n}{5280 \left(\frac{d}{t} - 1\right)^2} \text{ tons per sq. in.}$$

In a ring having a free gap of $3\frac{1}{2} \times t$, the change of gap (δ_1) is approximately equal to $4\frac{1}{2} \times t$ when putting the ring over the piston, and $3\frac{1}{2} \times t$ when putting it into the cylinder.

The load Q is calculated from $Q = \left(\frac{3\pi}{\pi + 4}\right) p.b.d.$,

where b and d are in inch units, p is expressed in lbs. per sq. in., and Q in lbs.

Ignition.

Two systems are in general use (*a*) magneto, (*b*) coil. The magneto may have either a rotating armature or a stationary one, in which case it is known as ' Polar Inductor Type.'

With a stationary armature the current reaches a maximum four times during each revolution of the rotating member or rotor, thus it is possible to produce two or four sparks per revolution of the magneto shaft. In the B.T.H. magneto of this type there are two cobalt-steel magnets fitted on the sides of the main body casting, making intimate contact with two laminated annular poles at each end of the body casting. The poles of the magnets are disposed axially so that the two laminated poles at the ends of the body casting are of opposite magnetic polarity. The flux enters the annular ends of the rotating inductors *via* the small air gap, passing through the inductors to the armature core.

The laminated core (fig. 16), is wound with primary and secondary windings, the beginning of the primary winding being earthed to one of the core flanges. The commencement of the secondary winding is connected to the end of the primary winding, and the end to a brass insert in the armature coil moulding. The condenser, which is carried on the core flanges, makes contact with the flange on one side and with the insulated screw, under which the end of the primary winding is located, on the other.

The low-tension lead from the insulated side of the condenser to the insulated contact of the contact breaker completes the primary circuit.

It is necessary to close and open the primary circuit, and this is effected by a cam mounted on the end of the rotor shaft. This cam operates a bell-crank lever, the end of which carries the movable contact.

In operation the primary winding is short-circuited when the contacts are closed and, as the inductors rotate, the current induced in the primary windings builds up until the cam separates the contacts at the instant when this current is a maximum.

The current induced in the primary winding sets up an intense magnetic field which links itself with the secondary winding, and when the primary current is suddenly broken by the action of the contact breaker this field collapses, with the result that a very high voltage is induced in the secondary winding, giving rise to a spark at the plug.

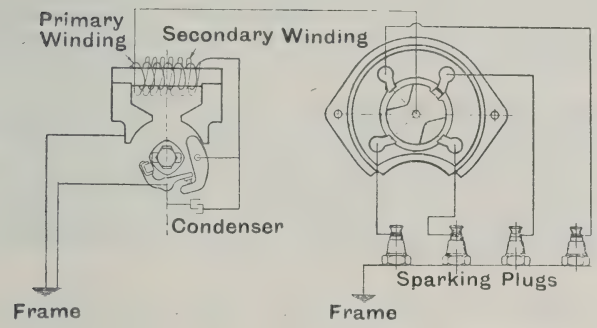


FIG. 16.

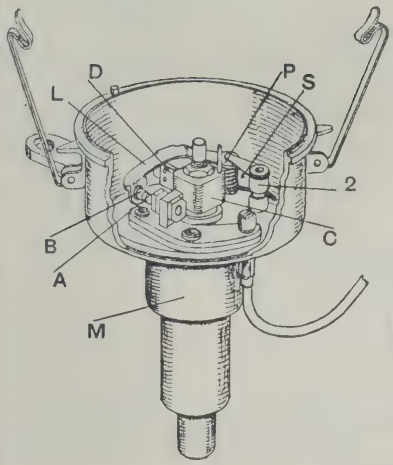


FIG. 17.

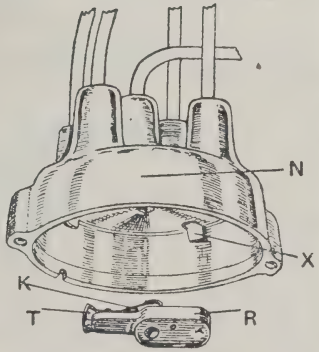


FIG. 18.

The condenser quenches the arc at the contacts, and results in a rapid collapse of the magnetic field linked with the secondary.

In a coil ignition system, such as the Delco-Remy, an intense spark is obtained by converting low voltage current, generated by a dynamo set, into high voltage. For this purpose an ignition coil, combined circuit breaker, and distributing mechanism form a single unit known as the distributor. See figs. 17 and 18.

Low-tension current from the primary coil winding is taken to contact A, which is insulated from the distributor body. Contact B, which is attached to lever L, pivoted at P, is held in contact with A by the spring S. The cam C, which is driven by the engine or dynamo, has as many lobes as there are cylinders to the engine. These lobes press against pad D attached to the lever L. In four-stroke engines the cam rotates at half-engine speed, and in two-stroke engines at engine speed.

The action of the distributor is as follows: The shaft driving the cam C carries an insulating block R, which rotates with it. A metal segment T, to which is attached a light spring K, is mounted on the top of the block R. The block, segment, and spring rotates within the distributor cap N which is made of insulating material and has four metal contacts, X, moulded into it. When the cap is in place the segment T passes very close to or actually touches each of these segments in turn as it rotates. A small carbon brush in the centre of the cap presses on spring K and makes contact with it all the time. This carbon brush is connected to one end of the secondary winding.

The positions of the various parts are arranged so that whenever the cam C separates the contacts A and B the segment T is just opposite one of the metal contacts in the cap N. The high-tension current then jumps across the small air gap between the segment and the contact, and is led away by means of a high-tension wire to the correct plug.

Horse-Power Rating Formulæ of Various Countries.

(S.M.M. & T. and R.A.C. 1928.)

Notation

H = horse-power treasury rating; d = bore of cylinder;
N = number of cylinders; s = stroke;
V = swept volume in one cylinder; r = engine revolutions.

Austria, *Latvia*, *Luxemburg*, *Norway*, *Russia*. $H = 0{\cdot}3Nd^2S$, where d is in cm., S is in metres.

Belgium. $H = KNd^2Sr$,

where

K = S for passenger and light goods cars of less than 6 cylinders;
K = 4·25 for heavy goods vehicles.

d is in mm., S is in metres, and r is in revs. per minute, taken as 1,200 for private cars and 1,000 for heavy goods vehicles and omnibuses.

England, *U.S.A.* $H = \frac{Nd^2}{2{\cdot}5}$, where d is in inches $= \frac{Nd^2}{1613}$, where d is in mm.

Denmark. As Austria for four-stroke engines. Special factory ratings are adopted for two-stroke engines.

France. $H = KNd^2Sr$,

where

K = 0·00020 for 1 cylinder; K = 0·00015 for 4 cylinders;
K = 0·00017 for 2 cylinders; K = 0·0013 for more than 4 cylinders.

d is in cm; s is in metres; r = max. engines revs. per second.

Germany. $H = 0{\cdot}3\ Nd^2S$ for four-stroke engines;
$H = 0{\cdot}45\ Nd^2S$ for two-stroke engines,

where d is in cm., S is in metres.

Hungary. $H = \frac{Na^2}{20}$, where d is in cm.

Irish Free State. $H = \frac{NV}{100}$, where v is in c.c.

Italy. $H = 0{\cdot}0878\ NV^{0{\cdot}654}$ for four-stroke engines;
$H = 1{\cdot}4\ (0{\cdot}0878\ NV^{0{\cdot}654})$ for two-stroke engines,

where V is in c.c.

(*Note*.—Fractions of a horse-power are neglected.)

Japan. $H = \frac{Nd^2}{3}$, where d is in inches.

Spain. $H = 0{\cdot}08\ (0{\cdot}785d^2S)^{0{\cdot}6}\ N$ for four-stroke engines;
$H = 0{\cdot}11\ (0{\cdot}785d^2S)^{0{\cdot}6}\ N$ for two-stroke engines,

where d and S are in cm.

Sweden.

$$H = \frac{Nd^2Sr}{K}$$

where

K = 220,000 for four-stroke engines using benzine;
K = 240,000 for four-stroke engines using petrol;
K = 180,000 for two-stroke engines.

d and S are in cm.; r = normal revs. per minute.

Switzerland.

$$H = 0{\cdot}4\ Nd^2S,$$

where d is in cm.; S is in metres.

Horse Power from Given Working Volume and M.E.P.

$$\text{Horse-power} = \frac{P_m \times Q \times N}{396000},$$

where

P_m = m.e.p. lbs. per sq. in. Q = working volume in cub. ins.; N = r.p.m.

Valve Area for Given Volume, Revolutions and Gas Velocity.

$$Vg = \frac{VcN}{360\ \Delta},$$

where

Vg = gas velocity, ft. per sec.; Vc = volume swept out per cylinder; cub. ins.
Δ = valve area in sq. ins.; N = r.p.m.

Inlet Pipe Diameter.

$$Vg = \frac{QN}{180\pi D_1^2},$$

where

Vg = gas velocity, feet per sec.; Q = working volume per rev.; cub. ins.
D_1 = diameter of inlet pipe; ins. N = r.p.m.

Compression Pressure.

$$\left(\frac{V_2}{V_1}\right)^n = R^n = \frac{P_2}{P_1},$$

where

V_1 = volume of clearance space; cub. ins. V_2 = volume of piston displacement + V_1 cub. ins.
P_1 = atmospheric pressure; lbs. per sq. inch. P_2 = compression pressure in lbs. per sq. in. (absolute).
$n = 1{\cdot}3$;
R = compression ratio.

Calculating Lift of Poppet Valves.

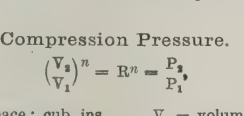


FIG. 19.

The gas passage is the area of the curved surface of the frustrum of the cone *abcd*. V = volume of gas to be discharged in cubic feet, and V_g = mean gas velocity in feet per second through valve opening, and A = area of valve opening, then,

$$A = \frac{144V}{Vg} \text{ (sq. ins.)}.$$

The lift l must be such as to produce this area for the gas to pass through.

$$h^2 = \frac{4A^2}{\pi^2(d_1 + d_2)^2} - \frac{(d_1 - d_2)^2}{4}.$$

Knowing the value of d_1, d_2, A and t, the lift $l = h + t$ (fig. 19).

DIMENSIONS OF POPPET VALVES.
(S.A.E. Standard.)

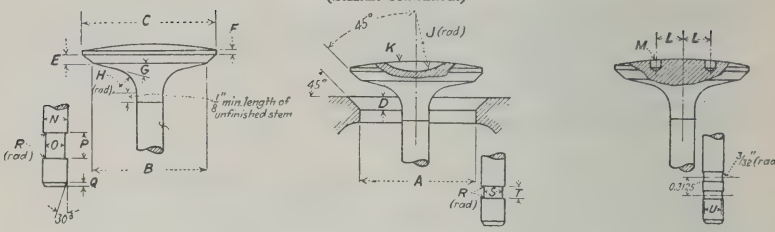


FIG. 20.

DIMENSIONS OF POPPET VALVES.
Heads and Seats.

Nom. Size. A.	Dimensions (ins.). Fig. 20. B.	C.	D.	E.	F.	G. Deg.	H.	J.	K. Width.	K. Depth.	L.	M. Diam.	M. Depth.
1	31/32	1 5/32	1/16	9/32	1/32	7	1/4	3/4	1/16	5/32	1/4	1/8	1/8
1 1/8	1 3/32	1 9/32	1/16	5/32	1/32	7	1/4	3/4	1/16	5/32	1/4	1/8	1/8
1 1/4	1 7/32	1 13/32	1/16	3/32	1/32	7	1/4	1	1/16	3/32	1/4	1/8	1/8
1 3/8	1 11/32	1 17/32	1/16	3/32	1/32	7	1/4	1	1/16	3/32	1/4	1/8	1/8
1 1/2	1 15/32	1 1/2	1/16	3/32	1/32	7	1/4	1 1/4	1/16	3/32	1/4	1/8	1/8
1 5/8	1 19/32	1 5/8	3/32	9/64	3/64	7	1/4	1 1/4	1/16	3/32	1/4	1/8	1/8
1 3/4	1 23/32	1 3/4	3/32	9/64	3/64	7	1/4	1 1/4	1/16	3/32	1/4	1/8	1/8
1 7/8	1 27/32	1 7/8	3/32	9/64	3/64	7	1/4	1 1/2	1/16	3/32	1/4	1/8	1/8
2	1 31/32	2	3/32	9/64	3/64	7	1/4	1 1/2	3/32	3/32	1/4	1/8	1/8
2	1 31/32	2 5/32	1/8	11/64	1/16	12	1/4	1 1/2	3/32	3/32	1/4	1/8	1/8
2 1/8	2 3/32	2 7/32	1/8	11/64	1/16	12	5/16	1 1/2	3/32	7/64	1/4	5/32	5/32
2 1/4	2 7/32	2 9/32	1/8	11/64	1/16	12	5/16	1 1/2	3/32	7/64	1/4	5/32	5/32
2 3/8	2 11/32	2 11/32	1/8	11/64	1/16	12	5/16	1 1/2	3/32	7/64	1/4	5/32	5/32
2 1/2	2 15/32	2 13/32	5/32	7/32	1/16	12	5/16	1 1/2	3/32	1/8	1/4	5/32	5/32
2 5/8	2 19/32	3 1/32	5/32	7/32	1/16	12	5/16	1 1/2	3/32	1/8	1/4	5/32	5/32
2 3/4	2 23/32	3 5/32	5/32	7/32	1/16	12	5/16	1 1/2	3/32	1/8	1/4	3/16	3/16
2 7/8	2 27/32	3 9/32	3/16	7/32	1/16	12	5/16	1 1/2	3/32	1/8	1/4	3/16	3/16
3	2 31/32	3 13/32	3/16	7/32	1/16	12	5/16	1 1/2	3/32	1/8	1/4	3/16	3/16

Radii for Curvature of Crown of Valve-Head.

Nominal size	1–1⅜	1⅜–1⅞	2–2½	2⅜–2⅝	2¾–3
Radius, ins.	5	6	7	8	9

VALVE STEMS AND GROOVES.
(Fig. 20.)

Nom. Size. A.	Dimensions (ins.). N. Max.	O. Max.	P. Max.	Q.	R. Max.	S. Max.	T. Max.	U. Max
1 –1⅜	0·3100	0·250	0·328	1/32	0·012	0·235	0·144	0·250
1⅜–1⅞	0·3725	0·309	0·391	1/32	0·012	0·253	0·144	0·309
2 –2⅜	0·4345	0·312	0·516	3/64	0·015	0·312	0·175	0·372
2½–3	0·5595	0·437	0·578	1/16	0·020	0·405	0·206	0·495

The above dimensions are for steel valves; they are considered equally applicable for cast-iron valves, except that it may be desirable to have slightly more metal in the latter under the heads.

Valve stem tips shall be hardened to give a scleroscope reading of 55–65 after finishing. Recommended that length of tips from end of stems to stem-grooves be not less than finished diameter of stem to prevent undue breakage after hardening.

Tolerance on fractional dimensions = ± 0·010 in.

Valve Springs.

(*Donkin.*)

Load in pounds per inch of deflection or rate R_r

$$R_r = \frac{Gd^4}{8D^3N},$$

where

G = rigidity modulus, lbs. per sq. in.; N = number of active coils;
D = mean diameter of spring, ins.; d = diameter of wire, ins.

The stress in the wire is determined from $S = \frac{8PD}{\pi d^3}$,

where

S = stress in lbs. per sq. in.; D = mean diameter of spring, ins.;
P = load in lbs.; d = diameter of wire, ins.

Experience has set a value of about 60,000 lbs. per sq. in. for the maximum stress, and 25,000 lbs. per sq. in. for the maximum stress range. The stress range is more important than the maximum stress up to the elastic limit. With a high stress range, the material is put through a large angle of twist which tends to fatigue it quickly and open up any discontinuity in the metal, causing early failure, therefore a high maximum stress with low stress range is less destructive.

Common practice is to make valve springs from oil-tempered spring wire. Tests made during a five-year period show that springs made from soft wire, heat-treated after coiling, are far superior to springs coiled from heat-treated wire.

Averages of life tests give the following comparative values:

(*a*) Carbon steel, heat-treated before coiling	1·0
(*b*) Electric furnace, chrome vanadium, heat-treated before coiling . .	0·9
(*c*) Carbon steel, heat-treated after coiling	5·0
(*d*) Electric furnace, chrome vanadium, heat-treated after coiling . .	10·0

Even though extreme care is taken with the material, etc., failures may occur due to the resonance between the natural frequency of the spring and the frequency of the forced vibration of the camshaft. When the forced vibration is of such a frequency that its wave-length is a simple multiple of the free wave-length of the spring a condition of resonance may be created. If the wave-length is long and is such that an integral number of the free wave-lengths of the spring are completed during the cycle of the valve, the application of the force will be in phase, and the resulting wave motion in the spring will be cumulative and resonant. The natural frequency of a spring $N = 531\sqrt{\frac{W}{R}}$ (Ricardo),

where

N = complete number of free vibrations of the spring per minute;
R = rate of spring in lbs. per sq. in
W = weight of the active mass of the spring in lbs.

If the vibration within the spring is excessive it may cause the valve to float open and thus impede the operation of the engine and cause excessive hum and noise. In all cases it causes a building up of stress which is destructive to the life of the spring.

Valve-Spring Pressure required with Tangential Cams.

For round-nosed cams with roller-footed tappets, see fig. 21.

$$\text{Minimum valve-spring force} = \frac{W \times N^2 \times D}{2{\cdot}256 \times 10^6} \left\{ 1 + \frac{D}{R + r} \right\} \text{ lbs.}$$

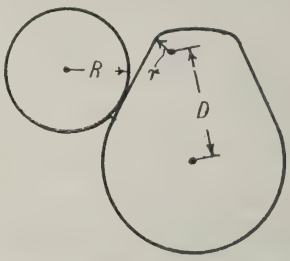


Fig. 21.—Valve Spring Pressure required with Tangential Cams.

where

W = equivalent weight of the valve system in ounces.
N = r.p.m. of engine.
D = distance between the centres of the cam-base circle and the cam-nose circle in inches.
R = radius of the roller on the foot of the tappet in inches.
r = radius of the cam-nose circle in inches.

Valve Timing.

Analysis of one hundred automobile engines (1927) showed the following average timing:

Inlet opens, 10° A.	Exhaust opens, 47° 50′ B.
Inlet closes, 40° A.	Exhaust closes, 7° 30′ A.
A = after T.D.C.	B = before T.D.C.

In 95 per cent. of the engines the inlet and exhaust valves were of the same diameter.

$$\frac{\text{Diameter of inlet valve } (\textit{clear})}{\text{Valve lift}} = 4{\cdot}75. \qquad \frac{\text{Cylinder Bore}}{\text{Valve diameter}} = 2{\cdot}2.$$

Material for Valves.

Valves operate under a tension-impact load, the temperature varying from practically atmospheric at the tappet end to about 1600° F. at the head. The temperature gradient is a function of the guides in which the valve reciprocates, and for any particular design is affected by the material in the guide, the degree in which it shrouds the stem, and the inner surface of the guide. The temperature of the head is affected by the width of valve seat and cross-sectional area of stem. The head and upper part of stem are subjected not only to high temperatures, but also to the corroding action of the mixture, and the corroding and eroding action of the hot gases passing through the valve at a velocity of 150–200 ft./sec.

The scaling of a valve not only reduces the cross-sectional area, but the loose scale may also become attached to the valve seat.

The strength of the valve material at ordinary temperature is important only in so far as it is a function of the hardness. For highly stressed valves the strength at high temperatures is important, and it is necessary to select a material which does not show a decrease in the original strength sufficient to cause distortion or failure.

Camshaft Drives.

(*H. E. Merritt*, '*Proc. I.A.E.*')

With side-by-side valves there are two general types—(*a*) chain; (*b*) spur or helical gear. Silence of operation is important, and the chief obstacle to this centres round the uneven torque of the drive. This may result in a wave motion in a chain drive, introducing considerable stresses and limiting the allowable velocity of the chain. With tooth gear the silence of operation depends largely on the accuracy of manufacture. In many cases helical teeth are used in place of the straight tooth, with improved performance. This introduces, however, an end-thrust on the crankshaft and camshaft, and a transverse torque on the intermediate gear. In correctly designed helical teeth the degree of overlap should not be less than one pitch, and with double helical gear end-thrust is eliminated.

Camshaft brakes may be used to secure uniformity of driving torque, in which case the breaking torque must be at least equal to the maximum forward accelerating torque, due to a closing valve spring.

Another method is to drive an auxiliary, such as the fan, water, or oil pump from the free end of the camshaft, thereby providing the necessary constant resistance.

The majority of camshaft drives are taken from the front end of the crankshaft with the object of accessibility, but in this position crankshaft torsional vibrations may produce severe fluctuations in tooth pressure. For minimising gear noise, the use of non-metallic gears is very common. Due to their resilience and high inter-molecular friction, the energy of tooth impact is absorbed, which prevents the bounce of the teeth.

Owing to the greater centre distance between an overhead camshaft and the crankshaft a variety of drives is possible, and representative arrangements are shown in fig. 22.

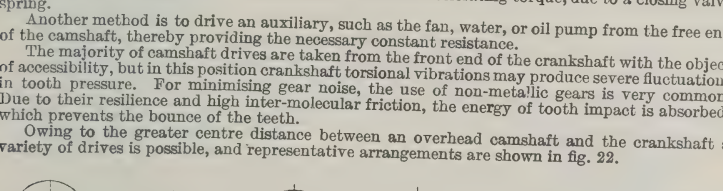


Fig. 22.—Types of Overhead Camshaft Drive.

A. Chain.
B. Spur Gear.
C. Bevel.
D. Chain-bevel.
E. Helical gear.
F. Bevel worm.

The train of spur gears (B) has a large number of points at which wear may occur, and is liable to be noisy. The bevel drive demands care in layout. Bevel gears require very accurate location in every direction, means for axial adjustment, and thrust bearings. Straight-tooth bevel gears are the most difficult of all gears to get to run quietly, and the tendency is to use spiral bevels, but this type calls for the greatest accuracy in cutting and assembly.

The unequal expansion of the cylinder block and the vertical shaft has an important effect on gear mesh and tappet clearance, and this can be provided for by the use of a telescopic vertical shaft. With the spiral and worm drives a wide range of gear diameters is possible. Another advantage is that there is no necessity for precise adjustment for end float on either crankshaft or camshaft, or expansion of the vertical shaft. On the other hand, the contact between spiral gears is only point contact and the intensity of tooth pressure is very high.

Bearings.

Crankshaft and big-end bearings are usually white-metal lined; gudgeon bushes of bronze.

Corresponding to a maximum explosion pressure of 300 lbs. per sq. in., the bearing pressure in lbs. per sq. in. of projected area is as follows:—

For 'big-end' bearings: 800–1,100 lbs. per sq. in.; more usually the latter.

For 'gudgeon' bearings: 1,500–2,700 lbs. per sq. in.; average about 2,000 lbs. per sq. in.

Strength of Splined Shafts.

From tests carried out on actual shafts, the elastic limit for a splined shaft is substantially less than for a plain round shaft of a diameter equal to the base diameter of the spline. On the other hand, the ultimate strength of the splined shaft is materially greater than that of the round shaft. It would appear that the reason the elastic limit is lower in the splined shaft is that the material in the splines adds very little to the torsional strength, but the strain on the fibres at the outside of the splines will be about the same as if the shaft were of solid round section and equal in diameter to the outside diameter of the splines. With fairly elastic material, if the distortion is continued beyond the elastic limit, the splines assume a helical form and gradually come into tension, thereby adding to the resistance of the shaft to further deformation. If the splines are supported, as in the case of the permanent fitting so that they are in more nearly direct shear from the beginning of the load application, their shearing strength is added to that of the body of the shaft.

Design of Gear-box Gears.

(*H. E. Merritt*, '*Proc. I.A.E.*').

To determine the surface stress and bending stress.

If

T = torque in lb.-in. units;
t = number of teeth;
M = module;
f = face width, ins.;
Y = strength factor;
Z = zone factor;
F_i = load (lbs.) per in. of face;
C = cone distance;
S_b = bending stress, lbs. per sq. in.;
S_c = surface stress, lbs. per sq. in.

For a spur gear

$F_i = 2\,T/M.t.f$;
$S^b = F_i/M.Y$;
$S_c = F_i/M.Z$.

For a straight bevel $F_i = 2\,T.C./[M.t.f(c - f)]$.

Surface stress S_c = load per in. of face per in. of relative radius of curvature

$$= \frac{\text{Tooth load}}{(\text{Length of line of contact}) \times (\text{Relative radius of curvature})}.$$

Values of S_b and S_c obtained in practice.

No.	Car.*	Constant Mesh Gears.		First-speed Gears.		Differential.		Spiral.	
		S_b.	S_c.	S_b.	S_c.	S_b.	S_c.	S_b.	S_c.
1	Austin 7	13,650	4,090	21,650	6,550	33,400	11,650	20,600	8,700
2	Austin 12	20,400	5,960	37,500	11,700	67,000	24,250	29,900	9,560
3	Morris 13·9	23,100	6,900	33,900	10,300	23,100	7,610	30,400	9,720
4	Morris 15·9	16,700	4,880	27,900	7,850	31,400	10,600	Worm	
5	Vauxhall 24	22,600	6,780	36,000	10,800	55,000	18,850	38,100	12,500

Front-Wheel Drives.

The front-wheel drive is receiving a considerable amount of attention. Some of the main advantages and disadvantages of such a drive are as follows: —

TRANSMISSION DATA.

(*Merritt.*)

(Representative British Practice.)

Make of Car.	No. of Cylinders.	Displacement c.c.	Engine Torque lbs. ins.	First Speed Ratio.	Rear Axle Ratio.	Constant Mesh.			First Speed.			Material.	Differential Material.
						Teeth.	M.	*f*.	Teeth.	M.*	*f*.*		
A.C. 15·7	6	1,990	975	13·2	4·5	16 : 30	0·125	0·5	18 : 28	0·125	0·688	—	AHNC
A.C. 11·8	4	1,495	692	14·7	5·0	16 : 26	0·125	0·432	15 : 27	0·125	0·560	—	AHNC
Argyll 12·5	4	1,630	876	18·0	4·5	15 : 33	0·125	0·625	17 : 31	0·125	0·625	CH5N	AHNC
Aston-Martin	4	1,488	990	9·3	3·5	17 : 29	0·116	0·83	18 : 28	0·116	0·394	AHNC	AHNC
Austin 7	4	749	315	18·4	5·66	14 : 28	0·125	0·44	16 : 26	0·125	0·44	CH5N	CHNS
Austin 12	4	1,820	893	19·9	5·17	17 : 34	0·127	0·56	13 : 25	0·167	0·56	CH5N	CHNS
Austin 20	4	3,620	1,630	20·1	4·66	19 : 41	0·125	0·75	15 : 30	0·167	0·75	CH5N	CHNS
Bayliss Thomas	4	1,495	770	17·2	4·55	13 : 31	0·125	0·56	17 : 27	0·125	0·56	OHNC	CHCS
Beardmore Taxi	4	2,400	1,000	18·5	4·5	20 : 40	0·125	0·75	15 : 30	0·167	0·75	CH5N	CHCS
Beverley Barnes	8	4,820	2,688	20·8	4·92	21 : 37	0·147	0·875	15 : 36	0·167	0·875	AHNC	AHNC
Crossley	6	3,180	1,600	18·4	4·92	15 : 29	0·143	0·56	15 : 29	0·143	0·78	CH3N	CH3N
Lea Francis	4	1,495	640	18·8	4·7	15 : 33	0·125	0·56	17 : 31	0·125	0·56	OHNC	—
Morris 13·9	4	1,800	807	15·2	4·75	16 : 30	0·128	0·50	17 : 29	0·127	0·57	CHNC	CHCS
Morris 15·9	4	2,510	1,140	15·6	4·5	18 : 39	0·125	0·81	22 : 35	0·125	0·81	CHNC	CHCS
Riley	4	1,650	760	18·3	4·7	17 : 29	0·125	0·625	14 : 32	0·125	0·625	OHNC	CHNS
Singer 7·8	4	857	390	17·1	5·0	16 : 31	0·10	0·50	17 : 30	0·100	0·50	AHNC	AHNC
Singer 14·8	6	1,775	830	17·1	5·0	16 : 31	0·125	0·625	17 : 30	0·125	0·687	AHNC	AHNC
Vauxhall 13·9	4	2,300	1,100	17·7	4·5	17 : 32	0·138	0·65	17 : 32	0·138	0·80	AHNC	CH3N
Vauxhall 24	4	4,230	2,640	11·1	3·33	17 : 31	0·167	0·875	17 : 31	0·167	1·00	AHNC	CH3N
Talbot 13·8	6	1,665	792	23·2	5·87	19 : 30	0·118	0·50	14 · 35	0·118	0·63	CH5N	CHCS

* M = module; *f* = face width, ins.

AHNC = air-hardening nickel-chrome steel;
CHCS = case-hardening carbon steel;
CH3N = 3 per cent. nickel case-hardening steel;
CH5N = 5 per cent. nickle case-hardening steel;
CHCN = nickel-chrome case-hardening steel;
OHNC = oil-hardened nickel-chrome steel.

Advantages.—(*a*) Relatively unencumbered space for a body of any design, floor, height, and dimensions; (*b*) elimination of the rear axle; (*c*) driving force always applied to the wheels in the direction of their motion; (*d*) elimination of long propeller shaft and substitution of shafts having lower speed and less tendency to vibrate; (*e*) greater quietness and freedom from body rumble; (*f*) ability to provide front-wheel braking without carrying brake drums and operating mechanism on wheels or axle; (*g*) a less expensive rear axle and a simplified frame construction; (*h*) decrease in the total weight of vehicle; (*i*) decreased thrust on front-wheel bearings; (*j*) front axle and spring can be relieved of all braking torque, and all springs can be relieved of driving torque reaction.

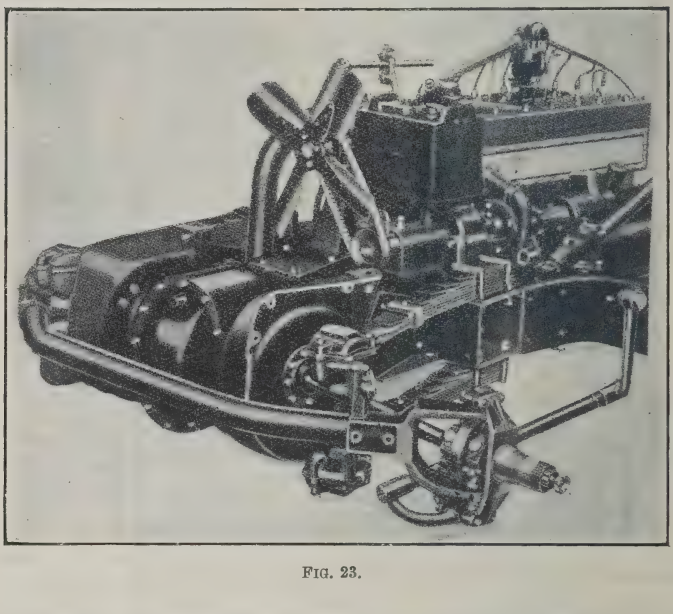


FIG. 23.

Disadvantages.—(*a*) Decreased traction, notably when hill-climbing and accelerating; (*b*) liability of greater damage to driving mechanism in the event of collision; (*c*) difficulty of obtaining satisfactory weight distribution, especially in trucks and coaches; (*d*) increased complication in the driving mechanism; (*e*) decreased space for radiator; (*f*) difficulty in obtaining a quiet drive, especially with a bevel drive mounted on the frame; (*g*) harder steering, due to increased load on front axle and steering pivots; (*h*) difficulty of obtaining an adequate steering angle.

A typical example of a front-wheel drive is shown in fig. 23.

Steering-Wheel Position.

The following formulæ (*W. Ferrier Brown*) are for the calculation of the steering-wheel position, a, b, and d, being known factors.

Given m and θ—to find c and e.

$$\phi = \tan^{-1}\frac{b-a}{m}; \quad n = \frac{m}{\cos\phi} = m\sec\phi;$$

$$\beta = \theta - \phi = \sin^{-1}\frac{c+d}{n}; \quad e = n\cos\beta;$$

Given c and e—to find m and θ.

$$\beta = \tan^{-1}\frac{c+d}{e}; \quad n = \frac{e}{\cos\beta} = e\sec\beta;$$

$$\phi = \sin^{-1}\frac{b-a}{n}; \quad m = n\cos\phi;$$

$c = n \sin \beta - d$.

To find h and l.

$h = m \sin \theta + a \cos \theta - d$; $l = \cos \theta (m - a \tan \theta)$.

$\theta = \phi + \beta$ = rake of column.

To find h and l.

$h = c + b \cos \theta$; $l = e - b \sin \theta$.

To find k (which may be + or −)

$$k = \frac{a}{\sin \theta} - \frac{d}{\tan \theta}.$$

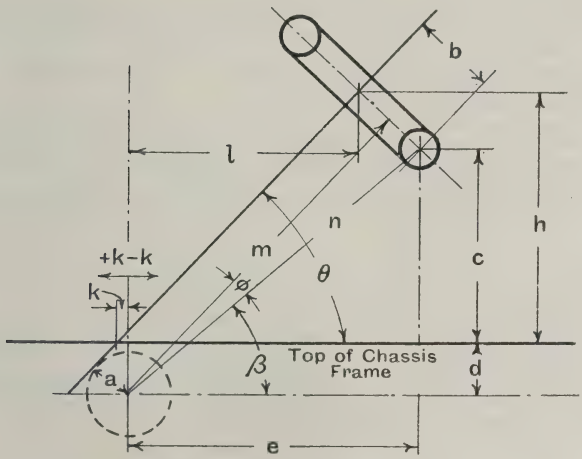


FIG. 24.—Steering-Wheel Position.

Lubrication.*

Carbon Formation.—The tendency towards carbon formation is due to an excess of oil working past the piston rings into the combustion chamber. Special means are used to limit the amount of oil reaching the combustion chamber, such as (*a*) baffle plates between the crank case and cylinder casting, (*b*) scraper ring on the piston, (*c*) effective oil relief groove on the piston or oil relief holes, (*d*) adequate control of the oil supply to meet the engine's requirements.

Sealing the Piston Rings.—The clearance between the piston and the cylinder wall controls the lubricant to seal the piston rings. When this is small, a light or medium bodied oil is sufficient. Larger clearances require a heavier-bodied oil.

Piston clearance must be considered in conjunction with the cylinder bore. A clearance of 0·005 in. on a small-diameter piston is relatively large compared with the same clearance on a piston of double the size.

Piston material affects the clearance allowed; thus heavier-bodied oils are necessary for aluminium pistons compared to cast iron. Other factors are: the number of piston rings and their position on the piston, the presence of oil grooves on the piston to assist in oil distribution, method of finishing cylinder bore.

Lubrication of Transmission and Differential.

The following characteristics of the lubricant are necessary:—

(*a*) Must be of a character and consistency to coat, follow, cushion, and adequately lubricate the gear mesh.

(*b*) Must be capable of thorough distribution to any ball, roller, or plain bearing, and must have no corroding effect on the balls, rollers, or raceways.

(*c*) Should be of a body and character to minimise resistance and power loss due to the churning action of the gears. In the case of transmission, it should offer the minimum of resistance to the manual act of gear changing.

(*d*) Should be capable of carrying off the greatest amount of heat, and of exerting a washing action on the gear teeth and bearing surfaces.

(*e*) Should be a body to prevent leakage from the gear-box or back axle under all normal operating conditions.

* See also Handbook published by the Vacuum Oil Co., Ltd.

Chassis Lubrication.

Advantages of Oil for Lubrication.

(O. T. Myers.)

(1) Sustains as heavy a bearing load as grease with less friction.

(2) Fed to one properly located point in a bearing will quickly spread over the entire surface.

(3) Will flow through a very small hole, and can be conducted a considerable distance by capillary attraction and surface tension.

(4) Carries much less dust than grease, can be filtered, and not so much exposed to dirt as grease, when in bulk or when being applied.

(5) Contains no inert matter to clog holes or channels.

(6) Engine oil can be used.

(7) Easier to apply automatically than grease.

(8) A flow of clean oil tends to clean a bearing, and bearings arranged for oil lubrication can be cleaned with petrol or paraffin.

(9) When oil is used on spring bolts, the seepage works down into the spring leaves, preventing rust.

Weight of Passenger Cars.

(*W. Ferrier Brown*, '*Proc. I.A.E.*')

The total weight of an average car is made up as follows:—
Chassis complete, 65 per. cent.; body complete, 30 per cent.; wings, valances, and step-boards, 5 per cent.

For the complete body in detail, the proportionate weights are:—
Body frame (steel), 14 per cent.; body framing complete with head-fittings, 42 per cent.; panels (steel), 10 per cent.; paint, glass and inside mouldings, 12 per cent.; trimming, including two extra seats complete, etc., 22 per cent.

Percentage Weight of Automobile Parts.

(P. M. Heldt.)

The following table shows the averaged values for the years 1917 and 1929. The percentage change gives an indication of the progress in design.

Component.	Percentage of Total Chassis Weight.		Change. Per cent.
	1917.	1929.	
Frame	9·12	13·0	+ 42·6
Front springs Rear springs	9·12	2·4 4·2	− 27·6
Front axle	3·04	5·0	+ 64·2
Rear axle	11·38	10·0	− 12·1
Front wheels	6·44	7·15	+ 11·0
Rear wheels	7·97	7·21	− 9·5
Engine	30·36	32·50	+ 7·0
Clutch	1·37	0·75	− 45·2
Transmission	5·3	4·45	− 16·0
Propeller shaft	1·67	1·03	− 38·3
Silencer and exhaust pipe	1·13	1·03	− 8·8
Under pan	0·68	0·52	− 23·5
Fuel tank	1·21	1·25	+ 3·3
Electrical equipment	5·38	4·28	− 20·4
Steering gear	2·11	1·83	− 13·3
Radiator	2·72	2·80	+ 2·9

COEFFICIENT OF FRICTION BETWEEN TYRE TREADS AND VARIOUS SURFACES AT SLOW SPEEDS.

(H. Chase.)

Tyre.	Surface.	Coefficient of Friction.			
		Ahead.		Crossways.	
		Start.	Slide.	Start.	Slide.
Average of 14 tyres with various treads	Dry concrete road	0·72	0·80	0·72	0·80
	Flooded concrete road	0·81	0·94	0·78	0·94
	Rough concrete (dry)	0·90	1·00	—	—
Non-skid tread	Dry concrete road	0·66	0·66	—	—
	Flat rubber	0·76	0·76	—	—
	Water on concrete	0·76	0·98	—	—
	Water on clean asphalt	0·80	0·90	—	—
	Dry clean asphalt	0·82	0·86	—	—
	Dry clean concrete	0·80	0·84	—	—
	Moist sandy clay	0·70	0·84	—	—
	Fine loose macadam	0·70	0·70	—	—
	Oiled asphalt	0·60	0·64	—	—
	Wet oiled asphalt	0·54	0·60	—	—
Ribbed tread	Flooded concrete road	0·76	0·94	0·72	0·96
	Dry concrete road	0·72	0·80	0·72	0·76
	Fine loose macadam	0·60	0·64	—	—

Automobile Brakes.

A common misapprehension is that which assumes that there is no slipping between tyre and road until wheels are locked. Actually slippage occurs immediately upon the sudden application of the brakes, and continues until the car is brought to rest whether the wheels are locked or not. Although the wheels continue to roll, their peripheral speed begins to decrease immediately and is therefore always less than that at which the road surface passes under the car.

The effectiveness of brakes can be determined directly by measuring the breaking force per unit of car weight *i.e.* per lb. per 1000 lbs. or per ton of car weight. The relation between the distance S in feet required to stop, the initial car speed V in m.p.h., and (1) the rate of slowing down in feet per sec.$^2 = a$; (2) the retarding force in lbs. per lb. $= f_1$, (3) the retarding force in lbs. per 1000 lbs. of car weight $= f_2$, is given by

$$(1)\ S = 1{\cdot}08\,\frac{V^2}{a}; \qquad (2)\ S = 0{\cdot}0336\,\frac{V^2}{f_1} \qquad (3)\ S = 33{\cdot}6\,\frac{V^2}{f_2}.$$

STOPPING DISTANCES FOR DIFFERENT TYPES OF BRAKE APPLICATION.

	Type of Brake.		
	Rear External Band Brake.	Four Brakes Internal Shoes.	Four Brakes, External Bands.
Total weight of car lbs.	6,000	6,000	6,000
Total surface of brake lining, sq. ins.	332·4	308·6	600
Weight per sq. in. of lining, lbs.	18·1	19·4	10
Stopping distance in feet at			
20 m.p.h.	35	23	15
30 „	70	45	35
40 „	113	74	60
50 „	175	114	90

STOPPING TESTS OF CAR FITTED WITH HYDRAULIC BRAKES.

Total weight of car with driver = 4,400 lbs.; Weight on front wheels = 2,000 lbs.; Weight on rear wheels = 2,400 lbs.

Tyres, cord, size 35 × 5, pressure 65 lbs. per sq. in.;
Road surface—dry concrete.
Speed, 30 m.p.h.
Stopping distance—brakes applied on rear wheels only = 89·2 feet.
" " " front " " = 67·6 "
" " " rear and front wheels = 36·8 feet.

BRAKES.

Four-wheel brakes have now become standard practice. Typical systems are illustrated in fig. 25.*

(*a*) *Six-brake Systems*, using two cross-shafts, as on Rolls-Royce, Bentley, Morris cars, etc. It is not much used on Continental or American cars. The hand-operated brakes operate a second pair of shoes side by side in each rear-wheel drum. The linkage employed may take several forms, of which the simplest and most popular is shown.

Compensation is effected by the use of swinging levers which interrupt the pull rods or cables near to each brake approximately at the point representing the centre around which the axle moves, as an arc when the springs deflect.

A second six-brake system is shown at B, fig. 25. In this case a single pedal controlled cross-shaft distributes the forces to four-wheel brakes as in (A), but instead of using separate shafts one is

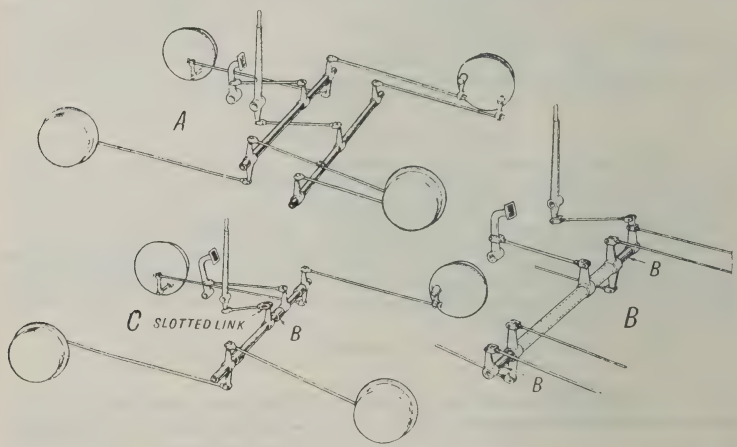


FIG. 25.—Four-Wheel Brake Systems.

placed inside the other. Thus the hand-lever may control a cross-shaft supported by end bearings BB, on which shaft is carried a tube operated by the pedal, and all the brake connections are taken to levers on this shaft and tube.

(*b*) *Five-brake Systems.*— In this system a hand-operated transmission brake is used. The brakes on the four wheels are connected by linkages to the foot pedal. A failure in a rod or pin may make the wheel brakes entirely inoperative.

(*c*) *Four-brake systems*, in which there are four brake units, one applied to each wheel, and all four or a pair of these units can be brought into operation by the use of either the pedal or hand-lever. A dangerous result is, however, obtained if this system is provided with thorough compensation, for the failure of any rod, pin, or cable would leave the brakes useless. The most widely used four-brake system is shown at C, fig. 25. There are two cross-shafts, one coupled to

* *Proc. I.A.E.*

the rear brakes and the other to the front brakes, and these shafts work in unison in virtue of a positive connection one to the other by rods or cables. The pedal operates the front shaft, and the hand-lever is connected to the rear shaft through the medium of a slotted yoke, cable, or chain, so as to be unaffected when the brakes are applied by the pedal.

Design of Brakes.

A retardation of 22·6 ft. sec.^{2} generally accepted as sufficient.
Retarding force F applied at the tyres = weight of car × 0·7.
Total outward force P on shoe lining,

$$P = (F.D)/(\mu.d),$$

where
D = tyre diameter, ins.
d = internal diameter of drum, ins.
μ = coefficient of friction of lining. 0·33 for general conditions.
Common to employ 1 sq. in. of lining on the service brakes per 28 lbs. of car weight.

Mean pressure on lining, 120 lbs. per sq. in. For internal expanding brakes not advisable for the arc of lining material to subtend an angle greater than 120°, 100° is generally sufficient.

Linkage leverage (overall) employed on current chassis varies between 35 to 1 and 45 to 1, made up as follows :—

(*a*) A cam and lever ratio at the brake itself of between 6 to 1 and 8 to 1.
(*b*) A ratio in the cross-shafts levers on other intermediate levers of between 1·2 to 1 and 1·7 to 1, sometimes unity.
(*c*) A pedal leverage of between 4 to 1 and 6 to 1.
For hand-lever, overall leverage 50 to 1.

Power Brakes.

'Servo' braking apparatus may be either 'hydraulic' or 'vacuum.' The Westinghouse vacuum system is shown in fig. 26. This system utilises, as its source of power, pressure of the atmosphere operating against the partial vacuum created by the pistons of the engine on the suction stroke, when the throttle is closed. A section of the 'Servo' motor is shown in fig. 26. The cylinder M contains a piston N with leather packing O, a hollow stem P guided in the end cover Q of the cylinder. Pull rod R has a spherical end to allow for angular motion of lever S. A spring, T, assists the prompt release of the brake.

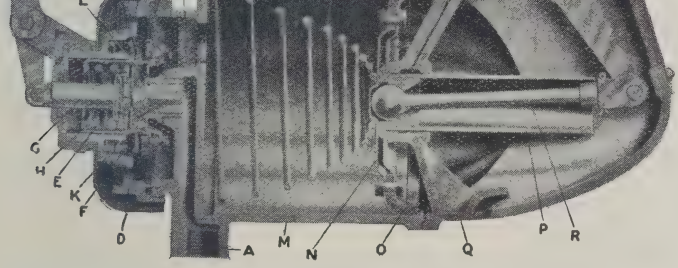


Fig. 26.

Shaft V is extended through the casing Q and carries a double-ended lever, one end of which may be connected to the front brakes and the other to the rear brakes.

The control valve consists of a diaphragm C open on one side to the atmosphere by orifices D, and on the other side with the brake cylinder by orifice B.

The hollow guide E, of which the interior always communicates with the atmosphere by orifices F and D, is fixed to diaphragm C. Valve K controls the passage between the induction pipe and the brake cylinder, and valve L controls the passage from brake cylinder to atmosphere.

In the normal or release position K is on its seat KS, while L is off its seat LS, the brake cylinder being open to the atmosphere by valve L and orifices F and D.

The pull exercised by the brake pedal on G is transmitted through spring H to guide E and diaphragm C which lifts seat LS against valve L thus closing communication between brake cylinder and atmosphere. Guide E continuing its upward movement, valve K is raised off its seat KS, and the end of brake cylinder connected to orifice B is put in communication with the induction pipe of the engine, and thereby subjected to a reduction of pressure, the degree of which determines the amount of pull on the brake rigging.

Laminated Springs.

Size and Number of Plates.

$$\text{Total load on spring W in cwts.} = \frac{sbt^2n}{L},$$

where,

s = maximum stress in plates = 460 cwts. per sq. in.; t = thickness of plates in ins.; b = breadth of plates in ins.; L = length of spring between eyes in ins.; n = number of plates.

Given W and L, and assuming n, values for b and t can be calculated. These must be checked against the rate of deflection, which is determined by :

$$\text{Deflection per ton of load} = \frac{L^3}{40{,}000bnt^2}.$$

From experimental data it appears desirable to so design leaf springs that the plate lengths are proportional to the second moment of area (moment of inertia) of the plate sections, except for strengthening near the eyes to take care of rebound.

Friction in Laminated Springs.

Friction tests show that the friction is a minimum with a lubricant consisting of equal volumes of cylinder oil and powdered graphite. The close proximity of the curve for static and sliding friction with this lubricant are of special note. An important conclusion drawn from the tests is that with graphite μ is substantially constant.

Values of μ under similar loading conditions.

	Light Oil.	Excess Oil wiped off.	Graphite.
Static . . .	0·24	0·34	0·08
Sliding . . .	0·10	0·18	0·06

Effect of End Taper.

Fig. 27 is a diagram showing the relative liability to fracture of various plates in springs as deduced from more than 500 fractures.

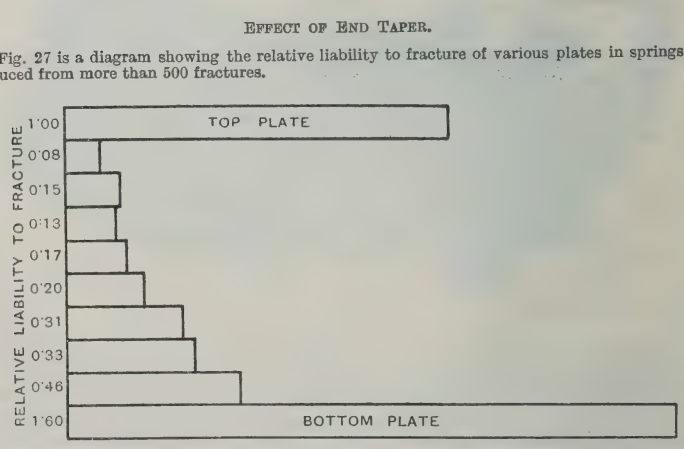


Fig. 27. Diagram showing the relative liability to fracture of spring plates. (*H. S. Rowell.*)

Plates which are reduced in width only are not appreciably different from square-ended plates, such tapering having no practical effect on the strengths, reactions, stresses, or flexibilities of the spring. On the other hand, taper in the thickness has an enormous effect. A combination of the two tapers—in the width and in the thickness—is sometimes of much importance, but generally the effect of tapering the width is very much less than of a taper in the thickness.

The life or fatigue of a spring is to a considerable extent governed by the stress reversal which occurs on account of nip and flick.

NOTE.—For a very complete survey of the subject, papers by Landau and Parr (*Journal Franklin Institute*, U.S.A.) and Rowell (*Proc. Institution of Automobile Engineers*) should be consulted.

Power Required by Motor Vehicles to Negotiate Gradients.

It is first necessary to determine the maximum engine torque T. If d = engine bore in ins., n = number of cylinders, s = stroke in ins., then $T = 5 \cdot 66 d^2 sn$ (approx.). Let S_R = total speed reduction ratio between engine and driving wheels and e = efficiency of driving mechanism, then the torque T_1 transmitted to the driving wheels is given by $T_1 = TS_R e$ (e may be taken as 0·9 for top gear and 0·75 to 0·85 for intermediate and low gears).

Total tractive force F in lbs. $= \frac{2TS_R e}{D}$, where D = diameter of driving wheels in ins.

This tractive force is available to overcome (*a*) resistance due to acceleration or gradients, (*b*) road resistance and wind resistance.

AVERAGE VALUES IN POUNDS PER TON FOR ROLLING PLUS AIR RESISTANCE FOR DIFFERENT CLASSES OF VEHICLES ON VARIOUS ROAD SURFACES.*

Road Surface.	Motor Buses and † Passenger Cars.	Commercial ‡ Vehicles.
Best paved surfaces	37	22
Paved surface in average condition	42	30
Best water-bound gravel	55	40
Ordinary gravel	65	50
Best earth roads	65	50
Fair to good earth roads	75	60

The size of tyre appears to affect the rolling resistance very little so long as the tyres conform to good practice as to size and load. Low-pressure cord tyres have somewhat higher rolling resistance than the high-pressure cords, but the difference decreases for the rougher surfaces.

If

G = gradient per cent. and R = rolling plus air resistance in lbs. per ton, and W = gross weight of vehicle in lbs.,

then, $\frac{2TS_R e}{D} = \frac{W.G}{100} + \frac{W.R}{2240}$; or, $\frac{4480TS_R e}{D} = W(24G + R)$;

from which the engine torque required to drive a vehicle up a given gradient can be determined; or knowing the engine torque, the maximum gradient that the vehicle can negotiate.

---

* Report of Committee on the Economic Theory of Highway Improvements, S.A.E. Highway Research Board, 1925.

† Speed 35 m.p.h., high-pressure cord tyres. For low-pressure cord tyres add 10 lbs. per ton.

‡ Speed 15 m.p.h., pneumatic tyres.

The following values of road resistances are given by G. W. Watson (*Proc. I.A.E.*):

Nature of Road.	Resistance, lbs. per Ton.	Nature of Road.	Resistance, lbs. per Ton.
Good asphalt	15	Soft macadam	97
Poor asphalt	29	Small cobbles	60
Good wood paving	30	Medium cobbles	130
Good granite sets	35	Large cobbles	240
Best macadam	43–46	Sand road	360
Ordinary macadam	50–60	Loose sand	560

Tyres.

Comparison of Tyre Sizes.

Range of wheel load, lbs.	675–1,120	1,000–1,500	1,400–1,900
High-pressure Tyres, ins.	4	4½	5
Inflation pressure, lbs. per sq. in.	45–60	45–60	45–65
Medium-pressure Tyres, ins.	4½	5½	6
Inflation-pressure, lbs. per sq. in.	30–50	38–54	42–58
Balloon Tyres, ins.	5½	6	6½
Inflation pressure, lbs. per sq. in.	27–35	26–38	30–40

Ideal Load and Inflation Pressure.

The following formulæ are based on the Goodyear Company's tables:

If d = section diameter ins. and P = inflation pressure lbs. per sq. in.:

For cord tyres $P = 12 \cdot 5 + 10d$; For canvas tyres $P = 17 \cdot 5 + 10d$.

The ideal load (W) per tyre is, $W = 32 \cdot 4d^{2 \cdot 24}$ lbs.

For any given inflation pressure the variation of permissible load with diameter is given by $W = cpd^{1 \cdot 8}$, where c = 1·81 for cord tyres and 1·63 for canvas tyres.

*Wear of Tyres.**

For high-pressure tyres, road tests show:

(1) Wear increases with speed.
(2) Increased wear with higher temperatures (high temperatures are very destructive to tyres).
(3) Relative wear of rear tyres to front varies from 200 to 118, probable average 150.

For balloon tyres:

(1) Wear is less affected by differences in road surfaces than for high-pressure tyres.
(2) Firm-surfaced roads (*i.e.* asphalt, cement) all have approximately the same effect on tread wear.
(3) Surfaces such as gravel, earth, dry and in good surface condition, cause less tread wear than firm-surfaced roads.
(4) Front tyre wear is greater than rear tyre wear on firm-surfaced roads, and is less than rear tread wear on loose-surfaced roads.

Power Losses in Tyres.

(W. H. Paull.)

The balloon tyre, due to its bigger running deflection, consumes slightly more power than the high-pressure tyre, which it replaces. At 25 m.p.h. the four tyres of a light car will consume about 1·8 h.p. if 3½-in. high-pressure tyres are used at 40 lbs. per sq. in., or 2·4 h.p. if 4·40-in. balloon tyres are used at 25 lbs. per sq. in. A 20 per cent. decrease in inflation pressure would cause these losses to be increased to 2·5 h.p. and 3·0 h.p. respectively. Four 5-in. high-pressure tyres fitted to a large car and running at 50 m.p.h. will consume 7·2 h.p.—6·75-in. balloon tyres on the same car will consume 11·9 h.p. under the same conditions.

* Highway Research Board, National Research Council, Washington.

RATE OF DEFLECTION OF PNEUMATIC TYRES.*

Standard Pneumatic Tyres.	Inflation Pressure. Lbs. per sq. inch.								
	15	20	25	30	40	50	60	80	100
650 × 65	700	525	450	404	341	302	—	—	—
26 × 3	642	492	420	373	297	232	195	—	—
28 × 3½	488	410	355	315	254	207	182	—	—
815 × 105	488	374	316	275	217	177	158	—	—
880 × 120	420	340	290	248	192	154	138	—	—
895 × 135	335	278	237	207	170	147	130	110	—
36 × 6	—	—	—	—	120	103	88	78	75
38 × 7	—	—	—	—	—	—	82	66	60
40 × 8	—	—	—	—	—	—	78	61	52
Balloon Tyre sizes.									
27 × 4·40	600	464	378	332	275	—	—	—	—
28 × 4·95	584	440	370	322	265	—	—	—	—
30 × 5·25	500	390	318	270	218	—	—	—	—
32 × 6·20	464	354	298	250	192	—	—	—	—
34 × 7·30	400	316	263	228	180	—	—	—	—

The above values represent the rate of deflection when any working load is already applied. The numbers are given in millimetres per cwt. increase in load, and in each case the result has been multiplied by 100 to avoid decimal points.

RATE OF DEFLECTION OF SOLID TYRES.

Solid Tyre Sizes.	Normal Load.	Normal Load less 25 per Cent.	Normal load plus 50 per Cent.
90 × 754	31	36	23
100 × 881	25	28	20
110 × 850	23	29	17
120 × 850	19	24	15
130 × 850	15	17	13
140 × 850	18	22	12
160 × 850	13	16	10
180 × 850	11	13	9
8″ × 850	11	13	10
10″ × 850	7	9	6
12″ × 850	6	7	5

THE MACINTOSH N.A.P. UNPUNCTURABLE TYRE.

This tyre, fig. 28, is so constructed that it possesses a large hermetically sealed core containing uncompressed air. It has a broad and flat base similar to that of the ordinary solid tyre, thick, strong, and abnormally flexible walls, and a narrow tread having a central row of triangular cavities about half an inch deep, which give definite non-skidding and side-slipping properties on slippery surfaces. This form of construction is such that it is claimed to need no attention whatever during the whole life of the tyre, which is guaranteed in the United Kingdom to be a minimum of 10,000 miles or twelve months, whichever expires the sooner. Another important feature is that the walls are entirely of rubber and contain no fabric, with the result that exceptionally cool running is obtained. This tyre seems to combine in a remarkably efficient manner the advantages both of the pneumatic and the solid variety. For such vehicles as Ford one-ton trucks it is particularly well adapted.

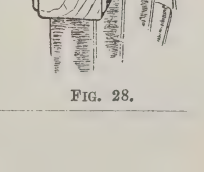


FIG. 28.

* A. Healey, *Proc. I.A.E.*, vol. xix.

MATERIALS USED IN AUTOMOBILE AND AERO-ENGINE CONSTRUCTION.

(*Dr. W. H. Hatfield, 'Proc. I.A.E.'*)

The following table gives the recommended materials for the various components of engine or chassis :—

Part.	Aero-Engines.	Touring and Racing Cars.	Heavy Slow-Speed Vehicles.
Cylinder	MCS	CI, MCS	CI
Cylinder liners	MCS	CI	CI when reqd.
Valve rocker rollers	5NCH	5NCH	5NCH
Valves :			
Inlet	HCr	HCr	3N, HCr
Exhaust	AHRS	AHRS	
Valve guides	HCr, THS	CI, THS	CI
Valve seats	CI, MCS, THS	CI, THS	CI
Valve springs	CVS	CVS, SMS	CVS, SMS
Valve rockers	5NCH, 3N, NCMV	5NCH, NCMV	5NCH
Connecting-rod	3NCS, NCV, 3½Ni	3NCS, NCV, 3½Ni	MS, 3N
Connecting-rod bolts	3N, 3NCS	3N, 3½Ni, 3NCS	3N
Fly-wheel	—	CI, MS	CI, MS
Gudgeon pin	5NCH	5NCH	CHC
Piston	Al, MCS	Al, CI	CI
Crankshaft	3NCS	3N, 3NCS	3N
Oil-pump shaft	CHC, HCr	CHC, HCr	MCS, CHC
Water-pump spindle	CHC, HCr, Nit.	CHC, HCr, Nit.	—
Rotary-pump gears	HCr	HCr	HCr
Keys	HCS	HCS	HCS
Bevel gear shaft for overhead camshaft	5NCH, AHNC	5NCH, AHNC	—
Camshaft and cams	5NCH	5NCH	5NCH
Tappets	5NCH	5NCH	5NCH
Timing wheels	5NCH, Nit.	5NCH, Nit.	MCS, Nit.
Prop. reducing gears	AHNC, NCCH	—	—
Differential spider	—	5NCH, CHC	5NCH, CHC
Gear-box shaft	—	5NCH, NCCH	3N
Transmission gears	AHNC, NCCH	AHNC, NCCH	AHNC, NCCH
Transmission worm	—	5NCH	5NCH
Worm shaft	—	3NCS, 5NCH	5NCH
Crown wheel	—	AHNC	—
Differential pinions	—	CHS, 5NCH	—
Rear axle	—	3NCS	3N, NCM
Front-wheel stubs	—	3N, 3NCS	3N
Steering-arm levers	—	3N	MCS
Steering-arm rods	—	3N	MCS
Steering column	—	Ni tube	MCS
Swivel fork	—	3N	3N
Steering pivot	—	AHNC	MCS
Steering worm	—	AHNC, 5NCH	AHNC
Brake brackets	—	MCI, MS	MCI
Brake drums	—	MCS	MCI, CI
Brake shoes	—	MCS	MCI
Brake operating cam	—	CHC, HCr, Nit.	CHC, Nit.
Clutch plates	—	HCS, Nit.	0·9 steel, MS
Clutch shaft	—	3NCS	3N
Clutch spring	—	CVS	CVS
Axle casing	—	Al, 3N	SC
Chassis frame	—	3N	MS, 3N
Chassis springs	—	CVS, SMS	CVS, SMS
Differential casing	—	Al, MS	MS
Differential gear-box	—	MS	MS
Dumb irons	—	MS	MS
Rear-wheel stubs	—	MS	MS
Front axle	—	3N	3N
Pedals	—	MCS	MCS
Rear-axle housing	—	MCS	MCS
Spring shackles	—	MCS	MCS
Shackle bolts	—	MC, 3NCS	MC, 3NCS
Torque tube	—	3N	3N
Universal joint pin	—	5NCHS	5NCHS

Abbreviations used :—

AHRS = austenitic heat-resisting steel.
AHNC = air-hardening nickel-chrome steel.
ANC = austenitic high nickel-chrome steel.
Al = aluminium alloy.
CI = cast iron.
CHC = case-hardening carbon steel.
CMS = chrome-molybdenum steel.
CVS = chrome-vanadium steel.
HCS = high-carbon steel.
HCr = high-chrome steel.
MCI = malleable cast iron.
MS = mild carbon steel.
MCS = medium carbon steel.
3N = 3 per cent. nickel steel.
NCH = 3 per cent. nickel case-hardening steel.
5NCH = 5 per cent. nickel case-hardening steel.
NCCH = nickel-chrome case-hardening steel.
NCM = nickel-chrome-molybdenum steel.
NCV = nickel-chrome-vanadium steel.
NCMV = nickel-chrome-molybdenum-vanadium steel.
3NCS = 3 per cent. nickel-chrome steel.
Nit. = Nitralloy.
SMS = silicon-manganese steel.
THS = tungsten high-speed steel.

For details of analyses and heat-treatments, see Metallurgical section.

Analysis of Breakdowns, 1926–1928.

(Royal Automobile Club.)

Cause of Breakdown.

(A) Engine.

	1926.	1928.		1926.	1928.
	Per Cent.			Per Cent.	
(1) Cylinders and piston	8·0	10·5	(5) Crankshaft	0·5	0·5
(2) Lubrication	2·3	2·5	(6) Starting	0·6	1·0
(3) Water circulation	1·7	2·2	(7) Valves	0·9	1·4
(4) Valve mechanism (including camshaft, layshaft, secondary and timing gears)	0·9	0·4	(8) Not stated	6·6	5·8

(B) Transmission and Brakes.

(1) Couplings, universal joints, and propeller shafts	4·6	4·1	(3) Gear Box	2·8	2·6
(2) Clutch	4·4	5·1	(4) Brakes	0·3	0·1

(C) Back Axle.

(1) Axle shafts	12·6	13·8	(3) Bevels and Worms	1·2	1·0
(2) Differential	1·2	0·7			
(D) *Ignition*	22·5	22·7	(H) *Lighting failures*	2·1	2·0
(E) *Carburation*	2·5	2·6	(J) *Accidents*	14·1	13·5
(F) *Front axle and steering*	4·3	3·3			
(G) *Road Wheels and Suspension*	5·9	4·2			

ADDENDUM.

Automobiles.

No radical change in general design is to be noted. Minor improvements have been incorporated from time to time. Front wheel drive has met with a certain degree of success in one or two instances; the adoption of this form of drive admits of a lower chassis and consequent overall height of body. These advantages can be obtained by placing the engine at the rear, which removes the objection of taking the drive through the steering swivels. Further, better

tractive adhesion is obtained by taking the drive through the rear wheels than through the front wheels. This is especially noticeable when getting away from rest or climbing a hill. In commercial vehicles the combination of rear and front wheel drive has proved very successful.

The 6-cylinder engine, except for small sizes, is predominant, owing to the smoother running that can be obtained than with a 4-cylinder engine. The 8-cylinder engine is only used on the more expensive cars owing to the added complications and cost.

Side-by-side valves (L-head) construction is again showing increased use, partly owing to the advantages it offers in manufacture, and generally speaking it provides quieter running.

The gear-box has been improved by the introduction of silent seconds and thirds. Central gear change is practically universal; automatic gear-changing devices have been introduced but are not likely to become general, at least in their present form. Free-wheel devices have extending application.

The trend in design as given on page 1935 may still be taken as representative.

The compression-ignition oil engine is receiving great attention, and although there is little prospects of its immediate adaptation to automobiles, it has been used in increasing number of commercial vehicles with definite success.

The advantages of this type of engine are: (*a*) higher thermal efficiency, (*b*) saving in fuel costs. Tests show a saving of over 70 per cent., (*c*) practical elimination of fire risks, (*d*) heavy fuel oil occupies about 16 per cent. less volume than an equal weight of petrol, a consideration where radius of action is concerned.

In comparing the relative merits of the heavy oil engine and the petrol engine, the following must also be taken into account: (*a*) m.e.p., (*b*) maximum fluid pressure in cylinders, (*c*) weight per h.p., (*d*) maintenance costs, (*e*) initial cost, (*f*) fuel consumption at reduced output.

With present practice the advantages as far as (*a*), (*b*) and (*c*) lies with the petrol engine. With regard to (*c*) the A.E.C. engine, which is rated at 95 h.p., weighs complete with all accessories and flywheel, 1,350 lbs., or 14 lbs. per h.p., which is claimed to be no more than a petrol engine of the corresponding size. Regarding maintenance, available information shows that it is no greater than for the petrol engine. The initial cost is at present much higher owing to the limited production, but given equal volumes of production the difference may prove slight.

The fuel consumption at reduced loads with the heavy oil engine is greatly superior to that of the petrol engine. Further, combustion is less affected by variations in r.p.m., so that the variation in m.e.p. or torque and fuel consumption with change of speed is relatively small. The heavy oil engine is more flexible than the petrol engine; sustaining its torque better at low speeds and assuming equal flywheel conditions, has a better acceleration.

The design of the combustion chamber is of great importance. With the petrol engine only sufficient air is taken into the working cylinder to deal with the fuel charge, whereas the oil engine always takes in a full charge of air. Four general types of combustion chamber design are in use: (*a*) open or non-turbulent type, in which the air is compressed into a relatively small space and the fuel introduced in a number of small finely atomised jets; (*b*) the combustion chamber is separated from the working cylinder by a neck or restriction; (*c*) pre-combustion type, in which a small auxiliary chamber is separated from the main combustion space by a number of small holes, and into this the fuel charge is introduced at a comparatively low pressure; (*d*) the rotational swirl, best used with a sleeve valve. In all systems a separate fuel pump is necessary to meter and time the charge to each cylinder, and in general there are two distinct types: (*a*) the ported type where the start and finish of injection are controlled by the pump plunger, (*b*) the valve type, where the operations are controlled by poppet valves. There appears to be little to choose between them as regards performance. There are two main types of injectors: the open type and the closed or differential area type.

The recommended fuel for this type of engine is gas oil, which distils after paraffin (between 200° and 350° C.), the remainder being known as fuel oil. A mixture of gas and fuel oil, known as Diesel oil, is obtainable, and is much cheaper than gas oil. The oil used should be free from solid matter and contain no asphalt. The following variations in properties appear to have little influence on the efficiency of operations: *

Specific gravity	0·84 to 0·91
Carbon content	up to 2·5 per cent.
Sulphur	up to 2 per cent.
Water and sediment content	up to 0·1 per cent.
Flash point	160° to 225° F.

* S. W. Nixon, *Proc. I.A.E.*

SECTION XXXIII

PART I

AIRCRAFT . . . . . pp. 1983–2043

(Contributed by John D. Frier, A.R.C.Sc., F.R.Ae.S.)

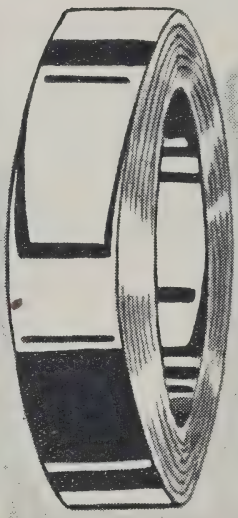
PART II

AERO ENGINES . . . . . pp. 2045–2073

(Contributed by Ernest W. Birch, A.M.I.A.E.)

STRIP STEEL.

IN ALL QUALITIES
IN WIDTHS FROM ¼″ to 30″
IN THICKNESSES ·0015″ AND UPWARDS



Specialities

DEEP DRAWING STEEL
STAINLESS STEELS
RUSTLESS IRON
NICKEL CHROME STEEL
ALLOY STEELS
and
STEEL TO ALL AIRCRAFT SPECIFICATIONS

J. J. HABERSHON & SONS
LIMITED
HOLMES MILLS, ROTHERHAM

SECTION XXXIII

PART I

TYPES OF AIRCRAFT—AERODYNAMICS—AIRSCREWS—SCALE EFFECT AND DYNAMIC SIMILARITY—PERFORMANCE—DESIGN.

(Contributed by J. D. Frier, A.R.C.Sc., D.I.C., F.R.Aë.S.)

A.—TYPES OF AIRCRAFT.

Aircraft fall naturally into two groups, viz. (*a*) Heavier- or (*b*) Lighter-than-air, craft.

The Aeroplane and Seaplane.

The type under group (*a*) is commonly known as the 'Aeroplane' when used for land purposes, or the 'Seaplane' or flying-boat when used for maritime purposes. Machines of this type are initially able to leave the ground or water and subsequently maintain themselves in flight by virtue of their forward motion relative to the surrounding air.

An aeroplane consists essentially of five components:—

1. The main planes, by means of which the aeroplane is sustained in flight.

2. The fuselage or hull, in which the pilot, passengers, and/or other load, such as cargo, petrol, and oil, are carried.

3. The tail, which is rigidly attached to the rear of the fuselage or hull and comprises a tail plane and elevators in a horizontal plane, and fins and rudders in a vertical plane. This component is mainly responsible for the control of the aircraft.

4. The undercarriage, which, in the case of land planes, consists of a braced structure connected to the underside of the fuselage near its front end, supporting an axle carrying wheels, by the aid of which the machine leaves or alights on the ground.

In the case of twin- or multi-engined aeroplanes, this component may be in duplicate.

In getting off and landing, the tail is protected by a small skid attached to the rear end of the fuselage.

In Arctic regions, the wheels are replaced satisfactorily by long skids, while some aeroplanes are convertible into seaplanes by replacing the undercarriage structure by a similar one having two floats instead of wheels.

In the case of flying-boats, the hull serves the dual purpose of the fuselage and the undercarriage. Present-day hulls are manufactured in duralumin, which have a very important advantage over wooden hulls owing to the absence of water soakage; which in the latter hulls may amount to several cubic feet; this equivalent weight is thus available for useful load.

5. The engine, being the unit controlling the height to which a machine may attain. In single-engined aeroplanes it is placed in the nose of the fuselage, so that the airscrew operates as a tractor; installations employing a pusher airscrew, with the exception of the Hill Pterodactyl, are now rarely seen, it being well known that the latter arrangement is less efficient than the former.

In the case of single-engined flying-boats the pusher airscrew is employed because of the necessity of keeping the airscrew well clear of both pilot and water.

When outboard engines are used, the mountings generally form part of the main plane structure.

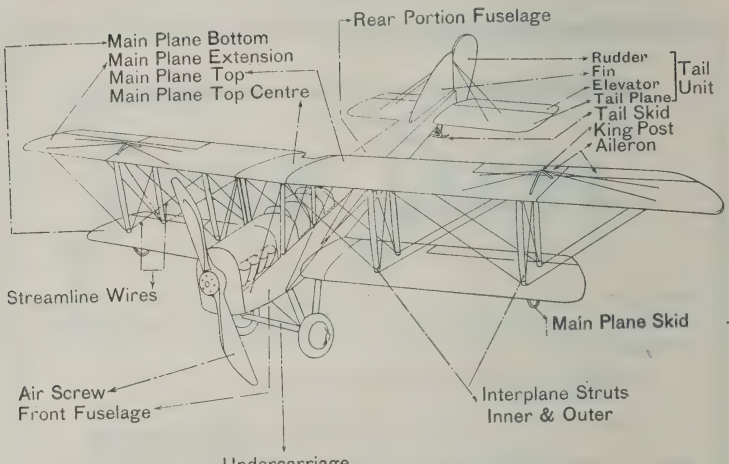


FIG. 1.—Aeroplane Components.

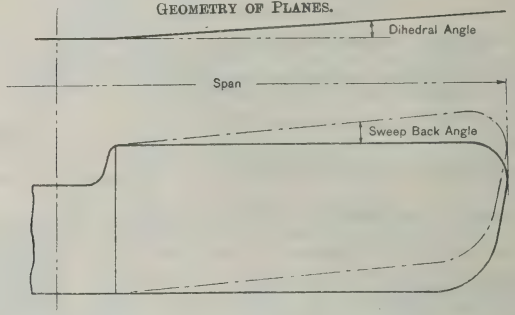


FIG. 2.

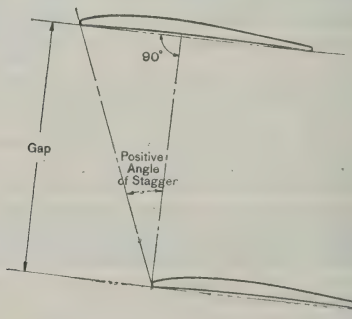


FIG. 3.

The several components are indicated in fig. 1, while the geometry of main planes is indicated in figs. 2 and 3.

Airships and Balloons.

Under group (*b*) aircrafts are classified either as Airships or Balloons, of which the former class is, of course, by far the most important.

Airships have experienced three stages of development, commencing with the non-rigid type, passing to the semi-rigid, and thence to the rigid type.

Both airships and balloons derive their total lift from two sources—viz. buoyancy and dynamic lift. The former is due to the difference in density of the gas in the envelope and the external air, while the latter is due to the relative air speed, and is also affected by the geometrical shape of the envelope.

Kite Balloons.

The free spherical balloon is now practically extinct. Balloons, however, of a certain kind, known as 'kite balloons,' are used for observation purposes. As their name would imply, they are either let up into the air from a fixed point on the ground by means of a cable, or towed behind a ship (as in the late war) for submarine spotting.

The height to which a balloon of this kind may be flown is determined largely by the weight of the cable to which it is attached; thus the balloon ceases to rise when its weight together with the weight of cable is equal to the total lift. Heights between 2000–3000 ft. may be reached.

The shape is maintained by means of the pressure of the gas it contains. When fully inflated, the fabric assumes a streamline shape. One or three fins are fitted, but these are inflated by air pressure, for which purpose a fabric pipe or scoop is directed into the wind.

Non-Rigid Airships.

As in the case of the kite balloon, the shape of a non-rigid airship is maintained solely by the internal gas pressure.

The relative speed of the ship being generally greater than that of a kite balloon, particular attention must be paid to its dynamic loading. Experiment shows that under flight conditions a very high pressure is exerted on the nose. It is not, therefore, surprising if at high speeds the nose is blown in. Fortunately the region of this high-pressure zone is quite local, the pressure falling off rapidly in a longitudinal direction.

The type of ship which survived others of more simple form was devised by Torres in Spain, and manufactured in France by the Astra Co., and known as the Astra-Torres type.

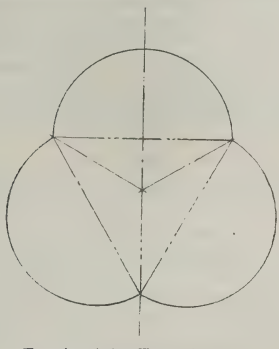


FIG. 4.—Astra-Torres Trefoil Section.

Although in outline the ship has a streamline shape, in cross-section a circular shape is distorted to form a trefoil section having three equal lobes, as in fig. 4. In this way an internal system of triangular bracing is possible, and to this the dead weights of the car and engine are carried by ropes.

The car is slung immediately underneath the envelope somewhat to the front of the ship, provision being made for the engine at the rear of the car, so that a pusher airscrew is employed, or separate engine units may be installed. To guard against any independent pitch or roll of the car, ropes are taken from it fore and aft, as well as port and starboard, to the two lower lobes. In these two lobes small compartments called balloonettes are made with fabric fore and aft, each being fitted with a scoop which may be adjusted to point into the airscrew slip stream. The function of these air balloonettes is to maintain the shape of the envelope when the internal gas pressure is varied. Thus air may be withdrawn or delivered to the balloonettes at the pilot's will. Should the nose be blown in for reasons already stated, the shape is regained by delivering air to

the forward balloonette. To assist the retention of the airship's shape, ribs are usually fitted at the nose in radial directions and taken back several feet behind the plane of the nose.

As is general, automatic gas valves are supplied apart from the pilot's control. The ship is fitted with relatively large rudder and elevator planes, to the rear of which the rudder and elevators are respectively attached.

Semi-Rigid Airships.

In order to relieve the envelope of a non-rigid airship from some of its strains arising from the connection of cabins, power cars, etc., this type of ship has been introduced permitting an envelope of circular cross-section to be used. For this purpose a keel consisting of a braced structure, usually triangular or rectangular in section, is suspended immediately under the envelope, and to this in turn the dead weights of the car and engine are transmitted.

In modern designs, as for instance the "Norge," the keel is practically enclosed in the envelope, in order to produce a good streamline form of low head resistance. In this way very little external rigging is therefore necessary.

Recently, it was suggested, in a proposed design,* to employ a form of twin keel, the two independent structures being manufactured in duralumin and braced together at intervals. The control cabin and a passenger saloon was arranged to be built between the two keels, and the wing power cars mounted on a transverse girder made integral with the twin keel.

Apart from the employment of a keel, a semi-rigid is in other respects similar to a non-rigid airship. The particular advantage of this type of ship over the rigid type is that in case of necessity the envelope may be deflated, the keel dismantled, and the whole transported without much difficulty.

Rigid Airships.

This type was developed in Germany by Count Zeppelin; it consists of a framework of girders whose external form corresponds with the shape of the airship, the covering being directly attached to them.

The early rigids were made with a comparatively long central parallel portion with a short head and tail; the parallel portion underwent successive shortening until the modern perfect streamline shape was evolved. Thus the fineness ratio was reduced from something in excess of 10 to about 8·5 in the case of R33, whose form is shown in fig. 5, while future ships may have this figure reduced further to about 5. In order to obtain minimum resistance, the actual lines of the hull are determined from a consideration of the rates of change of curvature.† The early shape had certain manufacturing advantages owing to its parallel centre portion, but with the new streamline form these advantages disappear.

Structurally, the framework consists of a series of transverse hoops or rings connected to a series of longitudinals. Both the rings and the longitudinals are similarly constructed, frequently

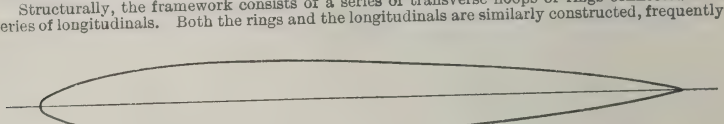


FIG. 5.—Envelope form of R33. Fineness ratio, 8·5.

being triangular in section. The rings are polygonal, having approximately twenty-five sides. Special provision is made for the efficient connection between the rings and longitudinals, while the whole structure is made rigid by a system of wire bracing. A central wire runs from nose to stern, and there are radial wires in the plane of each ring, which pick up the central cable. The space between two rings is occupied by a balloonette; there may be twenty of these balloonettes in all. About half the bags are fitted with a manœuvring valve on top, but each is supplied with an automatic valve, which maintains the hydrogen pressure at some predetermined amount, usually opening when the ship has reached a height of about 20,000 ft.

The navigation car is suspended near the nose of the ship, and is fitted with the usual controls and instruments. Another car placed aft is generally fitted up as an auxiliary control car in case of necessity. Engines may be installed in the rear of these cars. Other cars or gondolas may be employed to carry power units only. Such are suspended out of the plane of symmetry, in order that their airscrews may not operate in the slip stream of those attached to the navigating or after car.

---

* *Flight*, November 17, 1927, p. 790.

† 'Some Modern Developments in Rigid Airship Construction.' By V. C. Richmond, Inst.N.A., March 30, 1928.

Bumpers are generally provided under all cars.

Elevators and rudders are connected to large fin surfaces, which are, of course, fabric covered. The structure of these latter is integral with the hull, in order that the considerable lateral forces, due to the control surfaces, may be sustained and transmitted to it. In the past the control surfaces have been hand-controlled through suitable gearing.

RIGID AIRSHIP R101.*

This airship, intended for Empire communication on a passenger-carrying and commercial basis, embodies a number of features which are entirely new in airship construction.

The ship is capable of non-stop flights of about 3 days, the passengers being accommodated on two decks inside the hull near the centre of the ship.

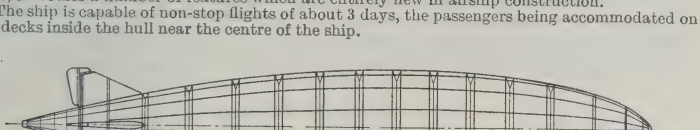


FIG. 6.

A skeleton diagram of the structure is given in fig. 6, brief particulars of the design and construction being as under :—

SHAPE OF ENVELOPE.

This is of streamline form of fineness ratio 5·5, the lowest on record. There is no parallel portion of the envelope as in previous designs.

The shape of the envelope was evolved to give a very low drag coefficient, the latter being very little more (about 2 per cent.) than that of a circular plate of diameter equal to the maximum diameter of the ship. The equation to the profile is :—

$$y = \left[0{\cdot}05549 \left\{ 400 - 0{\cdot}00076x^2 \right\}^{\frac{1}{2}} + 0{\cdot}09968 \right] \left[x \left\{ 20 - 0{\cdot}02762x \right\} \right]^{\frac{1}{2}}$$

where x = distance from bow in feet;
y = ordinate or radius in feet; from the longitudinal axis.

The total length of the ship is 732 ft., and the maximum diameter 132 ft. After initial trials the length was increased to 777 ft.

GAS BAGS.

Including one in the tail there are 16 gas bags, having a total capacity of 5×10^6 cu. ft. A gross lift of 150 tons is obtained.

FRAMES.

In the main, the structure has been manufactured by Messrs. Boulton and Paul, Ltd., Norwich. The frames are of the rigid 'ring' type, requiring no radial bracing in their own plane. The structure of each ring is triangular, there being two outer ridge girders and one inner, all braced together. Again, each girder is, in itself, of triangular form, whose booms and webs are made from stainless strip steel and suitably lightened duralumin respectively.

There are 15 frames, apart from the light conical structure ending with the tail cockpit.

LONGITUDINALS.

There are 15 main longitudinal members bolted to the frames, so as to be entirely outside them. By the introduction of 15 intermediate longitudinals the airship section is a polygon of 30 sides. The intermediate members are intended to assist in taking the lateral load from the envelope, reduce the tension in the latter, and thus help to keep a better shape of envelope.

The longitudinals are braced triangular structures, consisting of Boulton and Paul closed joint tubes of rolled stainless steel strip.

FINS.

The horizontal and vertical fins are built on continuous girders at two frames. To facilitate the fitting of the fins, the section of the ship changes between frames Nos. 12 and 13 (*i.e.* immediately in front of the leading edge of the tail fins) from 30 to 32 sides. To do this the top longitudinal is forked and the others skewed.

* Destroyed by fire at Beauvais in France, October 5, 1930. Official Report, April 1931.

Power Installation.

There are five identical power units, each housed in a car, which are quickly and easily detachable. The engines are of the Beardmore heavy fuel oil type, giving 585 h.p. (normal) and 650 h.p. (max.), at about 1,000 r.p.m.

The main engine is started by an auxiliary petrol engine through a Bendix drive.

There is room for an engineer in each car, and certain repairs may be effected during flight.

Evaporative cooling is employed.

The airscrews are required to be about 16 ft. diameter. Variable pitch hollow steel airscrews have suffered on test under torque variation, and solid, light alloy ones are being developed.

A windmill absorbing 15 h.p. drives an auxiliary generator and air-compressor. The latter is used for trimming the water-ballast and also to pump fuel from the main tanks to the engines, or to shift it from tank to tank to give trim.

About 30 tons of fuel is carried in all, being sufficient for about 50 hours' flight at 75 m.p.h.

Prevention of Corrosion.

All duralumin is treated by the anodic process, and the Boulton and Paul rust-proofing process is applied to steel parts.

B.—AERODYNAMICS.

Wind Channels.

Practically all aerodynamic research is performed with a wind channel, the forces and moments on a model suspended in an air stream being measured. Invariably it is impossible to apply the data thus obtained to the full-scale machine or component (as distinct from a detail such as a wheel or strut) without certain corrections, due to what is known as 'scale effect,' which is dealt with in Part D, p. 2011.

Up to 1923 two principal types of channel had been developed. At Göttingen, Professor Prandtl uses a channel of the 'racecourse' type, *i.e.* one in which the same air is caused by means of a fan to move along a closed circuit of rectangular plan form. The working portion occurs at one of the longer sides. A particular advantage of this type of channel is that it is independent of the size and shape of the room in which it is installed.

N.P.L. Type Channel.

In this country the staff of the National Physical Laboratory has been responsible for the development (since 1910) of the English type of channel. Fig. 7 shows in outline the general layout of a 2 ft. 6 ins. × 2 ft. 0 in. demonstration channel of modified N.P.L. type, constructed at the Northampton Polytechnic Institute, London, for the use of students in the Aeronautical Laboratory.

The N.P.L. type channel is built of wood and supported on a braced steel structure, so that the horizontal centre line is well above the floor. The working portion of the channel is situated towards the end of the front portion of the channel which is fitted with a carefully shaped intake. This part of the channel is generally square in section and about six diameters long. In order to further reduce any possible eddy motion which may have been set up at the intake, and to counteract the tendency of the fan to impart rotation to the air stream, a honeycomb is placed at each end of the channel. This is made of mild steel sheet, and is constructed to give cells 2 ins. to 4 ins. square and 1 ft. to 2 ft. long, according to the size of the channel.

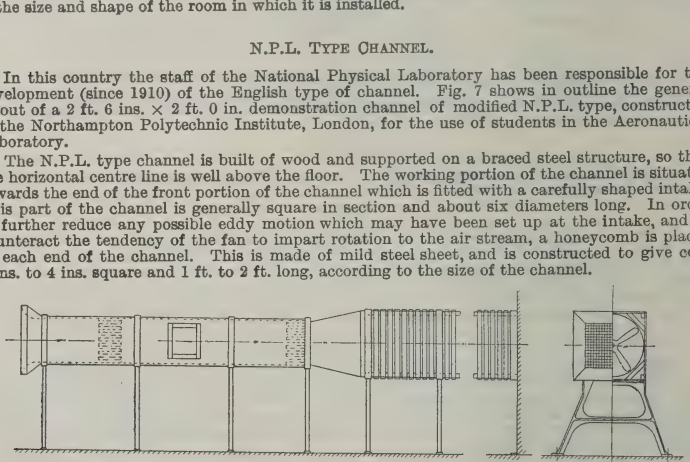


Fig. 7.—Modified N.P.L. Type, 2 ft. 6 ins. Wind Channel. End View of Channel.

The end of the channel at the rear honeycomb is rigidly connected to a metal fan chamber, so constructed that in a distance of a few feet its section changes from that of the channel, viz. square or rectangular, to circular. The four-bladed fan is driven directly by a motor mounted on independent supports, so that no vibration from it shall reach and affect the air stream.

The air thus drawn into the channel is discharged by the fan into another and somewhat larger channel known as the diffuser, which in length is shorter than the channel. It is constructed to have a number of transverse slots in horizontal and vertical planes. The slot widths are adjusted by experiment to give a constant outflow velocity along the length of the diffuser.

This type of wind channel, in which air enters at one end and is discharged through the diffuser at the other, must be installed in a large room free from obstacles which interfere with steady flow.

In the region of the working portion, glazed air-tight doors are fitted for convenient access and visibility.

Under working conditions, the air speed must be maintained as constant as possible. For this purpose the line voltage is regulated to within $\frac{1}{2}$ per cent., while in addition the motor speed is under the complete control of the experimenter. With a carefully shaped intake and a properly adjusted diffuser the velocity of flow in the channel at the working portion will be uniform to within $\pm$ 1 per cent.

Other Types of Channel.

It has long been realised that with the present-day high-speed aircraft, special care must be taken in order to reduce the scale effect corrections which should otherwise be applied to results of model tests. The criterion being that Reynold's number $\frac{vl}{\nu}$ (where v is speed, l some linear dimension, usually the wing chord, and ν the kinematic viscosity of the fluid medium) for the full-scale machine, and the model should be the same, suggests the possible courses:

(*a*) To increase the size of model and/or wind speed of test.

(*b*) To increase the wind speed and decrease the kinematic viscosity of the air by increasing its pressure, *ie.* its density.

The N.P.L. Duplex Channel.

During 1917 the existing 7-ft. wind channels at the N.P.L. were found to be inadequate, and in accordance with (*a*) above the now well-known 'Duplex' channel of section 14 ft. wide and 7 ft. high was designed and built on the lines suggested by experiments on models. In this channel the diffuser takes the special form of a cellular wall across the room. The air speed may be maintained at 120 ft. per sec. by two fans.

The Variable Density Channel.

In America it was decided to work upon the lines indicated in (*b*), and as a result Munk designed a channel to operate with air under a pressure of 20 atmospheres. Fig. 8 shows the form of the

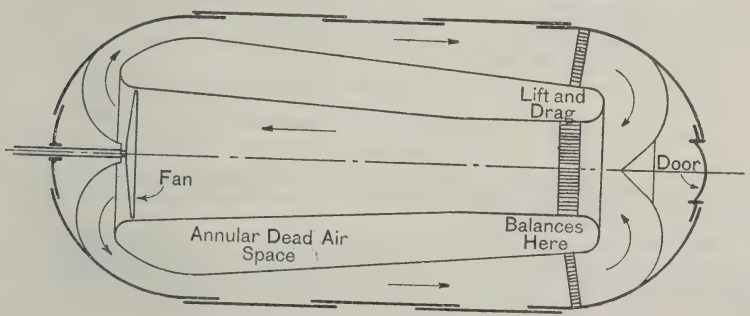


Fig. 8.—N.A.C.A., Variable Density Wind Channel.

channel constructed at Langley Field for the N.A.C.A. (National Advisory Committee for Aeronautics). It consists of a 5 ft. diameter working portion fitted with a honeycomb, the air being delivered by a fan at the 7 ft. diameter intake. Using the 'racecourse' principle, the channel proper is enclosed in a cylindrical steel shell fitted with hemispherical ends. The fan is driven by an externally placed motor, and observation panels are of course provided. Special balances are arranged in a dead air space and controlled from outside the channel. The casing rests on concrete supports. The air-compressor takes about half an hour to raise the channel pressure to the required amount. The channel operates at a Reynold's number of 5×10^6, which corresponds to a vl in free air of about 830. This is in excess of the vl for some full-scale machines.

Wind Channel Constraint.

The theory that the wind channel walls restrict the flow past bodies suspended in the channel for test, necessitating certain corrections to be made to the observed data, has been developed by Prandtl.* For an observed k_L the corrections, owing to the deflection of the air stream, involve (in the case of an aerofoil) small increases $\Delta\alpha$ and Δk_D to the observed α and k_D in order that the results may apply to a channel of infinite size—*i.e.* free air conditions.

The corrections take the form,

$$\Delta\alpha = \delta \frac{S}{C} k_L \text{ radians} = 57{\cdot}3\, \delta \frac{S}{C} k_L \text{ degrees} \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$\text{and } \Delta k_D = \delta \frac{S}{C} k_L^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where,

S = main plane area of model.
C = cross-sectional area of channel.
δ = a coefficient which depends on the shape of the channel.

Values of δ obtained mathematically are given in Table I.

Table I.

Shape of Channel.	Rectangular.			Circular.
Ratio $\frac{\text{height}}{\text{width}}$	$\frac{1}{2}$	$\frac{1}{\sqrt{2}}$	1	—
δ	0·274	0·238	0·274	0·25

Aerofoils.

When the main planes of an aeroplane are considered apart from the rest of the machine they may be referred to as 'aerofoils,' and the shape of a main plane section given by a plane parallel to the plane of symmetry is spoken of as the 'aerofoil section.'

A typical section is shown in fig. 9.

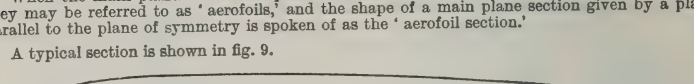


Fig. 9.—Aerofoil R.A.F. 15.

The upper and lower surfaces of an aerofoil are cambered, being arranged to meet in the front to form a blunt nose or leading edge, while at the rear the aerofoil is finished off with a somewhat elongated trailing edge.

The chord of an aerofoil is generally defined as the length of the common tangent to the lower cambered surface cut off by projectors from the leading and trailing edges. In the case of a 'thick' aerofoil section in which the lower camber is convex like the upper, the chord may then be defined as the length of the line passing through the centre of curvature of the leading and trailing edges cut off by the extremities of the aerofoil.

* 'Tragflügeltheorie,' *Göttingen Nachrichten*, 1919.

Fig. 10 shows an aerofoil inclined at an angle of incidence α to the direction of the wind. The resultant force R acting on the aerofoil depends upon α, the wind speed V, and the area of the aerofoil S. Let the line of action of R cut the chord AB in C. The resultant force R may be resolved normal to and along the direction of flight giving a lift force L and a drag force D. The angle between the direction of the resultant force and the lift component is γ. The position of C is determined in the case of model experiments by the measurement of the moment M acting at the point A, the sign of M being reckoned positive when acting in a clockwise direction.

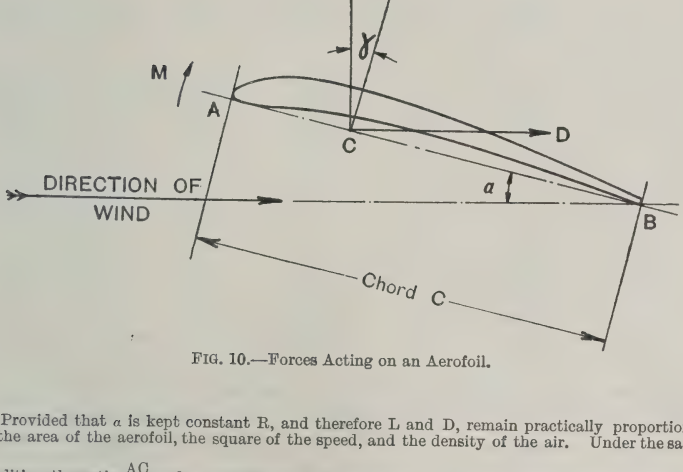


FIG. 10.—Forces Acting on an Aerofoil.

Provided that α is kept constant R, and therefore L and D, remain practically proportional to the area of the aerofoil, the square of the speed, and the density of the air. Under the same condition the ratio $\frac{AC}{AB}$ and γ remain practically constant. If ρ is the mass density * of air, the quantities $\frac{R}{\rho SV^2}$ and $\frac{AC}{AB} \cdot \gamma$ are non-dimensional and also practically independent of the linear dimensions of the model or wind speed during a given test.

The following definitions may now be made :—

$$\text{Lift coefficient } k_L = \frac{L}{\rho S V^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

$$\text{Drag coefficient } k_D = \frac{D}{\rho S V^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$\text{Moment coefficient } k_M = \frac{M}{\rho S c V^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Centre of pressure coefficient—

$$k_{CP} = \frac{AC}{AB} = -\frac{M}{c\,(L \cos \alpha + D \sin \alpha)} = -\frac{k_M}{k_L \cos \alpha + k_D \sin \alpha} = -\frac{k_M}{k_L} \text{ approx.} \quad . \quad (6)$$

* The mass density $= \frac{\text{weight density in lbs./ft.}^3}{g} = 0{\cdot}00237$ slug/ft.^{3} at ground level under standard conditions.

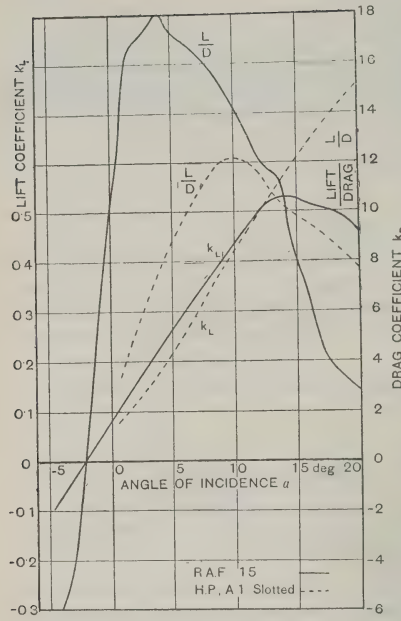


FIG. 11.

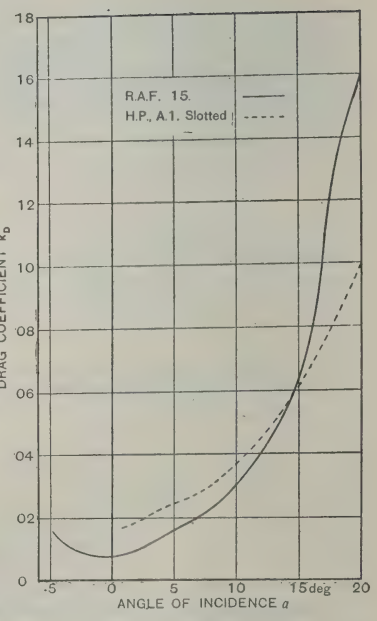


FIG. 12.

It is thus possible to express the aerodynamic characteristics of an aerofoil by plotting, as in figs. 11 and 12, the coefficients in equations (3) and (4) against the independent variable α, the angle of incidence. In experimental work α is given various values between $-5°$ and $+20°$, so that the whole flying range is more than covered.

TABLE II.—AEROFOIL R.A.F. 15.

	Contours.											
Distance from leading edge	0	0·01	0·02	0·03	0·04	0·05	0·06	0·08	0·12	0·16	0·22	0·30
Height of upper surface	0·013	0·027	0·035	0·041	0·045	0·049	0·052	0·057	0·063	0·066	0·068	0·067
Height of lower surface	0·013	0·008	0·005	0·003	0·002	0·001	0·001	0·000	0·001	0·003	0·006	0·008
Distance from leading edge	0·40	0·50	0·60	0·70	0·80	0·90	0·95	0·98	0·99	1·00	—	—
Height of upper surface	0·065	0·062	0·056	0·048	0·040	0·029	0·023	0·017	0·015	0·010	—	—
Height of lower surface	0·008	0·006	0·002	0·000	0·001	0·003	0·005	0·006	0·007	0·010	—	—

The above quantities are fractions of the chord. The shape of this aerofoil is shown in fig. 9.

TABLE II.—AEROFOIL R.A.F. 15 (*continued*).

Aerodynamic Characteristics (R. and M. 954).*

Model, 36 ins. × 6 ins. Tested at 60 ft. per sec. at N.P.L., 7 ft. No. 1 channel.

Wind Channel Results.				Corrected for Channel Interference.		
Lift Coefficient (k_L).	Angle of Incidence ($\alpha°$).	Drag Coefficient (k_D).	C.P. Coefficient ($k_{C.P.}$).	Angle of Incidence ($\alpha°$).	Drag Coefficient (k_D).	$\frac{L}{D}$
− 0·093	− 4·60	0·0144	+ 0·122	− 4·64	0·0145	− 6·41
− 0·063	− 3·60	0·0117	0·132	− 3·63	0·0117	− 5·38
− 0·021	− 2·55	0·0090	− 0·197	− 2·56	0·0090	− 2·33
+ 0·021	− 1·50	0·0080	+ 0·760	− 1·49	0·0080	+ 2·62
0·061	− 0·45	0·0073	0·431	− 0·42	0·0074	8·25
0·101	+ 0·60	0·0077	0·362	+ 0·65	0·0078	12·95
0·144	1·65	0·0084	0·306	1·72	0·0086	16·75
0·185	2·70	0·0104	0·300	2·79	0·0107	17·30
0·221	3·75	0·0119	0·296	3·86	0·0123	18·00
0·259	4·80	0·0143	0·277	4·92	0·0149	17·37
0·296	5·85	0·0168	0·265	5·99	0·0175	16·90
0·334	6·90	0·0198	0·269	7·06	0·0207	16·10
0·371	7·95	0·0236	0·262	8·13	0·0248	14·95
0·443	10·05	0·0306	0·260	10·26	0·0322	13·75
0·510	12·15	0·0410	0·252	12·39	0·0432	11·80
0·527	14·15	0·0553	0·254	14·40	0·0576	9·15
0·514	16·15	0·0821	0·281	16·40	0·0843	6·10
0·493	18·15	0·1373	0·301	18·39	0·1393	3·54
0·443	20·10	0·1619	0·334	20·31	0·1635	2·71

Correcting factors : $\Delta\alpha° = 0·48k_L$; $\Delta k_D = 0·00838k_L^2$.

Table II. gives the contours of aerofoil R.A.F. 15 and its aerodynamic characteristics, which have been corrected for channel constraint.

The correcting factors are obtained in the following way :—

For N.P.L. 7 ft. channel $h/b = 1$ $\therefore \delta = 0·274$ as given by Table I.

Area of model aerofoil $= \frac{36 \times 6}{144} = 1·5$ sq. ft.

Area of channel $C = 7 \times 7 = 49$ sq. ft.

then

$$\Delta\alpha = \frac{57·3 \times 0·274 \times 1·5}{49} k_L \text{ degrees}$$

$$\Delta\alpha° = 0·480\, k_L$$

and

$$\Delta k_D = \frac{0·274 \times 1·5}{49} k_L^2 = 0·00838\, k_L^2.$$

THE SLOTTED AEROFOIL.

The introduction of the slotted aerofoil has been described as the most important invention of the last decade. It was produced simultaneously by Handley Page in this country, and Lachmann in Germany in 1919, both working independently. At that time the primary object was to produce an aerofoil having a high lift coefficient in order to reduce the landing speed of a given machine.

The main aerofoil, throughout its entire length, was made with an adjustable nose of streamline form called an auxiliary aerofoil, which could be moved to or away from the main aerofoil by

* The abbreviation R. and M. refers to Reports and Memoranda of the Advisory Committee for Aeronautics. Published by H.M.S.O.

a hand-operated link motion. With the slot closed the main aerofoil is of normal section. With the auxiliary aerofoil in its forward position the slot is said to be open.

Later, it was found that good results were given if the auxiliary aerofoil were made of duralumin sheet, bent to suit the shape of the nose of the main aerofoil. Fig. 13 shows a leading edge slot of this type fully open. The main aerofoil section is the Handley-Page A.1, which is a slightly modified R.A.F. 15, shown in fig. 9, having increased camber to permit deeper spars being employed. It will be noticed that there is a rear slot provided between the back of the rear spar and the leading edge of the aileron. When the aileron is depressed the slot remains

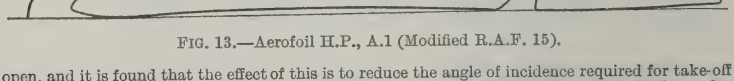


FIG. 13.—Aerofoil H.P., A.1 (Modified R.A.F. 15).

open, and it is found that the effect of this is to reduce the angle of incidence required for take-off or landing from an excessive figure above 20°, to one in the region of 18°. With the front slot open and the aileron at 0°, the results of wind-channel tests (for which k_D and α have been corrected for channel restraint) are shown compared with those for a standard R.A.F. 15 section in figs. 11 and 12. It will be seen that an increase in lift coefficient of nearly 100 per cent. is obtained, but at the expense of an increased angle of incidence. The model used for these observations was 36-in. span by 6-in. chord (6·4 ins. with slot open), and the experiments were performed in a channel 4 ft. × 4 ft. at a wind speed of 44 ft./sec.* It is clear, therefore, that by the use of a leading edge slot extending the whole main plane span, a greatly increased speed range is possible.

As a result of subsequent research, the principle of the slot may be applied to give full lateral control at and beyond the stalling speed of the aeroplane. In this case the leading edge slot is fitted near the main plane tips only, but slightly inboard, over a distance roughly corresponding with the length of the ailerons, but the leading edge of the latter is still so shaped as to produce a slot between itself and the back of the rear spar. Further, for control purposes, the slot need only be fitted to the top plane of a biplane combination. The type of control is known as the slot and aileron control, since the auxiliary aerofoil and the aileron must be operated together. For the successful operation of the gear, the hinge position of the auxiliary aerofoil is very important, as also is the path along which it travels. These points are satisfied experimentally as there is a very wide range of direction of the resultant force on it as the angle of incidence is varied, and it is therefore essential that the interconnection of the auxiliary aerofoil and aileron by means of rods and links is so arranged that there is a closing torque operating on the auxiliary aerofoil. The ailerons are rigged in the usual way, *i.e.* when one is depressed the other is elevated. The corresponding movements of the auxiliary aerofoil are as follows :—

(*a*) With ailerons normal, both front and rear slots are open ;
(*b*) with one aileron depressed, front slot is fully opened and rear slot open ;
(*c*) with one aileron elevated, front slot is closed and rear slot open.

The value of the slot and aileron control lies in the rolling and yawing movements set up. For further information a number of references are given below.†

The latest development is due to the work of the staffs of the Royal Aircraft Establishment and N.P.L. in collaboration with the Handley Page Company, and consists of an arrangement whereby the opening and closing of the auxiliary aerofoil is carried out automatically without the use of link mechanism operated by the pilot.

SLOT AND INTERCEPTOR CONTROL.

Although the automatic slot (which opens or closes according to the angle of incidence) gives lateral stability when the aeroplane is stalled, the ailerons are not powerful in operation.

To improve control, a large yawing moment of correct sign is required—a state of affairs which demands that the slot on one side of the machine is closed or inoperative in its action when the aileron is moved up on that side.

It has been pointed out‡ that to avoid the complicated interconnection of aileron and slot, the desired effect may be obtained by an 'interceptor' connected to the aileron.

Under stalled conditions the interceptor considerably increases the power of the ailerons. It consists of a narrow hinged plate, independent of and behind the slot, and arranged to coincide normally with the main plane section. Through its hinge, the interceptor is connected to the aileron, so that when the latter is moved up the interceptor plate also moves up and comes into a vertical position. In this way the air flow is disturbed and burbling commences as though the slot were closed, thus causing that particular part of the main plane to stall, but giving the desired effect on control.

* *Flight*, January 28, 1926. 'Tests on Slotted Aerofoil.' F. Handley Page.
† R. and M. 1047 and 1048.
‡ 'The Sixteenth Wilbur-Wright Lecture.' By F. Handley Page, *Journal R.Ae.S.*, August 1928.

Factors affecting the Aerodynamic Qualities of an Aerofoil.

Quite apart from its geometrical properties, the results of tests on an aerofoil lose much of their value unless certain data are specified. The model chord, aspect ratio, and wind speed should be quoted, as these quantities have been shown to have a considerable effect on the characteristics. Experiments at the N.P.L. have in the past been undertaken to ascertain the extent of the variations involved when an aerofoil of given section is tested as a monoplane.

(*a*) Changes of Aspect Ratio.

These tests were carried out with models of the same chord, but different spans. Both k_L and $\frac{L}{D}$ were found to be affected. The maximum valve of k_L increases with the aspect ratio, as does the gradient of the curve for which k_L is plotted against α. This effect is still in evidence for aspect ratios well in excess of 10 or 12, but it is most serious for low values up to about 5, becoming less important as the aspect ratio is increased above this amount.

The $\frac{L}{D}$ variation is very large for valves of α above 2° or 3°; below this region the effect is, however, small. For a particular model, by changing the aspect ratio from 4 to 6 it was possible to obtain a 20 per cent. increase in the value of $\frac{L}{D}$. Hence large aspect ratios appear to be desirable from an aerodynamic point of view, but as a rule structural considerations limit the aspect ratio to a figure between 5 and 6.

(*b*) Changes of Wind Speed.

The effect of speed changes may be very serious to both k_L and k_D. Generally both the coefficients increase with speed, though not necessarily over the whole range of the test.

The $\frac{L}{D}$ variation is much greater than that of k_L. Here, the effect tends to reach a limit as the speed is increased, say, from 20 ft. per sec. to 60 ft. per sec. This variation is due to scale effect, and in Part D it is shown that for a series of models of the same aerofoil, but having different chords, the same coefficients are obtained, if for each model the speed is so arranged that the product of speed and chord (*i.e.*, V.*l.*) is the same in every case.

BIPLANE AND TRIPLANE EFFECT.

The aerodynamic characteristics of a given aerofoil when tested as a monoplane are no longer realized when tested as a biplane arrangement. In the case of a triplane a further loss of efficiency is experienced. Approximate results are given in Table III., which show the falling off of both $k_{L\,max.}$ and $\frac{L}{D}_{max.}$ due to the mutual interference of the planes. The justification of the general use of the biplane arrangement in practice lies in its structural advantages.

Table III.—Biplane and Triplane Effect on R.A.F. 15 (R. and M. 432).

Each plane 3 ins. × 18 ins. No stagger. Gap/Chord = 0·75. Wind speed, 40 ft./sec.

	Monoplane.	Biplane.	Triplane.
$k_{L\,max.}$	0·54	0·48	0·44
$\frac{L}{D}_{max.}$	16·5	13·0	11·5

Experiment shows that in a biplane, rounded tips, dihedral angle (if less than 10°), and sweep-back (if less than 20°), have practically no effect on k_L and k_D. On the other hand, however, the gap/chord ratio and stagger are important.

(*a*) Changes of Gap/Chord Ratio.

In experiments on a biplane the gap/chord ratio was varied between 0·67 and 2·33, and in all cases there was no stagger. A summary of the results is shown in Table IV.

Table IV.—Changes of Gap/Chord Ratio for R.A.F. 6c Biplane (R. and M. 196).

Each plane, 3 ins. × 18 ins. Zero stagger. Wind speed, 40 ft./sec.

	Gap/Chord Ratio.					
	0·67	1·00	1·33	1·67	2·00	2·33
k_L max.	0·531	0·562	0·570	0·600	0·598	0·601
k_D min.	0·0131	0·0133	0·0132	0·0132	0·0128	0·0124
$\frac{L}{D}$ max.	14·1	15·3	16·0	16·1	16·7	17·2

It will be seen that the maximum lift coefficient definitely increases with the gap, while the minimum drag coefficient tends to fall.

When the angle of incidence is greater than 1°, for a given lift coefficient, $\frac{L}{D}$ falls as the gap/chord ratio is decreased.

On the whole, there seems little to gain by using a gap/chord ratio in excess of, say, 1·33. Here again, however, structural considerations alone prevent the use of long interplane struts; generally, in practice, a gap/chord ratio in the region of unity is adopted.

The necessity of using a relatively small gap with its attendant loss of efficiency may be partly overcome by the use of a lower plane of smaller chord than the upper.

(*b*) Variations of Stagger.

Using the same aerofoils as in the previous set of experiments, observations of lift and drag were taken for three angles of stagger. The results are shown in Table V.

Table V.—Changes of Stagger for R.A.F. 6c Biplane (R. and M. 196).

Each plane, 3 ins. × 18 ins. Gap/chord, 0·9. Wind speed, 40 ft./sec.

	Angle of Stagger.		
	+ 30°	0°.	− 29°.
k_L max.	0·585	0·555	0·525
k_D min.	0·0133	0·0133	0·0137
$\frac{L}{D}$ max.	14·8	14·6	14·9

For angles of incidence up to about 5° there is practically no aerodynamic difference between the three arrangements. At larger values of k_L the negatively staggered biplane appears to be inferior. The positive stagger increases the lift coefficient, while negative stagger with this aerofoil reduces it.

The Distribution of Forces between the Planes of a Biplane Combination.

Using a R.A.F. 15 combination, the lift, drag, and centre of pressure coefficients have been obtained for angles of stagger of + 30°, 0°, and − 30°.

Regarding the efficiency of a plane in a biplane combination as the ratio of its lift to that of the combination, fig. 14 is obtained by plotting the efficiency of the top plane for various angles of incidence for each stagger considered, and joining the corresponding points with a smooth curve. Assuming this procedure allowable (in the absence of other intermediate values), the following observations may be made :—

(*a*) For zero stagger, the upper plane is always the more efficient.

(*b*) Except for large angles of incidence of a combination employing a large negative stagger, the efficiency of the *forward* plane is the greater.

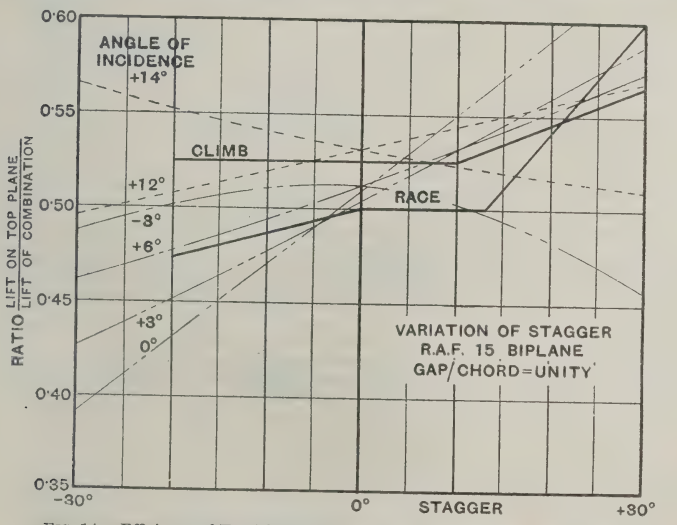


FIG. 14.—Efficiency of Top Plane in Biplane Combination. (R. and M. 589.)

The mean curves in fig. 14 for average conditions of race and climb correspond to those given in the Air Ministry's ' Handbook of Strength Calculations.' *

Reference to the original work in R. and M. 589 shows that the centre of pressure coefficient of each plane changes in the usual way, varying from about $0{\cdot}26\,c$ at $\alpha = 14°$ to $0{\cdot}40\,c$ at $\alpha = 1°$. For a given angle of incidence the C.P. coefficient for both planes is approximately the same.

Distribution of Pressure over the Chord of an Aerofoil.

In order to determine the manner in which the pressure varies over the chord of an aerofoil on both upper and lower surfaces, a special model is prepared, and measurements made of the fluid pressure at various places. The model is preferably made of metal, and the surface drilled deep enough at the desired places, so that all the holes are connected to a common tube, which is finally connected to a Chattock gauge. The holes are filled with plasticene, and one opened at a time, so that the local pressure may be taken. It has been shown that the size of the hole is unimportant; generally it is $\frac{1}{64}$ in. diameter. In this way the distribution of pressure over any chord may be obtained. Except for a region near the tips, the pressure distribution across the chord for the whole

* Air Publication 970, obtainable from H.M.S.O. Price 3*s.* 0*d.* net.

length of plane (of constant chord) is indicated in fig. 15 for two angles of incidence. This refers to experiments on R.A.F. 15 biplane of 6-in. chord, the results of which are given in R. and M. 355. Generally there is a suction on the upper surface and a smaller pressure on the lower surface. Both the positive and negative pressures reach their greatest values near the leading edge where the change in pressure is very rapid. At the point on the nose normal to the air stream, the pressure

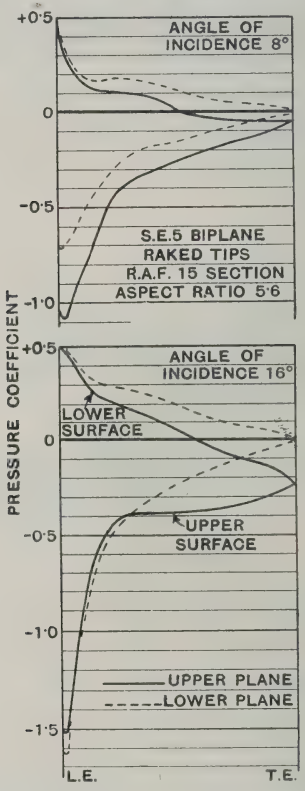


FIG. 15.—Distribution of Load over Chords of Biplane. (R. and M. 355.)

is known to be $\frac{1}{2}\rho V^2$ lbs. per sq. ft. In fig. 15 the coefficient of ρV^2 is plotted for each point, and it is clearly shown that near the nose of the upper surface of the aerofoil the pressure rapidly falls to a local value of approximately $-1{\cdot}5\rho V^2$.

Distribution of Pressure over the Span of an Aerofoil.

The investigation described in the previous paragraph was extended in order to find out how the intensity of load per foot-run of span varied over the whole of the top and bottom planes. For angles of incidence of 8° and 16° the results are exhibited in fig. 16, in which, for convenience, the ordinates are coefficients which represent the normal force per foot-run of span per foot-chord divided by ρV^2.

As would be expected, the normal force on the bottom plane is generally less than that for the top plane, while, with the exception of a temporary rise near the tips, there is a general falling off in loading as the tip is approached.

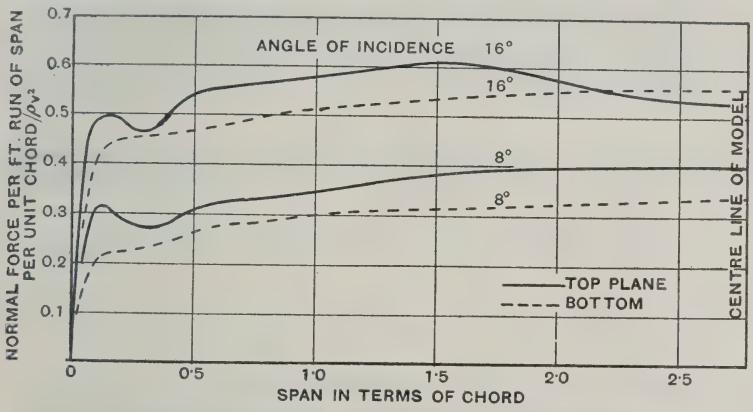


FIG. 16.—Variation of Normal Force along Span. (R. and M. 355.)

By comparison with a similar investigation on a R.A.F. 6 biplane with rounded tips, it appeared that the raked tips of the S.E. 5 biplane now under consideration gave approximately the same type of diminution of normal force towards the main plane tips.

Distribution of Pressure over Airship Envelope.

In the case of a model consisting of a solid of revolution, pressure plotting is carried out in the usual way by arranging holes along one generator, all being connected to a single tube. The

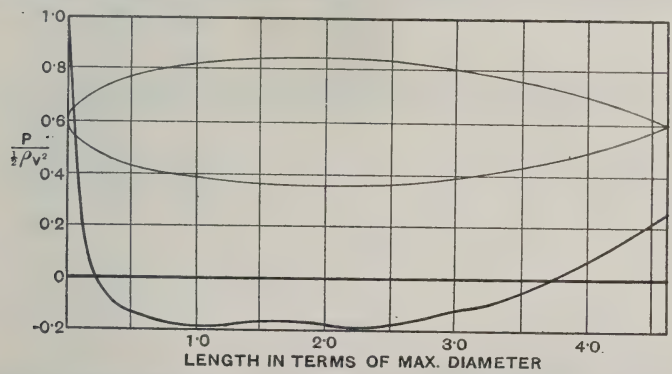


FIG. 17.—Pressure distribution on Model Airship Envelope. Axis along Wind. (R. and M. 600.)

difference in pressure between the point on the model and the suction hole in the channel wall are measured and corrected for the drop in static pressure which occurs along the direction of the axis of a model of this kind, whose length may be several feet.

For a model 39·7 ins. long and maximum diameter 8·59 ins. (fineness ratio, 4·62), at a wind speed of 40 ft./sec., the type of pressure distribution for the axis along the wind is shown in fig. 17.

As would be expected, the coefficient of ρV^2 at the nose is 0·5, which, however, rapidly falls to zero, and changes sign at a short distance behind the nose. The coefficient remains at a negative value until a point near the tail is reached. It will be seen that the maximum pressure coefficient at the tail is only 25 per cent. of that at the nose.

The diagram illustrates very clearly the necessity of strengthening the nose of airships.

When the direction of the wind is inclined to the axis of the model the pressure distribution undergoes important changes. At any cross-section the pressure is no longer uniform round the circumference but varies considerably.

A graphical representation of the kind of distribution is given in 'Applied Aerodynamics' (L. Bairstow), p. 211.

C.—THE AIRSCREW.

A typical wooden airscrew is shown in fig. 18. The shape of the blade changes continually along its length, but at all cross sections the resemblance to an aerofoil is maintained, except that near the boss it becomes more or less streamline.

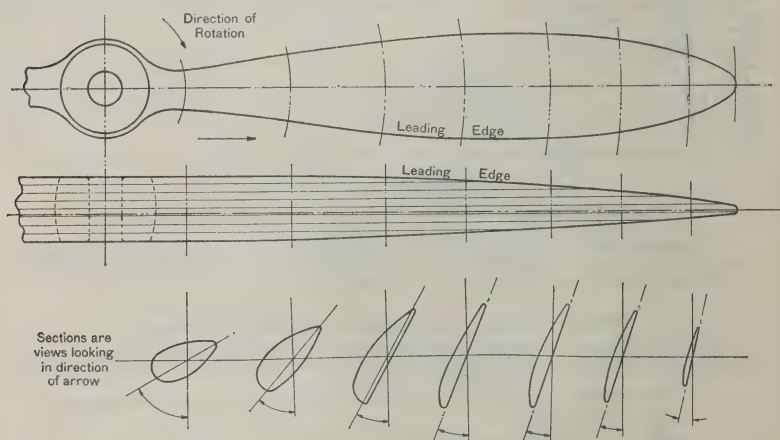


FIG. 18.—Typical Wooden Airscrew showing Laminations.

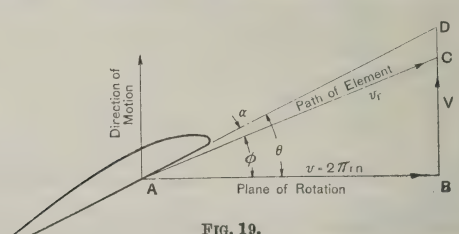


FIG. 19.

Consider an element of the blade as in fig. 19, and let

V = velocity of translation of element, *i.e.* forward velocity of aircraft;

n = rotational airscrew speed;

r = radial distance of element from axis of rotation;

α = angle of attack (or incidence) of blade element to helical path;

ϕ = angle between direction of relative wind at blade element, *i.e.*, the plane of rotation;

θ = blade angle of element, *i.e.*, the angle between chord of element and plane of rotation.

so that $\theta = \alpha + \phi$.

The triangle ABC is development of helical path of element, in which AB represents linear velocity ωr or $2\pi rn$ in the plane of rotation ; BC represents V ; and AC, the path of the element, represents the relative wind velocity v_r.

Thus $$\tan \phi = \frac{V}{2\pi rn}, \quad \text{or,} \quad 2\pi rn \tan \phi = V.$$

Now an airscrew is designed for a definite value of V corresponding to race or climb conditions, hence in a given airscrew $2\pi rn \tan \phi$ is constant at all points along the blade, and hence $\tan \phi$ varies inversely as r, and each element describes a helix of the same pitch.

Slip.

The analogy of a nut working on a bolt breaks down in the case of an airscrew for two reasons, viz. :

(*a*) For a given rotational engine speed the aeroplane, and therefore the airscrew, may have different translational speeds ; and

(*b*) As the air passes the plane of the airscrew it receives an additional backward velocity and the airscrew is said to slip and the stream of air behind the airscrew is called the slip stream.

Pitch.

Considering the airscrew as a whole the term ' pitch ' requires careful definition. Two pitches are in use.

(i) *The Geometrical Pitch.*—Suppose in a certain airscrew each blade element had no angle of attack as in fig. 20. In unit time each element moves forward V feet, but travels along its path

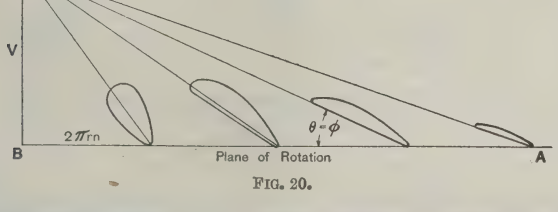


Fig. 20.

in the plane of rotation a distance $2\pi rn$. In these circumstances the advance BC of the airscrew per revolution is called the ' Geometrical Pitch ' P_g and the airscrew is said to have a constant pitch.

Usually, however, airscrews are made with varying angles of attack α along the blade (fig. 19), thus the geometrical pitch of each element varies, and the airscrew cannot be said to have a geometrical pitch.

In such a case the geometrical pitch of an element situated at two-thirds radius from the axis is called the ' Geometrical Mean Pitch ' for the airscrew, *i.e.*,

$$BD = P_g = 2\pi . \tfrac{2}{3}D \tan \theta = 2 \cdot 1 \ D \tan \theta,$$

where D = airscrew diameter.

(ii) *The Experimental Mean Pitch.*—Apart from other factors the thrust of an airscrew depends upon its rotational and translational speeds, n and V.

For any airscrew, at certain corresponding values of n and V the slip-stream entirely disappears, the air in the vicinity of the airscrew remaining practically undisturbed so that the thrust falls to zero. The value of $\frac{V}{n}$, or the advance per revolution when the airscrew thrust is zero, is called the ' Experimental Mean Pitch ' P.

Under working conditions when the airscrew is developing thrust at speeds V_1 and n_1 the slip is given by

$$\frac{V_1}{n} + \text{slip} = P.$$

Airscrew Characteristics.

The thrust T and torque Q of an airscrew are shown in Part D to depend upon ρ, V, n, and some linear dimension, generally taken to be the diameter D.

Provided that $\frac{V}{nD}$ (the advance per revolution as a fraction of the diameter) is constant, T is practically proportional to the mass density ρ, the square of the rotational speed n, and the fourth power of the airscrew diameter D.

∴ We may put

$$\text{Thrust coefficient } k_T = \frac{T}{\rho n^2 D^4} \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

similarly,

$$\text{Torque coefficient } k_Q = \frac{Q}{\rho n^2 D^5} \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

where k_T and k_Q are both non-dimensional, varying only with the value of $\frac{V}{nD}$ provided that the tip speed does not approach the velocity of sound.

Hence, in the same way that k_L and k_D depend upon the angle of incidence α for an aerofoil, so k_T and k_D depend upon $\frac{V}{nD}$ for an airscrew. The complete performance of an airscrew may therefore be expressed in terms of these quantities by plotting experimental results.

A typical set of characteristic curves is shown in fig. 21.

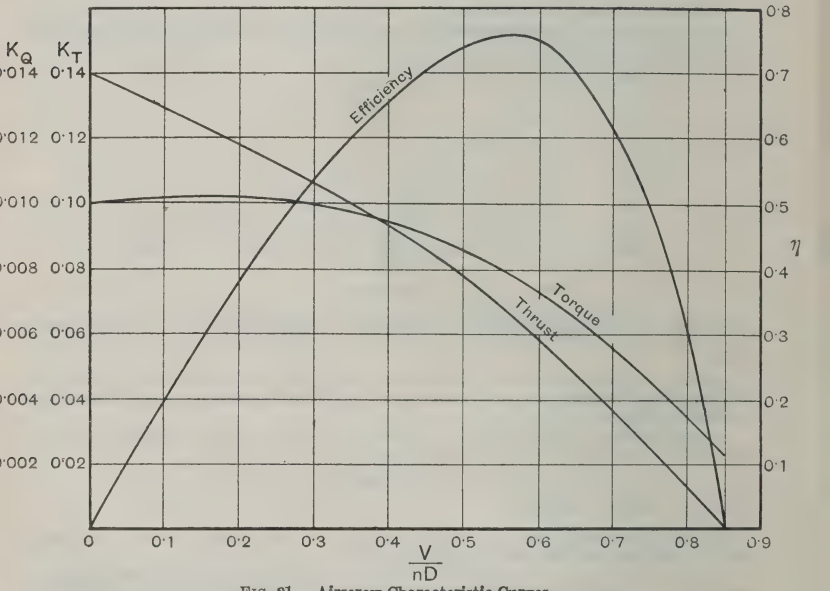


FIG. 21.—Airscrew Characteristic Curves.

The efficiency curve is obtained from the known values of k_T and k_Q from the relation

$$\eta = \frac{1}{2\pi} \cdot \frac{V}{nD} \cdot \frac{k_T}{k_Q} = \frac{1}{2\pi} \cdot \frac{k_T}{k_Q} \cdot J \quad . \quad . \quad . \quad . \quad . \quad (9)$$

where $J = \frac{V}{nD}$.

It is seen that the thrust is maximum when the airscrew has no translational speed V, falling off to zero at a certain value of J. This value corresponds to the experimental mean pitch P as a fraction of D.

The torque remains tolerably constant, first increasing slightly then falling off to a low value when the thrust is zero.

The efficiency is zero when $V = 0$ and also when $T = 0$.

The working range of this airscrew corresponds to values of J between 0·40 and 0·65, the maximum efficiency being reached when J is between these values. Except in special cases the value of J at top speed does not exceed unity.

Airscrew Theories.

A number of theories have been put forward by means of which expressions for thrust, torque, and efficiency may be obtained. Among the most important are :—

(*a*) The Froude momentum theory.
(*b*) The application of Bernouilli's theorem.
(*c*) The simple aerofoil theory.
(*d*) The inflow theory.

The Froude Momentum Theory.

In propounding this theory Froude supposed the actual airscrew replaced by an ideal one consisting of an imaginary disc in whose plane an instantaneous change of pressure occurred under working conditions.

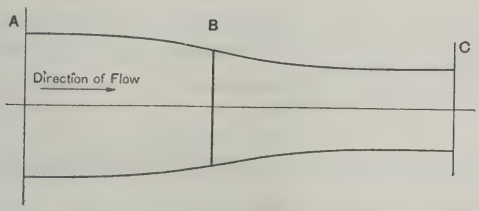


Fig. 22.

Regarding the airscrew as stationary, the ingoing and outgoing streams are indicated in fig. 22. On the upstream face of the disc there is a region of negative pressure which causes motion, and on the downstream face the pressure instantaneously changes to a positive value which increases the pressure of air in the outgoing stream or slip-stream.

Assuming frictionless streamline flow, and considering sections A and C sufficiently removed from the disc B such that the streamlines are parallel, tabulate the quantities concerned :—

Section.	Velocity.	Cross-sectional Area of Stream.	Pressure.
A	V_1	A_1	p_1
B	V_0	$A_0 = \frac{\pi}{4}D^2$	
C	V_2	$A_2 = KA_0$	p_1

The outgoing stream contracts, making the slip stream velocity V_2 greater than V_1.

Thus

$$V_2 = V_1(1 + b), \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

where b is the outflow factor.

Neglecting losses, it may be shown that,

$$V_0 = V_1\left(1 + \frac{b}{2}\right)$$
$$= V_1(1 + a_1) \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

where a_1 is the inflow factor.

Hence, in the ideal airscrew, half the added or slip stream velocity is imparted to the air immediately in front of the airscrew.

THRUST.

Since the thrust is given by the rate of change of momentum of the air stream, it may be deduced that,

$$T = \rho K A_0 (1 + b)\, b V_1^2 \quad . \quad . \quad . \quad . \quad . \quad (12)$$

where T is the thrust.

EFFICIENCY.

The ideal efficiency may be given as,

$$\eta_f = \frac{\text{work got out per sec.}}{\text{work got out per sec.} + \text{added kinetic energy of slip stream}}$$

from which

$$\eta_f = \frac{1}{1 + \frac{b}{2}} = \frac{2}{2 + b} = \frac{1}{1 + a_1}$$

which is known as the ideal or Froude efficiency. Generally the actual efficiency is between 80 per cent. and 85 per cent. of this ideal value.

THE APPLICATION OF BERNOUILLI'S THEOREM.

A consideration of the instantaneous pressure difference at the plane of the airscrew leads to a result identical with equation 11, so that in the ideal airscrew the theoretical inflow factor a_1 is half the outflow factor b. In the inflow theory it is assumed that under all conditions $a_1 = \lambda b$.

Actually λ is less than $\frac{1}{2}$ being more nearly $\frac{1}{2}$.

The Bernonilli and momentum theories may be combined to give results which agree fairly well with practice, so far as thrust is concerned.

The Simple Aerofoil Theory.

Considering the airscrew as made up of independent aerofoil elements moving along helical paths forms the basis of this theory proposed by Froude and subsequently developed by Lanchester and Drzewiecki.

Axial inflow as indicated in the momentum theory is neglected, the velocity of air relative to the airscrew and immediately in front of its plane being taken as V, the translational speed of the aircraft.

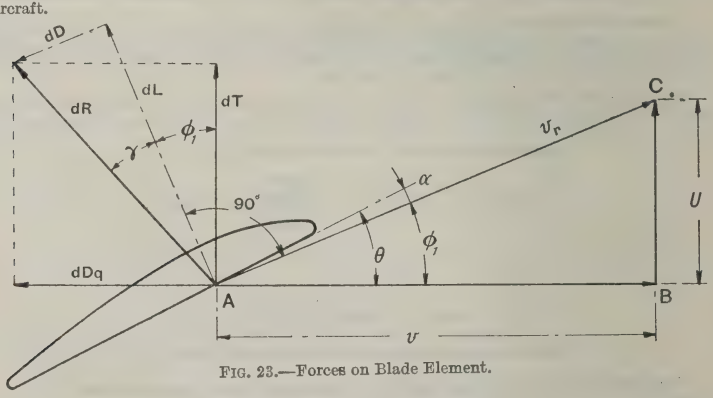


FIG. 23.—Forces on Blade Element.

FORCES ACTING ON BLADE ELEMENT.

Fig. 23 shows the forces acting on an element of blade at radius r from the axis, and substituting V for U and ϕ for ϕ_1, the diagram is applicable to the present case.

We have, treating the element as an aerofoil,

dR = Resultant force on element
dL = Lift " " "
dD = Drag " " "

where,

dL and dD act normal to and along CA respectively.

The direction of dL is in advance of dR by angle γ such that,

$$\gamma = \tan^{-1} \frac{k_D}{k_L}$$

if

dT = thrust on element, *i.e.*, the axial component of dR,
dD_q = transverse component acting at radius r from axis of rotation,
dQ = torque reaction on element,

then

$$dT = dR \cos(\phi + \gamma) \quad . \quad . \quad . \quad . \quad . \quad (13)$$
$$dD_q = dR \sin(\phi + \gamma)$$
$$dQ = r dR \sin(\phi + \gamma) \quad . \quad . \quad . \quad . \quad (14)$$

Efficiency of Whole Airscrew.

The efficiency of the whole airscrew is given by.

$$\eta = \frac{\text{work done by airscrew}}{\text{work done by engine}}$$

which reduces to

$$\eta = \frac{\tan \phi}{\tan(\phi + \gamma)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

On this basis, the efficiency of the element is maximum when $\phi = 45° - \frac{\gamma}{2}$.

Since $\frac{\gamma}{2}$ is small, this means that the maximum efficiency occurs when the rotational and translational speeds of the element are approximately equal.

The efficiency of the whole airscrew may be taken to be that of an element situated at a distance from the axis of about two-thirds tip radius. Thus, if an element so situated is working under the above condition, the airscrew would be working approximately at its maximum efficiency.

The Inflow Theory.

The simple theory, which was the first to be applied to design, generally gave efficiencies which were in excess of those obtained in practice. This led to the adoption of the inflow theory which allowed for the fact that the air immediately in front of the airscrew has a velocity greater than that of translation of the airscrew.

The axial inflow factor a_1 may be obtained experimentally for any given model or it may be calculated

There is experimental evidence that in addition to axial inflow, a rotational inflow also exists acting towards the boss. This is, however, small and is generally neglected.

Helix Angle of Blade Element.

Fig. 23 shows the forces acting on an element of blade, as already described, but in arriving at the helix angle ϕ_1, the velocity of the air passing the airscrew disc is now given by

$$U = V(1 + a_1)$$

and

$$\tan \phi_1 = \frac{U}{\omega r} = \frac{V(1 + a_1)}{2\pi r n}.$$

Efficiency of Blade Element.

The efficiency of the element now becomes

$$\eta_a = \frac{1}{1 + a_1} \cdot \frac{\tan \phi_1}{\tan(\phi_1 + \gamma)} \quad . \quad . \quad . \quad . \quad . \quad (16)$$

Thus the actual efficiency is the product of the Froude and Drzewiecki efficiencies except that U and not V is used in calculating ϕ_1.

The Total Thrust and Torque per Blade.

For any element of blade having chord c, treating the element as an aerofoil, the resultant force dR on it is $dR = k_L \rho c\, dr\, v_r^2 \sec \gamma$,

and

$$\text{Total thrust/blade } T_b = \int_0^R pc \sec\gamma \cos(\phi_1 + \gamma)\, dr \quad . \quad . \quad . \quad (17)$$

$$\text{Total torque/blade } Q_b = \int_0^R pc\, r \sec\gamma \sin(\phi_1 + \gamma)\, dr \quad . \quad . \quad (18)$$

where

p = pressure/ft.^{2} or air loading of blade $= k_L \rho v_r^2$.

In an analysis, the integrals in (17) and (18) are evaluated graphically. For this purpose about six sections or elements of blade are considered, approximately corresponding in position with those shown in fig. 18 and the thrust and torque grading for each element, viz.,

$$pc \sec\gamma \cos(\phi_1 + \gamma) \text{ and } pc\, r \sec\gamma \sin(\phi_1 + \gamma),$$

calculated, assuming $\sec\gamma$ to be unity. These are plotted on a base representing the blade radius, so that the area under each curve represents T_b and Q_b respectively.

The efficiency of the whole blade is,

$$\eta_b = \frac{V}{\omega} \cdot \frac{T_b}{Q_b}.$$

The efficiency of a well-designed airscrew is in the region of 80 per cent.

Plan Form of Blades.

The design process generally involves the choice of a particular plan form of blade, the shape of which is expressed in terms of the maximum chord or blade width c, at stated radii (as fractions of the tip radius) from the boss.

A shape of blade known as the Griffith plan form, in which torsional stresses are low, is practically symmetrical, and is shown in fig. 18.

Diameter of Airscrew Required.

It is generally necessary to fix the diameter of an airscrew and determine the necessary blade widths to absorb the available torque by the use of equations (17) and (18).

In order to avoid a trial and error process as far as possible, a preliminary determination of the required diameter may be obtained by formulæ proposed by Watts.* These are given below :—

Two-bladed airscrews

$$P = 1{\cdot}11 \times 10^{-10} D^4 n^2 V \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

Four-bladed airscrews

$$P = 2{\cdot}30 \times 10^{-10} D^4 n^2 V \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

where

P = h.p. absorbed ; n = r.p.m. ;
D = required diameter in ft. ; V = translational speed in m.p.h.

Note that here V is an estimated speed which will, among other things, depend on the efficiency of the airscrew when designed. In view of this there is every justification for looking upon the first airscrew for a particular case as experimental, making any necessary modifications after the initial trials have been completed.

In a particular design the diameter may be restricted to one smaller than that given by equations (19) or (20) above, for reasons of, say, ground clearance.

Horse-Power Absorbed.

The engine power curve, supplied by the engine manufacturers, gives the power to be absorbed at stated engine revolutions. The corresponding airscrew revolutions are known from the gear ratio of the engine, and at this speed the airscrew torque must be such as to absorb the power developed.

Thus, if

b = number of blades ; P = horse-power absorbed.

$$P = \frac{2\pi\, Q_b\, \text{n.b.}}{33{,}000} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

where Q_b is the torque absorbed per blade as given in equation (18).

In design, the unknown maximum blade width c is determined from equation (21) and the remaining chords at specified sections are immediately known from the plan form adopted.

* *Screw Propellers for Aircraft.* H. C. Watts.

Rotational Speed.

In order that it may develop maximum power, the engine, and therefore the airscrew, must be run at or near a certain rotational speed. For a given flight or translational speed, the airscrew must have its maximum efficiency when the engine is running at that rotational speed. It follows that every combination of engine and aircraft will require its own particular airscrew.

Neglecting inflow, the maximum efficiency of a blade element at radius r from the boss occurs when $\phi = 45°$ approximately.

From fig. 19, $\tan \phi = \dfrac{V}{2\pi rn} = 1$ when $\phi = 45°$.

This means that for the element, $\dfrac{V}{nd} = \pi$ for best results. In any case the whole blade cannot work under this condition of maximum efficiency, and in practice it is not possible to cause the airscrew to rotate slow enough to give a value of π for the pitch diameter ratio.

On the other hand, in order to obtain a light airscrew for a given diameter, the blade widths must be small and the rotational speed high.

For conditions of maximum efficiency, airscrews operate, as a rule, at a value of J between 0·6 and unity.

The question is one of compromise, but, other things being equal, a slow-running airscrew has a greater efficiency than a high-speed one.

Number of Blades.

Airscrews are made with either two or four blades. Attempts to produce three-bladed airscrews do not appear to have given satisfactory results.

The choice of the number of blades usually rests with the designer. The advantages of two- and four-bladed airscrews are given below :—

Two-bladed airscrew—

Weighs very much less than the corresponding four-bladed airscrew.
Easy to transport.
Easier to manufacture than four-bladed airscrew.
Stronger at boss " " "
Cheaper " " "

Four-bladed airscrew—

Smaller diameter than corresponding two-bladed airscrew, which may be an advantage when clearances are important. Better balancing effect so that engine tends to run better and without vibration. Generally somewhat more efficient than two-bladed airscrew.

Manufacture.

Airscrews are made in either wood or metal, the materials being American walnut, Honduras mahogany, duralumin, or steel. In the case of wood, the moisture content should average 13 per cent.

A very great disadvantage of wooden airscrews is that in tropical countries the blade warps, and consequently its efficiency seriously falls.

Metal airscrews, which do not give this trouble, are, on the other hand, much heavier than corresponding wooden ones. It is hoped that before long the present high weights of metal airscrews will be reduced by modifying the method of manufacture.

Wooden Airscrews.

Laminæ.—An airscrew is produced by gluing together the requisite number of laminations, each 20 mm. thick. The shape of each lamination is determined from the drawing, but for the gluing operation each lamination is cut longer than the finished size.

The timber must be straight-grained and free from knots and, in general, must be cut so that the grain runs lengthwise and parallel to the surfaces to be glued. The surfaces are toothed before gluing (about 25 to the inch).

In a four-bladed airscrew the laminations are half-lapped at the boss, but the joints in adjacent laminations are slightly staggered. It is, however, permissible to have one lamination full thickness right through and employ butt joints for the two remaining blades.

The heavy arms of each lamination are marked, so that on gluing up, a heavy arm is in contact with a light one ; this facilitates final balancing.

Gluing.—All laminations are warmed to 70° F., glued together, and left under pressure for 24 hours.

Bore.—It is convenient to bore the airscrew before shaping.

Shaping.—When the airscrew has been out of the press for 2 days it is roughed out in plan form and on the blade face by band-saw, after which it is left for 4 days. After this period it is taken down to size by using templates at the sections indicated on the drawing and the whole blade cleaned up.

Balancing.—The airscrew is arranged on a horizontal mandrel, and tested for static balance. Material may, if necessary, be removed from the face of the blade only. A 1-oz. weight in a bolt-hole must cause the airscrew to turn.

Fabric Covering.—The blade is covered with fabric for about two-thirds the radius from the tip. In the case of a local failure of the timber, the fabric may hold the material together and prevent possible serious damage to the aircraft.

Metal Sheathing.—It has been found advisable to protect the leading edge and tip from rain, hail, etc., by covering them with thin sheet-brass. This is connected to the blade by countersunk screws for which purpose the brass is drilled and punched with a conical depression to receive the screw.

Metal Airscrews.

Wooden airscrews are being rapidly replaced by metal ones, manufactured from light alloy (generally duralumin) or mild steel.

At present they are more expensive and heavier, especially in the case of mild steel.

There are, however, very definite advantages in the use of metal airscrews, among which are :—

(*a*) Damage to blades by rain and hail is very rare.

(*b*) They withstand large and rapid changes in temperature and variations in atmospheric conditions generally, whereas the blades of wooden airscrews twist in tropical countries, causing a falling off in efficiency.

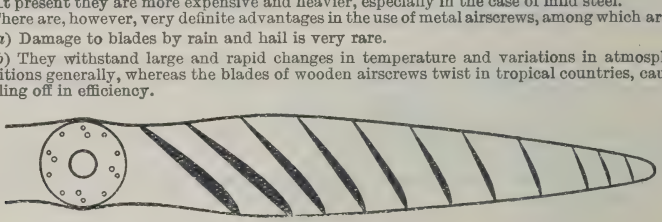


FIG. 24.

(*c*) As the blade tip sections can be made quite thin, as indicated in fig. 24, very high tip speeds may be employed, such as occur in high speed or racing aircraft.

DURALUMIN AIRSCREWS.

These airscrews were introduced by Dr. S. A. Reed in America, and developed and manufactured by the Fairey Aviation Co. in this country. Briefly, the manufacture of the standard D type airscrew is carried out in the following manner :—

1. A flat forged strip of duralumin, with a hub thickness up to 1½ ins., according to the size of airscrew, is sawn to suit the plan form of the blades.
2. By milling or shaping machines the blades are profiled to the desired aerofoil section.
3. The thick portion of the airscrew near the boss is twisted by a hydraulic press.
4. The outer portion of each blade is set by a twisting machine.
5. Blade angles checked.
6. Boss built up from wooden laminations or specially shaped metal blocks, the whole being held together by countersunk bolts.
7. Airscrew balanced on its own hub. A typical finished airscrew is shown in fig. 25.

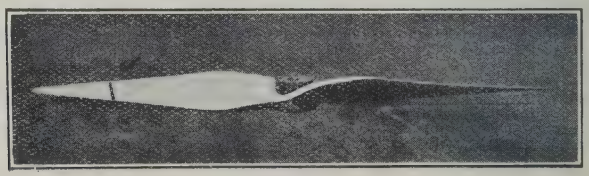


FIG. 25.—A Standard Fairey Metal Airscrew.

Advantages.

Particular advantages of this type of airscrew are :—

(*a*) After preliminary trials the blades may be adjusted to suit the aircraft and engine speeds. The blades may also be altered to suit any subsequent change in these speeds.

(*b*) The single forging avoids joints in the blade just where a joint is least desired. The extra complication of blade attachment fittings are also avoided.

(*c*) In cases of accident these airscrews bend where wooden ones would break. Damaged duralumin airscrews are frequently corrected and put into service again.

(*d*) Bent blades may be straightened up to 5° without thermal treatment.

Blade Stresses.

Cases have been known of airscrews working satisfactorily at a stress of 27,500 lbs./in.², corresponding with a factor of safety of 2.

Fatigue has not been known to occur.

Variable Pitch Airscrews

A variable pitch airscrew improves the performance of an aircraft, since by a small rotation of its blades about their own axis, its efficiency under varying flight conditions may be maintained.

There are three classes of variable pitch airscrews—namely, those in which the pitch is varied by :—

(1) Centrifugal force ;

(2) manual effort of pilot ;

(3) an entirely automatic action.

In (1) the pitch increases with the engine r.p.m., and there is the disadvantage that there is no control over the actual pitch. In take-off, therefore, when the engine speed is greatest and the aircraft speed small, the blades are rotated to give a large pitch when a small one is desired.

In (2) objections arise through the extra work thrown on the pilot, apart from any mechanical difficulties.

In (3) the problem has been solved by the Gloster Helo-Shaw Beacham variable pitch airscrew, which is entirely automatic in action, but under the control of the pilot, so that the engine r.p.m, may be varied to suit prevailing conditions.

The mechanism is so arranged that by means of a wheel or lever the pilot can set the airscrew to run at any desired rotational speed, which will be maintained under all flight conditions. This speed will not vary unless or until the pilot again operates his airscrew speed control.

In fig. 26 the airscrew hub A carries blades B_1B_2, which can rotate in the hub, while the centrifugal force set up by the blades is taken by ball thrust races. The airscrew shaft, C, is attached to the hub through a flanged coupling. A piston, D, is rigidly fixed to the front half of the hub, the former working in a cylinder, E, to which two tie rods, F_1F_2, are fixed. A sleeve, G, is connected to the other ends of these rods, the former being arranged concentrically with the airscrew shaft, so that it can slide fore and aft on bosses projecting from the hub. The sleeve and crank pins, I_1I_2 (which are fixed to the blades), are connected by short rods, H_1H_2. In this way, if cylinder E is displaced, the crank pin and blades are also displaced.

Three concentric oil pipes, 1, 2, and 3, run through the hollow airscrew shaft and rotate with it, for which purpose a rotating oil-tight joint, R, is provided. Pipes 1 and 2 communicate with opposite sides of piston D. The pitch of the blades is controlled, or maintained, by the oil pressure in these two pipes, the pressure variation causing cylinder E to move.

The oil pressure is obtained from a variable stroke pump, J, which, together with a governor are both driven by the engine. Engine lubricating oil is employed.

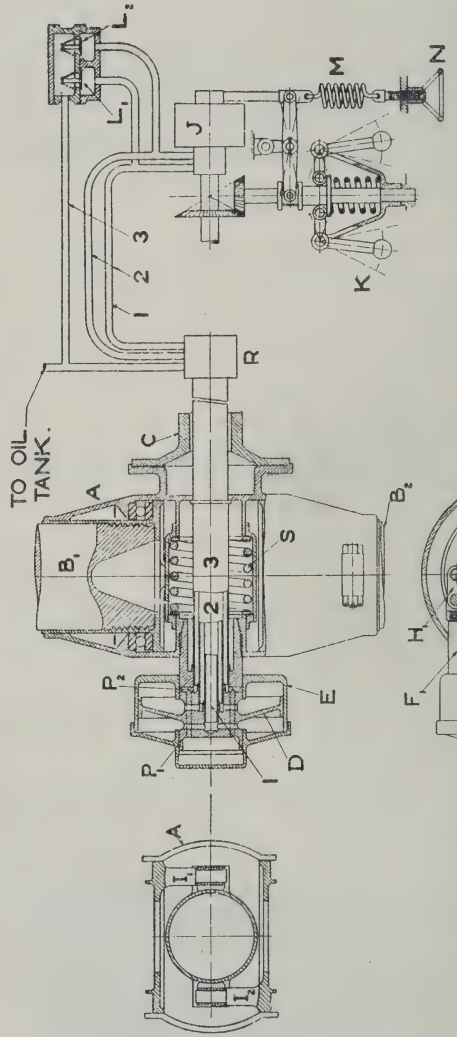


FIG. 26.—Gloster-Hele-Shaw Beacham Variable Pitch Airscrew.

The pump is of the type in which the pressure can be modified or changed to suction by altering the stroke of the pump. The governor is connected by links to the pump stroke rod and the controlling springs of the governor, being so arranged that at normal engine speed the pump has no stroke, and thus there is no oil delivery.

Pipes 1 and 2 are connected to the pump, so that either pipe 1 is on the pressure side and pipe 2 on the suction side, or *vice versa*.

Loss of oil is made up by leading branch pipes from pipes 1 and 2 to non-return valves, L_1L_2.

The speed at which the governor will float is controlled by a spring, M. It is adjusted by a handwheel, N, which also controls the normal speed at which the governor will take up its mid-position.

If the direction of rotation of the airscrew be such that a movement of cylinder E to the left will increase the pitch, then the pump is arranged so that when the governor commences to operate oil is pumped into pipe 1 and sucked from pipe 2.

Suppose the control wheel is set to any desired rotational speed with the airscrew at normal pitch. If, during flight, the forward speed of the machine increases, the r.p.m. will increase and the governor will cause oil to flow into pipe 1 to the front cylinder, thus increasing the pitch. This increases the torque, and the engine r.p.m. decreases, until the r.p.m. is again normal.

The maximum displacement of the cylinder from its normal position is limited in both directions by stops. When either limit is reached the oil pressure is kept normal by the opening of either of the relief ports, P_1 or P_2, in the piston communicating with pipe 3, which allows oil to reach the non-return valves and get back to the pump.

Should the hydraulic system fail a stiff spring, S, in the sleeve returns the airscrew to normal pitch.

D.—DYNAMIC SIMILARITY AND SCALE EFFECT.

Flight tests and experiments on full-scale aircraft are an expensive matter, but in many instances the required information may be obtained from a test on a geometrically similar model in a wind channel.

It does not necessarily follow, however, that because the model and full-scale machine are geometrically similar, there will be similarity of flow of fluids past these bodies, nor similarity of motion of the latter. In other words, there may not be dynamic similarity. The whole of the utility and application of model test results to full-scale depends on the conditions of dynamic similarity being known and satisfied.

Generally the model test cannot be carried out so as to satisfy all the required conditions simultaneously, but as a rule one particular condition has a predominating effect on the motion concerned, and if this is fulfilled the model results may generally be applied to full-scale. Owing to the neglect of the secondary conditions there is frequently a discrepancy between the full-scale results as predicted from model experiment (depending, for instance, on the shape of the body concerned) and those obtained directly from full-scale flight tests. Due to this 'scale effect' a correction has to be applied to the model test figures to bring them into agreement with full-scale results.

Sufficient full-scale tests have been carried out to enable certain scale effects to be determined.

The particular conditions which have to be satisfied when testing models of an aerofoil, a fuselage, flying-boat hull or an airscrew may be obtained by a method developed by Lord Rayleigh, and consists of an examination of the dimensions (in mass, length, and time) of the quantities thought or known to be involved.

A summary of the conditions governing the use of models in experimental work in order to predict the full-scale results is given below.

The Main Planes.

In the case of an aerofoil, the expression for the resultant force acting on it when in a stream of fluid is given by

$$R = \rho l^2 v^2 f\left(\frac{vl}{\nu}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

where

ρ = mass density of fluid (generally air) (ML^{-3});
l = some linear dimension (L);
ν = kinematic viscosity $(L^2 T^{-1})$.

The dimensions of these quantities are shown in brackets.
Also f is some function of the variable to be determined by experiment.

This result represents the resistance to steady motion arising from skin friction, due to the viscosity of the fluid but neglects the effect of compressibility of air which becomes important only when v approaches the velocity of sound v_s in the medium, viz., air. It may therefore be applied to aeroplanes and airships in steady flight.

In equation (22) the term l^2 in the case of an aerofoil is taken to be its area S, so that

$$R = \rho S v^2 f\left(\frac{vl}{\nu}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

Since for aerodynamic bodies, R is proportional to v^2, approximately, $f\left(\frac{vl}{\nu}\right)$ is practically constant for a given angle of incidence of the body.

$$\therefore \quad R = k\rho S v^2.$$

Resolving R vertically and horizontally

$$\text{Lift } L = k_L \rho S v^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

$$\text{Drag } D = k_D \rho S v^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

where k_L, k_D, the lift and drag coefficients, are non-dimensional, since $\frac{vl}{\nu}$ is non-dimensional, and depend upon α.

Owing to the slight inconstancy of $f\left(\frac{vl}{\nu}\right)$ there is only a small correction to be applied to model results in passing to full-scale.

Application of Law to Model and Full-Scale.

Consider model and full-scale geometrically similar machines, for which the effects of viscosity are of paramount importance.

It follows from equation (22) that the resultant force on full-scale machine is given by

$$R_1 = R_m \frac{l_1^2 v_1^2}{l_m^2 v_m^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

if

$$f\left(\frac{v_1 l_1}{\nu}\right) = f\left(\frac{v_m}{\nu} l_m\right) \text{ where suffix } m \text{ refers to the model.}$$

The condition is clearly satisfied if in this case (*a*) $v_1 l_1 = v_m l_m$, or more generally if (*b*) $\frac{vl}{\nu}$ is the same in each case.

A law of corresponding speeds is represented by condition (*a*) in that for no scale effect the required speed of test for a model is given by $v_m = \frac{v_1 l_1}{l_m}$ when the force on the model would be that on the full-sized machine, which may be undesirable.

If in a certain case $l_1 = 10 l_m$, then $v_m = 10 v_1$. This is a clearly impossible condition, for assuming a full-scale machine to have a speed of 150 m.p.h. the speed of air in the wind channel would have to be 1,500 m.p.h.!

In actual wind-channel work the model size and wind speed are arranged to work at a value of $v_m l_m$ as near as possible to $v_1 l_1$ for the full-scale machine.

Condition (*b*) suggests the variable density wind channel (see p. 1989) in which ν_m is decreased by increasing the air pressure.

The Flying Boat.

The resistance to motion of that part of the hull of a ship or flying boat in contact with water is made up of two parts viz. (*a*) skin friction (due to the viscosity of water), and (*b*) wave formation (due to the gravitational effect).

The general equation now takes the form

$$R = \rho v^2 l^2 f\left(\frac{vl}{\nu}, \frac{v^2}{lg}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

Assuming these two effects to be independent

then

$$R = \rho\, v^2 l^2 f_1\left(\frac{vl}{\nu}\right) + \rho\, v^2 l^2 f_2\left(\frac{v^2}{lg}\right) \quad . \quad . \quad (28)$$

Wave-making Resistance.

At high speeds the resistance due to skin friction is relatively unimportant so that the wave-making resistance alone may be considered.

In order to give the function f_2 the same value, Froude's law of comparison demands that the corresponding speed of the model should be

$$v_m = v_1 \sqrt{\frac{l_m}{l_1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

both for model and full-scale hull.

This means that for wave-making resistance

$$R_1 = R_m \frac{l_1^3}{l_m^3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (30)$$

Hence the resistances at corresponding speeds are as the displacements. This forms the basis of tests on flying-boat hulls.

Total Resistance.

As it is, however, impossible to satisfy the conditions for both skin and wave-making resistance simultaneously, the model must be made full scale.

Thrust and Torque of Airscrews.

Expressions for the thrust and torque of an airscrew as given below show that its performance depends upon the value of J $\left(i.e. \frac{V}{nD}\right)$ at which it is working.

Consider first the case of airscrews whose tips speeds are less than 800 ft./sec. Then the effect of compressibility of the air may be neglected, although it is then becoming important.

Static Airscrew.

Consider a static airscrew, *i.e.*, one which rotates in a fixed plane and has no forward motion

If the thrust T depends only upon airscrew diameter D, rotational speed n, mass density ρ, then

$$T = k_T \rho n^2 D^4 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (31)$$

where k_T is a constant called the thrust coefficient.

Similarly the torque,

$$Q = k_Q \rho n^2 D^5 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

where k_Q is a constant called the torque coefficient.

Moving Airscrew.

Let the airscrew be moving with translational velocity V. If the thrust now depends only upon V, D, n, and ρ,

$$T = \rho n^2 D^4 f_1\left(\frac{V}{nD}\right)$$

or

$$T = \rho n^2 D^4 f_1(J) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

where J is non-dimensional.

Similarly

$$Q = \rho\, n^2\, D^5 f_2(J) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

Equations (33) and (34) show both T and Q to be dependent on J. In a given airscrew, D is fixed so that T and Q depend upon $\frac{V}{n}$. If then $\frac{V}{n}$ be kept constant $f_1(J)$ and $f_2(J)$ are also constant, so that we have

$$f_1(J) = k_T = \frac{\text{Thrust}}{\rho\, n^2\, D^4} \quad . \quad . \quad . \quad . \quad . \quad . \quad (35)$$

$$f_2(J) = k_Q = \frac{\text{Torque}}{\rho\, n^2\, D^5} \quad . \quad . \quad . \quad . \quad . \quad . \quad (36)$$

High-Speed Airscrews.

From experiments on high-speed *static* airscrews, it appears that when the tip speed of such approaches the velocity of sound (in air) the airflow undergoes changes, until, if the tip speed be increased, the outflow ceases, and the slip-stream disappears when the velocity of sound is reached; the airscrew then appears to operate as a centrifugal fan. With high-speed airscrews this is serious for the phenomenon is associated with a falling off of the thrust.

It seems reasonable to suppose that at such high speeds the compressibility of the air plays an important part.

Considering compressibility effects only, if v_s is the velocity of sound in air, the resistance to motion takes the form

$$R = \rho l^2 v^2 f\left(\frac{v}{v_s}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (37)$$

and the effect of compressibility depends upon the velocity v as a fraction of the velocity of sound v_s in the undisturbed fluid. The relative importance of this may be obtained by applying Bernouilli's equation to a compressible fluid and assuming adiabatic transformation.

Let δp = increase in pressure at a stagnation point in the body.
v = velocity of undisturbed stream.
ρ = mass density where speed is v.

Then, if $\gamma = \frac{7}{5}$,

$$\delta p = \tfrac{1}{2}\rho v^2\left[1 + \tfrac{1}{4}\left(\frac{v}{v_s}\right)^2 + \tfrac{1}{40}\left(\frac{v}{v_s}\right)^4 + \ldots\right] = \lambda\rho v^2. \quad . \quad . \quad . \quad (38)$$

Since the series in equation (38) rapidly converges, the effect of compressibility may be neglected so long as $\frac{1}{4}\left(\frac{v}{v_s}\right)^2$ is small compared with unity.

The following table gives the approximate values of λ for the whole range of values of V from zero to v_s.

V *ft./sec.*	0	200	400	600	800	1,000	1,100 (v_s)
λ	1·000	1·008	1·033	1·077	1·149	1·224	1·275

When V = 220 ft./sec. (150 m.p.h.), $\lambda = 1{\cdot}01$, so that the neglect of compressibility at this speed would introduce an error of 1 per cent. On the other hand, it is clear that for velocities of 800 ft./sec. and over the effect of compressibility can no longer be disregarded.

Complete Expressions for Thrust and Torque.

In addition to the quantities already defined, let the following notation apply to the material of blade:

Δ = mass density (ML^{-3})
E = coefficient of elasticity, *i.e.* Young's modulus ($ML^{-1}T^{-2}$).

Assuming that T and Q depend upon

$$V, v_s, D, n, \nu, \rho, \Delta \text{ and } E,$$

it may be shown that

$$T = \rho n^2 D^4 f_1\left(\frac{V}{nD}, \frac{\nu}{nD^2}, \frac{v_s}{nD}, \frac{E}{n^2D^2\Delta}, \frac{\Delta}{\rho}\right) \quad . \quad . \quad . \quad (39)$$

and

$$Q = \rho n^2 D^5 f_2\left(\frac{V}{nD}, \frac{\nu}{nD^2}, \frac{v_s}{nD}, \frac{E}{n^2D^2\Delta}, \frac{\Delta}{\rho}\right) \quad . \quad . \quad . \quad (40)$$

For direct comparison of, say, the thrust, the function f_1 must be the same for both model and full scale, hence each of the five variables must be the same in each case. Of these variables it should be noted that:

(i) May be satisfied in that $\frac{V}{nD}$ may be made the same for both model and full scale.

(ii) The viscosity term is satisfied if nD^2 is the same if both airscrews are operating in the same medium. Thus, in order to ignore viscosity effects, a high rotational speed is necessary for the model.

(iii) The compressibility of the medium may be neglected if the tip speeds nD are the same.

(iv) Represents the blade distortion and if the material of the model and full-scale screws are the same, the distortion effects are the same if the tip speeds nD are the same.

(v) Is always satisfied if the material of the blade and density of the medium in which the screws are working are respectively the same.

Clearly these five conditions can only be satisfied simultaneously when the model is full size.

The question then arises as to the relative importance of these terms.

For airscrews of tip speeds not more than 700–800 *ft./sec.*—Experiment shows that the neglect of viscosity, compressibility, and distortion effects do not lead to serious error. With regard to the latter, the adoption of Griffith's symmetrical plan form of blade (fig. 18) greatly reduces the angle of twist. Under these circumstances equations (35) and (36) may be used for thrust and torque, as the variation of k_T and k_Q for a stated $\frac{V}{nD}$ is small, though definite.

For airscrews whose tip speeds are in excess of 800 *ft./sec. and approaching the velocity of sound.*—It has been shown in R. and M. 884 that in the case of model experiments:

(i) T increases up to V = 0·9 v_s, afterwards falling off rapidly.

(ii) Q increases more rapidly to V = 1·0 v_s, afterwards falling off slowly.

(iii) η falls away rapidly after V = 0·75 v_s (850 ft./sec.). In a particular case the efficiency fell from 68 per cent. at V = 850 ft./sec. to 40 per cent. at V = 1,300 ft./sec.

Full-scale experiments show that—

(i) Serious losses in efficiency occur at high tip speeds, resulting from a decreased thrust and increased torque.

(ii) Strength considerations are not likely to limit tip speeds.

(iii) At these speeds the noise emitted is very great and is apparently largely confined to the plane of the airscrew.

E.—PERFORMANCE.

In the estimation of aeroplane performance, two important quantities are involved, viz., horse-power required, and horse-power available. Both of these are usually required for all attitudes of the aeroplane over the flying range for a stated series of altitudes, from ground level to say, 10,000 ft. or 15,000 ft.

On the one hand, the horse-power required is calculated from the known or estimated drag coefficients of the various components and the aerodynamic characteristics of the main planes; on the other hand the horse-power available is a quantity which is not obtained without considerable calculation. Apart from the characteristics of the airscrew itself, its performance also depends on the particular combination of engine and aeroplane.

Account has to be taken of (*a*) the variation of engine power consequent upon the decrease in density, pressure, and temperature, with altitude; (*b*) the changing value of J under which the airscrew operates with varying attitudes of flight; (*c*) the continual change of atmospheric conditions from day to day and place to place.

The Standard Atmosphere.

In order to afford a basis of comparison of full scale tests, an International Standard Atmosphere has been defined * which corresponds with the average conditions in Western Europe.

It is assumed that:

(i) The air is dry and its chemical composition the same at all altitudes, by volume being 78·03 per cent. nitrogen, 20·99 per cent. oxygen, 0·94 per cent. argon, and 0·04 per cent. carbon dioxide.

(ii) The temperature at mean sea-level is 15° C., and the barometric height reduced to °0 C. is 760 mm.

(iii) The weight of air under these conditions is 1·2257 kg./m.^{3} (0·07656 lbs./ft.^{3}). The value of 'g' is taken to be uniform at 980·62 cms./sec.^{2} (32·17 ft./sec.^{2}). The mass density ρ_0 of air at mean sea-level is therefore 0·00237 slug/ft.^{3}

* Air Publication 1173.

(iv) For any altitude Z metres above mean sea-level up to a limit of 11,000 metres, *i.e.* to the bottom of the Stratosphere, the temperature (°C.) varies thus :—

$$\theta_z = 15 - 0{\cdot}0065\mathrm{Z} \quad . \quad . \quad . \quad . \quad . \quad (41)$$

(v) Above 11,000 metres θ is constant and equal to $-56{\cdot}5°$ C.

(vi) It follows that for any altitude Z less than 11,000 metres, where the barometric pressure is p_z and specific weight a_z and specific mass ρ_z, these quantities are connected by the equations :—

$$\frac{p_z}{p_o} = \left(\frac{T_z}{T_o}\right)^{5{\cdot}256} = \left(\frac{288 - 0{\cdot}0065\mathrm{Z}}{288}\right)^{5{\cdot}256} \quad . \quad . \quad . \quad . \quad (42)$$

and

$$\frac{\rho}{\rho_o} = \frac{a_z}{a_o} = \left(\frac{T_z}{T_o}\right)^{4{\cdot}256} = \left(\frac{288 - 0{\cdot}0065\mathrm{Z}}{288}\right)^{4{\cdot}256} \quad . \quad . \quad . \quad (43)$$

(vii) For altitudes above 11,000 metres the relations are :—

$$\log_{10}\frac{p_1}{p_z} = \log_{10}\frac{\rho_1}{\rho_z} = \log_{10}\frac{a_1}{a_z} = \frac{\mathrm{Z} - 11{,}000}{14{,}600} \quad . \quad . \quad . \quad (44)$$

where suffix 1 refers to conditions at Z = 11,000 metres.

Table VI. shows the approximate values of relative density and pressure and temperature for various heights as given by equations (42) and (43). For convenience, altitudes in feet have been tabulated.

TABLE VI.

International Standard Atmosphere.

Height, feet.	Relative Density $\frac{\rho}{\rho_o} = (\sigma)$	Relative Pressure $\frac{p}{p_o}$	$\left(\frac{p}{p_o}\right)^{1{\cdot}05} = f(h)$	Temperature. °C.
0	1·00	1·00	1·00	15·0
1,000	0·98	0·97	0·97	12·5
2,000	0·94	0·93	0·93	11·0
3,000	0·92	0·90	0·89	8·5
4,000	0·88	0·86	0·86	7·0
5,000	0·86	0·83	0·82	5·0
6,000	0·83	0·80	0·79	3·0
7,000	0·81	0·78	0·77	1·5
8,000	0·78	0·74	0·73	− 1·5
9,000	0·76	0·72	0·71	− 3·0
10,000	0·74	0·69	0·68	− 5·0
15,000	0·63	0·57	0·55	− 15·0
20,000	0·53	0·46	0·44	− 25·0
25,000	0·45	0·37	0·35	− 34·5
30,000	0·38	0·30	0·28	− 44·5
35,000	0·31	0·23	0·21	− 55·0

The Variation of Engine Power with Altitude.

The engine power is now thought to depend on atmospheric pressure only, and as a result of a large number of full-scale experiments the variation appears to be roughly proportional to the pressure. It has actually been found by H. L. Stevens (R. and M. 960), that the variation in power is approximately proportional to $\left(\frac{p}{p_o}\right)^{1{\cdot}05}$.

For a given r.p.m., therefore, the b.h.p. in a standard atmosphere at mean sea-level (generally referred to as ground-level) must be multiplied by a horse-power factor $f(h)$ in order to obtain the corresponding b.h.p. at altitude h.

In symbolic form

$$(\text{b.h.p.})_h = \text{Std. b.h.p.} \times f(h) \quad . \quad . \quad . \quad . \quad . \quad (45)$$

the r.p.m. being the same in each case;

where

$$(h) = \left(\frac{p}{p_o}\right)^{1{\cdot}05} \quad . \quad . \quad . \quad . \quad (46)$$

which is given in Table VI., col. 4.

Generalised Airscrew Curves.

From an examination of a large number of airscrew tests, Bairstow and Coales have shown in R. and M. 474, that the results relating to thrust and torque coefficients may be expressed with fair approximation by a single curve for each when referred to the advance per revolution as a fraction of the experimental mean pitch, *i.e.* $\frac{V}{nP}$.

For this purpose two coefficients T_c and Q_c are introduced such that by multiplying k_T and k_Q respectively, by them, the products are each unity when $\frac{V}{nP} = 0{\cdot}5$.

Let $$\frac{V}{nP} = J_1$$

then, for all airscrews plotted on a base of J_1 we have,

(i) A curve representing thrust in which $T_c k_T$ is $1{\cdot}0$ when $J_1 = 0{\cdot}5$;

(ii) A curve representing torque in which $Q_c k_Q$ is $1{\cdot}0$ when $J_1 = 0{\cdot}5$;

where, for a given airscrew, T_c and Q_c are constant coefficients.

Now the airscrew efficiency may be expressed

$$\eta = \frac{1}{2\pi} \cdot \frac{T_c k_T}{Q_c k_Q} \cdot \frac{Q_c}{T_c} \cdot \frac{P}{D} \cdot J_1 \quad . \quad . \quad . \quad . \quad . \quad (47)$$

Using the fact that η is $\eta_{\text{max.}}$ when $J_1 = 0{\cdot}725$, a third curve showing the relation between $\eta/\eta_{\text{max.}}$ and J_1 may be obtained. These curves, known as 'Generalised Airscrew Curves,' are shown in fig. 27.

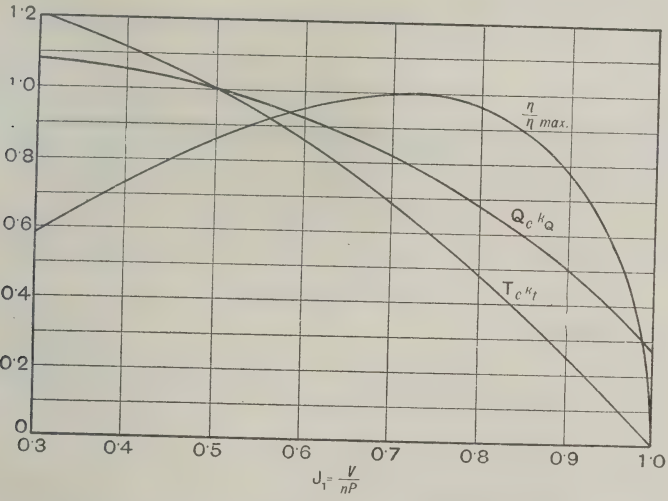


FIG. 27.—Generalised Airscrew Curves.

The curves of $T_c k_T$ and $Q_c k_Q$ were initially thought to be independent of $\frac{P}{D}$. Later work showed small variations in the value of the former with $\frac{P}{D}$. The curve in fig. 27 represents mean values. In the case of $Q_c k_Q$ larger variations were found with $\frac{P}{D}$. The curve shown in fig. 27

corresponds very approximately with $\frac{P}{D} = 1 \cdot 2$. For greater accuracy a family of curves should be drawn for $Q_c k_Q$.*

When $J_1 = 1 \cdot 0$ the corresponding values of $Q_c k_Q$ and $\frac{P}{D}$ are given in Table VII.

TABLE VII.

$\frac{P}{D}$	0·8	1·0	1·2	1·4	1·6
$Q_c k_Q$	0·48	0·38	0·30	0·25	0·21

Estimation of H.P. Available.

An approximate estimate of this quantity may be calculated if slip-stream effects of the airscrew are ignored. The problem then reduces to finding the airscrew efficiency for various engine speeds and therefore various flight speeds.

The generalised airscrew curves are used and the following data are required :—

(i) Engine power curve.

(ii) Estimated top speed $V_{max.}$ as given by some rapid method.

(iii) Airscrew diameter.

It is further necessary to assume that :—

(*a*) The engine runs at maximum permissible revolutions at maximum flight speed, and

(*b*) At this speed $J_1 = 0 \cdot 8$ say.

Rapid Estimation of Top Speed.

The probable maximum speed $V_{max.}$ at ground-level may be quickly obtained by using equations given in R. and M. 776. They are :—

For single-engine aircraft,

$$\left(\frac{V}{\sqrt{w}} - 25\right)^2 = 16 \cdot 6 \left(\frac{N}{\sqrt{w}} - 11 \cdot 8\right) \quad . \quad . \quad (48)$$

For multi-engine aircraft,

$$\left(\frac{V}{\sqrt{w}} - 23\right)^2 = 15 \cdot 7 \left(\frac{N}{\sqrt{w}} - 11 \cdot 6\right) \quad . \quad . \quad (49)$$

For boat seaplanes,

$$\left(\frac{V}{\sqrt{w}} - 20\right)^2 = 14 \cdot 4 \left(\frac{N}{\sqrt{w}} - 11 \cdot 35\right) \quad . \quad . \quad (50)$$

where

V = flight speed in m.p.h. at ground-level.

N = power per 1000 lb., i.e. $\frac{1000 \times \text{H.P.}}{\text{Total weight}}$.

w = main plane loading $\left(\frac{W}{S}\right)$ lbs./ft.^{2}

The value of $V_{max.}$ (in non-supercharged engines) falls as the altitude increases. For an aeroplane having a ceiling of about 20,000 ft., the values of $V'_{max.}$ at altitude h, may, with reserve, be taken to be the fractions of the ground-level value given in Table VIII.

* *Applied Aerodynamics*, Bairstow.

TABLE VIII.

h, ft.	5,000	10,000	15,000	17,500
$V'_{max.}$	97·5%	95%	90%	85%

Table VIII, in conjunction with the appropriate equation, 48, 49, or 50, therefore gives a rough estimate of the top speed at any desired altitude.

GENERAL PROCEDURE.

For flight at a given altitude h, proceed thus :—

(i) Calculate torque coefficient k_Q for a range of airscrew speeds n from the equation,

$$k_Q = \frac{550}{2\pi\rho_0 D^5} \cdot \frac{f(h)}{\sigma} \cdot \frac{\text{Std. b.h.p.}}{n^3} \quad . \quad . \quad . \quad . \quad (51)$$

Tabulate as in Table IX.

TABLE IX.—ESTIMATION OF H.P. AVAILABLE.

1	2	3	4	5	6	7	8	9	10
Engine speed r.p.m.	Airscrew speed r.p.s. n	Std. b.h.p.	k_Q	$Q_c k_Q$	$J_1 = \frac{V}{nP}$	$\frac{\eta}{\eta_{max.}}$	η	V ft./sec.	$H.P._A$
1,200									
1,400									
1,600									
1,800									
etc.									

(ii) Calculate experimental mean pitch P. From assumptions (a) and (b) we have

$$\frac{V_{max}}{n_{max} P} = 0{\cdot}8, \text{ so P is given at once.}$$

(iii) Calculate aircraft speed V for various airscrew speeds n.

Here $$V = \left(\frac{V}{nP}\right) \times n \times P.$$

This gives column 9.

(iv) Calculate Q_c.

For $J_1 = 0{\cdot}8$ corresponding to $n_{max.}$ and $V_{max.}$ read off $Q_c k_Q$ from fig. 27.

Then

$$Q_c = \frac{Q_c k_Q}{k_Q}, \text{ which is constant for all airscrew speeds.}$$

Column 5 may now be completed. Using the values of $Q_c k_Q$ thus found, read off and tabulate the corresponding values of J_1 and $\frac{\eta}{\eta_{max.}}$.

(v) Calculate $\eta_{max.}$.

Plot J_1 and n and read off n_1 when $J_1 = 0{\cdot}725$ which corresponds to $\eta_{max.}$.

Hence when airscrew is working at its maximum efficiency, aircraft speeed V_1 is given by,

$$V_1 = 0{\cdot}725\ n_1\ P.$$

The Froude efficiency η_F is then obtained by successive approximation in the following manner:

Assume $\eta_1 = 0{\cdot}8$ and calculate thrust T_1 from H.P.$_1$ and V_1.

From Froude's momentum theory

$$T_1 = \rho k A_0 (1 + b)\, b V_1^2,$$

where KA_0 = area of slip stream and $K = 0{\cdot}6$

$$A_0 = \frac{\pi}{4} D^2.$$

This gives b, and

$$\eta_F = \frac{2}{2 + b}.$$

If η_F is not equal to η_1, take η_F as a second approximation and repeat.

Finally, the actual efficiency $\eta_a = 0{\cdot}8\eta_F$ whichis the required $\eta_{\max}$.

The efficiencies at other aircraft speeds are then obtained from the tabulated ratios of $\frac{\eta}{\eta_{\max}}$.

(vi) Calculate the horse-power available, H.P.$_A$.

For each aircraft speed,

$$\text{H.P.}_A = \eta.\ \text{Std. b.h.p.} \quad . \quad . \quad . \quad . \quad (52)$$

Finally plot H.P.$_A$ against V as in fig. 27.

Horse-Power Required.

The amount of labour involved in the calculation of the horse-power required depends, as in the case of the horse-power available, upon the data available and the degree of accuracy required. The value of the result depends upon a fairly accurate knowledge of the resistance of the several parts or components as deduced from model tests. Fortunately, for most components the resistance is proportional to the square of the velocity over a very large range, but in the case of elongated aerodynamic bodies such as a fuselage or flying-boat hull, where skin friction effects are important, the index of V is less than two. Unless due care is taken serious errors in the estimated resistance may arise. It is found, however, that if the resistance of such bodies is known for a speed approximately mid-way over the flying range of speeds, the error at the extreme speeds in applying the V_1^2 law is small.

Resistances of Components.

The following list gives the resistances of various details or components at 100 m.p.h. at ground level which may be used in the absence of other data. In all cases the resistance may be assumed to vary as the square of the speed.

Fuselage (good streamline) when $\alpha = 0°$	2·6 lbs./sq. ft., projected area.
,, (two cockpits and nose radiator) when $\alpha = 0°$	8·0 ,, ,, ,, ,,
Tail plane and elevators	0·75 lbs./sq.ft. of plan area.
Fins and rudders	0·60 lbs./sq. ft. of side area.
Wheel, 800 × 150 mm.	13·5 lbs. with shield to tyre.
Struts, fineness ratios 2·5 to 5·0	2·3 lbs./sq. ft. projected area.
Add 0·4 lbs. for each end.	

Streamline wires.

4 B.A.	0·051	lbs./ft. run.	For end fittings add 2 ft. to the length of each wire.
2 B.A.	0·056	,, ,,	
$\frac{1}{4}$ in. B.S.F.	0·075	,, ,,	
$\frac{5}{16}$,, ,,	0·094	,, ,,	
$\frac{3}{8}$,, ,,	0·112	,, ,,	
$\frac{7}{16}$,, ,,	0·131	,, ,,	
$\frac{1}{2}$,, ,,	0·150	,, ,,	

Miscellaneous details. 5 lbs./sq. ft. projected area.

Simple Method of Calculation.

In an approximate method of calculating the horse-power required ($H.P._R$) the following assumptions are usually made :—

(i) The tail load may be ignored.
(ii) The variation in resistance of components due to changes of α may be neglected.
(iii) The line of thrust passes through the C.G.
(iv) The obliquity of the line of thrust may be neglected.
(v) The airscrew slip-stream diameter is 0·8 D.
(vi) The resistance of parts in the slip-stream is increased by 25 per cent. for any speed of flight (i.e., the slip-stream velocity is always $\sqrt{1{\cdot}25}$ V).

In view of condition (iii) the method outlined below cannot be applied to flying boats without modification.

The total resistance is composed of two parts (a) that due to main planes ; (b) that due to remainder.

For the former,
in steady flight,

$$L = W \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (53)$$

$$D = \frac{W}{L/D} = \frac{W}{\delta} \quad . \quad . \quad . \quad . \quad . \quad . \quad (54)$$

Let V_1 = flight speed in m.p.h.

then

$$V_{m.p.h.} = 1{\cdot}465\ V_{ft./sec.} \text{ and } (V_{m.p.h.})^2 = 2{\cdot}15\ (V_{ft./sec.})^2$$

and

$$V_1^2 = \frac{W}{2{\cdot}15\ k_L \rho S} \quad . \quad . \quad . \quad . \quad (55)$$

For the latter,

if R_1 = resistance of parts outside slip-stream at 100 m.p.h.
R_2 = " " inside " " " "
R = total resistance at 100 m.p.h.

then

$$R = R_1 + 1{\cdot}25\ R_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (56)$$

For a single-engined tractor, R_2 includes the fuselage, centre section struts and wires, part of tail unit, and undercarriage.
Hence, using equations (54) and (56).

At any speed V_1, the total drag of the aeroplane is given by

$$D_W + D_B = \frac{W}{\delta} + R\left(\frac{V_1}{100}\right)^2 \quad . \quad . \quad . \quad (57)$$

But for steady flight, using equation (3),

$$T = \frac{W}{\delta} + \frac{R}{10^4}\left(\frac{W}{2{\cdot}15\ k_L \rho S}\right) = \frac{W}{\delta} + D^1 \quad . \quad . \quad . \quad . \quad (58)$$

Finally,

$$H.P._R = \frac{V_1 T}{375} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (59)$$

Draw up Table X. for equal increments of k_L up to $k_{L\,max.}$ and plot the results as in fig. 27.

Table X.—Estimation of H.P. Required.

k_L	$\delta = \frac{L}{D}$	$\frac{W}{\delta}$	D^1	T	V_1	$H.P._R$

Deductions from Performance Curves.

Range of Flight Speeds.

In fig. 28, the curves of H.P.$_A$ and H.P.$_R$ both relate to the same height, viz., ground-level. The extreme flight speeds are given at the points of intersection of these curves. At A the stalling speed is obtained whilst at B the maximum flight speed is revealed.

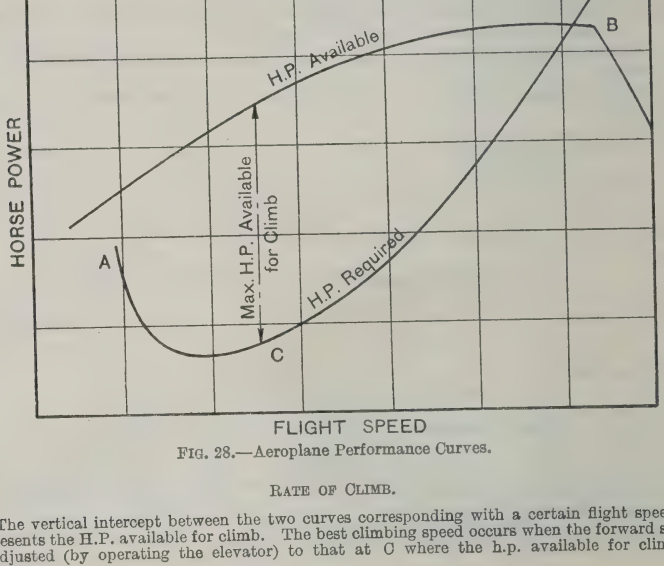


Fig. 28.—Aeroplane Performance Curves.

Rate of Climb.

The vertical intercept between the two curves corresponding with a certain flight speed V, represents the H.P. available for climb. The best climbing speed occurs when the forward speed is adjusted (by operating the elevator) to that at C where the h.p. available for climb is maximum.

It will be seen that the rate of climb r (or vertical velocity of ascent) will vary with the forward speed, in fact

$$r = \frac{(\text{H.P.}_A - \text{H.P.}_R)\,33{,}000}{W} \text{ ft./min.} \quad . \quad . \quad . \quad . \quad (60)$$

At the extreme speeds r is zero.

Flight at Altitude.

In order to obtain the horse-power required H.P.$_R$ for altitudes other than ground-level it is not necessary to go over a similar set of calculations.

If H.P.$_R$ and V refer to ground conditions, then at altitude h where the relative density is σ the corresponding powers H.P^1.$_R$ and speeds V^1 are given by

$$\text{H.P}^1.{}_R = \frac{\text{H.P.}_R}{\sqrt{\sigma}} \quad . \quad . \quad . \quad . \quad . \quad (61)$$

$$V^1 = \frac{V}{\sqrt{\sigma}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (62)$$

Thus all the powers and speeds in Table X. are divided by the appropriate $\sqrt{\sigma}$ to obtain the corresponding figures at any desired altitude.

Water Performance of Flying Boats

In a proposed design, although according to calculation the machine would be able to fly in the air with a reserve of power, it would not necessarily follow that there would be sufficient power to enable the boat to leave the water in getting off. The water resistance is very large, so that floats and hulls have to be designed to reduce this as much as possible. For fuller information on this subject reference should be made to available published data.*

F.—DESIGN.

All Service aircraft have to be manufactured in metal. It is not permissible to use wood for a structural member. Constructors use either steel, duralumin, or both.

Steel.

Steel may be used in either of two forms, viz., as seamless tubing, or rolled sheet.†

Seamless Tubing.

Apart from manufacturing difficulties,‡ the use of round tubing in aircraft construction is limited to those members subjected to end loads only, for the material in a round tube is not disposed at a sufficiently large distance from the neutral plane as to allow it to be used efficiently for the transference of bending moments.

Since fuselage members are generally subjected to end-loads only, 50-ton round or square section tubing is frequently used in fuselage construction.

Rolled Sheet.

Rolled sheet steel is obtainable in various widths and lengths in thicknesses approaching 0·01 in. The material will withstand a compressive stress between 75 and 100 tons per sq. in.

Much of existing knowledge and technique of the production of rolled steel spars is due to the early investigations of Low, Robertson, Wylie, and others.

Duralumin.

The remarkable mechanical properties of duralumin depend on its heat treatment. When normalized at 485° C. it develops an ultimate stress of 25 tons per sq. in. after four days.

Being about one third the density of steel, duralumin may be used in more reasonable thicknesses than that necessary for members in rolled steel sheet.

Actual Examples.

The source of the diagrams used to illustrate the following notes is in each case indicated. Thanks are due to the Bristol Aeroplane Co. and the Gloster Aircraft Co. for diagrams and technical data. Special thanks are due to the Technical Editor of '*Flight*' for permission to reproduce a number of illustrations from that journal representing various types of construction.

Sir W. G. Armstrong-Whitworth (Aircraft), Ltd.

Fuselage Structure.—Round steel tubes are used throughout. At all panel points the longerons are fitted with short sleeves slipped over them and bolted in position.

The vertical struts are fitted with a cupped socket to take the ball-head of a bolt through the longeron. A typical joint for the rear portion of the fuselage is shown in fig. 28 (1). The port and starboard sides are erected on jigs and the structure completed by connecting the horizontal transverse struts.

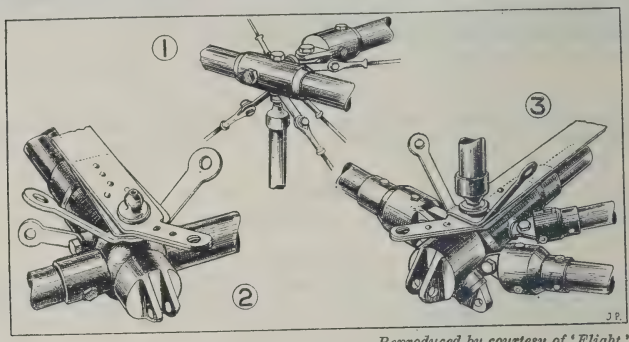
A joint on the bottom longeron at the point of attachment of the front spar and lift wire is shown in fig. 29 (2), while (3) shows the corresponding joint for the rear spar. In this particular design the longeron is connected to a fork-end at this joint so that the front and rear portions of the fuselage are detachable.

---

* 'Aeroplane Performance Calculations.' H. Booth.
† See 'Aeroplane Structural Design,' Appendix III. Jones and Frier.
‡ See 'Steel Tubes,' W. W. Hackett and A. G. Hackett, *Aeronautical Journal*, April, 1918.

MAIN PLANE STRUCTURE.—Strip steel is used throughout for both spars and ribs.

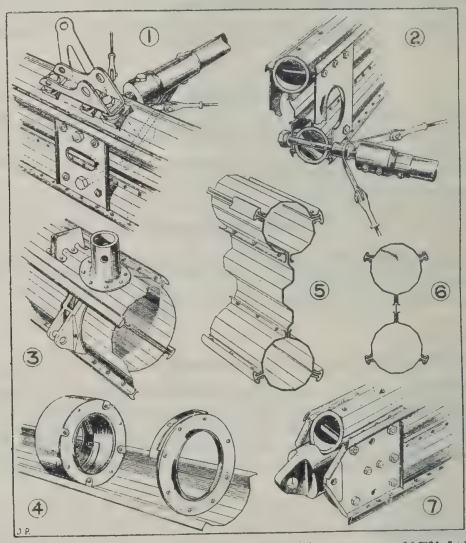
MAIN SPARS.—Two types of spar have been standardised. They are shown in fig. 30 (5) and (6). The section of the flanges or booms is polygonal and is built up in three equal sections with



Reproduced by courtesy of 'Flight.'

FIG. 29.—Armstrong-Whitworth Fuselage Construction.

flanged edges. Adjacent edges are held together by specially formed U strips, riveted at intervals. In the deep spar the web is corrugated as shown by flat surfaces joined by small curves. In the small spar the web consists of a narrow flat strip. Typical interplane and drag strut fittings for



Reproduced by courtesy of 'Flight.'

FIG. 30.—Armstrong-Whitworth Steel Strip Wing Spar Construction.

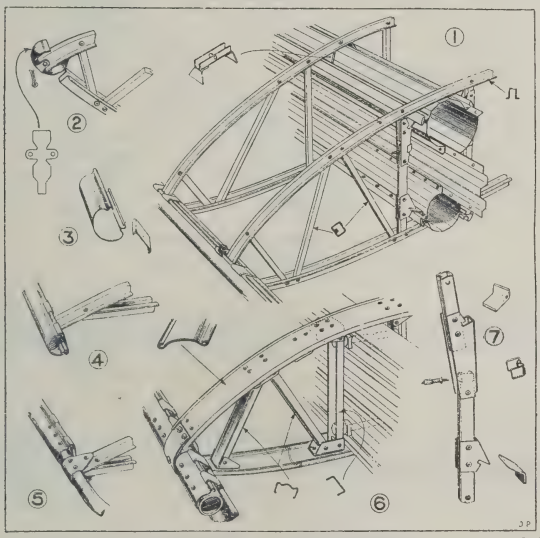
the deep type of spar are shown in (1) and (2). The spar is stiffened locally by short lengths of round tube inserted in the booms.

DRAG STRUTS.—The drag struts, which are of round seamless tube, are not attached at the neutral plane of the spar but to one of the booms. External flanged stiffeners are provided on each side of the spar.

End fittings for a shallow spar are shown in (7).

AILERON SPARS.—For members such as aileron spars which are required to transmit torque, a section approximating to a circle is necessary. Spar sections as in (5) and (6) are unsuitable. The type adopted is shown in (3) and consists of a polygonal form built up of four strips secured as in the case of the spar booms. Internal built-up stiffeners are provided where necessary, similar to those shown in (4). A rib clip, hinge, and strut socket are shown in (3).

RIBS.—Main plane ribs are built up in channel section for the flanges and square section for the bracing members, as indicated in fig. 31 (1). The ribs are not connected directly to the spars by riveting, but are held in position by the use of the toggle arrangement shown in (7).



Reproduced by courtesy of 'Flight.'

FIG. 31.—Armstrong-Whitworth Steel Wing Construction.

The leading edge is shown in (3). The ribs are connected to the leading edge by the upper rib boom only. For this purpose the end of this boom is fitted with a bent plate, shown in (2), which slides inside the leading edge strip. A friction grip between these two is obtained by sliding small plates up against each side of the rib. These are shown in postion in (1).

The fixing of the trailing edge to the rib is indicated in (5).

For strength ribs a further set of sections are used as illustrated in (6). The same section which is used for the flange of these ribs is employed for the fuselage fairing, to which the fabric is attached.

HEAT TREATMENT.—The operations of rolling or drawing are carried out on the flat strip while soft, and the necessary heat treatment given when the various units are made. The operation is done by using the finished unit as an electrical resistance. The scale which is subsequently formed on the material is removed by wire brushes.

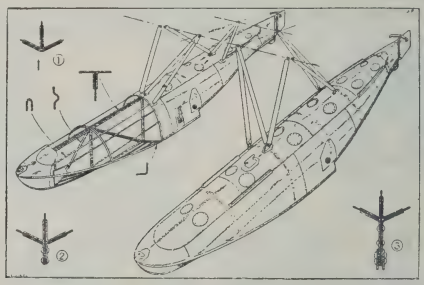
PREVENTION OF CORROSION.—The formation of rust is prevented by dipping in a specially constructed tank containing enamel, and stoving the complete units. Service conditions show that the process is entirely satisfactory.

The Blackburn Aeroplane and Motor Co.

BLACKBURN SPRAT.

The Blackburn 'Sprat,' fitted with a 270-h.p. Rolls-Royce 'Falcon' engine, is a training machine convertible into a seaplane or landplane.

FLOATS.—These are of duralumin and are fitted as shown in fig. 32. It will be seen that water rudders are fitted for good manœuvrability on the water. Each float has twelve inspection



Reproduced by courtesy of 'Flight.'

FIG. 32.—Floats (Blackburn 'Sprat').

holes, and the section of some of the members of the main framework is given. Inset (1) shows the joint of the keelson to keel near the bows; (2) Shows the same joint at some distance aft the step; and (3) the joint immediately aft the step, at which place the keel is fitted with a steel shoe as shown.

Boulton and Paul, Ltd.

A number of sections suitable for various structural elements have been standardised, both steel strip and duralumin being used.

THE B. AND P. PARTRIDGE.

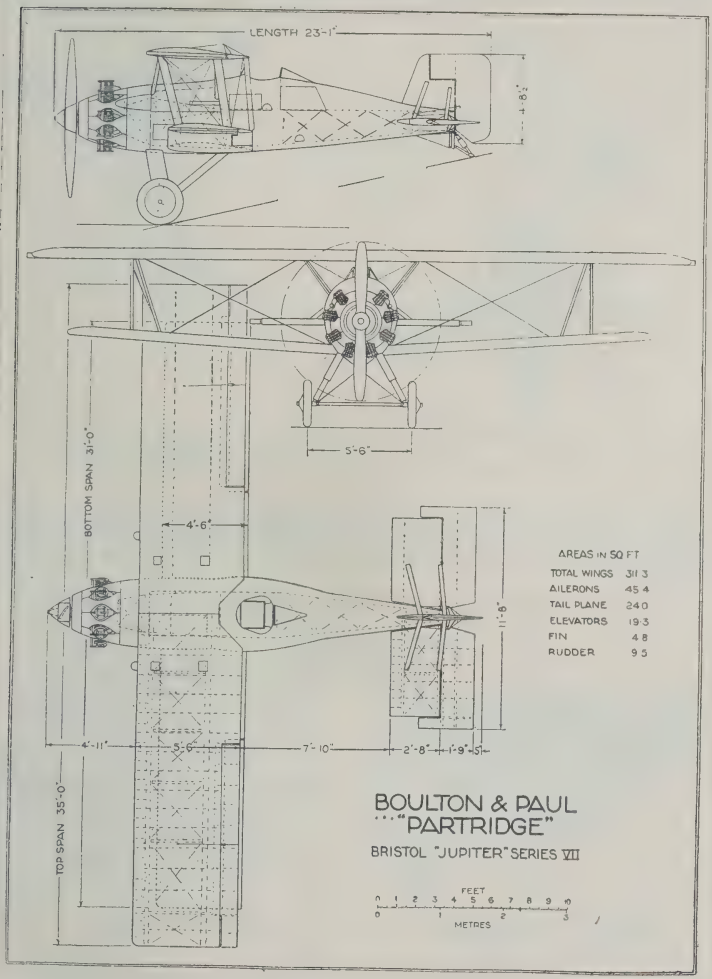
The latest product of this firm is named the 'Partridge,' which is a high-performance tractor biplane single-seater fighter. It is, of course, entirely in metal. The general arrangement of the machine is shown in fig. 33.

FUSELAGE STRUCTURE.—The front portion of the fuselage consists of seamless high-tensile steel tube. Fuselage struts are of four kinds, viz., solid drawn steel or duralumin tubes, or closed joint tubes made from steel or duralumin strip. In the manufacture of closed-joint tubes the longitudinal edges of the metal are brought together and rolled over inside the tube itself. This is seen in fig. 34 (3) at the lower broken end of the longeron.

The main plane spar joint is shown in (4), and it will be seen that the longeron is fitted with a light alloy sleeve. Two high-tensile steel discs are employed to receive and transmit the spar and lift-wire loads. The rear portion of the fuselage is carried out in locked joint tube longerons and duralumin struts. A typical joint is shown in (3).

MAIN PLANE STRUCTURE.—This is unusual in that there is no top centre section. The top spars are joined together on the centre line by fish plates.

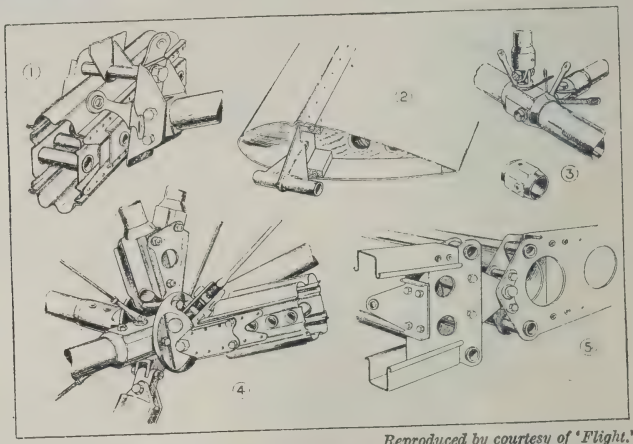
MAIN SPARS.—These are made from high-tensile strip steel and are of standard stock sections. Both flanges and webs are corrugated with frequent tubular stiffeners. Plate stiffeners are supplied at the various joints. A typical main plane joint is shown in fig. 33 (1).



Reproduced by courtesy of 'Flight.'

FIG. 33.

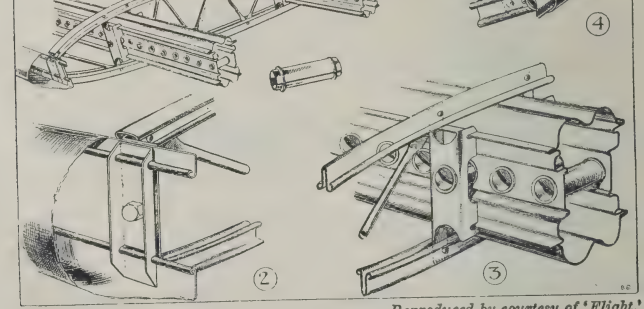
INTERPLANE STRUTS.—These employ standardised duralumin sections. The load is transmitted by a channel-shaped section connected to the spar by a **T** fitting shown in fig. 34 (1) and (2). The nose of the strut is in the form of a thin **U** section riveted to the main channel so that it takes part of the end load. The tail of the strut is faired off in wood.



Reproduced by courtesy of 'Flight.'

FIG. 34.—The Boulton & Paul 'Partridge' Wing and Fuselage Construction.

RIBS.—These are all made in duralumin, the sections used being shown in fig. 35. The ribs are attached to the spars in the following manner. Channel pressings are riveted to the top and bottom rib booms and the former are slotted to receive the riveted edges of the spar. The rib is then located by a long tubular rivet which passes through one of the tubular stiffeners of the spar. See fig. 35 (3).



Reproduced by courtesy of 'Flight.'

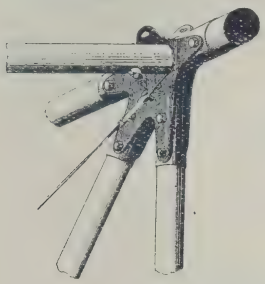
FIG. 35.—The Boulton & Paul 'Partridge' Wing Construction.

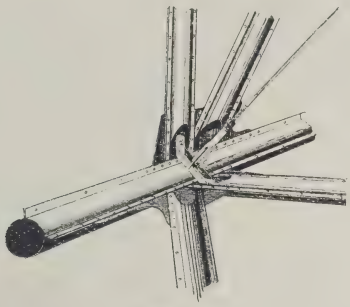
LEADING AND TRAILING EDGES.—The method of connection of ribs is indicated in fig. 35 (2) and (4). It will be seen that the leading edge is a **U** section with turned-in edges.

Bristol Aeroplane Co., Ltd.

FUSELAGE STRUCTURE.—For the front portion of the fuselage, which has to carry large loads, solid drawn high-tensile steel tubes are used, a typical joint being shown in fig. 36.

In the case of the rear portion, high-tensile strip steel is used.





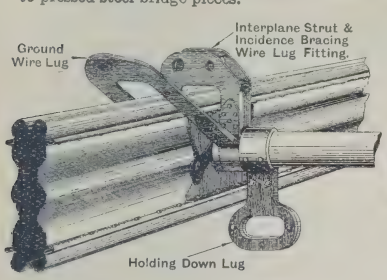
Reproduced by courtesy of Bristol Aeroplane Co.

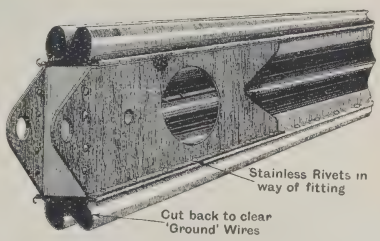
FIG. 36. FIG. 37.

LONGERONS.—These are built up from two sections, one of which subtends an angle of 90 degrees. In this way there are two riveted flanges.

PANEL STRUTS AND CROSS MEMBERS.—These are made from two similar sections riveted together.

TYPICAL REAR FUSELAGE JOINT.—In fig. 37 the manner of connecting the panel struts to the longerons by means of plates riveted in position is shown. Bulkhead bracing wires are connected to pressed steel bridge pieces.





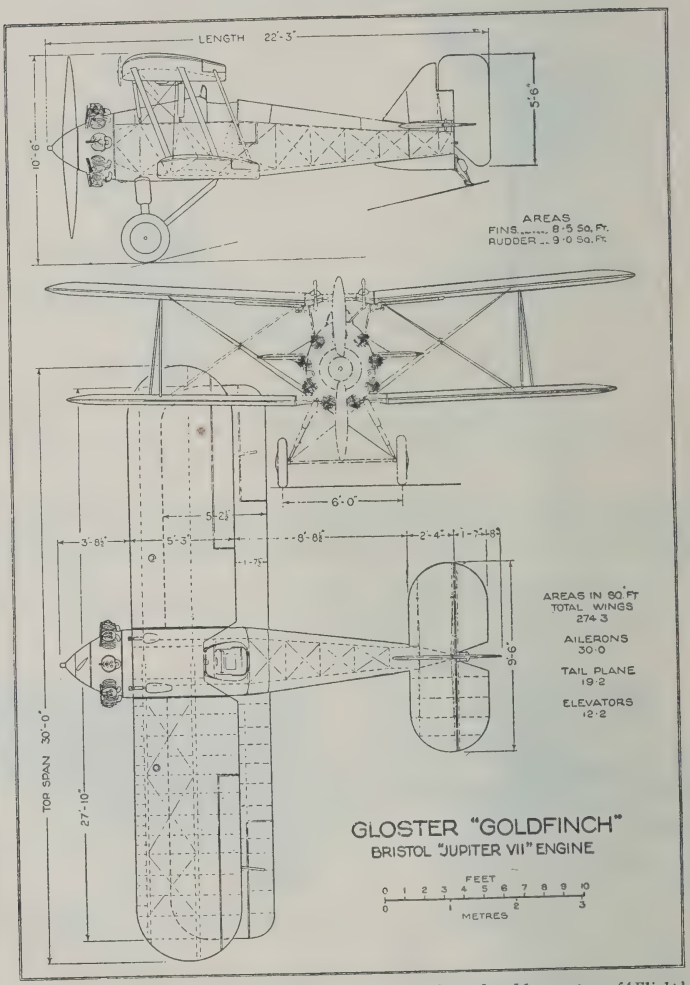
Reproduced by courtesy of Bristol Aeroplane Co.

FIG. 38. FIG. 39.

MAIN PLANE STRUCTURE.—Strip steel is used throughout.

MAIN SPARS.—The spars are made by riveting together two flange and two web sections, the upper and lower riveted edges being connected by internal plates running the whole length of the spar.

MAIN PLANE JOINTS.—Typical joints are shown in figs. 38 and 39.



Reproduced by courtesy of 'Flight.'

FIG. 40.

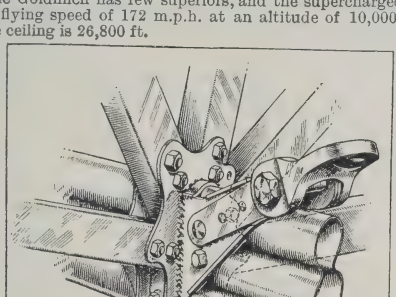
The Gloster Aircraft Co., Ltd.

The Company is well known as designers and constructors of high-speed aircraft.

GLOSTER GOLDFINCH.

This machine is an all-metal single-seater fighter, fitted with a Bristol Jupiter Mark VII supercharged engine (fig. 40). The Goldfinch is of the same family as the Gamecock, but embodies all the latest improvements on its predecessor and has many additional points worthy of note.

In performance the Goldfinch has few superiors, and the supercharged Jupiter enables it to develop a maximum flying speed of 172 m.p.h. at an altitude of 10,000 ft. and 157 m.p.h. at 20,000 ft. Its service ceiling is 26,800 ft.

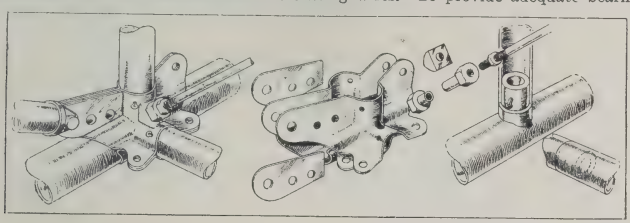


Reproduced by courtesy of 'Flight.'

FIG. 41.—The Gloster 'Goldfinch' Spar and Fuselage Joint.

The Goldfinch is constructed principally of steel with duralumin used for portions of the substructure and aluminium for the majority of the front fairing and cowling.

FUSELAGE STRUCTURE.—The front frame of the fuselage is a rigid structure built up of square tubes rigidly jointed by means of flat mild-steel plates bolted through the struts and longerons. Fig. 41 shows the joint at the spar attachment. In the rear frame the struts and longerons are round tubes braced together by means of tie rods. The joints are made in the form of pressed steel plates wrapping round the longeron and shaped to house the strut ends as shown in fig. 42. Provision also made for the attachment of bracing wires. To provide adequate bearing area



Reproduced by courtesy of 'Flight.'

FIG. 42.—The Gloster 'Goldfinch' Rear Fuselage Construction.

for the struts, hollow duralumin bushes are fitted into the ends of the dural struts which in turn go inside the longeron fitting—a small split pin retaining the strut in position during erection.

Bracing wires have special Gloster design high-tensile steel fork-ends and pins which allow of an appreciable weight reduction when used in conjunction with high-tensile steel wiring lug attachments. Robustness of design is always considered to be of primary importance to a doubtful gain in weight which would result from the using of ultra-light gauge members which would not stand up to handling conditions.

MAIN PLANE STRUCTURE.—The main planes are of high-tensile steel construction with the Gloster type of lattice girder in the thick upper wing, and in the lower wing a rolled high-tensile steel strip spar having continuous webs shown in fig. 41. To provide adequate aileron control, long, narrow chord ailerons have been fitted, supported on light false spars, the latter helping to stiffen up the main wing structure and cutting out excessive deflections of the spars under

maximum loading. Petrol tanks are fitted into each of the upper wings, following usual Gloster practice, and admit of quick detachment and a simplified petrol system.

THE UNDERCARRIAGE.—This is of the normal Gloster type. The pilot's seat is adjustable in use, being raised or lowered by the movement of a lever. The weight of the pilot is taken by rubber shock absorbers, thus facilitating the raising of the seat.

EQUIPMENT.—The Goldfinch is provided with a very full equipment, including wireless, parachute, two machine guns, ammunition, oxygen apparatus, and electrical equipment, as well as the usual navigation and engine instruments. Owing to its relatively low landing speed (under 60 m.p.h.) and wide wheel track, the Goldfinch might well be used as a night fighter. In that case, an exhaust ring is fitted. The installation of the two generators (wireless and lighting and heating) is unusual in that they are built into the top centre-section, thus giving a minimum of drag.

FLYING QUALITIES.—Pilots who have flown the machine report that the Goldfinch is very manœuvrable and easy to control, and that the cockpit is well arranged, roomy, warm, and comfortable, while the view is good for fighting, bombing, and ground machine gunning. It is stated to have no vices, and to be very easy to land.

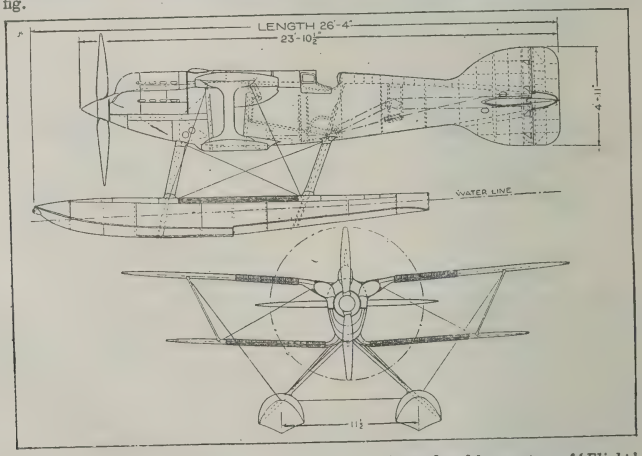
COMPONENT WEIGHTS.—The following details of item weights are interesting in showing how the total loaded weight of 3,200 lbs. (1,450 kg.) is made up :—

Component.	*Weight.* Lbs.	Kg.	*Component.*	*Weight.* Lbs.	Kg.	*Component.*	*Weight.* Lbs.	Kg.
Airscrew . .	69	31·4	Instruments .	19	8·64	Wireless equipment . .	25	11·35
Engine . .	798	363·0	Bottom wings .	157	71·4	Tail . . .	44	20·0
Starting magneto	8	3·63	Guns and mountings . .	67	30·45	Bottom fin .	7	3·18
Piping . .	15	6·82	Ammunition .	97	44·1	Top fin . .	10	4·55
Exhaust pipes .	10	4·55	Controls . .	35	15·9	Rudder . .	10	4·55
Top wings .	253	115·0	Pilot . .	180	81·8	Skid . . .	9	4·09
Fuel and tanks .	503	229·0	Seat and parachute . .	34	15·45	Undercarriage .	115	52·3
Struts and wires.	53	24·1	Very pistol .	7	3·18	Sights . .	4	1·82
Oil and tank .	67	30·45	Oxygen apparatus . .	5	2·27	Fuselage and equipment .	410	186·3
C.C. gear . .	17	7·73				Overload . .	150	68
Electrical equipment . .	22	10·0						

Total loaded weight 3,200 Lbs. 1,450 Kg.

GLOSTER IVB.

This is a racing seaplane fitted with an 875-h.p. Napier Lion racing engine. This machine (fig. 43) distinguished itself at Venice, 1927, on the occasion of the Schneider Trophy Race. A machine of this type requires a fuselage of the minimum cross section, while the main planes and engine cylinder blocks must be very carefully faired into it. Both these features are in evidence in the fig.



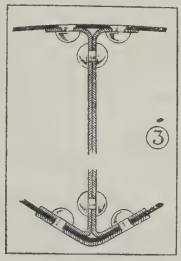
Reproduced by courtesy of 'Flight.'

FIG. 43.—Gloster IV P. Racing Seaplane (Napier Lion Racing Engine).

FLOATS.—The Gloster Company have gone very carefully into the design and construction of all-metal floats, both for use on their Schneider type racing seaplanes and on service type seaplanes.

From an aerodynamic viewpoint the research carried out has enabled a float to be designed which has a wind drag of 1·4 lbs./sq. ft. at 100 ft./sec. whilst still retaining good hydrodynamic qualities.

The form of construction now standardised consists of a main longitudinal duralumin girder lightened wherever possible, attached to which are transverse frame members of lightened sheet duralumin which support the decking of the float and the longitudinal planing bottom stiffeners,



Reproduced by courtesy of 'Flight.
FIG. 44—Gloster Float Joint.

which latter in turn take the skin. Bending is assumed to be taken by the skin and shear by the longitudinal girder. Attachment of the float to the undercarriage legs is made by means of watertight linings fitted to main transverse bulkheads at approximately one-third and two-thirds of the bulkhead measured from the nose.

For a float having a displacement of 5,700 lbs. the approximate weight would be of the order of 238 lbs.

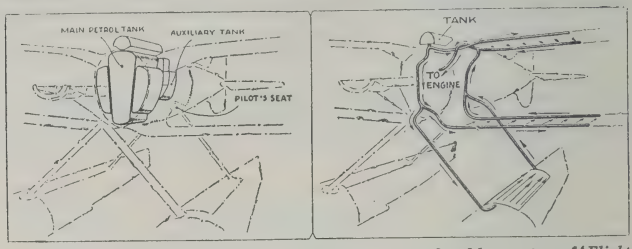
To provide the limit in cleanness the Gloster Company have developed a special form of joint whereby it is possible to have all the rivet heads flush with the skin, as indicated in fig. 44.

Reproduced by courtesy of 'Flight.'
FIG. 45.—The Gloster IVB. Wing Surface Radiators.

COOLING SYSTEM.—Surface radiators are employed, being installed on both top and bottom planes and on each float, and are shown in fig. 42. The radiators are made in thin corrugated copper sheet. Those mounted on the planes extend over the whole chord, brass waterways being placed at the leading and trailing edges. The outer end of the lower plane radiator and duralumin interplane strut are shown in fig. 44 (1) and the water-pipe entering the fuselage at the trailing edge of the lower plane is shown in fig. 45 (2).

Half of the complete radiator system is given in fig. 46.

PETROL TANKS.—These are housed in the fuselage and are shown in fig. 46.



Reproduced by courtesy of 'Flight.'

FIG. 46.—Petrol Tanks and Cooling System (Gloster IVB.).

Short Bros.

This firm for a number of years have been engaged in the design and production of duralumin marine craft.

THE 'CALCUTTA' FLYING BOAT.

This boat, ordered by Imperial Airways, has accommodation for 15 passengers. Drawings are given in fig. 48. It is the first commerical flying boat to be built in this country. The entire structure is in duralumin.

HULL.—The hull is of the type patented by this firm. It has no longitudinal members, being formed by frames connected together by duralumin plates. The latter are stiffened by V stringers so that the metal skin can transmit the necessary forces without crinkling.

MAIN PLANE STRUCTURE.—This is of normal biplane type with tip floats.

MAIN SPARS.—The spar construction is shown in fig. 47. In (1) the top rear centre section spar joint is shown together with steel fittings for struts and lift wires.

The spars are stiffened at such joints by additional plates shown in (2).

The bottom centre section spars are slightly different and are indicated in (4).

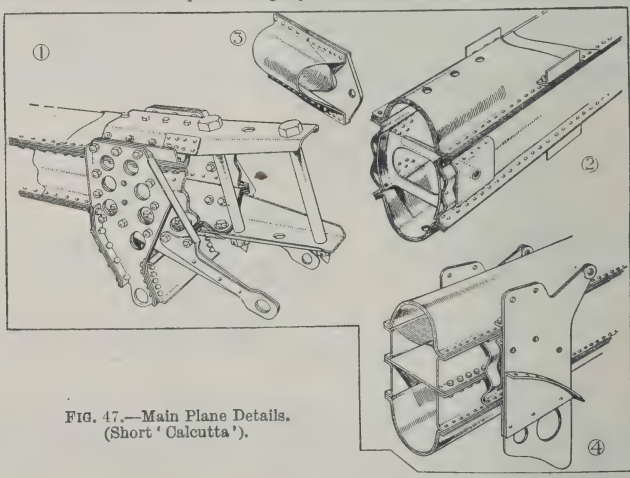


FIG. 47.—Main Plane Details. (Short 'Calcutta').

Reproduced by courtesy of 'Flight.'

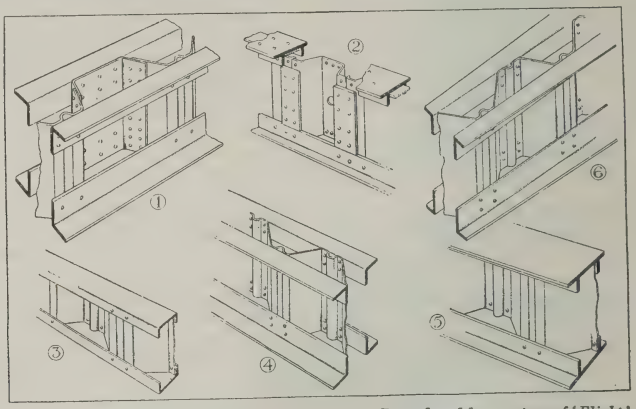
DRAG STRUTS.—These are built up from duralumin sheet and a stiff plate riveted at the ends for the spar connection. See fig. 47 (3).

Reproduced by courtesy of 'Flight.'

FIG. 48.

Vickers, Ltd.

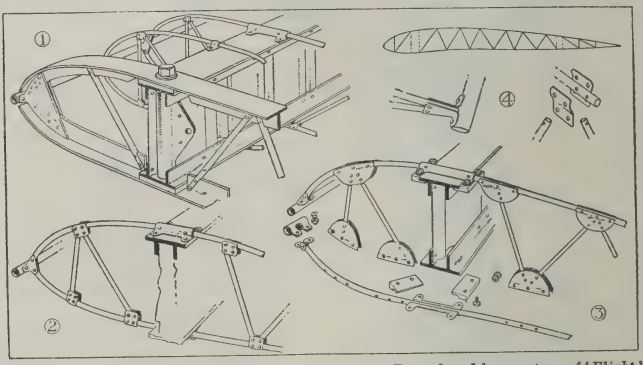
This firm, proprietors of the patents covering the manufacture of duralumin, have produced a number of sections which have been standardised for aircraft construction. The use of the metal in the form of sheets or strips is augmented by the introduction of extruded sections similar to those in steel used by structural engineers, including angles, ties, channels and a section having a comparatively wide flange and twin webs.



Reproduced by courtesy of 'Flight.'

FIG. 49.—Duralumin Spar Construction (Vickers).

MAIN PLANE STRUCTURE.—In the various types of spar shown in fig. 49 the flanges in each case consist of a rolled section or sections connected by a web of sheet metal so pressed that it is connected alternately to the back of the front flange to the front of the rear flange by rivets, and because of this has been called a 'wandering web.' Vertical flutings at the points of connec-



Reproduced by courtesy of 'Flight.'

FIG. 50.—Duralumin Construction of Main Planes (Vickers).

tion give the necessary stabilisation of the web. The web in general is a continuous strip, but a splice, if required, is carried out as in (1).

A particular advantage of this type of construction is that the spars may, in general, be tapered in both planes.

TABLE XI.—GEOMETRICAL PARTICULARS OF VARIOUS AEROPLANES. SINGLE SEATERS.

Aeroplane.	Purpose.	Engine.	Type.	Weight. Lbs.		Main Planes.					Tail.				Fin Area.	Rudder Area.	
				Empty.	Full.	Span. Ft.		Chord. Ft.		Total Area.	Type.	Span.	Chord.	Total Area.			
						Top.	Bottom.	Top.	Bottom.								
Avro 'Avian'	Long Range Sport	120-h.p. Gipsy	Land	1,100	2,225	30·0	30·0	4·75	4·75	Sq.ft. 262	Monoplane	Ft. 9·0	Ft. 4·0	Sq. 30·1	Sq.ft. 3·7	Sq.ft. 7·9	Range 1600 miles.
Armstrong-Whitworth 'Starling'	,,	Supercharged Jaguar	,,	—	—	31·33	23·83	5·0	4·3	Ailns. 23 240	,,	9·17	2·2	25·3	4·5	10·9	
Bristol 'Badmington'	Fast Sport	420-450-h.p.-520 Jupiter VI.	,,	1,840	2,460	24·07	22·07	5·0	5·0	Ailns. 12·5 211	,,	10·0	3·27	24·7	0·5	10·3	T. Wing Bristol 1a B.Wing. R.A.F. 25
Bristol 'Bristol'	Fighter	420-h.p. Jupiter	,,	1,990	3,125	33·83	27·5	6·33	4·75	307	,,	—	—	—	—	—	
Hawker 'Woodcock'	Night Fighter	380 Jupiter IV.	,,	2,080	3,040	32·5	32·5	6·0	6·0	346	,,	—	—	—	—	—	
Boulton & Paul 'Partridge'	Fighter	Jupiter VII. Supercharged	,,	2,021	3,100	35·0	31·0	5·5	4·5	311	,,	11·67	4 58	43·3	4·8	9·5	
Gloster 'Goldfinch'	,,	,,	,,	2,340	3,200	30·0	27·8	5·25	5·25	274	,,	9·5	3·9	31·4	8·5	9·0	
Armstrong-Whitworth 'Siskin'	,,	Jaguar Major, Geared and Supercharged	,,	—	3,180	33·17	—	7·0	3·5	293	—	—	—		—	—	

TABLE XI (*contd.*)—GEOMETRICAL PARTICULARS OF VARIOUS AEROPLANES. SINGLE ENGINE OR TWO SEATERS.

Aeroplane.	Purpose.	Engine.	Type.	Weight. Lbs.		Main Planes.					Tail.						
						Span. Ft.		Chord. Ft.									
				Empty.	Full.	Top.	Bottom.	Top.	Bottom.	Total Area.	Type.	Span.	Chord.	Total Area.	Fin Area.	Rudder Area.	
Westland 'Widgeon'	Sport	Cirrus II. or Genet	Land	852 775	1,400 1,323	36·0	—	6·0	—	Sq.ft. Ailns. 28 200	Monoplane	Ft. 8·33	Ft. 3·25	Sq.ft. 23·0	Sq.ft. 2·0	Sq.ft. 10·5	
Spartan 'Arrow'	Sport	120-h.p. Gipsy II.	,,	965	1,560	30·5	30·5	—	—	251	,,	—	—	—	—	—	2 Str.
A.W. 'Atlas'	Army Co-opn.	450-h.p. Jaguar	,,	2,550	4,000	39·6	—	6·6	5·6	391	,,	—	—	50·9	—	29·75	
De Haviland D.H. 60X. Moth	Sport	84 h.p. A.D.C. Cirrus	,,	885	1,550	30·0	30·0	4·37	4·37	—	,,	—	—	—	—	—	2 Str.
Gloster 'Goring I.'	H. Perf. Recon. or Bomber	450-h.p. Jupiter VIII.	,,	—	5,600	42·0	33·33	7·0	5·75	450	,,	—	—	—	—	—	W. Section Modified Jonkowski
Hawker 'Horsley'	Day Bomber	690-h.p. R.R.CondorIII.	,,	4,760	7,800	59·75	49·83	8·5	5·17	692	,,	—	—	—	—	—	W. Section Modified Göttingen
Parnall 'Imp'	Sport	87-h.p. A.S. Genet	,,	850	1,320	25·5	21·5	4·0	4·0	176	,,	—	—	—	—	—	W. Section Modified R.A.F. 31
Vickers 'Vivid'	Reconnaissance	590-h.p. Lion XI.	,,	3,560	5,550	45·08	39·58	8·25	7·0	593	—	—	—	—	—	—	
Westland 'Wapiti'	,,	450-h.p. Jupiter VI.	,,	2,760	4,568	46·42	46·42	5·75	5·75	488	,,	—	—	—	—	—	
Vickers 'Vellore'	Freight Carrier.	525-h.p. Jupiter IX.	,,	4,550	9,550	76·0	76·0	9·5	9·5	1,416	Biplane	17·17	3·5	105·5	—	44·5	

TABLE XI. (*contd.*)—GEOMETRICAL PARTICULARS OF VARIOUS AEROPLANES. MULTI-ENGINE.

Aeroplane.	Purpose.	Engine.	Type.	Weight. Lbs.		Main Planes.					Tail.				Fin Area.	Rudder Area.	
				Empty.	Full.	Span. Ft.		Chord. Ft.		Total Area.	Type.	Span.	Chord.	Total Area.			
						Top.	Bottom.	Top.	Bottom.								
										Sq.ft.		Ft.	Ft.	Sq.ft.	Sq.ft.	Sq.ft.	
Handley Page 'Hampstead'	Commercial	3–385-h.p. Jaguar IV.	Land	8,920	14,500	79·0	79·0	10·0	10·0	1,564	Mono-plane	24·0	7·5	164·0	50·0	40·0	
De Haviland D.H. 66	Commercial	3–450-h.p. Jupiter	,,	9,060	14,700	79·5	79·5	10·0	10·0	Ailns. 126 1,536	Bi-plane	T. 17·67 B. 23·5	2·75 6·5	178·0	23·7	44·9	7 passengers and goods
Westland 'Wessex'	Commercial	3–310-h.p. Genet Major	Land	3,425	5,750	57·5	—	9·5	—	490	Mono-plane	18.3	4·6	44·5	7·2	21·7	R.A.F. 34
Saunders 'Medina'	Commercial	2–450-h.p. Jupiter IV.	,,	8,060	11,560	52·0	58·0	9·33	9·33	1,000	,,	—	—	—	—	—	10 passengers
Supermarine 'Solent'	Torpedo Carrier	3–421-h.p. Jaguar	,,	9,840	16,300	75·0	75·0	11·0	11·0	1,576	—	—	—	—	—	—	Capacity, Normal, 210 galls. No Torpedoes, 560.
Blackburn 'Iris' II'	Coastal Patrol and Reconnaissance	3–665-h.p. Condor IIIA.	,,	17,350	27,000	95·5	95·5	13·5	13·5	2,460	Bi-plane	—	—	—	—	—	Capacity, Normal 680 galls. max. 1000 ,,
Short 'Scipio'	Commercial	4–525-h.p. Jupiter IX.	Boat	17,900	30,310	113·0	92·5	13·5	13·5	2,665	Mono-plane	32·7	11·0	319	—	—	16 pass. 3 crew. 720 galls.
Boulton & Paul 'Sidestrand'	Day Bomber	2–450-h.p. Jupiter VI.	,,	5,275	8,850	72·0	72·0	7·0	7·0	944	Mono-plane	17·67	7·5	119·3	11·0	31·8	
Short 'Calcutta'	Commercial	3–525-h.p. Jupiter IX.	Boat Sea-plane	12,600	20,200	93·0	76·5	11·5	11·5	1,825	,,	29·0	9·0	217·0	56·0	Main 49·0 Servo 7·6	
Supermarine 'Southampton'	Patrol or Bomber	2–450-h.p. Lion	,,	9,010	14,300	75·0	75·0	10·0	10·0	1,426	,,	24·0	7·4	170·0	55·5	45·5	

TABLE XII.—PEFORMANCE PARTICULARS OF VARIOUS AEROPLANES. SINGLE SEATERS.

Aeroplane.	Engine Horse Power.	Total Weight.	Main Plane Area.	Loading.		Speeds (m.p.h.) Ground.		Speeds (m.p.h.) at heights.					Ceiling. Ft.		Climbs (mins.) to—					Rate of Climb at G.L. ft./min.	
				Main Plane.	Power.	Landing	Max.	5,000 ft.	10,000 ft.	15,000 ft.	20,000 ft.	25,000 ft.	Service.	Absolute.	5,000 ft.	10,000 ft.	15,000 ft.	20,000 ft.	25,000 ft.		
		lbs.	sq. ft.	lbs./ft.²	lbs./h.p																
Avro 'Avian'	110	2,225	262	8·5	20·2	50	115	—	—	—	—		10,000	12,500	13·0	41·5		—	—	485	
Armstrong-Whitworth 'Starling'	450	—	—	—	—	50	—	—	—	150	—	—	—	30,000	—	—	—	—	—	—	
Bristol 'Badmington'	450	2,460	211	11·7	4·82	—	—	—	—	—	--	—	—	—	—	—	—	—	—	—	
Bristol 'Bristol'	420	3,125	307	10·17	7·44	—	148	156	176	176	172	166	29,200	30,300	2·9	5·4	8·2	12·1	18·8	—	V = 194 at 29,000 ft. Climb 29,200 ft. in 31 mins.
Hawker 'Woodcock'	380	3,040	346	—	—	47	143	—	138	128	115	—	21,500	—	—	8·8	16·0	27·9	—	—	
Boulton & Paul 'Partridge'	Jupiter VIII. Supercharged 420	3,100	311	9·9	7·35	61	—	—	167	—	164	—	29,000	—	—	6·5	—	15·1	—	—	
Gloster 'Goldfinch'	,,	3,200	274	—	—	59	—	—	172	167	157	—	26,800	28,000	3·25	6·5	10·3	16·1	28·0	—	
Armstrong-Whitworth 'Siskin'	500–550	3,180	—	—	—	60	—	—	181	186·5	185·5	180·5	32,700	33,800	—	6·5	9·25	12·75	17·75	—	

TABLE XII. (*contd.*)—PERFORMANCE PARTICULARS OF VARIOUS AEROPLANES. SINGLE ENGINE OR TWO-SEATERS.

Aeroplane.	Engine Horse Power.	Total Weight.	Main Plane Area.	Loading.		Speeds (m.p.h.) Ground.		Speeds (m.p.h.) at heights.					Ceiling. Ft.		Climbs (mins.) to—					Rate of Climb at G.L. ft./min.	
				Main Plane.	Power.	Landing.	Max.	5,000 ft.	10,000 ft.	15,000 ft.	20,000 ft.	25,000 ft.	Service.	Absolute.	5,000 ft.	10,000 ft.	15,000 ft.	20,000 ft.	25,000 ft.		
		lbs.	sq. ft.	lbs./ft.²	lbs./h.p															
Westland 'Widgeon'	95 75	1,400 1,323	200	—	—	42	100	—					14,000	—	11	28·0	—	—	—	560	
Spartan 'Arrow'	—	1,560	251	6·22	13·0	38	106	—	—	—	—	—	—	—	—	—	—	—	—	830	Range 280 m. with 22 galls.
A.W. 'Atlas'	450	4,000	391	10·03	9·0	54	143·5	140	134	125	—	—	17,300	19,000	5·25	12·5	26·0	—	—	—	Range 4 hrs. at cruising sp. of 120 m.p.h.
De Haviland D.H. 60 X. Moth	84	1,550	—	6·37	18·5	44	102	97	—	—	—	—	—	14,500	14	37·0	—	—	—	500	Range, 5 hrs 410 m.
Gloster 'Goring' I.	450	5,600	450	12·43	12·43	51	—	4000 ft. 136	133	122	—	—	16,500	—	4000 4·5	13·1	27·5	—	—	—	Range 6½ hrs. at 15,000 ft.
Hawker 'Horsley'	690	7,800	692	—	—	50	126	—	120	114	—	—	20,000	—	—	11·75	24·0	—	—	—	Range 10 hrs. Capacity 230 galls.
Parnell 'Imp'	87	1,320	176	—	—	—	—	—		—	—	—	—	—	—	—	—	—	—	—	U.C. travel 5 ins.
Vickers 'Vivid'	590	5,550	593	9·35	9·4	59	152	150	145	138	—	—	18,600	20,100	—	10·0	18	—	—	—	Capacity 210 galls.
Westland 'Wapiti'	450	4,568	488	9·36	10·15	47	—	135	129	123	—	—	22,000	24,000	—	10·6	19	—	—	—	Range 500 m. at full throttle
Vickers 'Velore'	525	9,500	1,416	6·70	18·1	50	110	106	101	—	—	—	12,000	14,700	10·5	29·0	—	—	—	—	

TABLE XII. (*contd.*)—PERFORMANCE PARTICULARS OF VARIOUS AEROPLANES. MULTI ENGINE.

Aeroplane.	Engine Horse Power.	Total Weight.	Main Plane Area.	Loading.		Speeds (m.p.h.) Ground.		Speeds (m.p.h.) at heights.				Ceiling. Ft.		Climbs (mins). to—					Rate of Climb at G.L. ft./min.	
				Main Plane.	Power.	Landing.	Max.	5,000 ft.	10,000 ft.	15,000 ft.	20,000 ft.	Service.	Absolute.	5,000 ft.	10,000 ft.	15,000 ft.	20,000 ft.	25,000 ft.		
		lbs.	sq. ft.	lbs./ft.²	lbs./h.p															
Handley Page 'Hampstead'	1,155	14,500	Ailns. 150 1,564	—	—	—	117	—	—	—	—	12,000	—	—	—	—	—	—	720	
De Haviland D.H. 66	1,500 / 1,350	14,700	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	
Westland 'Wessex'	930	5,750	490	11·7	18·6	52	108	—	—	—	—	10,000	12,300	—	—	—	—	—	530	Capacity 100 galls.
Saunders 'Medina'	900	11,560	1,000	11·56	12·85	—	115	—	—	—	—	—	—	—	—	—	—	—	—	Capacity 200 galls.
Supermarine 'Solent'	1,265	16,300	1,576	10·3	12·9	54	111	—	—	—	—	11,000	—	10·25	—	—	—	—	—	Range 920 max.
Blackburn 'Iris'	1,950	27,000	2,460	11·0	13·4	57	119	—	—	—	—	—	14,000	—	—	—	—	—	700	Capacity 680 normal
Short 'Scipio'	2,100	30,310	2,665	11·35	14·4	60	132	—	—	—	—	19,000	—	—	—	—	—	—	760	
Boulton & Paul 'Sidestrand'	900	8,850	944	9·4	9·8	49	125	130	129	122	—	21,500	23,000	—	10·5	19·0	—	—	—	
Short 'Calcutta'	1,575	20,200	1,825	11·0	12·8	58	120	—	—	—	—	15,700	—	6·5	16·0	—	—	—	937	
Supermarine 'Southampton'	900	14,300	1,426	10·0	15·9	56	108	—	—	—	—	—	14,000	—	—	—	—	—	610	

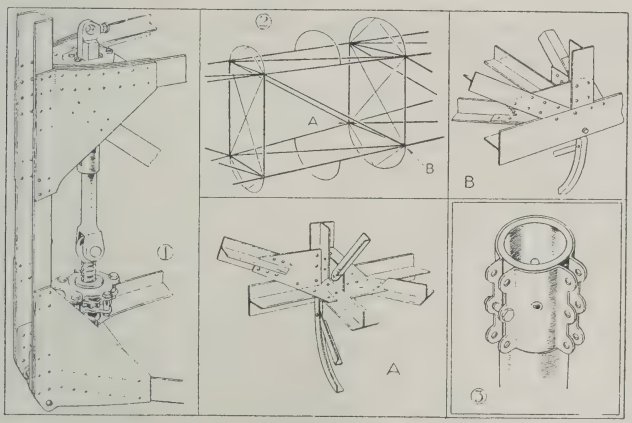
DRAG STRUTS.—These are solid drawn tubes and at the connection to the spar the web is stiffened locally as in fig. 49 (2).

MAIN PLANE RIBS.—Light and nose ribs are made from tubes, and are of the fully-braced type, while end ribs are made with T booms, as shown in figs. 50 (1) and (2).

For replacements, should a rib be damaged, a special rib with a detachable bottom boom has been produced and illustrated in (3). No riveting is required as it is designed to be fixed by bolts.

The rib lay-out is indicated in (4) and also shows the plates used for attaching the bracing diagonals to the booms.

INTERPLANE STRUTS.—Interplane struts of approximately streamline section are made by riveting together at the leading and trailing edges two curved sheets of heavy-gauge duralumin.



Reproduced by courtesy of 'Flight.'

FIG. 51.—Duralumin Fuselage Construction (Vickers).

FUSELAGE CONSTRUCTION.—This is carried out in tee and angle sections. Typical joints of the fuselage shown in skeleton in fig. 51 (2) are indicated at A and B.

The stern post and tail adjusting gear is indicated in (1), while (3) shows a fuselage fitting when tubular longerons are used.

Particulars of Various Aeroplanes.

Geometrical particulars of various types of aeroplanes are given in Table XI., pp. 2037–2039. The performance of these aeroplanes is given in Table XII., pp. 2040–2042.

Throughout this Section free use has been made of Reports of the Aeronautical Research Committee for which thanks are due to the Controller of H.M. Stationery Office from whom permission has been obtained.

ADDENDUM.

AIRSHIP DRAG.

In its general form the drag coefficient k_D is given in eqn. (4) on p. 1991.

In airship work it is convenient to use a coefficient which is related to the gross lift, *i.e.* to the airship volume or capacity.

Thus airship drag coefficient is given by

$$C = \frac{D}{\rho V^2 (\text{vol})^{\frac{2}{3}}}$$

where D is the drag in pounds.

The following table gives comparative predicted values of C for R 33 and R 101, based on model tests.*

Item.	R 33.	R 101.
Bare Hull	0·01150	0·00725
Control Car	—	0·00015
Engine cars	0·00277	0·00081
Fins	0·00251	0·00152
Interference between cars and hull	0·00057	0·00055
Miscellaneous	0·00121	0·00090
Radiators	0·00121	0·00038
	0·01977	0·01156

For R 33 and R 101 the values of (volume) $^{\frac{2}{3}}$ is 16,100 and 31,600 (before modification) respectively.

* *R.Ae.S. Jour.* 1929, p. 690.

SECTION XXXIII

PART II

AERO ENGINES.

(Contributed by Ernest W. Birch, A.M.I.A.E.)

Engines for Aircraft are either air- or liquid-cooled, and the majority operate on the four-stroke cycle. The various types in use are Vertical, Vee, 'Broad Arrow,' 'X,' Horizontal, Radial, and Rotary. Air-cooled engines are largely used, and the following advantages are claimed for this type of engine: (*a*) Less sensitive to sudden heat or cold; is easier to start and requires less warming up in cold weather; the oil does not heat unduly, and remains at working temperatures. (*b*) Top overhauls are easier to make. The grinding of valves involves only the removal of a cylinder, with no alteration to the timing—water-cooled engines, as a rule, require either the removal of the engine, or the dismantling of the camshafts and valve mechanism, and the whole engine requires retiming on assembly. (*c*) Due to the reduction in the number of parts, the radial air-cooled engine can usually be built cheaper than the water-cooled engine. The tendency in water-cooled engines is towards block construction to give a more rigid and lighter structure; this, however, involves the rejection of an entire block for the defect of a single cylinder.

Under certain conditions the liquid-cooled engine has definite advantages over the air-cooled engine, and at the present time both types have wide and varied application.

Air-cooled engines are not confined to fixed-radial and rotary types. In America the 'Liberty 12,' which is normally a water-cooled engine, has been developed as an air-cooled engine, and as such has developed slightly more than its rated output as a water-cooled engine.

In this country a four-cylinder in line air-cooled vertical engine has proved highly successful, and a six-cylinder in line with similar features is in production.

The most successful engines at the present time are really refinements of types available for some years past. Some of these improvements are greater compactness, resulting in lighter weight and increased rigidity; higher crankshaft speeds; improvements in valve operating mechanism, valve cooling and valve materials; improvement in bearing design, and improvement in volumetric efficiency. The success of the inverted 'Vee' type, and the static radial air-cooled engine, of reduction gearing and of the rotary inductor supercharger, has contributed a large share to recent progress in engines.

The Pressure-Ignition engine is now receiving great attention, and several engines have passed the experimental stage.

Rolls-Royce Engine, Type F.

Twelve-cylinder water-cooled type, having two rows of cylinders at 60° fitted with spur drive reduction gear for airscrew.

The engine can be supplied with either 0·632 to 1 or 0·553 to 1 ratio reduction gear, also with either 6 to 1 or 7 to 1 compression ratio pistons. With 6 to 1 compression pistons the engine will develop 490 b.h.p. at normal speed at ground level. If fitted with 7 to 1 compression pistons 480 b.h.p., and this power can be regained at an altitude of 3,000 ft. by opening the throttle to full open.

The engine is designated by four series numbers according to the gear ratio and ratio of compression, the leading particulars of each series being as follows:—

Series No.	F.XI.A.	F.XI.B.	F.XII.A.	F.XII.B.
No. of cylinders	12	12	12	12
Bore and stroke	5″ × 5·5″	5″ × 5·5″	5″ × 5·5″	5″ × 5·5″
Normal b.h.p.	490	480	490	480
Normal speed (crankshaft)	2,250	2,250	2,250	2,250
Maximum speed	2,500	2,500	2,500	2,500
Normal speed of propeller	1,421	1,421	1,244	1,244
Reduction gear ratio	0·632	0·632	0·552	0·552
Compression ratio	6 : 1	7 : 1	6 : 1	7 : 1
Fuel consumption at normal power and speed gallons/hour	30	28¼	30	28¼
Oil consumption pints/hour	5	5	5	5
Weight, including carburetters, magnetos, engine feet, reduction gear, and airscrew hub—lbs.	865	865	865	865

Cylinders in banks of six, mounted on crankcase at 60° with each other, are of cast aluminium provided with steel liners and renewable valve seats. Heads integral with cylinder blocks. Two inlet and two exhaust valves per cylinder, operated by camshaft and rockers mounted on top of cylinder heads, the whole being totally enclosed by means of a removable cylinder head cover.

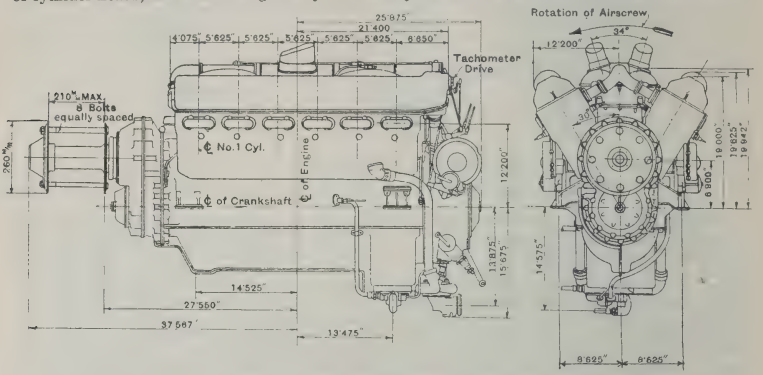
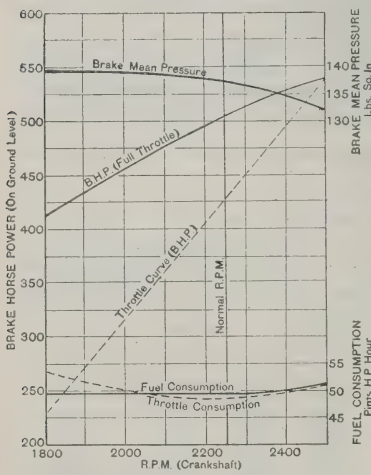


FIG. 1.—Rolls-Royce Engine (Type F).

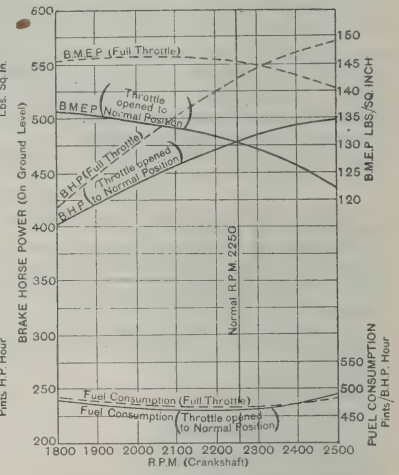
Valve mechanism positively lubricated. Inlet and exhaust valves of each cylinder block operated by one camshaft.

Gears for driving camshaft and all auxiliaries driven from rear end of crankshaft through the medium of a spring drive to eliminate crankshaft torsional vibrations. The camshafts are driven



Compression ratio = 6 : 1.
Atmospheric temperature = 16° C.
Atmospheric pressure = 29·97 ins. mercury.
Curves shown are correct to within 2½%.

FIG. 2.—Brake Horse Power and Consumption Curves of Rolls-Royce F. XI. A. and F. XII. A. Engines.



Compression ratio = 7 : 1.
Atmospheric temperature = 16° C.
Atmospheric pressure = 29·97 ins. mercury.
Curves shown are correct to within 2½%.

FIG. 3.—Brake Horse Power and Consumption Curves (on ground level) of Rolls-Royce F. XI. B. and F. XII. B. Engines.

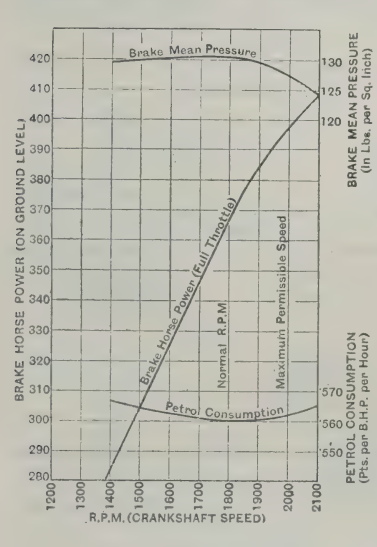
by means of inclined tubular driving shafts with bevel gears at upper and lower ends. Out-of-alignment and expansion effects are allowed for by hardened serrated couplings. Pistons, special aluminium alloy forgings, machined all over. Four piston rings, three arranged as compression rings above gudgeon pin and one scraper ring at base of piston skirt. The compression rings are prevented from rotating by means of stops.

Gudgeon pins 'floating' fit in piston bosses and connecting rod small end.

Connecting rods are 'H' section of the 'forked' type, machined all over to reduce weight variations. A divided white metal lined steel block is bolted to the forked rod. The plain rod works upon the centre portion of the steel block, the latter having white metal bearing surface. Small ends of both rods fitted with floating phosphor-bronze bushes. All bearings positively lubricated under pressure.

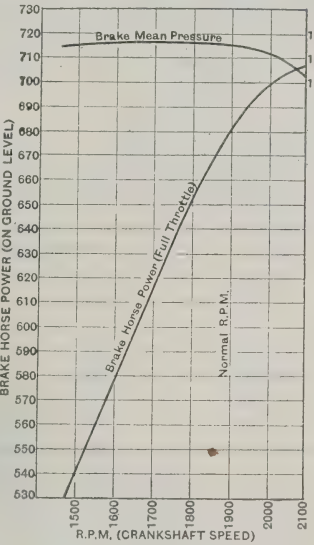
Six-throw crankshaft carried in seven bearings. All journals and crankpins bored for lightness and to convey oil to all bearings and connecting rods.

Crankcase of aluminium alloy, made in two halves of box section suitably stiffened by ribs.



Rolls-Royce 'Eagle,' Series IX.*

Compression ratio = 5·22.
Atmospheric temperature = 16° C.
Atmospheric pressure = 29·97″ Hg.



Rolls-Royce 'Condor,' Series III.

Compression ratio = 5·3.
Atmospheric temperature = 16° C.
Atmospheric pressure = 29·97″ Hg.

FIG. 4.—B.h.p. and Consumption Curves for Rolls-Royce Engines.

Main bearings, consisting of divided mild steel shells, white metal lined, are held in the upper half crankcase by special caps which are secured by bolts on each bearing. Facings are formed on either side of the caps which fit between corresponding facings formed on cheeks within the upper half crankcase. Transverse bolts pass through the crankcase and cups. This arrangement secures the rigidity of one in which the caps are integral with the lower half, while retaining the advantages of caps bolted only to the upper half.

A single spur gear reduction is fitted at the front end of crankshaft. The pinion is driven from the crankshaft through a short shaft, having teeth at the inner edge to engage an internally toothed flange bolted to the crankshaft, and at the outer end teeth engaging with teeth cut inside a part of the gear pinion. This shaft prevents loads from the gear pinion coming on to the crankshaft. A centrifugal water circulating pump is fitted below the auxiliary gear case. Lubrication is on the 'dry sump' systems.

* For details of Rolls-Royce 'Eagle' Series IX. engine, see *Engineer's Year Book*, 1928, pp. 2055–7.

The 'F' series of engines can be fitted with a supercharger of two different types, *i.e.* the moderate type, which permits a certain amount of supercharging at take-off and maintains the normal horse-power to 3,000 ft. or with a full supercharger, which is capable of maintaining the ground level rated power up to 11,500 ft. The supercharger consists of a high-speed centrifugal fan mounted co-axially with crankshaft at rear of engine and drawing its air through the carburettors. Aluminium alloy impeller with radial blades driven a speed multiplying gear comprising a system of three planets. These are frictionally driven from a pinion on crankshaft by means of slippers held in engagement with the insides of the gear rims by light springs. As the speed of rotation of crankshaft and slippers increases, the radial pressure of springs is augmented by centrifugal force, and consequently the torque which the planets can transmit increases as the square of the speed. As the torque required to drive impeller increases at a like rate, the gear is capable of driving the impeller with a predetermined margin of torque above that causing slip which is constant throughout the working range of speed. The object of this drive is to protect the gearing against damage on sudden acceleration or deceleration of the crankshaft due to inertia of impeller. Gear ratio between crankshaft and impeller is 1 : 5·5 or 1 : 10, giving calculated impeller speeds of 12,375 and 22,500 r.p.m.

Rolls-Royce 'H' Type Engine.

General design similar to 'F' type. Leading particulars as follows :

Number of cylinders	12	Normal speed propeller	954
Bore, in.	6·6	Gear ratio	0·477
Stroke, in.	6·6	Fuel consumption at normal power and speed, galls. per hour	53
Normal b.h.p.	825	Oil consumption, pints per hour	9
Normal speed (crankshaft), r.p.m.	2,000		
Maximum speed (crankshaft), r.p.m.	2,200		

Weight of engine, including carburettors, magnetos, engine feet, reduction gear, airscrew hub, but excluding exhaust boxes, radiator, airscrew, oil, fuel, and water = 1,460 lbs.

Performance.—Test at full throttle at ground level. Compression ratio, 5·25 : 1. Series HX.

R.P.M. Crankshaft.	B.H.P.	B.M.E.P. Lbs. per sq. in.	Fuel Consumption. Pints/b.h.p./hr.
1,730	700	143·0	0·540
1,800	740	146·0	0·535
1,900	784	146·5	0·530
2,000	830	147·5	0·525
2,100	866	146·5	0·530
2,200	900	145·0	0·537

Rolls-Royce Engines. ('Condor' Series IIIA.)

The 'Condor' engine is of the 12-cylinder water-cooled 'Vee' type, fitted with a spur drive reduction gear for the airscrew.

The leading particulars are :—

Number of cylinders, 12 ; bore, 5½ ins. ; stroke, 7½ ins. ; normal b.h.p., 665 ; normal speed (crankshaft), 1,900 r.p.m. ; maximum speed (crankshaft), 2,100 r.p.m. ; normal speed (propeller) with 0·477 reduction gear, 907 r.p.m.

Fuel consumption at normal power and speed, 41 galls. per hour.

Oil consumption, 1·3 galls. per hour.

Weight of engine, 1,182 lbs. (including carburettors, magnetos, engine feet, distributor, and pipes for power starter, etc., but excluding reduction gear, exhaust boxes, radiator, airscrew, oil, fuel, and water).

Weight of engine as above, but including reduction gear, 1,338 lbs.

Napier Engine. (Lion.) Series XI.

The leading particulars are as follows :

Number and arrangement of cylinders, 12, three blocks of four each, one vertical, two at 60° ; bore, 5½ ins. ; stroke, 5½ ins. ; Compression ratio, 6·0 : 1 ; normal b.h.p. and speed, 530 b.h.p. at 2,300 r.p.m. ; direction of rotation of crank, anti-clockwise, viewed from front ; direction of rotation of air screw, clockwise, viewed from front, *i.e.* L.H. tractor or R.H. pusher ; ratio of gear reduction to airscrew, 49–26 ; oil pressure, maximum 70 lbs./sq. in., normal 55–60 lbs./sq. in., minimum 40 lbs./sq. in. ; oil consumption, 6–12 pints per hour ; type of carburettor, special Napier Claudel Hobson Triplex (see page 2068).

Fuel consumption, 280 pints per hour at normal load near ground.

Type of magneto, two B.T.H. A.V. 12E (dual ignition), or A.V. 12F.

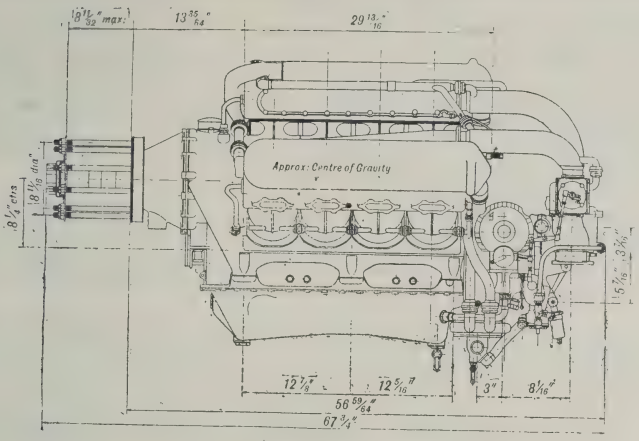


FIG. 5. Napier 'Lion' Engine.

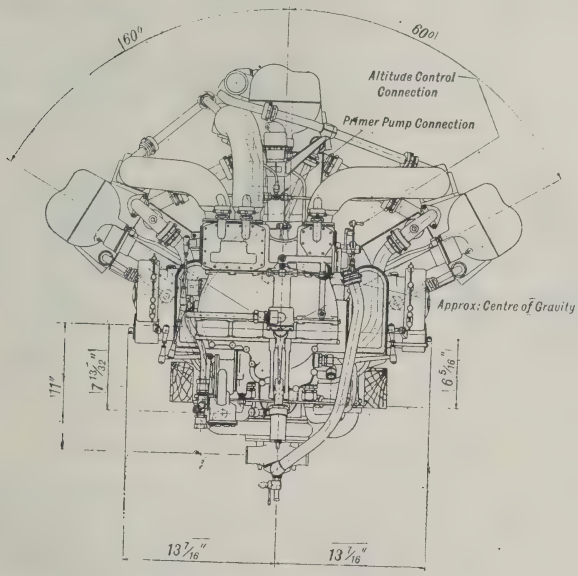


FIG. 6.—Napier 'Lion' Engine (Rear View).

Firing order, 1, 5, 9, 3, 7, 11, 4, 8, 12, 2, 6, 10.

Numbering of cylinders, airscrew { 12, 11, 10, 9 / 5, 6, 7, 8 / 4, 3, 2, 1

Speed of magneto, 1½ engine speed.

Magneto timing, 40° before t.d.c. at full advance on exhaust side, 45° before t.d.c. on inlet side.

Inlet valve opens (deg. on crank), 10° after t.d.c.

Inlet valve closes (deg. on crank), 51° after b.d.c.

Maximum lift of inlet valves, 0·344 ins.

Clearance of inlet valve, 0·020 in.

Exhaust valve opens (deg. on crank), 48° before b.d.c.

Exhaust valve closes (deg. on crank), 10° 44′ after t.d.c.

Maximum lift of exhaust valves, 0·354 in.

Clearance of exhaust valve, 0·020 in.

Weight of engine complete without water, fuel, or oil, 995 lbs. (approx.).

Maximum speed, under any conditions, 2,585 r.p.m.

Cylinders.—Machined from steel forgings with integral combustion heads. Separate sheet-steel jackets to each cylinder barrel. Cylinders are assembled in rows of four with a single aluminium head and camshaft casing combined for each row.

Pistons.—Forged aluminium alloy, trunk type, with flat tops. Hollow gudgeon pin floats in piston and is located sideways by bronze buttons. Two gas rings and two scraper rings fitted.

Connecting Rods.—The rods for central cylinders are master rods with two lugs on each side to accommodate wrist-pins for auxiliary rods. All rods of ' H ' section, split white-metal lined big ends for master rods and bronze-bushed for gudgeon and wrist-pins.

Crankshaft.—Of normal four-throw type, supported on five roller bearings and one plain bearing (fig. 7). The bearings between crank-throws are bored so that they may be threaded over the crankwebs and pins into place, and split packing bushes to make up diameter of shaft to fit inner races are then pressed into place.

Crankcase.—Aluminium alloy casting. Upper half provides three sets of cylinder facings, contains the crankshaft bearing supports, and includes a housing at forward end for airscrew reduction gear. Lower half of crankcase forms an oil sump.

Valve Gear.—Four symmetrically arranged valves per cylinder operated by two separate camshafts, one for inlet and one for exhaust. Two camshafts geared together by equal pinions at rear end of camshaft casing. One shaft is prolonged beyond pinion and carries a bevel pinion which mates with a bevel gear or camshaft drive shaft for that row of cylinders.

Airscrew Drive.—Airscrew shaft driven through spur reduction gear of 0·53 to 1 ratio.

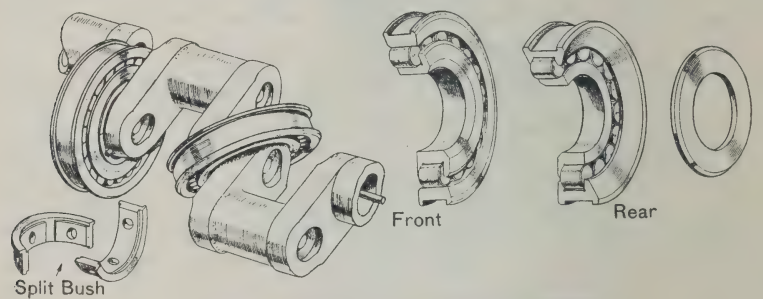


FIG. 7.—Crankshaft Bearings of Napier ' Lion ' Engine.

Auxiliary Drives.—Rear end of crankshaft drives a short extension shaft through a dog coupling. This carries a pair of bevel wheels back to back. The inner bevel drives the two camshaft drive-shafts for outer rows of cylinders. Vertical (central) camshaft drive is taken from a bevel on port shaft. A downward-projecting vertical shaft, driven from this first bevel drives water pump directly, and oil pumps through worm gears. Second bevel on crankshaft extension drives two horizontal shafts for magnetos. All auxiliary gears contained in separate casing bolted to rear end of crankcase.

Carburation.—A triple carburettor, carried on a bracket at rear of engine supplies gas to trough section aluminium induction channels, bolted to one side of each cylinder-head casting.

Lubrication.—Two scavenger pumps and one pressure pump fitted. Oil from pressure pump passes into the hollow crankshaft and is supplied direct to each big-end. A separate pressure lead taken to each camshaft.

Ignition.—Dual ignition provided by two 12-cylinder magnetos. Two sparking plugs per cylinder.

Weight Dry.—995 lbs. (452 kg.).

Performance.—Rated output 530 b.h.p. at 2,350 r.p.m. (1,247 airscrew r.p.m.). Average power actually developed, 540 b.h.p. at 2,350 r.p.m., and 575 b.h.p. at 2,585 r.p.m. Average fuel consumption at full load, 0·50 lbs. per b.h.p. hour, using a fuel mixture of Aviation spirit to B.E.S.A. spec. No. 121, part B, and Benzol. Average oil consumption, 0·0235 lb. per b.h.p. hour during Air Ministry 2-hour endurance test at $\frac{9}{10}$ full power.

Overall Dimensions.—Length to centre of airscrew, 5 ft. 1 in.; width, 3 ft. 6 ins.; height, 3 ft. 3 ins.

Napier 'Lion,' Series VIIa.

This is a direct drive high-compression engine specially developed for high speed. Main characteristics as for standard 'Lion' engine. Bore, 5½ ins.; stroke, 5½ ins. Chief modifications introduced to decrease head resistance, shorter connecting rods, and magnetos fitted at front end of engine. Compression ratio, 10 : 1. Maximum H.P. developed, 900 b.h.p. at 3,300 r.p.m. Weight, 850 lbs., or 0·944 lbs. per H.P.

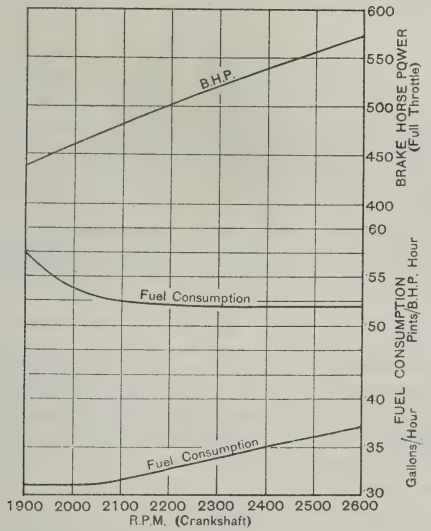
Napier 'Lion,' Series VIIb.

Similar to Series VIIa except that a double reduction gear is fitted. This reduction is obtained through a layshaft, mounted over airscrew shaft and crankshaft, which are co-axial. Reduction ratio, 1·306 to 1. Maximum power developed, 875 b.h.p. at 3,300 r.p.m. Weight, 930 lbs., or 1·063 lbs. per h.p.

Overall Dimensions.—Series VIIa and b. Length, 5 ft. 6½ ins.; width, 3 ft. 2⅜ ins.; height, 2ft. 10½ ins. A special Benzol-petrol fuel with admixture of Ethyl fluid is used. Consumption under racing conditions, 50 galls. per hour (approx.), or 0·46 pint per b.h.p. hour.

Napier 'Lion,' Series V.

Details of this engine are given in the 1929 edition, page 2154.



Consumption ratio = 6 : 1.
Fuel 77% British Standard Aviation Spirit Spec., No. 121—1923, part B, 23% Benzole.

Fig. 8.—Full Throttle Power and Consumption Curves. Napier-Lion Engine, Series XI.

Armstrong-Siddeley Engines.

A series of engines of the static radial type, known as *Leopard*, *Jaguar*, *Lynx*, *Mongoose*, and *Genet*, the general construction of which is similar.

Special functions of the engines are as follows :—

Leopard.—Fourteen cylinders, 700–805 h.p., for civilian use or for carrying troops or torpedoes.

Jaguar Major.—Fourteen cylinders, 500–550 h.p. Interchangeable with the 'Jaguar,' and recommended where extremely high performance is required. 'Jaguar,' 14 cylinders, 460–500 h.p. geared, for civil or service requirements. The supercharged 'Jaguar,' suitable for all types of military aircraft in which the maintenance of power at altitude is essential.

Lynx Major.—Seven cylinders, 260–280 h.p., for civil and service uses. 'Lynx,' 7 cylinders, 215–230 h.p., for use in land or sea-training machines, civil aircraft, photographic and survey duties, and for application to twin or multi-engined aircraft, two-seater reconnaissance training machines. Single seater fighter, deck landing, and twin-engined seaplanes. The geared or supercharged 'Lynx' serves the same purpose as the gear or supercharged 'Jaguar,' but the engine is only half the power.

Mongoose.—Five cylinders, 150–165 h.p., for training aircraft.

Genet Major.—Five cylinders, 100 h.p., for light multi-engined aircraft. 'Genet,' 5 cylinders, 80–88 h.p., for light aircraft.

Leading Particulars of Armstrong-Siddeley Aero Engines.

Engine.	Leopard. D	Jaguar.				Lynx.			Mongoose.	Genet.
		D	G	S	SG	D	G	S		
No. of cylinders	14	14	14	14	14	7	7	7	5	5
Bore, ins.	6	5	5	5	5	5	5	5	5	4
Stroke, ins.	7·5	5·5	5·5	5·5	5·5	5·5	5·5	5·5	5·5	4
Swept volume, c. ins.	2,969	1,513	1,513	1,513	1,513	757	757	757	540	251
Normal engine, r.p.m.	1,500	1,700	2,000	1,700	2,000	1,900	2,000	2,000	1,850	2,200
Normal b.h.p.(sea level)	700	400	460	—	—	215	215	—	150	80
Maximum b.h.p. (sea level)	805	440	510	380	380	230	235	190	165	88
Normal r.p.m., propeller	1,500	1,700	1,314	1,700	1,314	1,900	1,314	2,000	1,850	2,200
Rated altitude, ft.	—	—	—	9,000	14,500	—	—	14,500	—	—
Power at rated altitude	—	—	—	410	460	—	—	230	—	—
Engine weight, lbs.	1,465	805	890	850	930	505	590	550	365	210
Weight per b.h.p., lbs.	1·82	1·84	1·75	2·07	2·02	2·19	2·51	2·39	2·21	2·38

D = direct drive. G = geared. S = supercharged. SG = supercharged and geared.

All types LH tractors. Fixed ignition. Oil pressure, 60 lbs. per sq. in.

Maximum propeller and engine speed, 10 per cent. above normal.

Compression ratio, 5 : 1, except for Genet, 5·25 : 1.

A.D.C. Engines.

Horizontally opposed, air-cooled engines, specially designed for light aeroplanes. Two types, known as *Scorpion* and *Hornet*. *Scorpion*, two cylinders horizontally opposed. Crankcase in one piece with integrally cast induction manifold. Cylinders machined from the solid with fins formed integrally. Detachable cylinder head of cast iron. Inside of head and top of piston

of hemispherical form, with inlet and exhaust valves of same diameter. Two sparkling plugs to each head for dual ignition. Valves of high tensile steel operated by rockers and tubular tappet rods. Crankshaft and connecting rods machined from alloy steel stampings. Pistons of aluminium alloy with two rings at the top only. Camshaft with one exhaust and one inlet cam to serve both cylinders. Roller-type tappets.

Lubrication on dry sump principle, with pressure feed and scavenger pump. Ignition by B.T.H. magneto fitted with impulse starter. *Hornet* engine, four cylinder, horizontally opposed, and comprises the main elements of two *Scorpion* engines mounted in tandem about a common crankcase.

Leading particulars of engines are as follows :—

	Scorpion.	Hornet.
No. of cylinders	2	4
Bore, ins.	4·015	4·015
Stroke, ins.	3·6	4·8
Swept volume, c.c.s.	1,500	3,990
Normal b.h.p.	34	75
Normal r.p.m.	2,300	1,875
Maximum b.h.p.	40	82
Maximum r.p.m.	2,750	2,175
Fuel consumption, pts./b.h.p./hour	0·52	0·53
Oil " " "	0·04	0·035
Weight complete, lbs.	109	225
Specific weight (on normal power), lbs./h.p.	3·2	3·0

SUPERCHARGERS.

Four possible types available : (*a*) reciprocating ; (*b*) eccentric, or sliding vane ; (*c*) centrifugal ; (*d*) Root's blower. Owing to mechanical limitations, types (*a*) and (*b*) are not much used.

The centrifugal type may be driven by an exhaust turbine and this forms a compact unit, but the exhaust turbine involves difficulties in manufacture, and also it has been found that such a combination does not respond as quickly to the control as a mechanically driven supercharger. Further, if the driving ratio is constant, the proportional increase in the air temperature in passing through the supercharger will be constant, irrespective of the altitude, and at ground level the temperature rise will be excessive, necessitating the installation of coolers with a consequent increase in weight. At ground level, where no supercharging is required, the power necessary is as great as that required at high altitude.

With the Root's type a large volume of air is handled at ground level at small power consumption and temperature rise, as there is only slight compression. The power theoretically required to compress adiabatically 1 lb. of air per sec. from an absolute pressure P and temperature T to an absolute pressure P_1 is given by H.P. $= 0{\cdot}336T\left(\frac{P_1}{P}^{0{\cdot}289} - 1\right)$, and the efficiency of a supercharger is equal to the ratio of the theoretical horse-power, to the power actually absorbed (as determined by dynamometer). Tests made on a Root's type supercharger (*National Advisory Committee for Aeronautics, U.S.A.*), with rotors of 9¾ ins. diam. by 11 ins. long, showed a maximum efficiency of 82 per cent. at 600 r.p.m., compression ratio 1·2, the efficiency decreasing uniformly with the compression ratio to a little over 60 per cent. for a compression ratio of 2·3 for all speeds from 1,000–1,800 r.p.m. Volumetric efficiency decreased with increase in compression ratio. With a compression ratio of 2·25, volumetric efficiency was approximately 90 per cent. from 800 r.p.m. upwards. With a compression ratio of 1·5, the efficiency was 95 per cent. at less than 800 r.p.m. The maximum efficiency obtained was 97 per cent.

The volumetric efficiency decreases with the speed, in which respect the Root's type differt from the piston type. The loss in volumetric efficiency is due not so much to the wire drawing as the inlet, as to leakage past the rotors. An important point is the maintenance of a small clearance between rotor and casing.

Tests to destruction of the rotors showed that an aluminium alloy rotor failed at 9,600 r.p.m., and a magnesium alloy rotor at 13,500 r.p.m. The designed speed of the rotor was 2,550–3,200 r.p.m. Complete assembly weighed 151 lbs., being made up of supercharger 88 lbs., coupling 10 lbs., supercharger tank 24 lbs., ducts 29 lbs.

SERVICE DATA FOR AIR-COOLED ENGINES. (*Roy Fedden.*)

	A.	B.	C.	D.	E.
No. of engines operated	20	30	91	5	4
Period covered by data, years	2	3	—	2	2·5
Total hours operation	9,280	25,000	36,464	1,450	3,083
Average hours operation per engine	468	830	401	290	771
Max. recorded hours, one engine	960	1,860	2,000	403	826
Average hours between overhaul	400	250	290	360	280
Average man hours for overhaul	1,356	332	—	100	110
Average operating r.p.m.	1,500	1,550	1,625	1,800	1,780
Average fuel consumption, galls./hour	19·7	20	11·3	4·0	4·5
Average oil consumption, galls./hour	0·57	0·8	0·42	0·5	1·0
Cost per running hour, including maintenance, overhaul, and repair, shillings	—	17·5	—	1·1	—

A. Bristol Jupiter IV. engines. Imperial Airways. Middle East Route.

B. Gnome et Rhone Jupiter engines. K.L.M. Royal Dutch Air Route.

C. Wright Whirlwind engines (nine different operators).

D. Cirrus engines. Bristol and Wessex Aero Clubs.

E. Cirrus engines (from three aero clubs).

SERVICE DATA. GNOME AND RHONE JUPITER IV. ENGINES.
(Eleven engines of 16,120 total running hours. K.L.M. Royal Dutch Air Lines.)

Component.	Average Life to Date, Feb. 1929.
Crankcase	1,500
Crankshaft	1,000
Valve, inlet	1,000
Valve, exhaust	650
Valve guides, inlet	615
Valve guides, exhaust	650
Piston	200
Piston ring	165
Scraper ring	270
Cylinder	1,500
Main roller bearings	800
Master rod	750
Articulated rod	1,500
Gudgeon pins	360

HIGH-TEMPERATURE LIQUID COOLING.

For this purpose ethylene glycol $CH_2(OH)\cdot CH_2\cdot OH$ is used. Its boiling point is 388° F.; it has a relatively high specific heat, is non-injurious to metals and has excellent wetting properties. It is a clear, colourless, and odourless liquid, with a slightly sweet taste. It is also used as an anti-freeze for automobiles.

In the ordinary water-cooled engine the normal working temperature is about 180° F., and the most suitable operating temperature with ethylene glycol is 300° F. Using this liquid, the size of radiator required is approximately one-quarter that used for water, this being due to the greater temperature difference between the radiator and the cooling air. A modern military plane (American) has a capacity of 14·5 galls. of water. Using ethylene glycol, 8–5 galls. is sufficient, or a reduction of 41 per cent., and the plane weighs 3·5 per cent. less than with a water-cooled system. The normal radiator of such a plane offers approximately 15 per cent. of the total parasite resistance. Reducing frontal area of radiator 75 per cent. reduces total parasite resistance

of plane by about 12 per cent. When operating an aircraft engine at a cooling liquid outlet temperature of 300° F., it has been found that the following advantages are gained (*G. W. Frank*):—

(1) The amount of cooling area or size of radiator necessary is greatly reduced.

(2) A reduced amount of liquid is required in the cooling system.

(3) A considerable reduction in weight of the installed power plant, due to the reduction in size of radiator and the lesser amount of cooling liquid required.

(4) The parasite resistance of the plane is reduced, due to the reduction in size of radiator.

(5) It permits the engine combustion chamber walls and cylinder barrels to operate at temperatures comparable to those in an air-cooled engine, with the added advantage of uniform cooling.

Evaporative Cooling.

In this system the cooling fluid itself is not passed to the heat-dissipating surfaces but is circulated around the engine through a small tank, which is arranged so that by mechanical or other means the vapour formed is separated and conducted to an air-cooled condenser instead of the normal type of radiator, the condensate being returned from the condenser to the circulating tank by gravity or other means. This system has the following advantages :—

(*a*) Control of the cooling fluid temperature by the pilot is unnecessary.

(*b*) With correctly designed condensers overheating cannot take place.

(*c*) Time for engine warming greatly reduced.

(*d*) For fighting aircraft the exposed radiating surface, which is liable to be punctured by bullets, contains vapour under atmospheric pressure.

(*e*) The system can be so arranged as to provide for cockpit and passenger cabin heating.

Liquid Cooled Engines.

The advantages claimed for the liquid cooled engine may be briefly summarised as follows :—

(*a*) The necessary heat-dissipating surfaces can be provided in such a manner that they present the minimum head resistance or drag. In some cases the existing surfaces of the aircraft itself can be utilised as cooling surfaces.

(*b*) There are no projections around the nose of the fuselage to upset the air flow.

(*c*) Thermodynamic advantage obtained by the use of a cylinder in which it is possible to maintain an equalised temperature throughout the combustion walls while extracting heat at a very high rate.

(*d*) Due to absence of hot-spots a higher m.e.p. can be obtained. Due to this fact the fuel consumption of liquid cooled engines is at least 10 per cent. lower than for directly cooled engines.

(*e*) It is impossible to damage the water-cooled engine by over-heating during times when it is necessary to run at full throttle.

(*f*) After a long glide sufficient heat can be stored in the cooling system and carburetter jackets so as to ensure certain acceleration when the throttle is opened.

(*g*) All the working parts can be totally enclosed, thus minimising mechanical chatter. The silencing of the exhaust of a normal type engine is a comparatively simple matter.

(*h*) The normal type of twelve-cylinder engine with its rigid cylinder block construction and almost perfect balance can be produced, so that if well mounted, vibration is almost entirely absent.

(*i*) The liquid-cooled engine having every working part totally enclosed requires no attention from a grease gun or oil-can after comparatively short periods of running.

(*j*) Owing to the excellent cooling of valves and cylinders, life between top overhauls should be considerably longer than with other types of engines.

(*k*) Water is not the only cooling medium which can be employed. Ethylene glycol has been used which permits of the boiling point being raised to 388° F. The effect of this is to permit the use of smaller radiators, a saving in area of over 60 per cent. being possible. The freezing point is reduced to — 15° F., which removes the difficulties normally experienced with water in severe weather.

(*l*) By the use of supercharging and special fuels, outputs three and four times as great as are normally taken from the cylinders can be obtained. To withstand these increased outputs, larger bearing areas for a given cylinder capacity are necessary, and these can be obtained without seriously affecting the head resistance or drag.

Oil Engines for Aircraft.

Oil engines for aircraft have developed on two main lines : (*a*) reducing the weight of the usual type of Diesel engine ; (*b*) redesign of existing aero and lorry type petrol engines to run on oil fuel.

The air-cooled Vee engine is attractive for oil, as the cylinder spacing necessary for cooling coincides closely with crankshaft requirements. It may be inverted without difficulty, as problems of carburettor location and manifolding do not arise.

There are two principal objectives in the application of the high speed, heavy oil engine to aircraft : (*a*) reduction of fire risk ; (*b*) saving in cost of fuel in commercial operation. In addition, there is the elimination of electrical ignition which is often a source of failure, and also the magneto requires a complicated system of screening to prevent interference with the radio circuits.

The oil used does not give off an explosive mixture at ordinary temperatures, and can therefore be handled with freedom. There is no possibility of formation of an explosive mixture external to the working cylinders, such as always exists in the inlet manifold of a petrol engine. On the other hand, there is reason to believe that a heavy oil is more likely to ignite by flashing off a hot surface, such as the exhaust pipe.

Owing to the high compression ratio necessary to ensure ignition of the fuel, the oil engine has an inherently higher thermal efficiency than the petrol engine. Due to the necessity of promoting a rapid and complete mixing of the fuel with the air it is essential to provide a large excess of air with further improvement in the thermal efficiency. In a multi-cylinder engine the fuel distribution being mechanical can be tuned up more perfectly, especially under varying conditions of load and speed. This timing is more effective in service than carburettor adjustment. The oil engine has a lower mechanical efficiency, chiefly due to its larger size and increased inertia loading of its heavier reciprocating parts, which must deal with higher maximum pressures and the greater friction, especially of the piston and rings. The mechanical efficiency is about 84 per cent., compared with 90 per cent. for a corresponding petrol engine.

The petrol engine has a small pumping loss, but this is offset by the extra power required to drive the fuel pump in an oil engine.

The brake thermal efficiency of the oil engine is better than that of the petrol engine, especially at the lower load factors.

The distinctive feature of the oil engine is the fuel system, and experience goes to show that this can be made as reliable as the carburettor and manifold. Dust and water can be kept out of the fuel by suitable filter.

Ignition takes place by heat of compression alone, and the compression ratio varies between 13½ to 1 and 16 to 1, according to size of cylinder, and this necessitates a powerful starter for the engine. When once warmed up, the engine will continue to fire regularly in spite of a serious loss of compression or fall in air temperature. At altitude the maximum power falls off with decreasing air density, and it may be necessary to warm the inlet air. The oil engine does not suffer from sparking-plug failure, and lends itself to supercharging at all altitudes. Its bulk and frontal area are somewhat greater than those of the petrol engine.

The following figures (*P. E. Biggar*) give a comparison of a petrol and oil engine, being based on a modern British aero engine, and the performance of a number of experimental oil engines.

Rated output, 500 b.h.p.	Petrol.	Oil.
Engine size, twelve cylinder, ins.	5 × 5½	5½ × 7
Engine displacement, cu. ins.	1,296	1,995
Speed at rated output, r.p.m.	2,250	1,800
Piston speed ,, ,, ft./min.	2,060	2,100
B.M.E.P. ,, ,, lbs./sq. in.	136	110·5
Fuel consumption, lbs. per b.h.p./hr.	0·48	0·37
Corresponding brake thermal efficiency	28·5%	36·4%
375 b.h.p. cruising output.		
Estimated speed, r.p.m.	2,100	1,700
B.M.E.P., lbs./sq. in.	109	87·5
Fuel consumption lbs./b.h.p./hr.	0·455	0·35
Fuel consumption galls./hr.	22·5	14·75
Fuel cost (cruising) petrol at 1*s.* 8*d.* per gall.; fuel oil at 5*d.* per gall.	37*s.* 6*d.*	6*s.* 2*d.* per hr.

The lower B.M.E.P. for the oil engine will be raised as the technique of combustion improves. At present if more than 75 per cent. of the air charge is treated with fuel, the air becomes oversaturated with carbon, which is deposited as soot on cylinder walls and seriously interferes with lubrication and free working of piston rings. For non-supercharged engines the limiting value of the B.M.E.P. appears to be about 160 lbs. per sq. in.

The lower B.M.E.P. requires a larger engine for the same output. Owing to the higher maximum pressures of the oil engine, which normally reach 800 lbs. per sq. in. and 1,200 lbs. per sq. in. occasionally, with corresponding values of 450 and 700 lbs. per sq. in. for the petrol engine, the structure of the oil engine must be more robust and have larger and stiffer bearing surfaces. The petrol engine in the above table is specified to have a dry weight of 865 lbs., or 0·66 lb. per cu. in. An oil engine can be built for 0·73 lbs. per cu. in.

Armstrong Siddeley 'Jaguar' Engine.

Air-cooled **static radial type, having 14 cylinders arranged in two banks of seven, and working on two cranks set at 180° to each other.**

The body of the engine consists of a cast aluminium crank case B, which is attached to the aeroplane by means of a pressed steel bearer flange F, and which is bored to receive the cylinders C, and the valve tappet guides T. A back cover attached to the engine body carries the magnetos and carburettors, and also the rear crankshaft bearing R. The inside of the back cover forms a casing for an induction fan M, which is attached to the crankshaft.

The crankshaft is a one-piece alloy steel forging and is balanced by weights W, which are detachable. A master connecting-rod O is rigidly attached to a master ring N, and is attached to the piston of the top vertical cylinder in the front bank of cylinders and to the bottom vertical cylinder in the rear bank.

There are six auxiliary tubular connecting-rods P, which are free to hinge in the master ring, being attached to it by the hardened steel pins L.

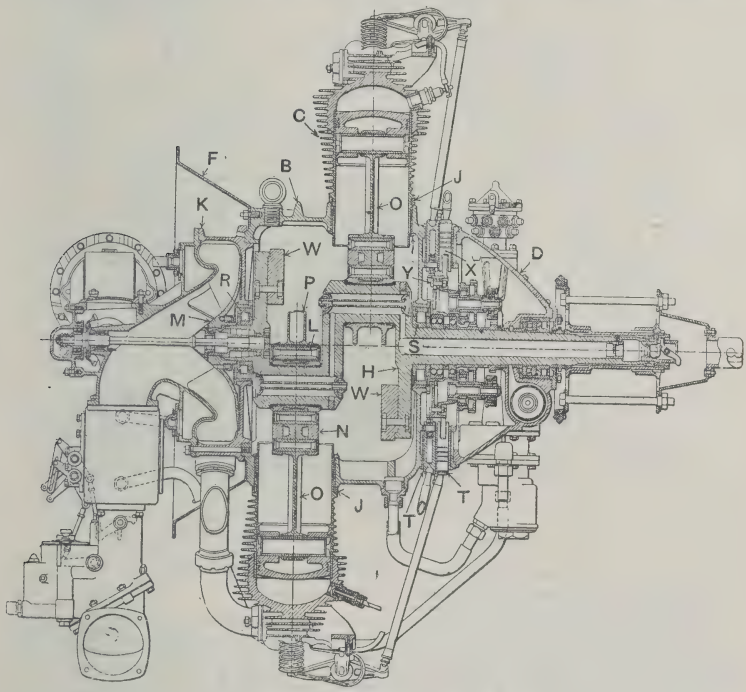


FIG. 9.—Armstrong Siddeley 'Jaguar' Engine.

Hollow floating gudgeon pins are used in the piston, the pistons being of forged 'Y' alloy with two cast-iron gas rings and two scraper rings. A hardened steel ring, Y, embodies three inlet and three exhaust cams, and is operated by planetary reduction gearing which drives it at one-sixth engine speed in an opposite direction to that of the crankshaft.

Lubrication is on the dry-sump system, two gear type pumps being fitted, one to feed and the other to scavenge the oil. Oil is supplied to the feed pump from a supply tank, and is then forced through a filter to the hollow crankshaft at a rate of about 2½ galls. per minute. As released from the engine the oil drains back to the sump, and is returned to the supply tank by the scavenge pump through oil-jackets surrounding the induction-fan casing and heater box. By these means the oil is cooled, and the incoming gas heated.

ARMSTRONG SIDDELEY 'GENET.'

General characteristics are: Five cylinder, air-cooled radial. Steel cylinders, screwed and shrunk on aluminium alloy heads with inclined valves operated by push rods. One piece crankshaft with counterbalance weights. Forged 'Y' alloy pistons. Oil pumps and magnetos mounted on front cover of crankcase, carburettor at back communicating with induction chamber containing a mixing fan. Dry sump lubrication and provision for dual ignition.

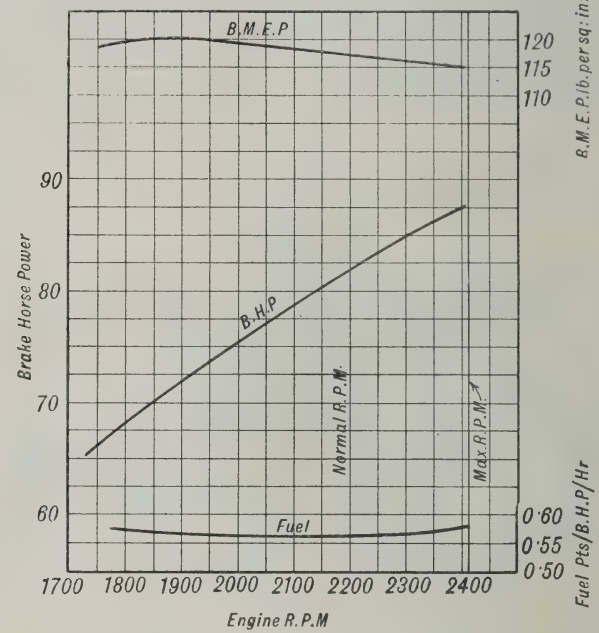


FIG. 10.—Power and Fuel Consumption. 'Genet' Engine.

WRIGHT 'WHIRLWIND' ENGINE. MODEL J–5 SERIES.

This engine (fig. 11) is of the air-cooled, static radial type.

The leading particulars are as follows:—

Number of cylinders, 9; bore, 4½ ins.; stroke, 5½ ins.; displacement, 788 cubic ins.; compression ratio, 5·2 : 1; guaranteed h.p. at sea level, 200 h.p. at 1,800 r.p.m. Average weight (dry), 508 lbs. Length overall (ex. starter), 34 ins. Diameter overall, 45 ins. Distance, mounting flange to end of propeller hub, $27\frac{17}{32}$ ins. Fuel consumption at rated power, 0·60 lb. per h.p. hour. Oil consumption at rated power, 0·025 lb. per h.p. hour.

The cylinder consists of a head H with integral fins cast in aluminium alloy screwed and shrunk on to a forged steel barrel B with integral fins and holding-down flange. Combustion chamber hemispherical with valves inclined at 35° to cylinder axis. Two sparking plugs per cylinder located in bronze bushings, diametrically opposite in the sides of the combustion chamber at an angle of 10° to crankshaft axis. Cylinder head contains two ports, the axes of which are at right angles to each other, one for attachment of inlet pipe, and the other for attachment of exhaust manifold. Cylinder attached to crankcase by eight studs.

Crankcase built up of five aluminium alloy castings. Front section S_1 includes thrust bearing housing, magneto brackets, magneto and cam-driving gears. Intermediate section S_2 contains cam and valve tappet mechanism and housing for front crankshaft bearing. Main section S_3, S_4 provided with pads for attachment of cylinders and contains the flange for mounting the engine, and the housing for rear crankshaft bearing and intake passages P. Rear section S_5

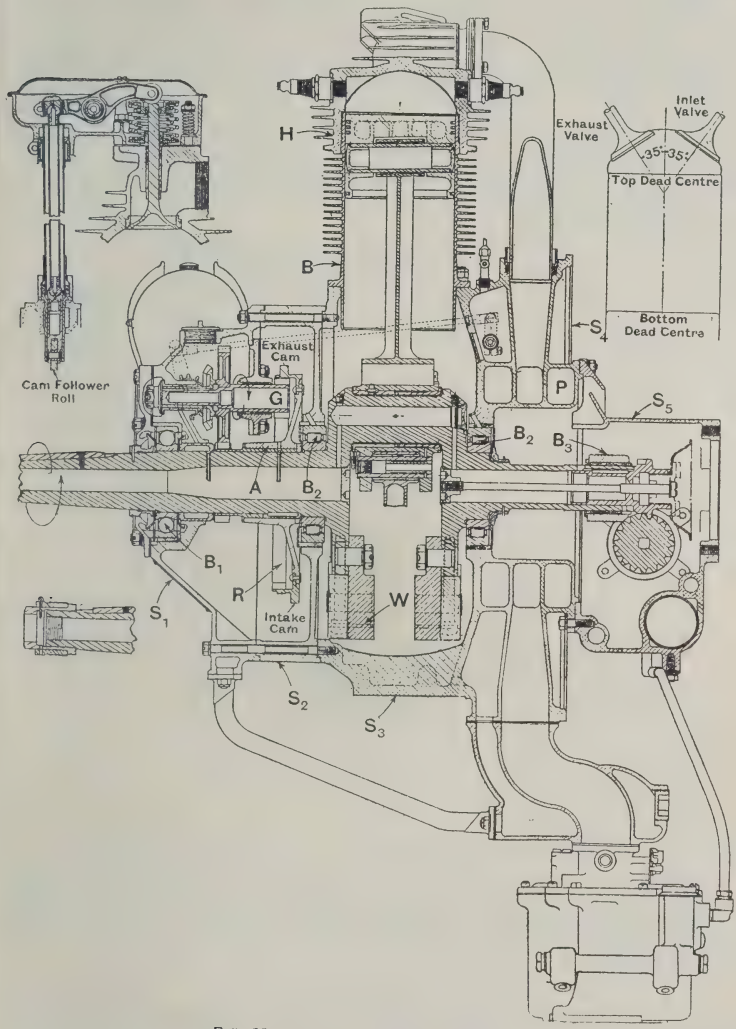


FIG. 11.—Wright 'Whirlwind' Engine.

Model J-5 Series.

contains drives for oil and fuel pumps, gun synchronisers and tachometers. It also provides a mounting for engine starter and housing for main oil strainer.

Crankshaft, single throw, counterbalanced type, chrome-nickel steel forging machined all over and hollow throughout its length. Counterweights W bolted to extensions of crank cheeks. Shaft runs in four bearings, ball thrust bearing B_1, two main ball bearings B_2, and a plain bearing B_3 in rear crankcase section.

Valve-operating mechanism actuated by cam located in intermediate crankcase section, and consists of a hardened steel ring R with eight lobes on the outside, and an internal gear G on the inside, which is riveted to an aluminium hub A riding on the sleeve portion of the cam drive gear, which is keyed to crankshaft. Cam followers of roller type. Push rods of nickel steel tubing with hardened steel balls pressed into either end. Valve gear enclosed by rocker arm housings, which form the support for rocker arm pins. Connecting rods consist of one master and eight link rods machined from chrome nickel steel forgings, master rod provided with steel-backed babbit-lined bearing at big end. Gudgeon and knuckle pin bearings of bronze.

Pistons cast aluminium alloy with three compression rings above gudgeon pin and one oil scraper ring at bottom of skirt.

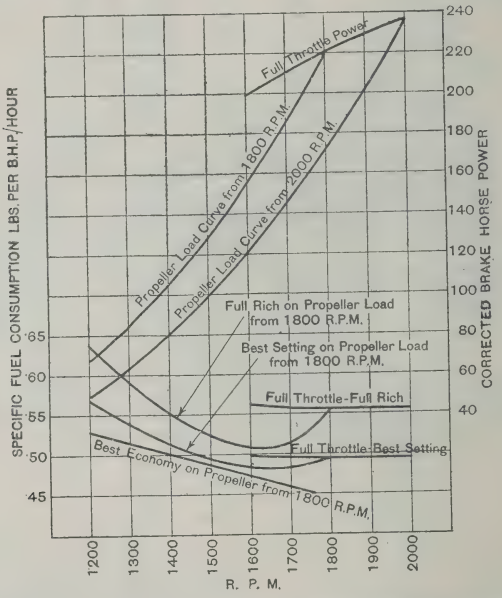


FIG. 12.—Power and Fuel Consumption for 'Whirlwind J-5' Engine.

Tulip-type valves of tungsten steel, exhaust valves being salt-cooled. Valve seats of aluminium bronze of rectangular cross section. Three concentric helical round steel wire springs per valve.

Ignition by two magnetos mounted on front section of crankcase. Carburettor attached to oil jacket, manifold sump attached to bottom of main crankcase section.

Pressure-type system of lubrication, except for cylinder walls and accessories drive gears, which is by spray.

WRIGHT 'CYCLONE' R—1,750 ENGINE.

General Specification.—Nine-cylinder radial. Rated output at sea level, 525 b.h.p. at 1,900 r.p.m. Weight (dry) without starter or hub 760 lbs. Weight per rated h.p., 1·52 lbs. Bore and stroke 6 in. × 6⅞ in., piston displacement 1,753 cu. ins. Outside diameter 54 ins., length over all, without starter 40⅝ in.

Guaranteed fuel consumption at rated h.p. 0·55 lbs. per h.p. hour. Oil consumption 0·035 lbs. per h.p. hour.

ISOTTA FRASCHINI (ITALIAN).

MODEL 'ASSO 1000.'

The engine is illustrated in fig. 13. Leading particulars are: 18 cylinders, water-cooled, three-banks 'W' formation at 40°; bore 150 mm.; stroke 180 mm.; compression ratio 5·3 : 1;

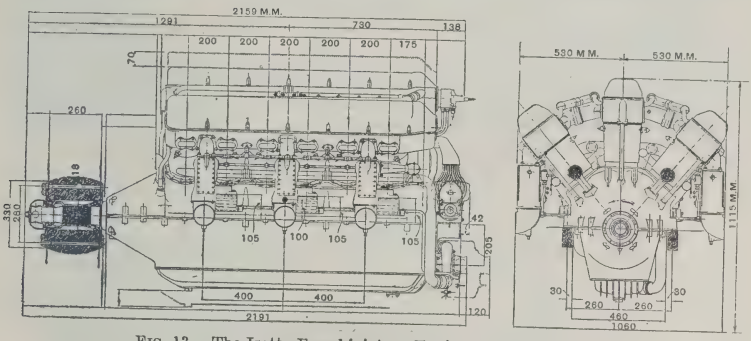


FIG. 13.—The Isotta Fraschini Aero Engine (type 'Asso 1000').

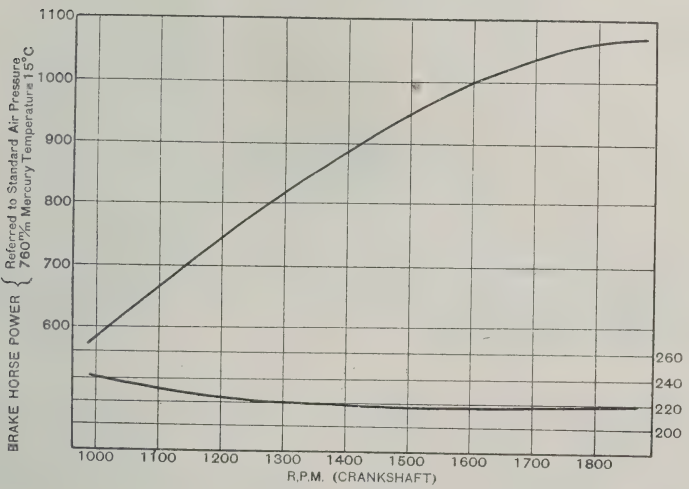


FIG. 14.—Isotta Fraschini Engine (type 'Asso 1000').

normal output 900 h.p. at 1,600 r.p.m.; maximum output 1,000 h.p. at 1,700 r.p.m.; weight (less airscrew) 1,770 lbs., weight per h.p., 1·77 lbs.; fuel consumption 0·485 lbs. per h.p. hour; oil consumption 0·032–0·04 lbs. per h.p. hour.

A.D.C. Aircraft Co.'s 'Nimbus' Engine.

This engine is one of the lightest of its type yet produced. Six-cylinder, vertical, water-cooled, direct drive.

Steel cylinder liners screwed into bottom of water-jacket blocks. Top of liners secured to cylinder head castings, which are in the form of aluminium alloy blocks, by the valve seatings. Water-jackets, aluminium castings, each block enclosing three cylinders. At the top the water-jacket blocks are secured to the cylinder head blocks by bolts. One inlet and two exhaust valves per cylinder, the former being operated from the camshaft by a rocker lever, and the latter direct.

Aluminium alloy pistons, with four rings and floating gudgeon pin.

Crankshaft, direct drive to airscrew, located in seven plain bearings and one roller bearing supporting forward end of shaft.

Camshaft located in a casing which protects the valve gear, and also forms a bridge over the cylinder heads, rigidly locking the blocks together. The shaft is driven by two trains of bevels.

Water circulation by centrifugal pump driven by an extension of the vertical shaft. Oil circulation to crankshaft, camshaft and vertical shaft, by dual oil pump that is combined with water pump.

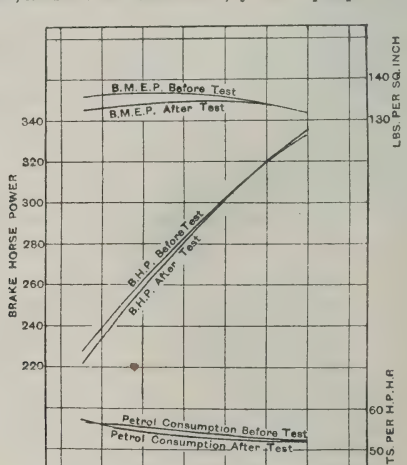


Fig. 15.—Power and Fuel Consumption Curves. 'Nimbus' Engine.

Bore, 152 mm.; stroke, 190 mm.; normal output, 305 b.h.p. at 1,450 r.p.m.; maximum output, 332 b.h.p. at 1,600 r.p.m.; petrol consumption at normal output, 0·52 pint per h.p. hour; oil consumption, 0·017 pint per h.p. hour; compression ratio, 5·4 : 1; b.m.e.p. 132 lbs. per sq. in.; weight dry (complete with airscrew boss, but less radiator and water), 670 lbs. ± 10 lbs. Dry weight per normal h.p. 2·19 lbs.

'Cirrus' Engines (Cirrus Aero Engines, Ltd.).

These engines are 4-cylinder in line, vertical type, cylinders and heads separate, the former being cast-iron and cylinder heads aluminium alloy, with air-cooling fins. Two sparking plugs per cylinder.

One inlet and one exhaust valve in each cylinder head, valves operated by rocking levers actuated by push rods to tappets in crankcase.

Pistons aluminium alloy with three castiron rings, connecting rods duralumin forgings of H section. Hollow gudgeon pin of full floating type. Crankshaft direct drive to airscrew. Located in three intermediate plain bearings and roller bearings at each end.

Airscrew shaft bolted to crankshaft and located in a radial thrust bearing.

Camshaft actuates direct to tappets, supported in three phosphor-bronze and one ball bearing.

Driven by steel gears housed in crankcase and timing gear cover.

Leading particulars are as follows:—

	Hermes.	Cirrus Mark III.	Mark II.
Bore	114 mm.	110 mm.	110 mm.
Stroke	140 mm.	130 mm.	130 mm.
Total swept volume	5,717 c.c.	4,939 c.c.	4,939 c.c.
Normal b.h.p.	105	90	75
Max. b.h.p.	115	94	80
Normal r.p.m.	1,900	1,900	1,800
Max. r.p.m.	2,100	2,100	2,000
Petrol consumption	0·58 pts./h.p. hour	0·6 pts./h.p. hour	0·6 pts./h.p. hour
Oil consumption	0·01 pts./h.p. hour	0·018 pts./h.p. hour	0·019 pts./h.p. hour
Weight dry	300	285 lbs.	280 lbs.
Compression ratio	5·1 : 1	5·4 : 1	4·9 : 1

'BRISTOL' ENGINES.

A series of engines of the static radial type, comprising the following:—

Jupiter.—Nine cylinders. Mark VI F. Bore, $5\frac{3}{4}$ ins.; stroke, $7\frac{1}{2}$ ins. New cylinder construction. Mark VII F, ditto, supercharged.

Series VI, direct drive. Mark H, M, or L (high, medium, or low compression).

Series VIII, IX, XI, geared engines, high, medium, low compression.

Both direct drive and geared type high compression engines known as 'Standard Service' type, compression ratio, 5·8 : 1. 'General Purpose' type, medium compression, 5·3 : 1. 'Commercial' type, low compression, 5·1 : 1. Supercharged engine compression ratio, 5·3 : 1.

In the new cylinder construction the cylinder is an open-ended barrel, machined from carbon steel forging, fins integral and concentric with bore, screwed at upper end to take cylinder head, which is a 'Y' alloy forging.

Four inclined valves per cylinder, and the combustion head is of the 'pent-house' form, each slope having one inlet and one exhaust valve seating and sparking-plug socket.

Valve seats bronze alloy, valves operated by push rods, and provision for neutralising effect of cylinder expansion incorporated. Each valve has three springs, the inner one being a volute of thin spring steel strip.

Crankcase machined from duralumin drop forging, the centre portion being split on the plane containing the cylinders.

Built up alloy steel crankshaft, airscrew shaft, forward crank cheek and crankpin being formed integral, rear crank cheek formed integral with a short shaft for driving auxiliaries is keyed and clamped to rear end of crankpin. Balance weights fitted to extension of crank cheeks.

Connecting rod assembly consists of master rod with integral big end and eight auxiliary rods secured by articulated joints.

Pistons, slipper type of cast aluminium alloy, gudgeon pins being supported by bosses formed integral with two transverse webs, the skirt being cut away at sides. Two piston and one scraper ring fitted.

Hollow gudgeon pins of full floating type.

Two four-lobed cam rings driven at one-eighth crankshaft speed by means of epicyclic gear. The cams actuate the push rods through roller type tappets. Tappet guides of duralumin.

Rear end of crankshaft drives gear-type oil pumps, and magnetos through bevel gearing at $1\frac{1}{8}$th crankshaft speed.

Lubrication on dry-sump system, the crankcase being drained by scavenger pump.

Carburettor of special design and air intake jacketed by warm oil from scavenger pump.

Supercharger, high-speed centrifugal type, placed at rear of engine between carburettor and induction pipes. The mixture is drawn from the carburettor and passes axially to the impeller,

whence it is discharged at high velocity through a ring of fixed diffuser vanes and then passes into the annular induction chamber, thence to the cylinders.

Rating.—Series VII F (supercharged).

Rated power, ground level, 500 h.p. at 1,775 r.p.m.; at 9,000 ft., 480 h.p. at 1,775 r.p.m. Maximum speed, 1,950 r.p.m.

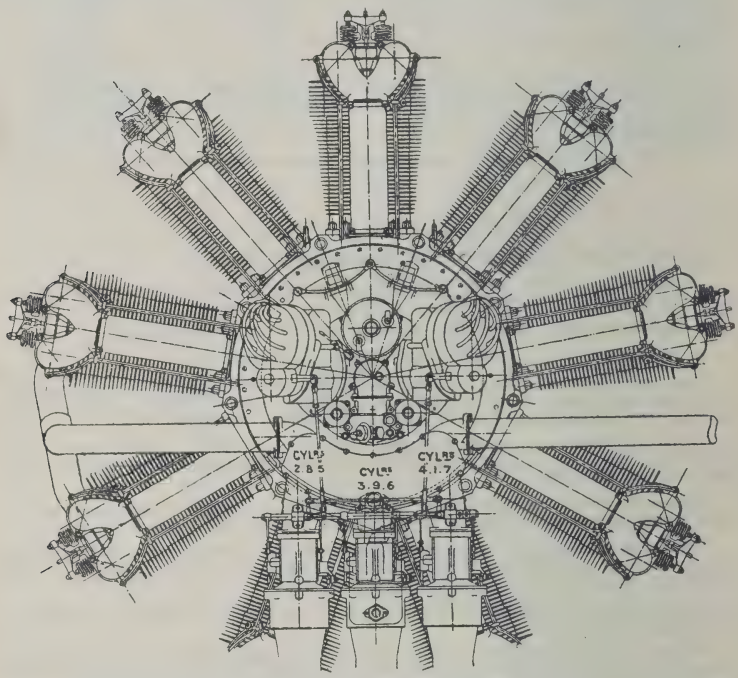


FIG. 16.—The 'Bristol' Jupiter Engine (Rear View).

Series VIII F (standard service engine), geared.

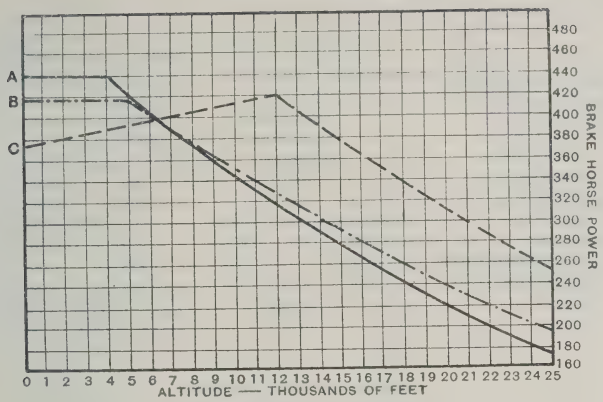
Rated power, ground level, 460 h.p. at 2,900 r.p.m. Output maintained up to 4,000 ft. Airscrew, half engine speed.

Series IX F (general purposes engine), 515 h.p. at 2,000 r.p.m.

Series IX F (commercial engine), 490 h.p. at 2,000 r.p.m.

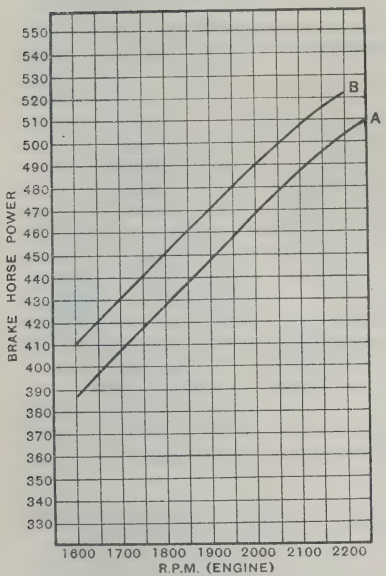
Neptune.—Series I. engine. Seven-cylinder radial, bore, 5·75 ins.; stroke, 6·5 ins. Normal rated output, 270 b.h.p. at 1,700 r.p.m. Maximum output, 315 b.h.p. at 1,870 r.p.m. Compression ratio, 5 : 1; general construction similar to Jupiter engine. Airscrew driven direct by forward extension of crankshaft.

Titan.—Series II. engine. Five-cylinder radial normal rated output, 205 b.h.p. at 1,700 r.p.m. Maximum output, 220 b.h.p. at 1,870 r.p.m.

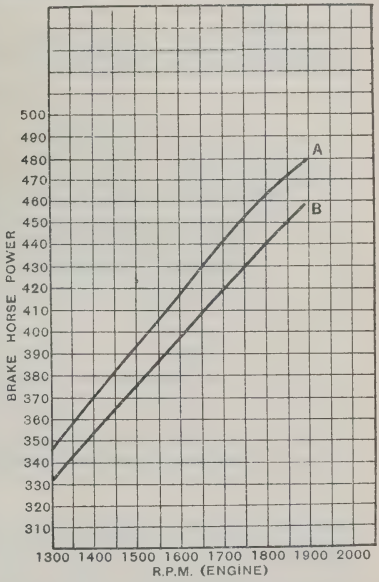


Curve A Jupiter VIII (5:8) b.h.p. at 2,000 Eng. r.p.m.
„ B „ VI A (4:3) „ 1,700 „
„ C „ VII „ 1,775 „

FIG. 17.—Bristol Jupiter Engine. Comparative Power at Altitude.
Max. power available at normal r.p.m. with standard 80/20 fuel.



Curve A Jupiter XI (5:1).
„ B „ IX (5:3).



Curve A Jupiter VI AM (5:3).
„ B „ VI AL (5:1).

FIG. 18.—Full Throttle Power Curves, Jupiter Engines.

Leading particulars of these engines are as follows :—

	VI. A.	VI. A.M.	VI. A.L.	VII.	VIII, IX, XI
Series					
Bore × stroke	5·75″ × 7·5″			5·75″ × 7·5″	5·75″ × 7·5
Total stroke volume, cu. ins.	1,753			1,753	1,753
Normal engine speed	1,700 r.p.m.			1,755	2,000
Maximum engine speed	1,870 r.p.m.			1,950	2,200
Propeller	direct drive LH tractor			direct drive LH tractor	LH tractor *
Oil pressure lbs./sq. in.	40			60	60
Standard weight, bare. lbs.	720			760	880
	VI. A.	VI. A.M.	VI. A.L.		VIII. IX. XI
Compression ratio	6·3	5·3	5·0	5·3	5·8 5·3 5·0
Rated b.h.p. at normal r.p.m.	415	440	420	420(a)	440 485 460
Rated b.h.p. at max. r.p.m.	455	485	460	440(b)	480 525 500
	at 5,000 ft.			(a) at 12,000 ft. (b) at 15,000 ft.	at 4,000 ft.

PACKARD ENGINES (AMERICAN).

Model 3A–1,500 and 3A–2,500.

These engines are of the 60° V, 12 cylinder, water-cooled type, rated at 600 h.p. at 2,500 r.p.m. for the 3A–1,500, and 800 b.h.p. for the 3A–2,500.

Cylinder construction of composite design, and cylinder and valve housing assembly mounted on engine as a unit.

Crankshaft follows usual practice, main journals and crankpins bored and plugged. Connecting rods of articulated type with babbitt bearings applied directly to big end. Piston pin and link pin bushings of phosphor bronze.

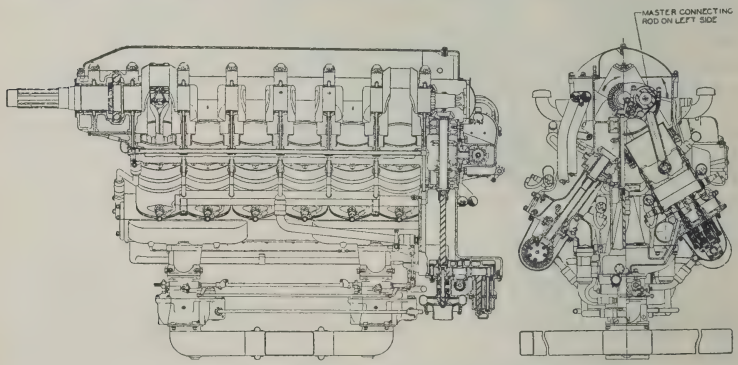


FIG. 19.— The Packard Aircraft Engine Model 3A–1,500 Inverted.

Piston of slipper type, permanent mould aluminium alloy castings. Three rings fitted above piston pin and lower ring groove fitted with oil drain relief. Piston pins of full floating type.

Cylinder of all-steel construction and design provides for water on contact with all heated

* Propeller gear ratio, 2 : 1.

surfaces, the cylinder head being of a double plate construction with water spaces between the two plates.

Leading particulars are as follows :—

Type :	3A—1,500.	3A—2,500.
Rating at 2,000 r.p.m.	—	800 b.h.p. 5·8 : 1 770 b.h.p. 5·1 : 1
Rating at 2,500 r.p.m.	600 b.h.p.	
Bore and stroke	$5\frac{3}{8}$″ × $5\frac{1}{2}$″	$6\frac{3}{8}$″ × $6\frac{1}{2}$″
Total piston displacement	1,530·4 cu. ins.	2,539·55 cu. ins.
Compression volume ratio	5·1 : 1 or 5·5 : 1 or 6 : 1	5·1 : 1 or 5·8 : 1
Firing order	1L—6R—5L—2R—3L—4R—6L—1R—2L—5R—4L—3R	
Overall length (direct and inverted)	$53\frac{7}{32}$″	$63\frac{7}{8}$″
Overall length (geared)	$39\frac{5}{32}$″	$46\frac{17}{32}$″
Journal diameter	3·00″	3·50″
Crankpin diameter	2·500″	3·250″
No. of bearings	7	7
No. of crank throws	6	6
Camshaft. Length	$38\frac{3}{8}$″	46″
Diameter	1	$1\frac{3}{32}$″
Valves * No. per cylinder	4	4
Seat angle	45°	45°
Lift inlet	$\frac{7}{16}$″	$\frac{1}{2}$″
Lift exhaust	$\frac{3}{8}$″	$\frac{1}{2}$″
Diameter inlet	$1\frac{15}{32}$″	$2\frac{7}{16}$″
Diameter exhaust	$1\frac{1}{2}$″	$2\frac{11}{16}$″
No. of springs per valve	7	10
Valve timing. Inlet opens	10° A.T.D.C.	10° A.T.D.C.
Inlet closes	45° A.B.D.C.	45° A.B.D.C.
Exhaust opens	48° B.B.D.C.	48° B.B.D.C.
Exhaust closes	8° A.T.D.C.	8° A.T.D.C.

Materials used.—	
Connecting rods	Chrome-vanadium steel.
Pistons	Aluminium alloy.
Cylinders	Chrome-molybdenum steel forgings.
Valve seats (exhaust)	High chrome steel.
Valves. Inlet	Tungsten steel.
Valves. Exhaust	Silchrome
Valve springs	Piano wire.
Valve housing	Aluminium alloy casting.
Bearing cups	Duralumin forgings.

Model 3A—1,500 inverted type, is shown in fig. 19.

Beardmore Engines.

Four engines produced known as ' Cyclone,' ' Typhoon,' ' Simoon,' and ' Tornado.'

The ' Cyclone ' and ' Typhoon ' are of the 6-cylinder vertical water-cooled type, the ' Typhoon ' being inverted. The ' Tornado ' is the first practical aero engine to use heavy oil fuel, and is of the compression-ignition type. In general design and construction it resembles the ' Cyclone.' It has, however, eight cylinders of $8\frac{1}{2}$-in. bore, and normal speed of 1,000 r.p.m. Normal b.h.p. 650, maximum b.h.p. 720. The weight, 3,000 pounds, is at present rather high for aeroplane work, but is particularly suitable for airship work when allowance is made for the very low fuel consumption and for the avoidance of fire risk.

* Exhaust valve stems are hollow and fitted with a $\frac{3}{16}$ in. steel tube. Three horizontal holes are drilled near lower end of tube and the valve stem proper is provided with two horizontal holes just below the necked down portion. Oil introduced under pressure at the upper end of the valve stem flows down inside the tube out through the holes at the base and up around the outside of the tube to the two outlet holes in valve stem. When the cam-shaft revolves, the exhaust cam depresses the cam follower and the oil which is trapped in the cam follower guide is forced through the exhaust valves thus cooling them and greatly prolonging their life.

Leading particulars of Beardmore engines are as follows :—

	Engine.		
	'Typhoon.'	'Cyclone.'	'Simoon.'
Normal b.h.p. and r.p.m.	925–1,350	925–1,350	1,100–1,250
No. of cylinders	6	6	8
Bore and stroke	$8\frac{5}{8}$″ × 12″	$8\frac{5}{8}$″ × 12″	$8\frac{3}{16}$″ × 12″
Compression ratio	5·25 : 1	5·25 : 1	5·25 : 1
Fuel consumption, lbs. per b.h.p./hour	0·48	0·48	0·48
Oil consumption, lbs. per b.h.p./hour	0·010	0·010	0·010
Weight complete (dry) lbs.	2,233	2,150	2,770
Overall length, ins.	80·3	80·3	98
Overall width, ins.	38·5	35	37·6

Carburation.

With radial and rotary engines it is usual to fit a simple type of carburettor, there being no float chamber, the petrol flow being hand adjusted to suit engine speed and atmospheric conditions.

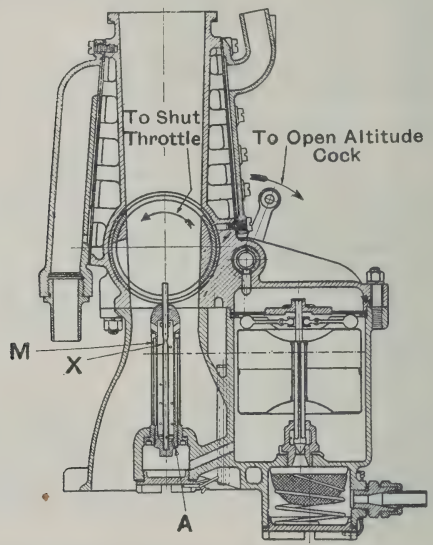


FIG. 20.—Napier Claudel-Hobson Water Jacketed Carburettor.

With stationary engines two or more carburettors are fitted according to the size of the engine. Representative types are the Zenith and Claudel-Hobson, which are fitted with float chambers and designed to give a practically constant mixture under the usual variations of load and speed.

These carburettors require to be carefully adjusted to each engine so as to give the same results, the operation of adjusting being known as 'synchronising.'

Aero engine carburettors must operate under variable circumstances. As the machine rises the density of the atmosphere decreases, and the quantity of fuel required per cu. ft. of free air becomes less. The desired reduction in petrol flow is controlled by hand adjustment, either by throttling the supply to the jet or, in the case of carburettors with float chambers, by reducing the pressure difference between the float chamber and the jet.

The Napier-Claudel Water Jacketed Carburettor, shown in fig. 20, is designed for use with the 'Lion' engine. It operates on the standard Claudel diffuser principle, the holes A and X being the main and pilot jets respectively. Water is delivered into the jackets on the induction pipes through pipes connected to the outlet flanges on the cylinder heads. From the induction jackets the water is led through short pipes to the water jackets of the carburettors, and thence through a return pipe to the water pump.

On tractor type engines the series parallel type main and slow running jets are used, whereas on pusher type engines the series type jets are used.

The jets shown in fig. 20 are the series type; the series parallel type main jets have an additional pilot jet hole drilled through their base central with the pilot jet, the pilot jet being specially calibrated to function correctly with this type of main jet.

The altitude compensation is of the 'Vacuum control' type. The areas of the various passages are specially proportioned for the 'Lion' engine. The functioning of the pressure balance and altitude control systems depends on the maintenance of an air pressure or depression above the petrol in the float chamber.

The arrangement of the induction piping is a most important factor in getting good carburation; sudden changes in the speed of the charge, and sharp bends, cause a considerable loss. Also there must be no dip in the pipes. The usual methods of heating are by water circulation, by exhaust gases, and by hot air taken into the carburettor. The purpose of the choke tube is to obtain the correct velocity of air around the jets in order to get the best mixture at all speeds. The main jet has most influence at high speeds, and the compensator, which corrects the irregularities of the main jet, has the greatest influence at low speeds.

For the Series XI engine a Napier-Claudel-Hobson Triplex Carburettor is fitted. The carburettor works on the standard Claudel diffuser principle.

Water Cooling.

Water cooling of aircraft engines has followed automobile practice in general principles, but there has been great diversity of opinion as to radiator design. The honeycomb type may now be considered as standard, with round tubes and hexagonally bulged ends. This gives a maximum cooling surface with minimum solid frontal area and therefore minimum head resistance.

From a series of tests on different radiators (Aeronautical Research Committee R. & M. No. 934) it was found that:

(*a*) Heat dissipation can be represented as a function of the cooling area ratio $\left(i.e.\ \frac{\text{cooling area}}{\text{frontal area}}\right)$ and drag as a function of cooling area ratio and diameter.

(*b*) A round bulge gave a better performance than the hexagonal, mainly on account of the smaller drag. For tubes of the same length : diameter ratio, large diameters are better than small, mainly on account of the drag.

(*c*) Shutters have comparatively little effect on heat dissipation until they are more than half shut.

(*d*) The shuttered results suggest the use of longer tubes in order to cut down the frontal area.

(*e*) Geometrical considerations suggest the development of hexagonal tube radiators, as they would have greater cooling area for the same tube length and a smaller capacity.

Opinions of designers vary greatly as to the best position of the radiator. The 'nose' position, in which the radiator is placed immediately behind the propeller, has many adherents, but excellent results are also obtained when the radiator is placed in two halves on the sides of the fuselage or below the fuselage. The radiator may also be placed in the top plane centre section.

Radiator Shutters.

In machines designed for flying at high altitudes, it is necessary to provide a variable cooling device to obviate the danger of the radiator freezing when high speeds are obtained as in diving. The device most generally used is a Venetian-blind type of shutter, which can be closed to prevent the air having access to the radiator. It is controlled by the pilot, who is also provided with a radiator thermometer, to indicate the circulating water temperature.

Average Weight of Power Plants.

	Lbs./h.p.		Lbs./h.p.
*Water-cooled engines	2·50	Air-cooled engine accessories	0·383
*Air-cooled engines	2·33	Water-cooled engine + accessories	3·50
Water-cooled engine accessories	1·00	Twin water-cooled engine + accessories	3·75
Twin water-cooled engines accessories	1·25	Air-cooled engines + accessories	2·50

Engine accessories include: radiator and shutters, cooling water, piping and cocks for fuel, oil, and water, exhaust pipes, pumps, starting gear, propeller. Fuel and oil tanks *not included*, as these vary with duration of flight.

* These figures can be considerably reduced with large power units.

Weight as a Percentage of Engine Weight.

	Per cent.
Water-cooled engine accessories	43
Twin water-cooled engine accessories	47
Air-cooled engine accessories	16

Weight of Honeycomb Radiators.

Weight of honeycomb radiators in lbs. = 0·4 × radiating surface in sq. ft. (approx.).

Weight of Tanks.

Weight of Fuel, Oil, or Water in lbs.	Approximate Weight of Tanks in lbs.		
	Steel. (Fuel.)	Self Sealing. (Fuel.)	Oil or Water.
200	30	50	27
400	50	75	40
600	65	95	50
800	75	110	—
1000	87	125	—

Weight of Power Plant as Percentage of Gross Weight of Complete Plane.

	Average value. Per cent.
Engine	18·5
Fuel and oil tanks	3·0
Radiator, shutters, and cowling	2·75
Cooling water	2·75
Airscrew	2·50
Piping, pumps, cocks, etc.	0·50
Engine starting gear	0·50
Total	30·50

DIMENSIONS, LOADS, RUBBING FACTORS, ETC., OF REPRESENTATIVE AERO ENGINES.

Type.	Engine.	Bore and Stroke.	Normal R.P.M.	Approximate H.P. delivered per Crank Pin.	No. Cylinders per Crank Pin.	Centrifugal Weight per Crank Pin.	Reciprocating Weight per Cylinder.	Crank Pin Diameter.	Effective Bearing Length.	Projected Bearing Area.	Bearing Pressure. Lbs. per Sq. In.		Rubbing Velocity.	Rubbing Factor.	Usual Oil Pressure.
											Max.	Mean.			
		Ins.				Lbs.	Lbs.	Ins.	Ins.	Sq.Ins.			Ft. Sec.		Lbs. Sq. in.
6-Cylinder Vertical	Maybach	6·5 × 7·1	1,400	47	1	5·14	17·35	2·6	2·6	6·76	1,730	520	15·85	8,250	6
	Rausie—E6	5 × 6	1,650	29	1	3·73	5·95	$2\frac{1}{4}$	$2\frac{1}{4}$	4·52	1,640	454	15·3	6,950	45
	Liberty—6	5 × 7	1,800	40	1	3·7	5·9	$2\frac{1}{16}$	$2\frac{1}{16}$	5·2	1,060	468	18·66	8,750	40
8-Cylinder Vee	Hispano—300	5·51 × 5·9	1,800	80	2–90° V.	6·2	7·2	$2\frac{1}{4}$	$2\frac{3}{8}$	5·04	1,586	856	16·7	14,360	70
	Wright—H2	5·51 × 5·9	2,000	100	2–90° V.	6·2	6·06	$2\frac{1}{8}$	$2\frac{7}{16}$	5·19	1,835	944	18·6	17,520	70
12-Cylinder Vee	Rolls Royce—Eagle, Series IX	$4\frac{1}{2}$ × $6\frac{1}{2}$	1,800	59	2–60° V.	3·95	3·75	$2\frac{1}{4}$	$1\frac{7}{8}$	4·23	1,180	624	17·7	11,050	40
	Packard—2025	$5\frac{3}{4}$ × $6\frac{1}{2}$	1,800	90	2–60° V.	8·02	8·06	$2\frac{1}{4}$	$2\frac{7}{8}$	6·08	1,220	840	19·7	16,500	55
	Packard—1237	5 × $5\frac{1}{4}$	1,800	50	2–60° V.	5·01	5·83	$2\frac{1}{4}$	$2\frac{1}{4}$	5·06	1,080	580	17·5	10,150	40
	Liberty—12	5 × 7	1,700	70	2–45° V.	6·3	6·2	$2\frac{1}{8}$	$2\frac{1}{16}$	5·2	1,035	750	17·6	13,200	40
	Curtiss—K12	$4\frac{1}{2}$ × 6	2,250	70	2–60° V.	4·4	3·25	$2\frac{1}{2}$	$1\frac{7}{8}$	4·73	1,036	759	24·6	18,675	85
12-Cylinder W.	Napier Lion	$5\frac{1}{2}$ × $5\frac{1}{8}$	1,925	115	3–60° W.	6·4	5·5	$2\frac{3}{4}$	$2\frac{3}{4}$	7·55	1,100	629	23·1	14,500	50
18-Cylinder W.	U.S.A.—W—1	$5\frac{1}{2}$ × $6\frac{1}{2}$	1,800	133	3–40° W.	11·2	7·4	3	$2\frac{7}{16}$	7·31	1,250	872	23·5	20,500	50
	U.S.A.—W—2	$6\frac{1}{2}$ × $7\frac{1}{2}$	1,400	167	3–40° W.	16·3	11·4	$3\frac{3}{8}$	$2\frac{1}{2}$	8·43	1,145	819	20·6	16,900	40
9-Cylinder Radial	Lawrance	$4\frac{1}{2}$ × $5\frac{1}{2}$	1,800	220	9 Radial	17·2	4·16	2	3·45	6·9	1,188	971	15·7	15,250	30

DETAILS OF AERO ENGINES.

Engine.	A	B	C	D	E	F	G
BRITISH							
A.D.C. Cirrus Hermes	Vt	A	4	$4\frac{1}{2} \times 5\frac{1}{2}$	345	5·1	112
A.D.C. Nimbus	Vt	W	6	5·98 × 7·48	1,261	5·4	132
Armstrong S. Genet II	R	A	5	4 × 4	251	5·25	118
Armstrong S. Mongoose II	R	A	5	$5 \times 5\frac{1}{2}$	540	5·0	123
Armstrong S. Lynx IV.	R	A	7	$5\frac{1}{2} \times 5\frac{1}{2}$	756	5·0	120
Armstrong S. Jaguar IV	R	A	14	$5 \times 5\frac{1}{4}$	1,512	5·0	123
Bristol Lucifer IVA.	R	A	3	$5\frac{1}{2} \times 6\frac{1}{2}$	487	5·3	122
Bristol Jupiter VIA	R	A	9	$5\frac{3}{4} \times 7\frac{1}{2}$	1,753	5·3	118
Bristol Jupiter VIII.	R	A	9	$5\frac{3}{4} \times 7\frac{1}{2}$	1,753	5·3	117
Napier Lion XI	W	W	12	$5\frac{1}{2} \times 5\frac{1}{8}$	1,461	6·0	121
Rolls-Royce Condor IIIA	V.60	W	12	$5\frac{1}{2} \times 7\frac{1}{2}$	2,138	5·3	126
Rolls-Royce Eagle IX.	V.60	W	12	$4\frac{1}{2} \times 6\frac{1}{2}$	1,240	5·22	127
Rolls-Royce F.10	V.60	W	12	$5 \times 5\frac{1}{2}$	1,296	6·0	130
AMERICAN							
Allison V–1,410	I.V.45	A	12	$4\frac{5}{8} \times 7$	1,410	5·3	128
Curtis D-12–D.	V.60	W	12	$4\frac{1}{2} \times 6$	1,145	5·3	135
Curtis C–6–A.	Vt	W	6	$4\frac{1}{2} \times 6$	573	5·2	133
Curtis OX–5	Y.90	W	8	4 × 5	503	4·9	112
Curtis H–1,640	R	A	12	$5\frac{5}{8} \times 5\frac{1}{2}$	1,640	5·4	130
Detroit. Air Cat	R	A	5	$4\frac{1}{2} \times 3\frac{3}{4}$	250	5·4	105
Packard 3A–1,500	V.60	W	12	$5\frac{3}{8} \times 5\frac{1}{2}$	1,530	5·5	140
Packard 3A–2,500 G	V.60	W	12	$6\frac{3}{8} \times 6\frac{1}{2}$	2,540	5·1	135
Packard IA–2,775.	X	W	24	$5\frac{3}{8} \times 5$	2,775	7·0	150
Wright J–5	R	A	9	$4\frac{1}{2} \times 5\frac{1}{2}$	788	5·2	125
Wright R–1,750	R	A	9	$6 \times 6\frac{7}{8}$	1,753	5·0	130
CONTINENTAL							
Farman 18 WD.	W.40	W	18	5·12 × 7·1	2,631	5·5	—
Farman 12 WE.	W.60	W	12	5·12 × 6·3	1,556	5·3	—
Hispano S. 8 F.	V.90	W	8	5·51 × 5·91	1,130	5·3	—
Hispano S.12 Hb.	V.60	W	12	5·51 × 5·91	1,691	6·0	—
Lorraine D. 14	R	A	14	5·32 × 5·91	1,839	5·0	—
Lorraine D.12 Ee.	W	W	12	4·72 × 7·09	1,489	6·5	—
Lorraine D. 18 Kd.	W	W	18	4·72 × 7·09	2,233	6·0	—
Renault 12 Md.	V.60	W	12	6·3 × 7·09	2,652	5·3	124
Renault 12 J.b.	V.60	W	12	4·92 × 6·7	1,528	5·6	129
Salmson 7 A.C.	R	A	7	3·94 × 5·12	437	5·0	—
Salmson 18.A.B.	R	A	18	4·92 × 6·70	2,293	5·4	—
B.M.W. VI.	V.60	W	12	6·3 × 7·49	2,800	6·3	—
Junkers L.5	Vt	W	6	6·32 × 7·49	1,410	5·5	111
Junkers L.55	V.60	W	12	6·32 × 7·49	2,820	5·5	111
Siemens & H. Sh.11	R	A	7	3·94 × 4·73	403	5·6	107
Siemens & H. Jupiter	R	A	9	5·75 × 7·5	1,748	5·3	107
Fiat A.20	V.60	W	12	4·58 × 5·91	1,159	5·7	143
Fiat A.25	V.60	W	12	6·70 × 7·88	3,374	5·25	134
Isotta. F. 200	Vt	W	6	5·51 × 6·3	901	5·65	132
Isotta. F. 500	Vt	W	12	5·52 × 5·92	1,690	5·5	128

A. Type: V = Vee; V_t = Vertical; R = radial.
B. Cooled by: W = water, A = air.
C. Number of cylinders.
D. Bore and stroke, inches.
E. Piston Displacement, cubic inches.
F. Compression ratio.
G. B.M.E.P. lbs. per sq. in.
H. Rated b.h.p. and r.p.m.
I. Max. b.h.p. at r.p.m.
J. Normal r.p.m. propeller.

DETAILS OF AERO ENGINES—*Cont.*

H	I	J	K	L	M	N	O	P	Q
105–1,900	115–2,100	1,900	·58	·010	300	2·85	38½	36½	19
305–1,450	332–1,600	1,450	·52	·017	670	2·25	69½	43¾	20½
80–2,200	88–2,380	2,200	·46	·02	210	2·38	28½	34	—
150–1,850	165–2,050	1,850	·46	·02	365	2·21	36	45½	—
215–1,900	230–2,090	1,900	·46	·02	505	2·19	40	45½	—
400–1,700	440–1,870	1,700	·46	·02	805	1·84	47	45½	—
120–1,700	140–1,870	1,700	·5	·03	330	2·3	19	48	—
440–1,700	485–1,870	1,700	·5	·03	730	1·5	18½	53	—
445–2,000	480–2,200	1,000	·51	·03	850	1·5	28	53	—
530–2,350	567–2,585	1,250	·53	·025	995	1·8	68	39	42
665–1,900	—	907	·50	·027	1,182	1·86	69⅛	43½	30½
360–1,800	—	1,080	·51	·026	965	2·63	73	46	32
490–2,250	—	1,421	·48	·012	865	1·58	60⅜	34½	24¼
430–1,900	450–2,000	1,900	·55	·03	1,000	2·33	78½	46¾	34½
435–2,300	460–2,500	2,300	·53	·015	680	1·56	56¾	34¾	28½
160–1,750	175–1,900	1,750	·50	·01	420	2·55	57¾	40⅜	23½
90–1,400	115–1,700	1,400	·50	·02	390	3·90	55⅜	35⅛	30
600–2,200	—	2,200	·55	·02	890	1·48	31¼	45	—
60–1,800	80–2,400	1,800	·52	·03	200	3·3	—	32	27¾
600–2,500	650–2,700	2,500	·50	·035	800	1·33	65	38¾	26⅞
800–2,000	835–2,100	1,000	·50	·025	1,435	1·80	69¾	42⅛	29½
1,200–2,600	1,300–2,700	2,500	·52	·03	1,500	1·15	78	45	28½
200–1,800	220–1,800	1,800	·53	·025	510	2·27	34	45	—
525–1,900	540–1,900	1,900	·55	·035	760	1·40	19⅜	54	—
700–1,850	820–1,920	825	·56	·022	1,600	2·28	96⅛	50	42¼
550–2,250	580–2,300	1,125	·50	·017	1,234	2·20	65¾	45¾	51
300–1,800	345–2,100	1,800	·50	·02	594	2·0	59	39	35
588–2,000	615–2,100	2,000	·48	·02	913	2·1	78	35½	29
420–1,800	535–1,800	1,800	·53	·026	960	2·10	52½	48¾	—
480–2,000	505–2,000	2,000	·51	·01	852	1·78	54⅛	44½	47¾
650–2,000	740–2,000	1,290	·51	·017	1,365	2·10	84	42¼	36¼
750–1,900	810–1,900	950	·50	·022	1,450	1·79	79⅛	45	38½
500–2,020	510–2,050	1,080	·49	·022	902	1·76	66	42	33⅛
95–1,800	—	—	·54	·04	286	—	32¼	37	—
460–1,700	—	—	·54	·04	1,012	—	54	47½	—
500–1,410	640–1,530	1,410	·48	·022	1,110	2·20	78¾	56¾	65½
280–1,275	310–1,450	1,275	·50	·02	695	2·4	69	49¾	22⅛
560–1,275	620–1,450	1,275	·47	·02	1,190	2·10	69	33⅜	45¾
89–1,500	96–1,750	1,500	·53	·026	326	3·38	32⅛	40½	—
420–1,720	490–1,870	1,730	·53	·044	812	1·65	30½	55¾	—
410–2,060	460–2,300	2,060	·52	·033	748	1·83	57½	33⅜	25½
950–1,700	1,000–1,700	1,700	·52	·052	1,850	1·95	92⅛	46¾	36⅛
240–1,600	290–2,000	1,600	·48	·026	616	2·12	76½	40	22
500–1,850	543–2,000	1,850	·48	·026	935	1·72	79½	37¼	32

K. Fuel consumption lbs./b.h.p./hour.
L. Oil consumption lbs./b.h.p./hour.
M. Weight dry, lbs.
N. Weight per h.p., lbs.
O. Length overall.
P. Height
Q. Width
} Outside diameter of cylinders for radial engines.

SECTION XXXIV

FUELS: SOLID, LIQUID, AND GASEOUS

By Harold Moore, M.Sc.(Tech.), M.I.Pet.Tech., F.C.S., A.I.M.E.,
(Author of 'Liquid Fuels for Internal Combustion Engines'),
R. B. Hobson, A.M.C.T., A.M.I.Pet.Tech. and W. Newton Booth.

(With Contributions by Arthur Grounds, B.Sc., A.I.C.,
L. C. Harvey, Alwyne Meade, Assoc.M.Inst.C.E., E. W. L. Nicol,
M.I.Mar.E., A.M.I.E.E., A. L. Summers, Chas. F. Wade, A.M.I.Mech.E.,
A.M.I.E.E. and other Specialists.)

SECTION XXXIV

FUELS: SOLID, LIQUID, AND GASEOUS.

(Contributed by Harold Moore, M.Sc.(Tech.), M.I.Pet.Tech., F.C.S., R. B. Hobson, A.M.C.T., and W. Newton Booth.)

Commercial fuels owe their heating value to two main combustible elements, carbon and hydrogen. Carbon is capable of forming two oxides, the monoxide CO formed when the oxygen supply is deficient and itself a gaseous fuel and the Dioxide CO_2; Sulphur, which is also contained in many fuels is itself combustible, but its effect on the total fuel value is relatively very small and it has the disadvantage that its oxide SO_2 has strongly acid properties and is thus liable to set up corrosion in metallic parts of the fuel consuming plant.

Provided that a fuel is completely burned, the carbon to CO_2, the hydrogen to steam, and the sulphur to SO_2, the amount of heat generated per unit weight of the fuel is perfectly definite and independent of the speed or method of combustion and is known as the calorific value, usually expressed in calories per gram or B.Th.U. per 1 lb. of the fuel.

(1 cal. per grm. = 1·8 B.Th.U. per 1 lb.).

If the steam produced from the combustion of the hydrogen is condensed, a further definite amount of heat is given out and there are thus two calorific values for any fuel containing hydrogen: the gross or higher cal. val. (H.C.V.) when the produced water is condensed and cooled and the nett or lower cal. val. (L.C.V.) when the produced water is not condensed.

The difference between the two values is

$$\text{H.C.V.} - \text{L.C.V.} = \text{H} \times 95{\cdot}13 \text{ B.Th.U. per 1 lb.}$$
$$= \text{H} \times 52{\cdot}8 \text{ Cal. per gm.}$$

where H is the percentage of hydrogen in the fuel.

In most commerical fuels there is generally some oxygen, and the carbon and hydrogen are combined in a more or less complicated manner. Such combinations may have resulted in the generation or absorbtion of heat and thus the actual calorific value of a fuel is generally different from the sum of the calorific values of its constituent elements.

Certain empirical formulæ have been evolved applicable to particular classes of fuel which give fairly satisfactory results *for those classes*. It is, however, safest to secure an experimental determination preferably by means of a bomb type of calorimeter of the C.V. of actual fuel to be used.

C.V. of Bituminous Coal is given roughly by the following—

Net calorific value = $[8{,}137\ C + 29{,}100\ (H - \frac{1}{8}O) + 2{,}500\ S - 600\ H_2O] \div 100$, where C, H, and O are the percentages of carbon, hydrogen, and oxygen respectively present, and H_2O the percentage of hygroscopic water present.

When the ultimate analysis is not known and it is desired to obtain a rough estimate of the calorific power from the proximate analysis, this can be obtained by the formula:

Calorific value = 1·8 (82C + aV) B.Th.U.

in which C = per cent. fixed carbon, V = per cent. volatile matter, and a is a constant computed from the following : $a_1 = 100V/(C + V)$.

a_1	5	10	15	20	25	30	35	38	40
a	145	130	117	109	103	98	94	85	80

COMBUSTION.

When a fuel is raised to its ignition temperature in contact with oxygen, combustion commences. It consists essentially of the chemical combination of the combustible elements with the oxygen and is accompanied with the generation of considerable quantities of heat, raising the whole mass to incandescence. The chemical processes may be very complicated, but the final results, if combustion is complete, are represented by the following equations :

(a) If oxygen supply is insufficient

$$2C + O_2 = 2CO.$$

(b) If oxygen supply is adequate

$$2CO + O_2 = 2CO_2$$
$$C + O_2 = CO_2$$
$$2H_2 + O_2 = 2H_2O.$$

The amounts of heat generated and the volumes of the products of combustion are given in Table I., p. 2079.

The following are recent figures for the spontaneous ignition temperatures of fuels :—

SPONTANEOUS IGNITION TEMPERATURES (IN OXYGEN).

Liquids.	Sp. Gr.	S.I.T.	
Pratt's perfection spirit No. I.	0·710	272 C.*	522 F.*
,, ,, No. II.	0·724	270	518
Paraffin oil	0·782	253	487
Gas oil (petroleum)	—	254	489
Various crude petroleums and residua	—	254 to 275	489 to 527
Shale oil	—	253	487
Commercial 100 per cent. benzol	0·875	566	1,051
Creosote oil	1·010	415	779
Coke-oven tar oil	1·046	478	892

(*Moore*, '*J.S.C.I.*,' Feb. 15, 1917.)

Gaseous Fuels (in Air).

Hydrogen	758 C.°	1,396 F.°
Carbon monoxide	910	1,670
Ethylene	895 to 905	1,643 to 1,661
Coal gas	850	1,562
Petrol gas	900	1,652
Methane (No ignition below 1,000)	1,000	1,832

(*McDavid*, '*J.C.S.*,' *Nov.* 1917, p. 1003.)

When a fuel contains oxygen this is, of course, available, and a correspondingly decreased quantity of external oxygen is required. The amount of external oxygen necessary for the combustion of 1 lb. of a fuel whose ultimate composition is known would thus be

$$\text{Oxygen required} = 8H + \frac{32}{12}C - O \quad \text{or} \quad 2{\cdot}66\left\{C + 3\left(H - \frac{O}{8}\right)\right\}$$

where HC and O are respectively the proportions of hydrogen, carbon and oxygen present. A dry atmosphere air contains 21 per cent. by volume of oxygen (23·3 per cent. by weight), the weight of air required is :

$$11{\cdot}6\left\{C + 3\left(H - \frac{O}{8}\right)\right\}.$$

In accurate calculations it is necessary to make allowance for the moisture contained in the fuel and in the air used for combustion. The latter depends on its temperature and humidity. (See Table on p. 2079.)

TABLE I.

Reaction.	Oxygen Required.		Air Required.		Cal. Value. B.Th.U.				Products of Comb.			
	Per 1 Lb. Fuel.		Per 1 Lb. Fuel.		Gross.		Nett.		Vol.	Wt.	Added N_2 if in Air.	
	Vol. Cft.	Wt. Lbs.	Vol. Cft.	Wt. Lbs.	Per 1 Lb.	Per Cft.	Per 1 Lb.	Per Cft.	Cft. per 1 Lb. Fuel.	Per 1 Lb. Fuel.	Cft.	Lbs.
$2C + O_2 = 2CO$ . . .	14·90	1·33	71·0	5·725	4,450	—	4,450	—	CO. 29·80	CO. 2·33	56·1	4·392
$C + O_2 = CO_2$ . . .	29·8	2·66	142·0	11·45	14,500	—	14,500	—	CO_2 14·90	CO_2 3 66	112·2	8·783
$2H_2 + O_2 = 2H_2O$ . .	89·1	8·0	424·0	34·18	62,100	325·2	52,290	271·9	178·2	9·0	334·9	26·18
$2CO + O_2 = 2CO_2$ . . .	14·90	1·33	71·0	5·725	10,050	323·5*	10,050	323·5*	14·90	3·66	112·2†	8·783†

* Per cub. ft. CO. Other figures per 1 lb. carbon.
† Assume preliminary oxidation in Air.

The formula assumes that all the carbon is burned to CO_2, no CO being formed, and that combustible constituents other than those named are negligible.

Rich coals require more air than those low in calorific value, and, in fact, for coals such as those used in steam raising, over a wide range of qualities, the 'theoretical' air required may be taken with fair accuracy as proportional to the calorific value, and calculated by the formula

$$\text{Lbs. air per lb. coal} = \frac{\text{higher calorific value} \times 7{\cdot}2}{10{,}000},$$

the calorific value being in B.Th.U. per lb.

Perfect regulation is never attained in practice, and in order to secure that at least sufficient air is present in every part of the combustion chamber it is necessary to provide an average excess. This mixes with and dilutes the products of combustion, lowering their average temperature and carrying away heat when it finally leaves, and it is thus advisable to reduce its amount to the lowest which will avoid waste of fuel and smoke. This necessary excess will vary from 100 per cent. in the case of small direct hand-fired furnaces, 50 per cent. in the case of good mechanically stoked boiler installations, 25 per cent. in anthracite or coke burning furnaces, 30 per cent. in good powdered fuel firing, to practically nothing in the case of carefully regulated gaseous fuel fired furnaces.

If too little air is used, unburned fuel will escape with the chimney gases and cause serious reduction in efficiency.

Quantity of Flue Gases.

In order to specify suitable fans for induced and forced draught, it is necessary to know the weight of flue gases and the weight of air actually used per lb. of fuel burned. From this the volume to be dealt with by the fans at the appropriate temperatures can be ascertained.

The air used influences the percentage of CO_2 in the flue gases. For typical coals, the relation between air used and CO_2 may be expressed by the following approximate formula :—

$$\text{Lbs. air per lb. coal} = \frac{132 \times \text{calorific value of coal}}{CO_2 \text{ percentage} \times 10{,}000}.$$

The weight of flue gases per lb. coal may be taken as (lbs. air + 1).

Excess Air.

The percentage of excess air is approximately

$$\left(\frac{18{\cdot}9}{CO_2 \text{ per cent}} - 1\right) \times 100.$$

Actually the 'constant,' 18·9, should vary for coals in which the proportion of hydrogen to carbon is different. The formula gives therefore merely an approximate figure for a typical coal, also it does not apply if there is any CO in the flue gases.

Combustion of Hydrocarbon Fuels.

In the case of fuels containing hydrocarbon compounds many very complicated physical and chemical changes may take place before complete combustion is effected. Upon heating such hydrocarbons, *e.g.* those in bituminous coal, many commence to melt, or to distil or decompose at temperatures well below their ignition point. In such cases there is a primary evolution of combustible vapours and gases, and these when subjected to the radiant heat in a furnace undergo further decomposition. In an ordinary furnace it is generally not possible to give such a regulated air supply and to mix it sufficiently intimately with the fuel as to give rapid and complete combustion. The combustion is thus a relatively slow and heterogeneous process taking place in long flames, in which fuel vapours and gases and air are flowing in more or less parallel streams which only mix gradually. It is thus necessary to provide combustion chambers sufficiently large to contain the whole of the flames produced.

In addition to these physical effects it must be remembered that the combustion of a hydrocarbon is a stage by stage process somewhat similar in nature to the low temperature oxidation by chemical means, *e.g.* ethane would be oxidised successively to ethyl alcohol, acetaldehyde and acetic acid, which latter would then suffer thermal decomposition to hydrogen and carbonic oxide before being finally oxidised to steam and carbon dioxide. Heat is given out at each step, resulting in a rise in temperature, and a stage is soon reached when the

hydrocarbon molecules suffer thermal decomposition to simpler bodies and free carbon. If there is shortage of oxygen in intimate contact or if the flame is cooled below the ignition point, combustion is stopped and smoke produced. It will be noted also that the final products of the incomplete combustion contain considerable proportions of free hydrogen and carbonic oxide. It will be seen therefore how important it is to secure rapid and intimate mixture of the flame gases with sufficient oxygen if we are to secure complete and smokeless combustion and avoid the loss of fuel gases.

Internal Combustion Engines' Exhaust Gases.

The exhaust gases from internal combustion engines are subject to the same considerations as flue gases. They, however, cannot be regulated so as to produce a definite CO_2 percentage, as to obtain this result it would be necessary to throttle down the incoming air, thus loading the engine by increasing the difference in pressure between exhaust and inlet, in which case a

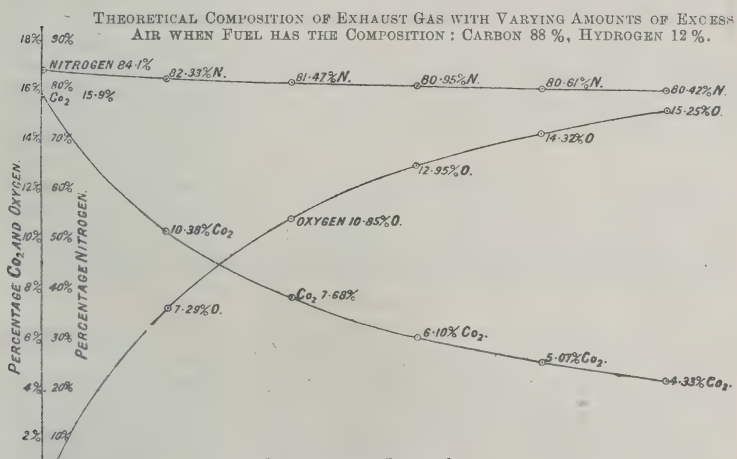


Fig. 1.

loss of efficiency would occur. The graphs in fig. 1 show the composition of the exhaust gases of an engine burning heavy petroleum (about 88 per cent. C and 12 per cent. H) with various quantities of excess air. Diesel oil engines running at full load generally take about 100 per cent. excess air; semi-Diesel engines a somewhat greater percentage excess air.

Heat Losses in Flue Gases.

The heat generated by the combustion of the fuel is communicated to the products of combustion which are thus raised to incandescence. The temperature of the flames thus depends upon the quantity and specific heat of these products and is thus affected greatly by the quantity of excess air admitted to secure complete combustion. As this temperature determines the 'temperature head' available for transmission of the heat, it will be seen what an important effect correct regulation of the air for combustion has upon the capacity of any given furnace plant, as well as upon its thermal efficiency.

When the combustion gases leave the furnace (at some temperature higher than that of the entering air, they carry away a certain proportion of the heat of the fuel and this is wasted. Its amount may be calculated if we know the leaving temperature of the waste gases and the composition of both fuel and gases.

The result with a typical bituminous coal is shown below :—

Temp. of Exit Gases.		Percentage of CO_2 in Exit Gases.											
		4%	5%	6%	7%	8%	9%	10%	11%	12%	13%	14%	15%
		Corresponding Percentage of Excess Air.											
		372%	278%	215%	170%	136%	110%	89%	72%	57%	45%	35%	26%
F.°	C.°	Total Percentage of Fuel wasted.											
400	204	32·4	26·2	21·8	18·9	16·4	14·9	13·4	12·3	11·4	10·6	9·8	9·2
500	260	40·5	32·7	27·3	23·6	20·8	18·6	16·8	15·4	14·2	13·2	12·3	11·6
600	316	48·6	39·3	32·8	28·3	24·9	22·3	20·2	18·5	17·1	15·9	14·8	13·9
700	371	56·7	45·8	38·2	33·1	29·1	26·0	23·5	21·6	19·9	18·5	17·2	16·2
800	427	64·8	52·4	43·7	37·8	33·2	29·7	26·9	24·7	22·8	21·2	19·7	18·5
		For each 1% Combustible Gases add approximate.											
—	—	25·8	21·5	18·4	16·1	14·3	12·9	11·7	10·7	9·9	9·2	8·6	8·1

Ash and Moisture Content.

The question of ash content is a serious one, as its removal from boilers involves labour and expense. The clinker to be removed from a steam plant is greater in quantity than the weight of ash content obtained by calculation from the results obtained in the laboratory.

As the ashes contain from 1 per cent. to 25 per cent. of combustible matter depending upon the class of coal and the efficiency of the stoking arrangements. In addition they leave the furnace hot. For the purposes of assessing losses the C.V. of the combustible matter may be taken as that of carbon (14,500 B.Th.U. per lb.) and the S.H. of the ashes as approximate 0·2.

High moisture content not only shows that water is being bought for coal. It must be remembered that this moisture has to be driven off in the form of steam, and in this way takes up heat from the combustion zone, and is liable to cause smoking, with consequent reduction of efficiency.

Combustion Computation.

The total heat per pound of water vapour as found in air and in flue gases may be computed, within one B.Th.U., by the formula $H = 1{,}057 + 0{\cdot}46t$, where t is the Fahrenheit temperature. This formula frees the computer from all need for tables so far as computing moisture loss is concerned, while at the same time ensures a degree of accuracy sufficient for all practical purposes.

Analysis of Flue Gases.

It is now standard practice to check the regulation of combustion by analysis of the flue gases. The simplest estimation is that of CO_2, but as the 'theoretical' percentage in ideal conditions varies with the hydrogen content of the fuel a knowledge of the O_2 content is really a better guide to the quantity of excess air admitted. For the purpose of checking waste of fuel a knowledge of the amount (if any) of combustible gases is very valuable, especially as there is a likelihood of these being produced when the excess air is being cut down to a minimum.

Most methods for the estimation of carbon dioxide are based on the absorption of this gas by potassium hydroxide (caustic potash) solution, the process consisting in measuring a volume of flue gas at a definite temperature, and then absorbing the CO_2 by means of a 35 per cent. solution of caustic potash, after which the volume of gas is again measured, the reduction in volume indicating the volume of CO_2 originally present. Many simple forms of chemical apparatus are available for this purpose.

It is frequently desirable in gas analysis to estimate carbon monoxide and oxygen as well as carbon dioxide. Similar apparatus with other absorbent solutions for the respective gases may be used. In the common form of Orsat apparatus, pipettes containing absorption solutions of caustic potash (for CO_2), acid cuprous chloride (for carbon monoxide), and alkaline pyrogallol (for oxygen) are used. The apparatus also contains a measuring burette and levelling bottle, and is generally built up in a wooden case to make the complete set easily portable.

The estimation of combustible gases may also be made by differential combustion of the hydrogen at comparatively low temperatures, and of the CO and hydrocarbon gases at a higher temperature in a tube filled with copper oxide, or by exploding the residual gas with oxygen and estimating the resultant contraction in volume and the CO_2 produced. Special forms of apparatus are available for this work.

Precautions in taking Test Samples of Flue Gas.

It is the practice of many engineers when obtaining accumulative test samples of flue gases to collect these samples over water. Owing to the solubility of the component gases, especially carbon dioxide, in water and the variations of these solubilities with the temperature and pressure, this liquid should not be used to collect the sample if accurate results are desired. If water be used, it will absorb carbon dioxide, for example, until it becomes saturated with this gas with respect to that gas mixture. This results in a loss of some of the carbon dioxide that was in the gas. Then if a gas cleaner in carbon dioxide is collected over the same water, the water will give up carbon dioxide and increase the carbon dioxide content of the gas. If, instead of water, a saturated solution of sodium chloride is used, these errors will be greatly reduced, as the gas is much less soluble in this solution.

The common practice of collecting flue gas from the outlet of a boiler by the use of common iron pipe for the sampling tube is objectionable if the temperature of the gas being sampled is sufficiently high to raise the temperature of the sampling tube to 750° F. When the temperature of the iron reaches 750° F. it starts to react with the carbon dioxide in the gas, forming iron oxide and reducing the carbon dioxide to carbon monoxide. Thus a sample of gas drawn through an iron pipe when at or above this temperature would have its composition completely changed, appearing to contain high carbon monoxide and low carbon dioxide. For sampling at these high temperatures the tube should be made of silica or clay, or be fitted with a water jacket so that it may be cooled. (*Power.*)

Automatic CO_2 Recorders.

In all large installations it is customary to instal apparatus to analyse and record continuously the composition of the flue gases. The majority of these record only the percentage of CO_2, but instruments are available for recording also the percentage of oxygen or combustible gases. They are arranged either to—

(1) record the reduction in volume after absorption of the constituent from a regular series of measured samples; or

(2) to measure some physical property of the gas from which the desired result can be inferred.

Typical instruments in Class 1 are:

(*a*) The 'Auto' CO_2 recorder.—This records CO_2 percentage only.

The 'Auto' CO_2 Recorder.

In fig. 2 is shown, in diagrammatic form, the construction of the instrument known as the 'Auto' CO_2 Recorder, which has been designed and constructed by the *Auto Recorder Company*, and is now in successful operation in connection with a large number of boiler installations in this country and on the Continent. The action of the apparatus is as follows:

A small stream of water is allowed to run into the tank A through the small funnel B, which only allows sufficient water to pass to work the instrument at a suitable speed. The water runs from the tank A through the valve C into the chamber D. When the chamber D is filled up to the pipe E, the water overflows down the pipe F and under the float G, which it raises, thereby moving the valve C upwards and shutting off the water supply from the tank A into the chamber D. This motion of the valve C opens a communication from the chamber D through the pipe H, the water rising over the float G until it flows down the pipe I, and the chamber D is empty; the pipe J then acts as a syphon and drains the water from under the float G, which is then allowed to fall again, and the chamber D to be once more filled. By this means the chamber D is alternately filled and emptied, and, as the float is connected by a system of levers to the distributing slide valve K, the water flowing through is the prime mover of the instrument.

The operation of gas analysis is very simple. The pipe L is connected to the supply of gas to be analysed and, as the end of the pipe dips slightly into the water seal in M, no gas can be drawn back. When the water in the chamber D is running out, there is a slight vacuum in this chamber which draws the gas through the pipes L and E, and, as the first measuring vessel N is in communication with the chamber D through the passages O and P and the slide valve K (this valve having moved over to the right by the rising of the float G), it is entirely filled with flue gas. Much more gas is drawn in than is analysed, in order that the sample

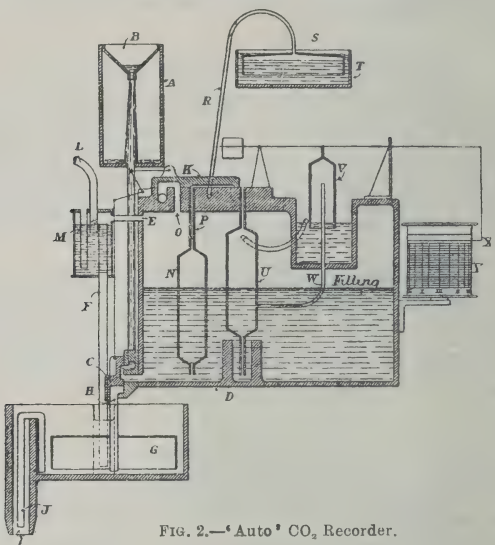


FIG. 2.—'Auto' CO_2 Recorder.

may be as fresh as possible. When the water reverses and begins again to run into D (the valve K having moved over to the *left*, by the sinking of the float G), the gas enclosed in the vessel N is sent through the pipes P and R, and under the dome S, which stands immersed in a solution of caustic potash contained in the vessel T. This dome is of large area, and the gas is in a thin layer between the wet dome and the top of the solution, so that the absorption of the CO_2 is rapid. After the water has risen and expelled all the gas from the vessel N into the absorber, and has again begun to run out, the valve K having moved over to the *right*, the water, as it descends in the second measuring vessel U, draws in from the absorber S the gas which has not been absorbed. As the measurer U is of the same capacity as the measurer N, it follows that, if there is not sufficient gas received from the absorber S to fill it, it will, after emptying the absorber, draw the remainder from under the bell V through the pipe W, and cause the bell to descend; the greater the amount of CO_2 absorbed the lower the bell V will be pulled down. As the bell is suspended from a balanced lever, which has a pen X attached to it, the pen marks the chart Y, which is placed round a clockwork drum, the lines made by the pen being of varying lengths, according to the amount of CO_2 which was in the quantity of gas in the first measurer N.

It has been found that one of the greatest troubles with recorders of this kind is to keep the pipes from the flue to the instrument clean; in the 'Auto' Recorder this is effected by a special filter.

(*b*) The Duplex, Mono Recorder.—This records in addition to the percentage of CO_2 the presence and amount of combustible gases. (Fig. 3.)

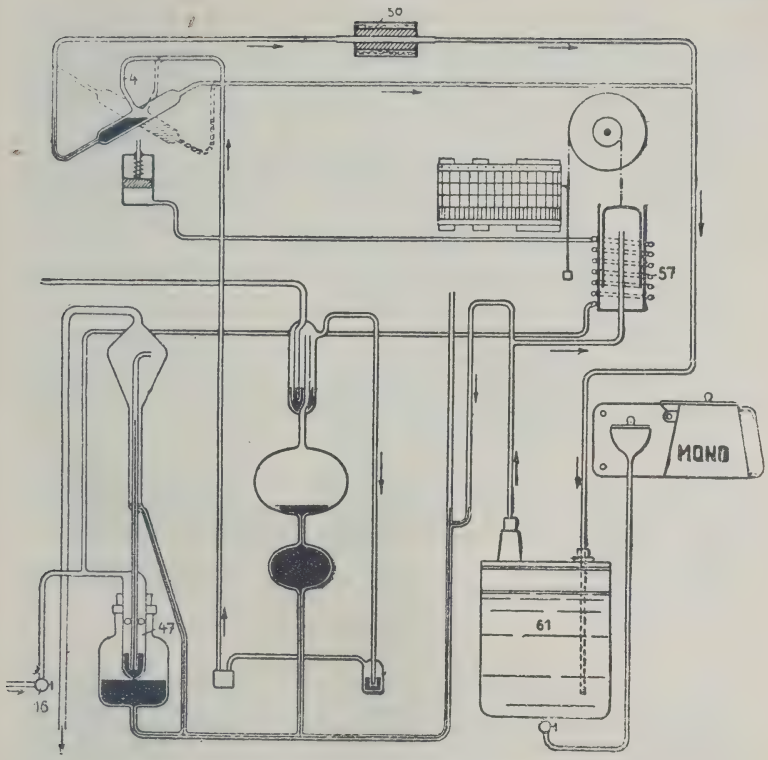


FIG. 3.—DUPLEX MONO RECORDER.

3. Connection from directing valve to furnace lead.
4. Connection from directing valve to volumeter lead.
6. Connection from directing valve to absorption tank lead.
7–9. Directing valve cylinder and piston.
11. Volumeter measuring chamber.
12. ,, lower chamber.
13. Mercury bottle.
14. Water pressure pipe.
15. Water trap.
16. Regulating valve.
19. Top mercury seal.
23. Connection between lower mercury seal and water trap.
24. Volumeter lower pipe.
25. Compression tube mercury drain.
27. Drum carrying chart.
29. Pipe line from directing valve through furnace to absorption tank.
30. Drum carrying chart.
31. Pipe line from directing valve to absorption tank.
32. Lower mercury seal tube.
37. Clock drum.
42. Water connection to directing valve.
43. Compression tube.
44. Stop tube.
46. Water discharge to waste.
47. Compression tube mercury reservoir.
48. Lower mercury seal.
50. Electric Furnace.
51. Combustion tube.
52–53. Pen operation ≤ heels.
57. Gasometer and measuring bell.
59. Stop tube connection to absorption tank.
61. Absorption tank.
64–65. Filling device.

Operation.

The operation of the *Duplex-Mono* is briefly as follows:—

The gas sample is taken from the place where the most characteristic sample can be obtained through the ½-in. sampling tube and then through a special filter which removes all suspended matter and moisture, thus obviating the possibility of the piping to the instrument choking, or water-traps forming. The gas then passes through a lead pipe ¼ in. in diameter to the instrument; this sample is taken, and driven, through the instrument by a mercury pump of special design which is an integral part of the apparatus and is operated by either water or compressed air, the pressure required being not less than 14 lbs.

The gas is drawn from the flue by the suction stroke of the mercury pump into a volumeter, which on the pressure stroke accurately measures the sample, after which it is forced through an absorption tank containing caustic potash, where the CO_2 is absorbed. The residue passes into a measuring bell, floating in glycerine, which, as the gas passes into it, rises and turns a balanced wheel, which, through a pin, actuates the pen wheel. This, in turn, operates the pen and causes it to draw a line on the chart indicating the percentage of CO_2.

When the mercury pump has completed its pressure stroke its suction stroke commences, and the measuring bell is put in communication with the atmosphere, and ejects by its own weight the residual gases, whereby the pen returns to zero.

On the next stroke of the mercury pump the gas sample is diverted by a directing valve operated by the water pressure) after being measured in the volumeter, and passed through a small electric furnace where the combustible gases are burnt to CO_2 and water vapour, in the presence of copper oxide, which supplies any required oxygen and also acts as a catalytic agent; from this furnace the gas passes through the absorption chamber where the water vapour and the total CO_2 (the original CO_2 in the flue gas and the CO_2 formed by the complete combustion in the furnace) is absorbed, and the residue measured in the measuring bell and the total percentage of CO_2 and combustible gases registered on the chart.

This process of *alternate analysis* continues automatically, about thirty analyses being made each hour.

Owing to the special design of the gas filter the pipes cannot choke, and in consequence the maintenance is negligible. The gasometer being cooled by a coil in which the operating water circulates and the rest of the tubes being kept at the same temperature by the mercury which is also in constant contact with the water, the gas before analysis, in the volumeter, and after analysis, in the measuring bell, is always at the same temperature, thus the measurements are extremely accurate.

The instrument is supplied with a two months' chart, self-filling pen, and with a special filling device for replenishing the caustic potash in the absorption tank; as well as a filter for filtering the operating water before admission to the instrument.

Typical Instruments in Class 2.

The 'W.R.' Patent CO_2 Indicator.

(International Gas Detectors, Ltd. (Late W.R. Patents, Ltd.), Great Wilson Street, Leeds.)

(*a*) The W.R. CO_2 or Combustion Indicator is an apparatus which continually, automatically, and accurately indicates the percentage of CO_2 which is passing through the flues at any hour of the day and night. With this Indicator installed on the boiler plant the percentage of CO_2 obtained can be seen at once, and the necessary steps taken to increase this amount and thus save fuel.

Fig. 4 shows a new type recently introduced and placed on the market. It can be used in all cases of combustion where any class of fuel is used, and where CO_2 is a product of this combustion. The arrangement adopted differs materially from other CO_2 indicators.

The method employed in the 'W.R.' Patent CO_2 Recorder depends on the difference of pressure generated on the two sides of a porous pot, one side of which is exposed to flue gas whilst the other contains soda lime granules contained in a paper cylinder which absorbs the incoming CO_2.

An aspirator, S, worked by the natural draught which it increases, is fixed at a point as near the chimney base as possible, and continually aspirates gases from the flue it is desired to keep under observation. The path of gases is shown by the arrows. They are first drawn through a filter F, and pass from thence into a chamber containing a porous pot, inside which is a dry reagent, R. The flow of gas is shown by continuous bubbling through water in vessel B. A pipe connects the chamber with a vessel containing water into which dips one end of another pipe, G, the other end of which is taken into the inside of the porous pot. Some of the gases penetrate into the interior of this pot, and are absorbed by the reagent, with the result that a partial vacuum is formed, and the water is forced up the pipe G. The latter is provided with a scale so graduated that the percentage of CO_2 in the flue gases can be read.

The W.R. Indicator is prompt in its action, having a lag of three minutes only. Its deviation from absolute accuracy has been found to be only 0·1 per cent. It is well constructed; has no mechanically moving parts; is fireproof and foolproof, and can be worked by any boy of average intelligence. With each Indicator the manufacturers furnish a blue print showing the method of fitting to each particular boiler, also special detailed instructions for installing the apparatus, which operation can be completed in from four to six hours by any works' engineer

or fitter. In cases where a recording graph is desired, one of a specially approved form can be supplied by W.R. Patents, Ltd. The cartridges last 24 hours of continuous working.

'Galvanised Iron Pipe' should never be used for connecting a CO_2 machine, that draws a continuous sample, to the sampling point at the boiler. The sulphur dioxide in the flue gas attacks the zinc coating on the pipe, forming zinc sulphate. This soon plugs the pipe and prevents the flue gas sample from being drawn through it.

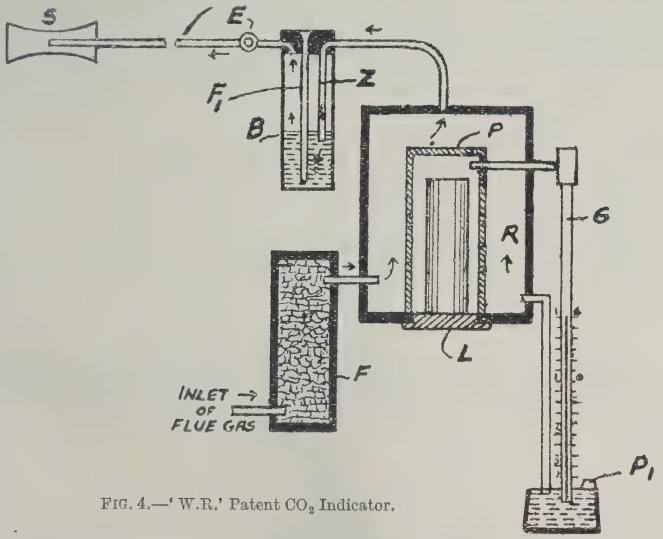


FIG. 4.—'W.R.' Patent CO_2 Indicator.

(*b*) **The Shakespear Electrical CO_2 Recorder** made by the Cambridge Instrument Co., Ltd.

The principle employed in this instrument is one which was devised by Dr. G. A. Shakespear of the University of Birmingham, for the testing of the purity of gases, and has been applied to many other purposes in addition to the recording of CO_2 content of flue gases.

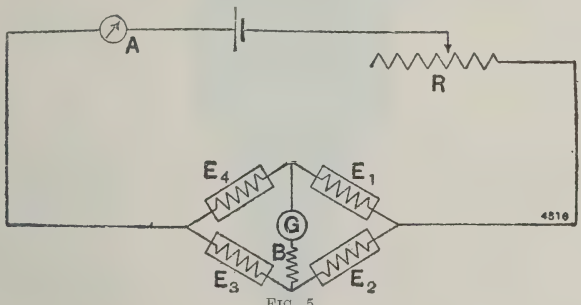


FIG. 5.

In this CO_2 meter, four identical spirals of platinum wire are enclosed in separate cells E_1 E_2, E_3, E_4, in a metal block, as shown diagrammatically in Fig. 5. Each of these spirals forms one arm of a Wheatstone bridge circuit. A definite electric current is allowed to flow through the bridge, thereby causing the spirals to become heated and to lose heat to the walls

of the cells. If two gases having different thermal conductivities are introduced, one into two of the cells (say, E_1 and E_3), and the other gas into the other two cells (E_2 and E_4), the spirals E_1 and E_3 will cool at a different rate to E_2 and E_4, and will therefore be maintained at a different temperature. The consequent difference in the resistance of the spirals will throw the Wheatstone bridge out of balance, causing a deflection of the galvanometer G, the extent of which depends upon the difference in the conductivity of the two gases. The construction is such that changes in the temperature of the gases affect both sides of the bridge equally. If, therefore, the cells E_2 and E_4 contain a pure gas and the cells E_1 and E_3 the same gas mixed with some other constituent, the extent of the deflection will be an indication of the amount of the second gas present, and the galvanometer can be calibrated to show directly the percentage composition of the mixture.

The difference in conductivity between air and carbon-dioxide enables the method to be used to determine the percentage of carbon dioxide in flue gases. These gases consist chiefly of nitrogen and carbon dioxide mixed with small amounts of oxygen, water vapour, and carbon monoxide. Since carbon monoxide, oxygen and nitrogen have all nearly the same thermal conductivity, the small variations of these gases do not affect the readings. The effect of the water vapour can be counteracted by keeping the gases in all the cells saturated.

By passing the gases through an electric furnace in which combustible gases are oxidised the instrument will indicate the change in composition so caused and can therefore be calibrated to read the percentage of combustible gases.

In practice, two of the cells are filled with air saturated with moisture, and the other two are exposed to the gas under test. The difference in conductivity of the gases in the cells then depends solely on the percentage of carbon dioxide in the gas.

The current in the Wheatstone bridge circuit, as shown by an ammeter (marked A in Fig. 5), is adjusted to a constant amount by a rheostat R, while the resistance B in the galvanometer circuit enables the electrical zero of the CO_2 meter to be adjusted.

CO_2 Metering Unit.

The metal block in which are mounted the four cells containing the platinum spirals is combined with a water-bubbler and aspirator to form a complete all-metal metering unit. This unit, which is illustrated in figs. 6 and 7, is so designed as to give accurate readings in

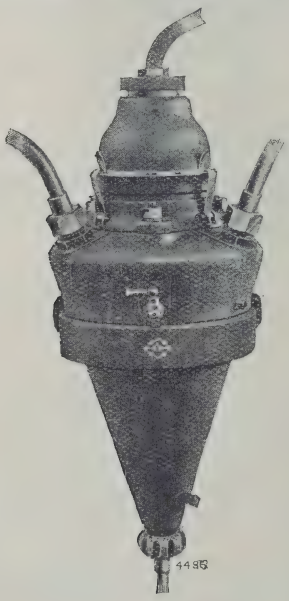


Fig. 6.—7 × 7 × 16 ins.

widely varying boilerhouse conditions, and is exceptionally robust and dependable. In fig. 7 the complete metering unit is shown diagrammatically in section. As mentioned two of the cells in the meter block A contain air. These air cells are sealed, but communicate with a central compensating chamber which contains a wick moistened with clean water, so that the air in the cells is saturated. The wick carries sufficient water to ensure complete saturation for a long period.

The ends of the platinum spirals are connected to terminals T in the head of the meter, which is provided with an easily detachable cover C. All meters of the same type are electrically interchangeable. Four-way cable is taken from these terminals through the screwed gland D in the cover to the indicator or to the recorder, which consists of a special indicating or recording galvanometer. The current is either supplied by an accumulator or the outfits are arranged to run off the local main electric supply.

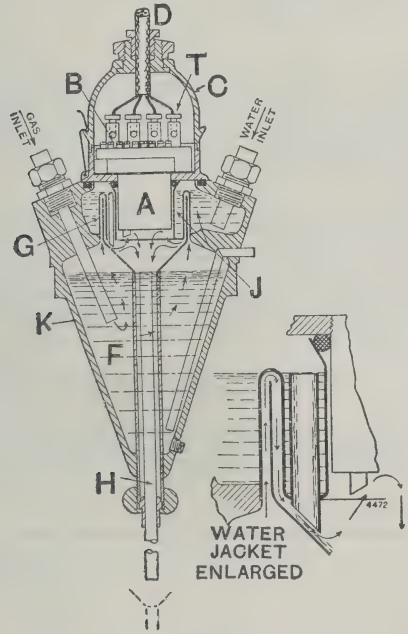


FIG. 7.

The CO_2 meter A is bolted to the top of the gunmetal casting K comprising the bubbler and aspirator, the joint being made gas-tight by a rubber gasket. In Fig. 7 the arrows show the direction of flow of the flue gases. After bubbling through the water in the bubbler vessel F, the gases flow through an annular surface cooler G, and past the CO_2 meter A to the central pipe H which forms the aspirator. The gases diffuse into the remaining two cells of the meter through a shrouded hole drilled in the end of the meter block. The meter block is provided with a water-jacket J, through which the water supply passes to the bubbler; the annular cooler G is immersed in this water-jacket, so that the meter block and the flue gases are brought to the same temperature, thus eliminating temperature gradients. Any excess water vapour present in the gases after their passage through the bubbler is condensed out in the surface cooler and drained back into the bubbler. The gases are thereby brought to saturation at the temperature of the meter block. The bubbler is so designed that there is a continual slow circulation of water through the bubbler vessel F, which ensures that any SO_2 in the flue gases is washed out without any appreciable quantity of CO_2 being removed. A device B provided in the head of the meter enables the electrical zero to be adjusted.

SOLID FUELS.

The most important naturally occurring solid fuels are :—

Coal ; Lignite ; Peat ; Wood.

These may be burned in the raw state, or may be subjected to destructive distillation. In the latter case solid, liquid, and gaseous products are evolved. The more important solid fuels which may be prepared by this treatment are coke (of two kinds, metallurgical and gas-works coke), lignite coke (of little value), peat charcoal, and wood charcoal.

Powdered coal is sometimes made into briquettes by the addition of a small percentage of coal-tar pitch or other binding material, and pressing in moulds.

COAL.

Coal consists of the mineralised remains of ancient vegetation and varies considerably in composition according to the particular portions of the paleolithic plants preserved, the age of the deposit, its depth, and the properties of the surrounding strata. The younger formations are more nearly allied to cellulose, as the lignites and brown coals, and contain a relatively large proportion of both known as coke. The matter distilled off is known as the volatile bituminous matter, and may amount to as little as 1 per cent. or as much as 40 per cent. of the coal according to its nature. (See Table *below*.)

It will be realised that bituminous coal is a very complicated mixture of heavy hydrocarbons, and much work has been done in attempts to determine its constitution. Prof. Phillips Bedson of Armstrong College, Newcastle, uses the selective action of various solvents of which the most useful appear to be chloroform and pyridene, which dissolve out certain constituents.

Workers at Owens College, Manchester, have attempted to separate mineralogical constituents, and recognise four main kinds, viz. Fusain, Vitrain, Clarain, and Durain, but it is doubtful if these constituents are definite compounds, and their composition as well as their proportions vary from one seam to another.

The preparation of microscope sections of coal and the examination thereof by transmitted light has been found to give valuable information as to the nature of the coal and its suitability for particular processes.

The composition of coal from various fields varies considerably according to the conditions to which the coal has been subjected during its formation. Its variation in composition can be seen from the following table of coal analyses :—

Ultimate Composition of Coal (*Brame*).*

Type of Coal.	Carbon.	Hydrogen.	Oxygen and Nitrogen.	Fixed Carbon.	Volatile Matter.
Lignites and brown coals . . .	69·5	5·5	25·0	52·0	48·0
Splint coal (Fife)	82·0	5·0	12·8	61·0	39·0
Gas coal (Durham)	85·0	5·5	8·2	66·0	34·0
Coking coal	87·3	5·05	6·9	73·5	26·5
Smokeless steam (Welsh) . . .	91·3	4·05	3·9	85·5	14·5
Anthracite (Scotch)	91·1	3·5	4·65	88·5	11·5
Anthracite (Welsh)	91·0	3·9	4·28	93·0	7·0

Coal possesses a sp. gr. of from 1·2 to 1·5 (water = 1) mainly depending on its composition. It also varies considerably in hardness and texture. The combustible portion of coal usually consists of from 73 to 93 per cent. carbon, 3 to 6 per cent. hydrogen, 2 to 20 per cent. oxygen, 1 to 2·5 per cent. nitrogen, and 0·5 to 5 per cent. sulphur. When completely burned coals yield from 1 to 15 per cent. ash. The moisture content varies within very wide limits. The heat value of coal is generally from 12,000 to 16,000 B.Th.U. gross. A comparison of the properties of coal with those of other solid fuels may be seen in the following table :—

* 'Fuels: Solid, Liquid, and Gaseous,' p. 32.

CALORIFIC PROPERTIES OF VARIOUS FUELS.

Substance.	Calorific Power.		Observed by
	Calories.	B.Th.U.	
Scotch peat briquettes (16 per cent. H_2O)	4,754	8,558	F. Mollwo Perkin.
Coke from Scotch peat	7,600	13,680	"
Oak (containing 13·3 per cent. H_2O)	3,990	7,182	Gottlieb.
" " 11·8 "	4,155	7,479	"
Yoke elm " 12·2 "	4,151	7,490	"
Beech " 12·95 "	4,168	7,502	"
" " 13·95 "	4,101	7,382	"
" " 13·75 "	4,114	7,405	"
Birch " 11·83 "	4,207	7,573	"
Pine " 12·17 "	4,422	7,960	"
Fir " 11·80 "	4,485	8,073	"
Anthracite	8,400	15,120	Dosch.
Anthracite from Westphalia	7,575	13,635	Langbein.
Short flame sinter coal	8,200	14,760	Dosch.
Kentucky coal (U.S.A.)	7,785	14,013	U.S. Geological Survey.
Illinois coal "	7,056	12,701	"
Pennsylvania coal "	7,875	14,175	"
Coke	6,900	12,420	Dosch.
"	6,919	12,454	Langbein.
Lignite	4,000	7,200	Dosch.
Earthy lignite	3,000	5,400	"
Lignite briquettes	4,668	8,402	Langbein.
Lignite from Saxony (moist)	2,622	4,720	"
Peat	4,230	7,614	"
Sawdust briquettes	4,065	7,317	"
Spanish lignite	5,761	10,369	H. Moore.
Asphaltum (extracted from petroleum)	9,550	17,190	"

Analysis of Coal.

The chemical tests usually made upon coal are to ascertain the calorific value, coke value, ash content, moisture content, and ultimate composition. The volatile matter per cent. is obtained by deducting the coke value of the dry coal from 100. The 'fixed carbon' is obtained by deducting the ash content from the coke value.

ANALYSES OF COALS. (*Gray & Robinson.*)

Constituents.	1.	2.	3.	4.	5.	6.	7.	8.	9.	10.	11.	12.
Carbon	62·55	65·50	68·64	69·50	72·14	73 77	74·99	74·57	77·00	76·71	78·80	85·70
Hydrogen	3·87	4·23	4·33	4·42	4·40	4·55	4·66	4·74	4·51	4·67	4·92	2·97
Sulphur	1·62	0·67	1·33	0·67	0·61	0·41	0·55	0·47	0·78	0·73	0·50	0·62
Water	8·63	7·92	8·06	9·28	8·34	7·99	7·21	6·95	2·31	1·94	2·72	3·15
Ash	11·78	10·15	6·90	5·97	4·22	1·76	2·68	1·37	7·17	8·49	4·18	3·59
Oxygen and nitrogen	11·55	11·53	9·84	10·16	10·29	11·52	9·91	11·90	8·23	7·46	8·88	3·90
Coke	61·74	61·06	57·53	59·12	58·47	56·93	56·59	56·84	74·61	72·32	66·43	92·03
Fixed carbon	49·96	50·92	50·63	53·15	54·25	55·17	53·91	55·47	67·44	63·83	62·25	88·44

1. Splint coal, Lochgelly, Fife.
2. Boiler fuel (a mixture), Fife.
3. Ell coal, Bothwell.
4. Main coal, Bothwell.
5. Poytshaw coal, Bothwell.
6. Dunfermline splint, Cowdenbeath, Pit 10.
7. Splint coal, Bothwell.
8. Dunfermline splint, Cowdenbeath, Pit 7.
9. Boiler fuel, Aitken Pit, Kelty.
10. Haughrigg coking coal, Kilsyth.
11. Coking coal, Kilsyth.
12. Anthracite, Kilsyth.

It is manifest that in selecting coal for a fuel considerable care is necessary to ensure that coal of the right class is chosen.

Ash of Fuel.

In addition to the mineral matter naturally occurring in the vegetable detritus from which the coal was formed, there is usually more or less inclusion of further mineral matter in the seam itself, and this may be changed considerably in chemical composition by the infiltration of water containing mineral matter in solution. Such mineral matter is incombustible, and is left behind as an ash or clinker. Not only its amount but its composition and melting point are important matters, as a readily fusible ash may melt into a treacly mass, interfering with the air supply and so impeding combustion, and also rendering ash removal a troublesome operation.

Analysis of Coal.

Before attempting the analysis of a coal it is necessary to secure a truly representative sample, and this is by no means an easy matter. In the case of coal delivered by railway wagon it will suffice to take, say, 14 lbs. from each wagon, the sample being chosen to be as representative as possible. In the case of steamer or barge delivery, similar carefully taken samples should be collected at approximately equal intervals as unloading proceeds. The samples should be collected together, subjected to any screening test desired, and then broken roughly on a clean floor till there is no piece bigger than a walnut; the coal should then be well mixed, quartered, and the selected quarter further broken, mixed, and requartered until a suitable size of sample remains. Messrs. Sturtevant make a useful sampling machine which crushes, mixes, and divides out a sample mechanically.

Great care should be taken to avoid loss or gain of moisture by the coal during this sampling process, as this, of course, vitiates the accuracy of the analysis.

A committee of the British Engineering Standards Association has carefully considered the conditions necessary to be observed in order to ensure that samples taken should be within certain prescribed limits of error from the true mean, and finds that the size of the individual portions taken as samples and the number of such samples necessary to be taken in order to ensure the desired accuracy is dependent upon the sizes of individual pieces of coal and upon the ash content of the coal much more than upon the total quantity of coal in the consignment to be sampled. Their findings are contained in B.E.S.A. Report No. 403 of 1930, entitled, ' Report on the Sampling of Small Fuel up to 3 ins.,' embodying some general principles of sampling. Based on such investigation they have also issued Report No. 404, ' British Standard Specification for the Sampling and Analysis of Coal for Export.'

In the preparation of coal for the market the run of mine coal is first screened to separate the large (over 1 in. or so) from the smalls, and the former is then passed over a picking belt where visible impurities—stone, shale, pyrites, &c.—are picked out by hand. The large coal may be further screened for certain markets into nuts, cobbles, and large screened coals. The small coal may either be sold without further preparation or further screened into various sizes of smalls and duff, and where reduction in the ash content of smalls is considered desirable these may be subjected to washing or dry cleaning processes by which the particles of high specific gravity are separated from the lighter coal. When purchasing coals, therefore, the sizes of screen to be used and the washing or other purifying processes used in its preparation should be known.

The Commercial Selection of Coals.

(Contributed by Chas. F. Wade, A.M.I.Mech.E., A.M.I.E.E.)

The commercial value of a coal for any specific purpose and for direct combustion purposes will be dependent upon the following factors:—

(i) The lower or net calorific value.
(ii) The amount of volatile matter and nature of coke.
(iii) The amount and nature of the ash content.
(iv) Its physical suitability for the particular conditions under which it is to be burned.
(v) The price per ton delivered on site.
(vi) Local cost of labour for handling, firing, and removal of ashes.

The net calorific value (i) calls for both a test for calorific value and also either an ultimate analysis or calculation in order to arrive at the hydrogen content of the fuel. The latter calculation or estimate is made from the percentage of volatile matter in the coal as disclosed by proximate analysis.

The net commercial calorific value will vary with the conditions obtaining in individual installations, and will depend upon the final temperature of the gases leaving the boilers or furnaces, together with the total hydrogen in the fuel as fired.

The total hydrogen will be the sum of the hydrogen combined in the fuel and that forming one-ninth of the moisture. Lower commercial calorific value for any specific plant will be

$$HCV = \left\{ 9H \left([212 - t] + 970{\cdot}4 + 0{\cdot}47\ [T - 212] \right) \right\},$$

where,

HCV = higher or calorimeter value of fuel in B.Th.U. per lb. as fired;
H = hydrogen fraction of the fuel as fired;
t = temperature of the fuel in °F., as fired;
T = final temperature of waste gases.

In the absence of direct ultimate analysis of the coal, the percentage of hydrogen in the fuel may be closely estimated by the following formula (Diederichs'):—

$$H = V \left(\frac{7{\cdot}35}{V + 10} \right) - 0{\cdot}013,$$

where,

H = percentage of hydrogen in the combustible; V = percentage of volatile matter in the combustible.

Combustible = coal, less sum of moisture and ash.

(ii) The amount of volatile matter and nature of the coke are found simultaneously in the proximate analysis. This test will give the correct classification of the coal as to its qualifying characteristics for special uses. The ratio of volatile matter to fixed carbon is of the greatest importance in deciding the suitability or otherwise of a coal for use on any particular plant; all other things being equal, the fuel with the lowest volatile content which can be efficiently consumed in the specific furnace conditions will be the most economical in use. This can only be definitely determined by the careful working test of a sample delivery of the fuel, and careful note made of the percentage of CO_2 in the waste gases, together with the final temperature of the gases and the percentage of combustible on the ash heap. The following general classification of coals will be of assistance in selecting suitable fuels for given purposes from the commercial viewpoint of economy in use.

Class of Coal.	Per Cent. Volatile Matter in Combustible.	Economical Application.
Short flame non-caking, semi-bituminous, or Welsh steam coals	12–20	Steam raising with strong draught.
Short flame caking coals	20–25	As above and coke manufacture.
Medium flame, moderately caking coals	25–32	Steam raising, domestic grates.
Long flame caking coals	32–40	Gas manufacture.
Long flame non-caking North Country coals	over 40	Furnaces, gas producers.

Burning a coal in a furnace which has insufficient draught for complete combustion will result in excessive waste of unburned carbon as coke in the ashes. Burning a coal of an unnecessarily high volatile content will produce an unnecessarily high loss in the waste gases, due to the hydrogen content, smoke will be difficult to avoid.

The amount of ash and moisture will not affect the classification given above.

(iii) The amount of the ash in the coal will not only reduce its commercial value by the reduction in actual calorific value, but also by its physical effect will reduce the value at a greater rate than that of the percentage of ash. For ordinary conditions of firing and draught, without steam jets, the rate of fall in practical value with the ash content will gradually increase until at 40 per cent. ash the fuel will be unburnable and valueless. This is for average ash.

The nature of the ash is chiefly affected by the percentage of iron in its composition, the fluxing effect of ferric oxide in the ferruginous ash coals causing the formation of liquid slag at comparatively low temperatures, which closes up the fire-bar spaces, assists in the fusion of firebars, and entraps appreciable quantities of unburned combustible. Ferruginous ash is distinguished by its red colour, the percentage of iron being roughly indicated by the depth of the coloration of the ash in the crucible after proximate analysis for ash.

(iv) The physical suitability of the coal for particular conditions depends upon the construction of the firegrates, type of mechanical stoker, and the amount of draught available. Under practically every condition of use mixed coals as regards size should be avoided. For hand-fired boilers and furnaces screened cobbles or nuts should always be selected. Large steam coal is

to be avoided; if fired as received, in lumps, efficient combustion is impossible, while breaking down results in uneven fires and waste due to the formation of a quantity of slack in the process.

Fine slacks are unsuited for hand-firing on plain grates, but can be burned very efficiently on special forced-draught grates fitted with steam jets and closely spaced firebars. High ash coals may be more efficiently burned on these grates than with plain natural or induced draught.

(v) The price per ton of coal, delivered, together with the cost of labour in handling and incidental work, is the last test of the value of the coal to be applied after all the technical deductions already referred to have been made, and is to be utilised to arrive at the net number of B.Th.U.'s per unit of cost under practical conditions of work.

Wherever possible, the following procedure should be adopted in selecting the most economically suitable coals for contract purposes:—

A sample delivery should be ordered of sufficient size to allow of observation under practical working conditions for a long enough period to ascertain the maximum hourly evaporation under forced and economical conditions of work, probable ash-heap loss, chimney loss, etc. This cannot be carried out without the proper equipment of testing instruments. This working observation will allow of a close estimate of labour and power costs. A low-grade slack of about 10,000 B.Th.U. per lb. will probably call for a power and labour cost quite double that involved in the use of good quality washed nuts of 14,000 B.Th.U. per lb. This shows the uselessness of evaluation on the calorific value alone. Standing losses on the plant, such as conduction and convection, etc., vary directly as the number of boilers, etc., used.

Final evaluation should be made as follows:—

Gross or higher calorific value of the coal, air-dried, then deduct the following:—

Hydrogen loss under practical conditions of waste gas temperature;

Heat loss in waste gases due to excess air as per CO_2 and temperature readings,

Loss due to unburned carbon in ash-heap per lb. of fuel used.

The final figure thus obtained gives the practical working value of the fuel in B.Th.U. per lb.

To the price of the coal must be added the following costs:—

Cost of attendance for firing per ton of coal used;

" power, such as stoker, fan, conveyor, economiser drive, etc., per ton of coal used;

" ash removal per ton of coal used.

From the data thus obtained the ultimate cost of, say, 100,000 B.Th.U. is obtainable for the actual conditions existing with the particular plant upon which the coal is to be used.

The system which has been used with success is to reduce the value of each coal to a figure of merit consisting of its comparative value with pure carbon as a standard, the latter being taken as '100.' The average final temperature of the gases is ascertained in the usual manner. The chimney loss for pure carbon burned perfectly without excess air is then calculated for the conditions of waste gas temperature obtaining on the plant. Taking the calorific value of pure carbon as 14,544 B.Th.U. per lb., the '100' figure of merit for this plant will be 14,544 minus the gas loss per lb. of carbon. This represents theoretical perfection with the plant. The actual results obtained for the coal under examination are expressed as a percentage of the pure carbon theoretical figure, and this is taken as the relative figure of merit for the fuel.

(*C. F. Wade.*)

Economic Effects of Impurities in Coal.

Each per cent. of sulphur reduces the boiler capacity by about 1·5 per cent. In experiments made in America with a chain grate stoker, using Illinois coal mixed with various percentages of ash, the efficiency dropped gradually as the ash increased up to 35 per cent., after which the efficiency dropped to zero rapidly with a fuel containing 40 per cent. of ash.

Loss of Thermal Value of Coal by Storage.

An average of 741 tests, in America, of a particular coal showed, after three years in the open air, a reduction of from 35 to 28·3 per cent. of volatile combustible matter, indicating a thermal value loss of 20 per cent.

Coal in storage should be inspected regularly. If the temperature reaches 150° F. the pile should be carefully watched, and if the temperature rises to 175° or 180° F. the coal should be removed as promptly as possible. The coal should be thoroughly cooled before being replaced in storage.

Temperature of Decomposition of Coal.

The temperatures at which coals begin to decompose are of considerable importance, both for technical purposes and as an aid to the study of the constitution of coal. There is little published information of a precise nature on this point, and it is frequently assumed that coal may be heated to 250° C., for purposes such as drying, without causing any modification in the coal substance. A general investigation of the behaviour of coal on heating is now being carried out at H.M. Fuel Research Station, and a report recently issued by H.M. Stationery Office deals with the results of experiments to determine the temperatures at which products of decomposition of various types of coal are first given off when the coal is heated. It is shown that when bituminous coals are heated, oil fumes may appear at temperatures as low as 215° C. The investigations are continuing and further reports will be issued in due course.

Utilisation of Anthracite Wastes.

A series of tests made in America, consisting of 30, 50, and 70 per cent. run-of-mine coking coal mixed with anthracite culm, showed that the pulverised bituminous coal alone produced better results, except the mixture of 30 per cent. culm with 70 per cent. bituminous, which was equal to that of the bituminous coal alone.

Stowage Space of Coal.

The stowage space occupied by coal is 40 to 42 cubic feet for one ton of Welsh steam coal, 45 cubic feet for one ton of North country coal, and 45 to 50 cubic feet per ton of Scotch coal.

EVAPORATIVE POWER OF FUELS.

The heat required to evaporate one pound of water at boiling point is 967 B.Th.U.

To raise water from 100° F. to boiling point and evaporate into steam at 212° F., 1,078 B.Th.U. per pound of water are required.

By dividing the calorific power (in B.Th.U.) of a fuel by 1,078 we obtain the *theoretical* quantity of water which it will raise from a feed-water temperature of 100° F. and evaporate at 212° F.

In practice a good boiler develops a thermal efficiency of about 70 per cent., and the actual evaporation obtained is therefore about 70 per cent. of the theoretical figure.

Evaporative Power of Coal.

Coal.	Calorific value. B.Th.U.	Evaporative power from 100° F. (37·7° C.) to Saturated Steam at 212° F. (100° C.).	
		Theory.	Practice. (70 % efficiency.)
Welsh anthracite . .	15,000 to 15,500	13·9–14·4 lbs.	9·7–10·1 lbs.
Scotch anthracite . .	14,500 to 15,000	13·45–13·9 ,,	9·4–9·7 ,,
Welsh navigation coal .	14,000 to 15,000	13·00–13·9 ,,	9·1–9·7 ,,
Average British steam coal	12,000 to 13,500	11·1–12·5 ,,	7·8–8·75 ,,

(See also Table of Evaporative Values, page 2103.)

Evaporation Factors.

To reduce results to ' from and at 212° F.'

Temp. of Feed Water in F.°	Boiler Pressure in Pounds per Square Inch.												
	60	80	100	120	140	160	180	200	220	240	260	280	300
40	1·209	1·214	1·218	1·222	1·226	1·229	1·232	1·234	1·237	1·239	1·241	1·243	1·245
50	1·198	1·204	1·208	1·212	1·215	1·218	1·221	1·224	1·226	1·229	1·231	1·233	1·235
60	1·188	1·193	1·198	1·202	1·205	1·208	1·211	1·214	1·216	1·218	1·220	1·222	1·224
70	1·178	1·183	1·187	1·191	1·194	1·197	1·200	1·203	1·206	1·208	1·210	1·212	1·214
80	1·167	1·173	1·177	1·181	1·184	1·187	1·190	1·193	1·195	1·198	1·200	1·202	1·204
90	1·157	1·162	1·167	1·170	1·174	1·177	1·180	1·183	1·185	1·187	1·189	1·191	1·193
100	1·147	1·152	1·156	1·160	1·164	1·167	1·170	1·172	1·175	1·177	1·179	1·181	1·183
110	1·136	1·142	1·146	1·150	1·153	1·156	1·159	1·162	1·164	1·167	1·169	1·171	1·173
120	1·126	1·131	1·136	1·140	1·143	1·146	1·149	1·151	1·154	1·156	1·158	1·160	1·162
130	1·116	1·121	1·125	1·129	1·132	1·136	1·138	1·141	1·144	1·146	1·148	1·150	1·152
140	1·105	1·110	1·115	1·119	1·122	1·125	1·128	1·131	1·133	1·135	1·137	1·139	1·141
150	1·095	1·100	1·104	1·108	1·111	1·115	1·118	1·120	1·123	1·125	1·127	1·129	1·131
160	1·084	1·090	1·094	1·098	1·101	1·104	1·107	1·110	1·112	1·115	1·117	1·119	1·121
170	1·074	1·079	1·083	1·087	1·091	1·094	1·097	1·099	1·102	1·104	1·106	1·108	1·110
180	1·063	1·069	1·073	1·077	1·080	1·083	1·086	1·089	1·091	1·094	1·096	1·098	1·100
190	1·053	1·058	1·062	1·066	1·070	1·073	1·076	1·078	1·081	1·083	1·085	1·087	1·089
200	1·042	1·048	1·052	1·056	1·059	1·063	1·065	1·068	1·071	1·073	1·075	1·077	1·079
210	1·032	1·037	1·042	1·046	1·049	1·052	1·055	1·057	1·060	1·062	1·064	1·066	1·068

For every 10° superheat, 0·0055 is to be added to the above factors.

Coke and Coking.

(Contributed by A. H. Middleton and C. Bride, M.A.(Oxon.), B.Sc.(Lond.), A.I.C.)

At the present time nearly all metallurgical coke is made in by-product ovens, where the chief product is still coke, but where other valuable products are recovered. These are: (1) ammonia, marketed mainly as sulphate of ammonia for use as a fertiliser; (2) tar, which is distilled or refined at the coke works, or sold to tar distillers working privately or on a co-operative basis; (3) benzole and its homologues, which either alone or mixed with petrol is sold as motor fuel. The greater proportion of the benzole produced is sold as 'National Benzole Mixture'; (4) (*a*) coke-oven gas, which is essentially the same gas as that produced from gasworks' retorts; or (*b*) waste heat, which serves for steam generation. The type of oven used determines whether gas or waste heat is the by-product produced.

The largest waste which occurs in almost all coke works is in the sensible heat of the coke when discharged from the ovens, and from which it is possible to recover heat equivalent to 0·4 lb. of steam per lb. of coke. The extra plant necessary for the recovery of this heat is costly and not likely to be generally employed unless it can be simplified and cheapened. Sometimes, owing to the unfavourable situation of a coking plant, the whole of the surplus gas cannot be disposed of with advantage.

In 1929, 13,421,998 tons of coke were made, yielding per ton of coal carbonised (according to circumstances) from 12 to 16 cwts. of coke; 20 to 26 lbs. (and occasionally up to 36 lbs.) of sulphate of ammonia; 2 to 3½ gals. of 65 per cent. crude benzole; 85 to 110 lbs. of tar; 10,000 to 13,000 cub. ft. of gas, a large proportion of which is available for external use either as gas or as its equivalent in waste heat.

Coking Coals.

In contrast to gasworks practice, great attention is paid to the condition and properties of the coal used. Coking coals are bituminous in type and pass when heated through a plastic stage between 250° and 450° C., and must possess sufficient coking power to leave a hard and compact residue on further heating.

Usually more than 8 per cent. of oxygen on an ash- and moisture-free basis denotes insufficient coking power. Oxidation or heating destroys coking power, and thus storage of coal, by encouraging this action, often tends to depreciate coking value. The extent of depreciation varies greatly with different classes of coking coals, some such as certain Derbyshire coals becoming practically non-coking after only two or three days in the coke-ovens' service bunkers, whereas certain types of Durham coal have been kept in stock for five or six years without appreciable loss of coking power. Other properties which distinguish coals most suitable for coking are:—

(*a*) Low content of ash and sulphur.
(*b*) Volatile matter between 24 and 29 per cent., since high volatile matter causes the formation of a 'spongy' coke unless special carbonising conditions are adopted.
(*c*) Negligible amount of alkalies, as these have an injurious effect on oven linings.
(*d*) Less than 4 per cent. of moisture as extra heat above that required for carbonising must be applied to evaporate moisture from the ovens. Thus carbonisation is retarded and the steam produced throws extra work on the gas cooling plant, as it has to be subsequently condensed.

In most coke works the coal is crushed more or less finely before being charged into the ovens and only in a few cases are run-of-mine coking smalls used directly. The effect of crushing is to disseminate shale impurities through the charge, so that their effect as sources of cracks in the coke is reduced. The average size and strength of the coke made is increased, with a corresponding reduction in the production of small coke and ballast. Crushing, however, decreases the packing density of the charge, and so reduces oven capacity. A density variation of from 45 lbs. to 52 lbs. per cub. ft. of coal is typical of varying the degree of crushing within the limits possible in the average coke works crushing plant.

Charging the coal in a loose condition through charge-holes in the oven tops is the usual method of loading. The once common practice of using stamped charges of finely ground moist coal in the form of a solid cake is now confined to a few isolated cases where an exceptionally feebly coking or highly volatile coal is used. Under ordinary conditions this method has nothing to recommend it.

Cleaning processes are used in many coke works to reduce ash and sulphur before crushing the coal. Wet washing plants are common, but require large drainage bunkers to reduce the water taken up by the coal to workable amount; they also necessitate screening of the coal at some part of the process to secure efficient washing. In recent years dry cleaning plants have come into use. In these, coal is separated from its impurities by passing it over oscillating tables through which a strong current of air is blown. This method eliminates drainage bunkers and desiccates the coal.

Ovens.

'Beehive' ovens, formerly largely used, are now almost superseded by the 'By-Product' type; in the latter type two systems of oven heating are in use:—

(*a*) The waste-heat system, in which upwards of 80 per cent. of the total gas evolved is burnt in the wall flues with air at atmospheric temperature, or at the most only slightly preheated by

passage under the hot ovens to the flues. Burnt gases leave the flues at a temperature of about 1100° C., and are used principally for steam-raising, and as in beehive practice, up to 1 lb. of steam is obtained per lb. of coal carbonised. This type of oven is simple in construction and operation.

(*b*) The regenerator system, in which chambers under the ovens, filled with chequer brickwork, as in Siemen's open-hearth steel furnaces, are employed to transfer heat from the burnt gas leaving the flues to air entering them. In this way the waste gas is cooled to 250° C. before it leaves the regenerator chambers, and the air entering the flues is preheated to 900° C. approximately. Only 35 to 50 per cent. of the total gas is required for heating the ovens, the remainder being surplus and available for other uses. The thermal efficiency of the two systems is about the same.

The majority of by-product ovens in use to-day are of pre-war construction and have an average length of 35 ft., a width of 20 ins. to 21 ins., and a height varying from 6 ft. to 7½ ft. Their capacity varies from 6 to 9 tons of coal. Working temperatures of the flues are from 1100° to 1200° C., being limited by the fusion point of the semi-silica (75–85 per cent. SiO_2) refractories used for oven linings and flues. On dry coal a carbonising time of 26 to 32 hours is attained, which gives a daily throughput of 5½ to 7½ tons per oven. Heating flues are arranged either horizontally or vertically in the oven walls. In waste-heat batteries combustion occurs continuously in each flue, and the main difference between the various constructions is in the spacing of the flues to obtain uniform heating, and in the methods adopted to regulate air and gas to each flue. In regenerative ovens, the periodic reversal of gas and air flow through the regenerator chambers causes a periodic change in the conditions in the combustion flues. In ovens with horizontal flues this consists of a reversal of the path of travel of the burning gases in the flues with a change in the points at which air is admitted, but necessitates no alteration in the points at which gas is admitted. In vertical flue ovens, which predominate in number, only half the flues are used at a time for combustion, the other half being used for carrying away the burnt gas. Flue designs in existing ovens vary in the arrangement used to eliminate uneven heating from this cause. They differ mainly in the way in which flues are grouped and cross connected to form separate reversal units.

Quenching coke by water in front of the ovens is usual, either on an inclined bench running the whole length of the battery or in some form of car movable from oven to oven. Approximately 1 ton of water is used per ton of coke quenched, and the coke is very slowly pushed from the ovens to allow sufficient time for quenching. Success of quenching depends largely on the skill of the workmen.

A typical modern post-war battery consists of 40 to 60 ovens, each 43 to 44 ft. long, 13 to 14 ft. high, and 16 to 18 ins. wide, heated regeneratively so as to have a maximum yield of surplus gas. In order to withstand the higher flue temperatures used (1200°–1400° C.), flues and oven linings are constructed in 95 per cent. silica material which will withstand temperatures up to 1650° C. without distortion. In some types the heating flues are built vertically and arranged on the 'hairpin' system, in which reversal of air and gas takes place between pairs of adjacent flues. Alternate flues are fed with air and gas, so that combustion takes place up one flue and down the next, at the bottom of which burnt gases are withdrawn. Another method of flue arrangement is by cross connection of groups of flues in opposite walls of the ovens through flues in the oven tops, so that reversal of combustion takes place from one side of each oven to the other. Regenerator chambers filled with firebrick chequerwork, built under each oven, are subdivided, so that each set of flues has its own regenerator system. The use of silica refractories increases very considerably the safety factor when working at high temperatures, with the additional advantage of better thermal conductivity through the oven walls. They are also much more resistant to the action of alkalies than other coke-oven refractories.

Recently erected ovens have a capacity of 14 to 17 tons of coal, which with a carbonising time of 16 hours gives a daily throughput of about 20 to 26 tons per oven.

Subsidiary, but necessary improvements incorporated in post-war ovens are :—

(1) High-speed pushing machines by which a 44-ft. oven is discharged in 30 secs., or approximately in one-tenth of the time required to push the shorter pre-war ovens.

(2) Electrically driven coal cars, in which the volume of the coal charged can be accurately gauged.

(3) Self-sealing and plug doors, which are removed and replaced mechanically and which require at most an extremely small amount of luting to render them gas-tight.

(4) Remote coke quenching, by which the coke pushed from an oven at 900° to 1000° C. is discharged into a steel car which carries it rapidly from the front of the battery to a water-tower some distance from the ovens, where a measured volume of water is sprayed on to it. After draining the coke is taken to a coke bench and discharged to steam and cool off before being taken by high-speed rubber belts to the screening plant. This system of quenching makes work in front of the oven tolerable under high-speed conditions and prevents damage to the hot oven walls by contact with steam and water, such as is possible from quenching in front of the ovens. Rather more water is required than with other systems (from 1¾ to 2 tons per ton of coke), but quenching is more uniform and coke moisture more constant.

Narrow ovens are better suited for the treatment of inferior coking coals. In extreme cases of feebly coking or highly volatile coals, 12-in. to 14-in. ovens are used, carbonising in 12 hours and producing a satisfactory coke, but with the average range of coking coals a 16-in. or 18-in. oven, carbonising in 18 hours, seems to give most satisfactory results. The amount of

heat supplied to the ovens *via* the combustion flues is usually from 850 to 1100 B.Th.U.'s per lb. of coal carbonised, but varies with the speed and temperature of carbonising and with the efficiency of the system. Under average conditions 60 to 65 per cent. of surplus gas is produced. For maximum yield of gas, compound regenerative ovens are built in which the flues may be heated by combustion of preheated producer, water, or blast-furnace gas, the whole of the coal-gas made being available for other use.

Unless the coke-oven gas can be sold for at least 7½*d.* per 1,000 cub. ft., it does not pay to heat the ovens with producer-gas and liberate for other uses a greater quantity of coke-oven gas.

Coke.

This varies in characteristics according to the coal used and carbonising procedure. Size is dependent on the thickness of the charge carbonised. In beehive practice the largest pieces are 30 ins. to 36 ins. long, approximating to the depth of coal in the ovens. With by-product coke the largest pieces are slightly less than half the oven width in length. Typical characteristics are :—

	True Sp. Gr.	Apparent Sp. Gr.	True Porosity. %	Apparent Porosity. %
Beehive coke	1·88–1·98	0·87–0·90	52–56	45–50
By-product coke	1·90–2·00	0·95–1·10	42–52	38–45

Crushing strength varies from 1,000 to 3,500 lbs. per sq. in., and volatile matter from 0·5 to 3 or 4 per cent. The volatile matter is greatest in high-speed practice where no over-coking takes place. Most of the coke made is used in metallurgical furnaces, and from 85 to 95 per cent. of the product of a coke works is in pieces large enough for such purposes, the remainder being in the form of coke nuts down to ⅝ in. size and coke ballast below this size.

For smelting use a hard dense coke is required strong enough to resist the severe crushing and abrasion it receives Beehive coke is still preferred by some foundry managers on account of its size and bulkiness, although equally good results are obtained from properly sized by-product coke. In British blast-furnaces the greater part of the coke used is the over-coked variety made in pre-war by-product plants, and screened to eliminate sizes below 2 ins. or 3 ins., but American blast-furnacemen appear to obtain greater efficiency by using the smaller but evenly sized and uniform coke which is produced by high-speed ovens.

Correct sizing and efficient screening is established as a means of obtaining greater efficiency in its use as a fuel. Structure and porosity have a definite bearing on the combustibility, while volatile content governs the ignition temperature, so that there are great possibilities in adapting oven conditions for various types of coke production. Screened coke nuts are to some extent used in domestic applications. A small uniform coke with low ash, up to 6 per cent. volatile matter and an open but strong structure is desirable, so that satisfactory ignition and combustion results under the feeble draught conditions of open ranges and fire grates. A new and increasing use of coke nuts is in the synthetic chemical industry where, as producer fuel, they supply the hydrogen and carbon monoxide required for ammonia and methanol syntheses. Other uses are in lime-kilns and for steam-raising. Coke ballast under ⅝ in. size is usually consumed under boilers by means of mechanical stokers, chiefly to supply steam at the coke works and adjacent plants.

4½ to 5 lbs. of steam can be raised per lb. of coke ballast.

By-Product Recovery.

This follows the main lines of gasworks recovery, with the following exceptions. Individual ascension pipes are not liquid sealed in the collecting main, but are shut off by means of valves. The gas-collecting main is kept clear by a flush of tar or ammoniacal liquor. In some cases, cold liquor is sprayed into the collecting main at the junction with each ascension pipe and gives additional gas cooling.

In coke-oven practice ammonia recovery has reached a higher stage of efficiency than in gas-works practice, and benzole is always recovered. Actual yields depend on the coal, and also on the extent of degradation of volatile products in the ovens caused by contact with the hot walls and coke. Ovens are designed with a view to keeping the tops reasonably cool and to allow the rapid removal of gaseous products in order to reduce the time of contact with heated surfaces.

	Average Gas Temperatures.
Gas leaving ovens	600°–700° C.
„ entering collecting main	300°–400° C.
„ leaving collecting main	150°–200° C.
„ entering by-product coolers	70°– 80° C.

Tar.

This is removed partly by condensation from the gas in various parts of the coolers, and is completely removed by passage through tar extractors before the ammonia recovery plant. Three types of extractors are in common use—(*a*) the impact type, in which tar mist is precipitated by contact with metal screens ; (*b*) the cyclone type, in which gas is constrained to follow a rapid spiral

course, and tar mist is separated centrifugally ; (*c*) the direct type, in which tar is precipitated by means of high-pressure ammoniacal liquor sprays.

Composition of Crude Coke-Oven Tar.

Ammoniacal liquor	2·5 – 5·0%	Free carbon in pitch, 6%–15%.
Naphthas	Trace – 0·5%	
Light oils (to 180° C.)	1·0 – 2·0%	Tar acids in carbolic oils, 1%–2·5%.
Carbolic oils (180°–230° C.)	6·0 – 8·0%	Naphthalene in carbolic and creosote oils, 5%–10%.
Creosote oils (230°–270° C.)	8·0 –11·0%	
Anthracene oils (270°–360° C.)	16·0 –22·0%	
Pitch	50·0 –63·0%	
Sp. Gr. at 60° F.	1·150– 1·205	

Ammonia Recovery.

Three systems are used :—

1. *Indirect System.*—Gas is cooled to atmospheric temperatures and sprayed with water. The resulting liquor mixed with liquor condensates from the gas-coolers has a concentration of about 1 per cent. ammonia, of which only 0·1 to 0·2 per cent. is fixed ammonia. The liquor is distilled with live steam and a little milk of lime in a continuous still and passed through a saturator containing hot mother liquor and 5–10 per cent. free sulphuric acid. Ammonium sulphate is formed and salts out, being removed by steam or air blowers at intervals. Draining and centrifuging reduces the moisture in the salt to about 1½ per cent. and the acidity to 0·3–0·5 per cent. Complete drying and neutralisation of the salt is performed in a separate plant. The indirect system is confined to early coke works. It is flexible and can be readily turned over to the manufacture of concentrated ammonia liquor should the sulphate market be unprofitable, but results in the formation of about 9,000 gals. of noxious waste liquor per ton of sulphate, which is often disposed of with difficulty. The steam consumption is about 9 lbs. per lb. of sulphate.

2. *Direct System.*—Gas is cooled to a temperature of 75°–80° C., that is, just above its dew-point, to avoid condensation of ammoniacal liquor. After tar extraction the whole of the gas is passed through a closed saturator for ammonia recovery. Heat of reaction between the ammonia and the acid causes so much evaporation in the saturator that, theoretically, the process uses no steam and makes no waste effluent. Actually a small amount of fixed liquor is condensed from the gas and must be distilled into the saturator, or else evaporated to give crude ammonium chloride. This system followed the indirect system, but it is rather difficult to work for complete tar extraction and elimination of liquor condensation, and is not now built into new plants.

3. *Semi-direct System.*—This embodies most of the advantages of previous systems, and is generally used in modern practice. Gas is cooled to 40°–50° C. with complete tar separation and partial liquor condensation. The remainder of the ammonia is recovered by passage of the gas through the saturator into which the previously condensed liquor is also distilled. Approximately 3–3½ lbs. of steam are required per lb. of sulphate, and the amount of waste liquor formed is only one-quarter of that of the indirect method.

Benzole Recovery.

Oil scrubbing of the gas at 20° C. is used to remove hydrocarbons of the benzene series, the benzolated oil being stripped by live steam to give a crude benzole before being cooled and returned to the scrubbers. The crude benzole, of which approximately 65 per cent. distils at 120° C., is redistilled to eliminate any wash oil, washed with concentrated sulphuric acid, neutralised and then fractionated into various products, motor benzole, pure benzole, toluole, xylene, naphtha, and so on, as required. Typical analyses of oils used for scrubbing, taken from the 1928 Report of the Research Committee of The National Benzole Association, are :—

Oil.	Absorption Capacity.	Viscosity (20° C. Redwood).	Sp. Gr. 20° C.	Drop Point.	Distillation Range.
Light creosote	4·8–4·5	34·6	1·013	200° C.	95% at 300° C.
Medium creosote	4·5–4·0	37·6	1·031	200° C.	„ „ 350° C.
Washed blast-furnace creosote	4·0	39·8	0·935	210° C.	„ „ 340° C.
Green oil (anthracene)	3·75	69·9	1·088	250° C.	„ „ 385° C.
Gas oil, American petroleum	3·0–2·8	40·0	0·864	210° C.	„ „ 363° C.

(The absorption capacity is the per cent. volume concentration of benzole in an oil solution in equilibrium with coal-gas containing 2·4 gals. of benzole per 12,000 cub. ft.)

A circulation of 80–120 gals. of oil through the scrubbers is required per ton of coal carbonised. Creosote oils are most generally used on account of their high absorption powers, but thicken quickly in use and lose this desirable property. They are also rather volatile, so that stripping

results in the loss of up to 6 gals. of oil per 100 gals. of benzole made. Blast-furnace creosote is used in Scotland and retains its fluidity well, but tends to give troublesome sludges in the oil coolers and heat exchangers. Gas oil, though of lower initial absorption power than creosote, is very satisfactory as it retains its fluidity, and on account of its low volatility loses little in stripping. Its low specific gravity makes it easily separable from water in the stripping stills and avoids troublesome emulsions. However, its paraffinoid nature causes contamination of the benzole. This is of little consequence in the manufacture of motor spirit, but it is undesirable when the products are to be used in chemical processes. Other methods tried for benzole recovery are absorption in absorbent charcoal or silica gel, but these have not yet proved commercially successful.

Coke-Oven Gas.

This is practically identical with coal-gas from gasworks' retorts, a typical analysis after stripping of tar and benzole being :—

CO_2. %	CnHm. %	O_2. %	CO. %	CH_2. %	H_2. %	N_2. %	Gross Calorific Value. (60° F. and 30 ins. pressure.)
1·1	2·4	Nil	5·2	26·2	58·4	6·2	524 B.Th.U.'s per cu. ft.

A surplus of roughly 60,000 to 65,000 million cu. ft. is made annually. Of this a small but increasing proportion is sold to gasworks for domestic supply, the amount in 1929 being 10,948,241,000 cu. ft.

In favourable circumstances the gas is used mixed with producer or blast-furnace gas in steel melting and reheating furnaces, but the largest use at present is for steam-raising and generation of electricity.

Low-Temperature Distillation of Coal.

This consists in the gradual heating of bituminous coal to a comparatively low maximum temperature (500° to 600° C.). The retorting vessels employed are designed in such a way as to ensure the rapid removal of the volatile products. By such a process a low yield of very rich gas is obtained, and a high yield of thin tar. The tar is richer in low boiling-point fractions than the tars produced by the gasworks or coke-oven plants. The residual coke contains approximately 10 per cent. of volatile matter, and may be marketed for use in domestic fires, for which purpose it is claimed to be easily ignitable, and at the same time to give off the major portion of its heat as radiant heat, and to burn without generating smoke. Until recently the process has not met with success, largely on account of mechanical difficulties in the operation of the retorts; these difficulties, however, appear to be now overcome by 'Low-temperature Carbonisation, Ltd.,' in their production of 'Coalite.'

Coalite.

The following results were obtained in a test by the Government Fuel Research Board * :

The coal used in the test was a good average Yorkshire coal, and one which has been frequently tested in every way at the Fuel Research Station.

The yield of products per ton of coal carbonised was as follows :—

'Coalite'	13·92 cwts.
Gas	5,620 cu. ft., or 39·6 therms, after stripping of motor spirit. (See †.)
'Coalite' crude oil	18·62 gals.
Ammonia liquor	26·00 gals.
Crude motor spirit from scrubbing the gas	1·78 gals.
Equivalent ammonium sulphate	13·55 lbs.

The 'Coalite' produced was of a very suitable size (1 to 3-in. pieces), containing only 4·6 per cent. of breeze, and was quite suitable to withstand the rough handling natural to rail or road transport. Analysis of this fuel showed a slightly lower percentage of volatile matter than usual, but when burnt in a household grate it readily ignited and gave a good hot fire.

The yield of crude oil was high, showing 68 per cent. of that obtained in the standard assay apparatus developed by the Fuel Research Board, and upon examination proved to be a normal low-temperature coal oil.

The yield of gas was greater than that anticipated from this particular coal, and averaged

$$\left(\frac{39{\cdot}6 \times 100{,}000}{5{,}620}\right)^{\dagger} = 705 \text{ B.Th.U. per cubic foot after being stripped of its crude motor spirit.}$$

The motor spirit obtained by scrubbing the gas was 1·78 gals per ton, but as the crude oil also contained 1·09 gals., the total yield of this product amounted to 2·87 gals. per ton; a most satisfactory result.

The ammonia was also higher than expected, and, when converted into ammonium sulphate, yielded 13·55 lbs.

* Report published by H.M. Stationery Office, Kingsway, London, W.C. 2, price 9*d*.

YIELD FROM ONE TON OF AVERAGE COAL (say 25-30 per cent. volatile matter).

	Low-temperature Carbonisation.	High-temperature Carbonisation.	
		Gasworks.	Coke Ovens.
Temperature of carbonisation	1,000° F.	About 1,800° F.	About 1,800° F.
Gas	6,000-6,500 cubic feet of rich gas, 700-750 B.Th.U. per cubic foot.	12,000 cubic feet of medium quality (town's) gas, 550 B.Th.U. per cubic foot.	11,500 cubic feet of poor quality coke-oven gas, 450 B.Th.U. per cubic foot.
Liquid	20 gallons coalite oil (fractionated to, say, 3 galls. motor spirit, 8-9 galls. Diesel or fuel oil, and 8-9 galls. lubricating oil).	10 gallons coal tar. (Fractionated to about half-gallon motor spirit, carbolic acid, a large amount of naphthalene, fuel oil, &c.)	8 gallons coal tar.
Sulphate of ammonia (from the ammonia)	15 lbs.	25 lbs.	28 lbs.
Residue in retort	14 cwt. of smokeless fuel coalite containing about 9-10 per cent. volatile matter.	13½ cwt. of soft coke, containing about 1 per cent. volatile matter.	14-14½ cwt. of hard coke, containing less than ½ per cent. volatile matter.

(*David Brownlie, A.I.Mech.E.*)

From one ton (American, 2,000 lbs.) of bituminous coal, the Colorado Fuel and Iron Company of Denver, U.S.A., obtains in actual practice the following products, viz.:

Coke	1,400 lbs.	Pure benzol	1¼ gals.
Coke breeze	60 lbs.	Toluol	¼ gal.
Gas	7,500 cu. ft.	Zylol	2/10 gal.
Ammonium sulphate	25 lbs.	Pitch	5 lbs.
Tar	10 gals.	Naphthalene	½ lb.
Light oil	3½ gals.	Creosote	1 qt.
Motor benzol	2¼ gals.	Paint	1 qt.

[Annual capacity of ovens, 540,000 tons (American) coke.]

Coke as a Boiler Fuel.

(*Contributed by E. W. L. Nicol, M.I.Mar.E., A.M.I.E.E.*)

With a view to cheapening the cost of steam raising by improving steam-boiler efficiency, and to eliminate the emission of visible smoke, gas coke is now largely used as fuel in lieu of crude coal. In hand-fired boilers, coke exclusively is used under ordinary natural draught conditions. With a draught of 0·25 in. W.G. over the fire, the average rate of combustion is 12 lbs. to 14 lbs. of suitably graded coke per sq. ft. of grate area per hour. Some form of impelled draught is necessary if this rate is insufficient. The forced draught apparatus usually employed is some form of steam-jet blower applied to the enclosed ashpit of the boiler, and used to maintain a 'balanced' draught over the fire (fig. 8). The steam consumption of the jets should not exceed 3 per cent. of the total steam generated when burning graded coke at a rate of 20 lbs. per grate-foot-hour. For hand-fired boilers the minimum allowance of grate area should be 1 sq. ft. per 22 to 25 lbs. of coke consumed per hour. In practice it is found that this rate can be exceeded only at the expense of efficiency.

Unlike coal, the combustion of coke is practically completed on the grate, a supply of secondary air over the fire being unnecessary. This tends to localise the heating effect to the boiler furnace area. The introduction of a circulator, *b*, tends to neutralise this effect and set up active circulation of the water in the boiler. The operation of cleaning fires, and the losses incidental thereto, due to inrush of cold air and consequent loss of steam pressure, are minimised by the use of the Gallagher Crompton baffle bridge *a*, also shown in the figure. During the cleaning operation the live fuel is heaped up on the baffle bridge.

In mechanically-stoked boilers fitted with both forced and induced draught, average rates of combustion exceeding 40 lbs. of graded coke per grate-foot-hour are maintained. Types of

mechanical stokers specially adapted for use with coke or coke breeze include the 'Underfeed travelling grate' and the 'Babcock balanced draught chain grate.' Existing chain grate

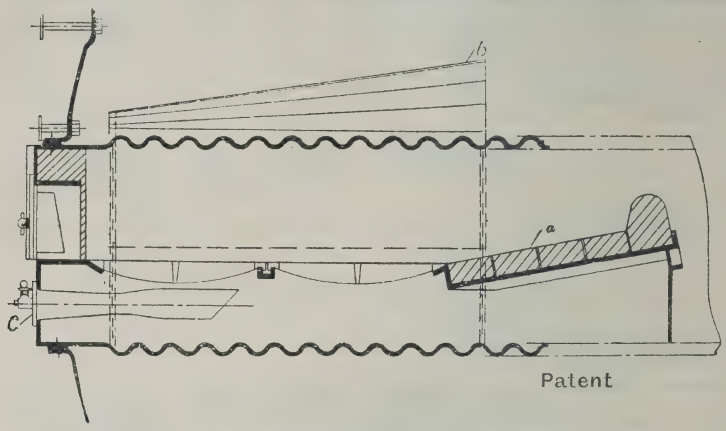


FIG. 8.—*a*, Gallagher Crompton baffle bridge; *b*, Ross Schofield circulator; *c*, Steam Jet forced draught.

mechanical stokers have largely been adapted to use admixtures of coke or breeze up to 75 per cent. at full normal rates under ordinary natural draught conditions, by means of the patent 'Sandwich' system (fig. 9), which is designed to feed, from a double hopper, well-defined layers

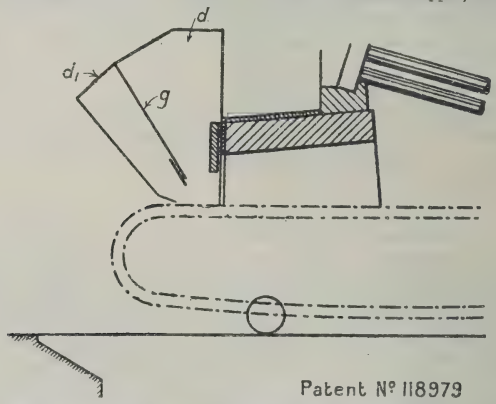


FIG. 9.—Sandwich System as applied to Chain-grate Stoker. d_1, coal hopper; *d*, coke hopper; *g*, separating diaphragm.

of coal slack and suitably graded coke of any desired thickness, the coal being uppermost. Using this system at the L.C.C. Greenwich Power Station, the General Manager gives the following data relating to the year 1919–1920 :—

Tons of fuel burnt	176,400	Average percentage of ash	10·0
Average price for coal	41*s.* 11·3*d.*	,, calorific value of breeze	9,000 B.Th.U.
,, calorific value	10,780 B.Th.U.	,, percentage of ash	18 to 20
,, percentage of ash	14·59	Coke and breeze consumed	25,000 tons
,, calorific value of coke	11,500 B.Th.U.		

With coal at 42*s.* per ton, and coke breeze at 22*s.* 6*d.* per ton, the fuel cost per 1,000 gallons evaporated was 19*s.* 6*d.* with equal quantities of each fuel, as against 23*s.* 2*d.* with Scotch nuts alone, a saving of about 16 per cent. The official report says: 'Apart from the saving in money, low grade classes of fuel can be dealt with on the Sandwich system at a much higher efficiency than would be the case if burnt separately. The results have been extremely satisfactory and a saving of about £8,000 a year can be effected.' Average evaporation obtained with 50 per cent. mixture of coal and breeze, 7·25 lbs.; with coal and coke 9 lbs. from and at 212° Fahr. Gas coke is also used extensively as fuel for road transport vehicles of both the under-type ('Sentinel,' 'Leyland,' and 'Atkinson') and the over-type wagon ('Foden' and 'Garrett,' etc.). The following table shows the fuel cost of evaporation from and at 212° Fahr. with common solid fuels at varying prices:—

APPROXIMATE FUEL COST OF EVAPORATION FROM AND AT 212° FAHR.

(*Compiled by E. W. L. Nicol, M.I.Mar.E., A.M.I.E.E., Fuel Expert to the London Coke Committee.*)

Coal Slack.			Hard Steam Coal.			Smokeless Welsh Coal.			Gas Coke.		
Mechanically Stoked.			Hand Fired: Natural Draught.			Hand Fired: Natural Draught.			Hand Fired: Forced Draught.		
At per Ton.		Evaporative Value. 7·5 lbs.	At per Ton.		Evaporative Value. 9·5 lbs.	At per Ton.		Evaporative Value. 10·5 lbs.	At per Ton.		Evaporative Value. 9 lbs. net.
		per 1,000 gallons evaporated.			per 1,000 gallons evaporated.			per 1,000 gallons evaporated.			per 1,000 gallons evaporated.
s.	*d.*	*s.* *d.*	*s.*	*d.*	*s.* *d.*	*s.*	*d.*	*s.* *d.*	*s.*	*d.*	*s.* *d.*
19	0	11 4	26	0	12 2	28	0	11 10	25	0	12 4
20	0	11 11	27	0	12 8	29	0	12 4	26	0	12 10
21	0	12 6	28	0	13 1	30	0	12 9	27	0	13 5
22	0	13 1	29	0	13 7	31	0	13 2	28	0	13 11
23	0	13 8	30	0	14 0	32	0	13 7	29	0	14 5
24	0	14 4	31	0	14 6	33	0	14 0	30	0	14 10
25	0	14 11	32	0	15 0	34	0	14 5	31	0	15 4
26	0	15 6	33	0	15 6	35	0	14 10	32	0	15 10
27	0	16 1	34	0	16 0	36	0	15 4	33	0	16 4
28	0	16 8	35	0	16 5	37	0	15 9	34	0	16 10
29	0	17 4	36	0	16 11	38	0	16 1	35	0	17 4
30	0	17 11	37	0	17 5	39	0	16 6	36	0	17 10
31	0	18 6	38	0	17 10	40	0	16 11	37	0	18 4
32	0	19 1	39	0	18 4	41	0	17 4	38	0	18 10
33	0	19 8	40	0	18 9	42	0	17 9	39	0	19 5
34	0	20 4	41	0	19 2	43	0	18 2	40	0	19 11
35	0	20 11	42	0	19 8	44	0	18 7	41	0	20 5
36	0	21 6	43	0	20 2	45	0	19 0	42	0	20 10
37	0	22 1	44	0	20 8	46	0	19 5	43	0	21 4
38	0	22 8	45	0	21 1	47	0	19 10	44	0	21 10
39	0	23 4	46	0	21 7	48	0	20 4	45	0	22 4
40	0	23 11	47	0	22 0	49	0	20 9	46	0	22 10
41	0	24 6	48	0	22 6	50	0	21 2	47	0	23 4
42	0	25 1	49	0	23 0	51	0	27 7	48	0	23 10

Anthracite.

(*Contributed by A. Leonard Summers, Author of 'The Anthracite Industry,' &c.*)

The bulk of the anthracite produced in Great Britain is mined in the western portion of the South Wales coalfield, the best qualities in Carmarthenshire and Glamorganshire. It consists virtually of carbon—as much as 94 per cent. in the better qualities—with only a small proportion of the constituents of bitumen. It is jet-black, with a metallic lustre, very hard, slow burning, absolutely smokeless, gives intense heat, and burns with a bright red glow. These qualities fit it for many purposes. It is used largely for smelting; in the burning of lime and brick; for the generation of steam; and in the United States and on the Continent of Europe extensively for domestic heating and cooking. For malting and horticultural purposes, owing to its purity and

freedom from arsenic, etc., it is in immense demand; while in Norway, Sweden, and Italy, Welsh anthracite is used for making carbide of calcium.

The American coalfield (Pennsylvania) produces the largest amount of anthracite—about 100 million tons per annum—the Welsh output being only about 5,000,000 tons a year at present, though capable of a much greater yield, the major part of the field being still untouched.

One ton of anthracite used in a pressure or suction-gas plant will give power equal to 10 tons of the best steam coal used under boilers. Some typical figures for a suction-gas plant using anthracite were produced by a test carried out at the Metropolitan Reservoirs, Chingford, under the direction of their chief engineer. A guarantee had been given that the consumption would not exceed 1·1 lb. of anthracite per actual b.h.p. hour, and in the report it was stated that the average fuel consumption for the four main pumps was 0·913 lb. per b.h.p. hour, whilst the small pump gave the figure of 0·8 lb. of anthracite. The general average for the whole of the pumps was 0·904 lb. of anthracite per actual b.h.p. hour, and the guarantee lift of the pumps was exceeded by almost 20 per cent. The producer tests showed that 1 lb. of anthracite yields 82·1 cu. ft. of gas at standard temperature and pressure, the net calorific value being 146·3 B.Th.U. per cu. ft., giving a yield of 12,011 B.Th.U. per lb. of anthracite. The calorific value of the anthracite supplied was 14,600 B.Th.U. The fuel used under the boilers which supplied steam for the producer was taken into account.

A ton of Welsh anthracite will produce from 170,000 to 220,000 cu. ft. of gas for power purposes. The following is a typical analysis of good Welsh anthracite:

Carbon	92·73	per cent.
Hydrogen	3·37	,,
Oxygen and nitrogen	2·69	,,
Sulphur	0·45	,,
Ash	0·76	,,
	100·00	,,

In furnace installations where crude gas can be used bituminous fuels are generally used, as the gas is supplied direct to the furnace along with the tar and soot. In small furnaces, where small burners are used, or where necessary to convey gas a long distance, it is generally not possible to use crude gas, and anthracite is more convenient than bituminous coal. The average consumption in a pressure gas-producer is about 13 lbs. of anthracite per 1,000 cu. ft. of gas, and at normal prices this would work out between 2*d.* and 3*d.* per 1,000 cu. ft. of gas, and the overall consumption for producer and engine should be about 1 lb. of anthracite per b.h.p. hour.

The principal anthracite collieries are equipped with electrical and other machinery, the preparation of the coal being very elaborate—breaking, screening, washing, etc. A revolving breaker deals with the large coal; this coal travels to the various 'screens' for sizing. These sizing screens are placed one above another at an angle and worked on a shaking principle. Thus the coal is kept steadily on the move to obviate congestion. The large collieries break and screen their product into at least eleven different sizes. The screens are perforated with round holes, through or over which the coal passes in sizes from 2½ ins. by 4 ins. (the largest cobbles) to ¼ in. by ¼ in. ('grains'). Similar moving belts, but not perforated, convey the screened coal to the pickers, who remove by hand whatever shale it may contain. Impurities are picked from the cobbles and large coal prior to the coal going to the breaker. These picking belts also load the coal into the railway trucks, but only convey the larger sizes, viz. machine-made cobbles and machine-made 'French' nuts. Sizes below the French nut cannot be conveniently picked and sorted by hand, and are therefore transferred to a washer. The 'washery' belts work with the same shaking movement, carrying the small coals, such as stove nuts, peas, beans, pea-nuts, etc., through tanks of water to remove the small particles of stone. To ensure the maintenance of a maximum standard of quality, a coal examiner usually twice daily takes and tests samples of coal from the screens and washery, and the percentage of waste material detected rarely exceeds 1 per cent.

What is known in the trade as 'rubbly culm' is the rough small coal which passes between longitudinal bars 1¼ in. apart. It is used principally for lime-burning and steam-raising.

The 'duff,' or very fine, small anthracite, is used in the manufacture of patent fuel and cement making.

In regard to ordinary boiler furnace utilisation of anthracite not much advance has been made, due to the narrow conception of boiler-house practice, coupled with the question of availability. When anthracite is used for this purpose it must not be disturbed, and should be stoked with a

thin bed, very evenly, and if this minimum attention is given an evaporation of from 8–10 lbs. of water per hour can be easily maintained. Tests made in London in which anthracite peas were used showed an evaporation of 12·2 lbs. of water per lb. of coal in one test, and 11·3 lbs. in another. Sometimes the draught or other conditions render the use of anthracite alone impossible, but its blending with other fuels gives very satisfactory results. Where mechanical stoking is used a larger proportion of anthracite—say 2 to 1—can be used, as distinct from equal quantities where hand-firing is adopted.

An important aspect of anthracite has been the recent utilisation of the large accumulations of duff and washery refuse. Hitherto, anthracite dust could not be satisfactorily burnt on standard furnaces, and considering that this fuel has a calorific value of 12,000 to 13,000, with over 80 per cent. fixed carbon, and about 10 to 12 per cent. of ash, it was obvious that valuable material was lying dormant in the dumps. Therefore a new furnace was introduced, with air spaces subdivided into a large number of small holes, and other modifications. The following results of tests at a Welsh colliery show the great increase made in evaporation :

Test No.	Class of Fuel.	Evaporation in Lbs. Steam per Hour.	Remarks.
1	Duff, good quality . .	3,500	Jet pressure 60 lbs.
2	'Grains' . . .	4,400	,, ,, 30 ,,
3	Duff	3 125	,, ,, 45 ,,
4	Duff	3,000	,, ,, 25–30 ,,
5	'Grains' . . .	5,750	,, ,, 30 ,,
6	Duff	4,250	,, ,, 40–50 ,,
7	Duff	4,875	,, ,, 55–85 ,,

The first four tests were carried out on the old type, and tests 5, 6, and 7 on the newly designed bar-furnace.

Burnt under proper conditions, there is no better coal for steam-raising than Welsh anthracite. At one works using English coal it was necessary to employ two Lancashire boilers 30 ft. by 7 ft. 6 in., but after a patent furnace was fitted and anthracite coal introduced, *one* of these boilers was sufficient to do the work previously done by the two boilers.

Below are the standard sizes of Welsh anthracite :—

Cobbles	$2\frac{1}{2}'' \times 4''$	'Beans'	$\frac{1}{2}'' \times \frac{3}{4}''$
'French' Nuts . .	$1\frac{3}{4}'' \times 2\frac{1}{2}''$	'Peas'	$\frac{1}{4}'' \times \frac{3}{4}''$
Stove nuts . . .	$1'' \times 1\frac{3}{4}''$	'Grains'	$\frac{3}{8}'' \times \frac{1}{4}''$
'Pea-nuts' . . .	$\frac{1}{2}'' \times 1\frac{1}{4}''$		

Lignite.

The composition of lignite, and its properties, lie between those of wood and coal.

Lignite occurs in most European countries and is abundantly found in the Colonies, but does not occur in quantities in the British Isles. Lignites vary widely in composition, but generally between the following limits : carbon 55 to 80 per cent., hydrogen 5 to 7 per cent., oxygen 10 to 35 per cent. (in the dry state). The calorific value is from 9,000 to 13,000 B.Th.U. on dry sample. The high ash and water contents of lignite frequently make it unsuitable for direct combustion, but good grades may be used for steam-raising. Lignite is frequently subjected to destructive distillation, whereby valuable liquid products are obtained. It may also be used for the manufacture of fuel briquettes, or be utilised in gas producers.

Peat.

Large deposits of peat occur within the British Isles, but little use has been made of them hitherto, the high water content (80 to 96 per cent. when in bog, and 30 to 60 per cent. after air drying) being a great drawback to their utilisation. Peat is suitable for use in gas producers and is sometimes carbonised, when peat charcoal is the main fuel product.

Anderson and Dillner (*Jour. Soc. Chem. Ind.*, 1902, 459) give the following calorific powers for peat (in the dry state) :—

Average from bog	8,100 to 11,000* B.Th.U.
Reed grass peat	7,450 to 9,820 „
Mud peat	7,825 to 8,200 „

Wood.

When freshly cut, wood contains up to 50 per cent. of water, but this can be reduced to about 20 per cent. by air drying. The composition of dry wood is about 49·5 per cent. carbon, 6 per cent. hydrogen, 44 per cent. oxygen, and ·5 per cent. ash, and the heat value is generally a little below 9,000 B.Th.U. It only finds use in countries where large forests exist and where coal is not readily obtainable.

Wood Charcoal.

Wood charcoal consists mainly of carbon and possesses a calorific value of 11,000 to 13,500 B.Th.U. (Brame). Its use as a fuel is very limited.

Briquette Fuel.

(*Contributed by Arthur Grounds, B.Sc., A.I.C., Assoc. M.I.Min.E., Consulting Chemist.*)

Briquettes can be used on hand-fired boilers, without the necessity of installing special stokers to deal with the fuel. The composition of the fuel can be varied from an anthracitic fuel, suitable for consumption in domestic grates, and giving a high degree of radiant heat, to a briquette of higher volatile matter, suitable for industrial boiler firing, and giving the long flame necessary for the heating of Lancashire, Cornish, and water-tube boilers. The shape, size, and weight of the briquettes must be suitable for the purpose for which they are to be used, and for the nature and duration of the transport and storage to which they will be subjected. Their degree of cohesion should be at least equal to that of good coal, and the loss by attrition and crumbling during ordinary handling and transport should not exceed 5 per cent. by weight. Their mean specific gravity should not be below 1·24, whilst the ash content of the dry fuel should not exceed 12 per cent., and is preferably below 10 per cent. They should not contain over 5 per cent. of moisture, nor should they absorb water, whilst they should not be affected in physical character by immersion in water over a long period. Their calorific value should be at least equal to that of good coal.

Artificial fuel is made in many sizes and shapes, from the 'boulet' of 30–40 grms. weight (1–1½ ozs.) to the large block of 28 lbs. weight. The latter block is chiefly made for export to the West Indies and Egypt, and to other tropical and sub-tropical countries, as a weight of 28 lbs. corresponds roughly to that which a native labourer or coolie can carry. Between these two sizes of fuel there are the normal ovoid, about the size of a hen's egg, weighing 70–100 grms., and the larger ovoid of 200–250 grms., which has lately been developed. These ovoids are chiefly used for domestic fuel, and are largely anthracitic, whilst for steam raising the block form is most popular, the common size being the 10-lb. block, measuring 10 ins. by 5 ins. by 4 ins. Ovoid fuel has lately been made in the 'boulet' size for use in suction gas producers, and for use with automatic stokers, in place of screened nuts. The blocks are made in various shapes, either as bricks with rounded corners, quadratic flattened cube shape, or grooved in order to facilitate their breaking up in the stokehold. They may be marked with the makers' name, or with some distinguishing mark, or may be quite plain.

Domestic ovoids may be made from anthracite and a binder alone, although this has the disadvantage that, as pitch is the binder generally used, when the fuel is placed on the fire, the pitch burns off very easily at first, and then the anthracite, being a non-caking coal, falls to powder in the fire, and the result is a dull, black-looking mass. To avoid this, these ovoids are generally made from a mixture of about 80 per cent. of anthracite duff, with 10 per cent. of a good coking coal and 10 per cent. of pitch.

When placed on the fire, the pitch at first burns off and the heat of combustion leads to the carbonisation of the bituminous coking coal, which cakes and acts as an agglutinant, binding the non-caking anthracite together, and this burns very slowly and gives out a large amount of radiant heat per unit weight. This is the cheapest form of artificial fuel, as anthracite duff is the cheapest form of raw material and forms 80 per cent. of the total weight of the fuel. The binder generally used for the process is coke-oven or gas-works pitch, the grade used being known in the trade as 'medium-soft coal-tar pitch, suitable for the manufacture of briquettes.' Blast-furnace pitch is used to a certain extent, but it is inferior to coke-oven or gas-works pitch in binding power, and contains a much higher percentage of ash, and, on carbonisation, it gives

* Exceptionally high.

a swollen, instead of a sintered coke. The ash in a good pitch should not exceed 0·5 per cent., whilst that in blast-furnace pitch varies from 5 per cent. to 25 per cent., or even more, according to the dust-catching arrangements at the blast-furnace from which it has been obtained. The volatile organic matter of the pitch is determined by heating 1 grm. of the pitch in a platinum or porcelain crucible over a bunsen flame for seven minutes, the flame being 18 cms. high, and the crucible being held on a triangle of fine wire, about 7 cms. above the mouth of the burner. The crucible is provided with a lid, and is weighed after heating for seven minutes, with subsequent cooling, the loss in weight giving the volatile matter. The ash can be determined by igniting the coke obtained as above until all the combustible matter has been burnt off. The residual coke from the volatile matter estimation should be in the form of a flat button. If the button is not flat, but conical in shape or puffed up, the pitch will not be suitable for the purpose. The twisting point is measured by immersing a bar of pitch, 2 ins. by $\frac{1}{2}$ in. cross section, in water at gradually increasing temperatures for a period of two minutes. The pitch is then withdrawn and a very slight pressure exerted with the fingers in the attempt to twist it round. If the pitch is still too hard, the water is heated again until it has risen in temperature another 5° C., and the pitch is again immersed for two minutes, and then tested to see if it will twist. When this is possible, the temperature of the water is taken, and this temperature is known as the 'twisting point.'

A suitable pitch for briquette manufacture should have a twisting point of 50–60° C. The melting point is determined by immersing a cube of pitch, of $\frac{3}{4}$-in. side, stuck on a No. 12 copper wire, bent horizontally, in a beaker of water. The water is gradually heated so that a rise in temperature of 5° C. or 9° F. per minute is obtained. When the cube of pitch softens and drops to the bottom of the beaker, the temperature is noted, and this is taken as the melting point. For a coke-oven or gas-works pitch of suitable quality, the melting point is usually about 85°–100° C.

The solubility in carbon disulphide is determined by weighing out 0·5 grm. of the pitch into a flask and adding 200 c.c. of carbon disulphide. The flask is heated on a water bath or an electric heater, under a reflux condenser, for an hour, and is then removed and cooled. The free carbon is filtered off through two filter papers, and is well washed with carbon disulphide, the washings being added to the filtrate. The carbon disulphide is then distilled off from the filtrate, which has been caught in a weighed flask, when the increase in weight of the flask gives the amount of matter soluble in carbon disulphide. This figure is a useful indication of the binding power of the pitch, as those pitches which give high solubility figures are generally found to be excellent binders. This figure should be in the region of 66–84 per cent. The difference between the percentage of matter soluble in carbon disulphide and 100 is the 'free carbon.'

Other tests are necessary for special purposes, but those cited are the usual tests adopted. A typical analysis of a suitable pitch would be: Volatile organic matter, 70·4 per cent.; coke, 29·6 per cent.; ash, 0·3 per cent.; fixed carbon, 29·3 per cent.; soluble in carbon disulphide, 75·5 per cent.; free carbon, 24·5 per cent. Twisting point, 56·5° C.; melting point, 97·0° C.

The amount of binder necessary varies with the coals being used, but, generally speaking, it increases with the hardness of the coals being briquetted. Thus, about 10–11 per cent. is necessary for hard anthracites, about 8·5–9 per cent. for steam coals, and 6·5–8·5 per cent. for bituminous coals, according to their volatile matter content. The hardness of the coal is roughly proportional to the volatile matter content of the coal, and the more bituminous coals, with a volatile matter content of 33–40 per cent., can be briquetted fairly easily by fine grinding (97 per cent. of the material should pass a 200-mesh screen) and pressing, taking care that the air is extracted from the fine coal before pressing. The pressure necessary for the production of briquettes made in this way is about 10 tons per sq. in. The cohesion is then about equal to that of a pitch-bound briquette. The fuel is said to stand handling well and to be weather-proof. The fuel can also be made from all coals possessing a volatile matter content of 11–12 per cent. and over, but the more bituminous coals already described are the most suitable for this purpose. Anthracite cannot be briquetted in this way. The great advantage of this process is the elimination of the binder, as the cost of pitch is a very serious item in the fuel-works balance sheet. For example, at the end of 1920, the cost of pitch had risen to £12 per ton, and assuming the pitch content of fuel to be 10 per cent., the cost of *pitch alone* in 1 ton of the fuel would be £1 4*s*. In the manufacture of the usual briquettes, with pitch as a binder, creosote and anthracene oil fractions are frequently used for the purpose of obtaining the optimum binding power of the pitch. The oil is mixed with the pitch in the pug mill in which the materials are mixed thoroughly, and the oil dissolves the pitch, which then flows more evenly over the grains of coal and coats them evenly without undue waste. By this means it is possible to cut down the pitch consumption by about 10–15 per cent., a very considerable amount when cost is taken into account.

As already mentioned, anthracitic fuel would not be suitable for steam-raising, and the coals most generally used for the manufacture of fuel intended for this purpose are blends of bituminous, anthracite, and steam coals. The following table shows the analysis of these coals, from which it will be seen that the steam coals contain more volatile matter than anthracite, whilst the bituminous coals contain still more volatile matter than the steam coals.

Fuel.	Moisture.*	On Dry Coal.			
		Volatile Organic Matter.	Coke.	Fixed Carbon.	Ash.
	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.
Anthracite . . .	0·2– 3·5	5·0–10·0	90·0–95·0	83·0–92·0	3–7
Dry steam coals . .	1·0– 4·0	10·0–13·0	87·0–90·0	80·0–85·0	5–7
True steam coals . .	2·5– 5·0	13·0–16·5	83·5–87·0	76·5–82·0	5–7
Semi-bituminous coals .	4·0– 6·0	17·0–20·0	80·0–83·0	71·0–77·0	6–9
True bituminous coking coals . . .	4·0–10·0	20·0–28·0	72·0–80·0	60·0–74·0	6–12
Caking coals . . .	4·0–10·0	28·0–40·0	60·0–72·0	48·0–66·0	6–12
Lignites, or brown coals	10·0–40·0	40·0–55·0	45·0–60·0	Varies	Varies

A good mixture for the manufacture of briquettes for steam-raising would consist of 50 per cent. steam coal, 20 per cent. bituminous coal, and 20 per cent. of anthracite duff with 10 per cent. of pitch as a binder. This would give a finished fuel of approximately the following composition :—Volatile organic matter, 21·0 per cent.; coke, 79·0 per cent.; fixed carbon, 73·0 per cent.; ash, 6·0 per cent. This is assuming that clean coals had been used in its manufacture, and such coals could be produced by washing the small coals in one or other of the washers on the market, snch as the Draper, Hoyle, British Baum, Notanos, or other plant.

Briquettes can easily be prepared to specification by varying the percentages of the constituent coals used for any particular purpose. Thus, if a briquette is desired for furnace or other purposes which shall give a long flame, this can be obtained by putting in an extra large percentage of bituminous coal, whilst we have seen that for domestic fuel anthracite is introduced in order to make the fuel burn slowly and to give radiant heat, instead of throwing a large number of B.Th.U.'s up the chimney in the form of volatile constituents. Resin is sometimes used as a binder, in addition to pitch, and this substance has, roughly, twice the binding power of medium soft pitch, but briquettes made with resin alone fall and soften in the fire and are therefore unsuitable.

A large number of inorganic binders have been tried with varying degrees of success, but these are objectionable on account of the fact that they introduce still more ash into the fuel than is due to the coal alone. Such materials are Portland cement, silicate of soda, lime, clay, spent tanning materials, and sulphite cellulose liquor. Coal tar has also been used, but the resultant briquette is too weak and softens in tropical countries. Various organic materials have also been tried, such as molasses and sugar residues, jelly from Irish moss, mucilage, waste starch, etc., but the only satisfactory binder seems to be pitch.

In the manufacture of blocks, the coal is generally crushed in a disintegrator of the squirrel-cage type, whilst the pitch is broken up in a pitch-cracker, falling on to a grating provided with longitudinal slots, through which revolving arms protrude, which arms carry the pitch forward and crush it against the breaker plate. The mixture of coal and pitch is sometimes subjected to a crushing either in a second disintegrator or in a mill of the Jeffrey swing-hammer, or fixed beater type, such as the Alexander Pulvergrinder, made by Messrs. Briquette Machinery Ltd., of Leeds. The coal should not be too finely ground, as a large amount of fine coal leads to the breaking down of the binding power of the pitch. The pitch, however, should be ground to as fine a degree as possible, so as to obtain the maximum spreading effect with the least possible quantity. The coal and pitch mixture is elevated into a pug, or cooking chamber, above the press, where it is thoroughly admixed, steam being blown in at the same time. Using steam at 400° F., the steam consumption for a press turning out 17–20 tons per hour of briquettes is about 2,200 lbs. Such a press absorbs about 45–50 B.H.P., and runs at 30–40 strokes per minute. The pressure necessary for briquetting steam and bituminous coal mixtures with 9–10 per cent. pitch is about 30–35 cwt. per sq. in. Ovoid fuel is made in roll presses, the fuel mixture being fed between the rolls, after cooking, at a temperature just above that of the twisting point of the pitch used, *i.e.*, about 60–70° C., with a normal pitch. The ovoids are caught on a shaker screen as they emerge from the press, so that all fines are removed, and are then loaded direct into either wagons or ships, provision being made for cooling, to prevent sticking of the fuel, and possible fusion into a solid mass.

* The moisture varies according to the amount of exposure to which the coal has been subjected and to the degree of humidity of the atmosphere, but the above are given as the average of a large number of results. The ash content in general increases with increase of volatile matter content.

In some of the processes recently devised for the purpose of low-temperature carbonisation, the low-temperature coke is fed direct from the retort to a press and is briquetted without the addition of any further binder.

The Smith continuous system of carbonisation, which produces 'Carbocoal,' is a combination of low- and high-temperature carbonisation with briquetting. The coal, sized to pass a $\frac{3}{8}$-in. mesh screen, is subjected to carbonisation at 480° C., the resulting product being a friable residue called semi-carbocoal, which contains 7–10 per cent. of volatile matter. This process takes some two to three hours for completion, but is continuous. The semi-carbocoal is reground, mixed with pitch, and briquetted. The briquettes contain about 15 per cent. volatile matter, and these are carbonised at 1,000–1,100° C., the residue being a hard coke containing 3–4 per cent. volatile matter. The tar yield from the primary carbonisation contains the products usually obtained from low-temperature processes, *i.e.*, paraffin bodies predominate, whilst the tar yield from the high-temperature process gives heavy oils and pitch, with an increased yield of ammonium sulphate. The gas yield from the first stage is 5,000–6,700 cub. ft. per ton, with a calorific value of 650–700 B.Th.U.'s. per cub. ft. whilst from the second stage there are obtained 4,480 cub. ft. of gas of calorific value 350–400 B.Th.U.'s. per cub. ft. The average yield is therefore about 10,630 cub. ft. of gas of calorific value 584 B.Th.U.'s per cub. ft.

Other processes merely briquette the low-temperature coke whilst hot, and either recarbonise at high temperature or sell the product as a smokeless fuel for domestic purposes.

A great amount of work has been done of late in Canada by the Lignite Utilisation Board on the briquetting of lignite. The Souris lignites are of very low grade in the raw state, but by carbonising at 600° C. a residue is obtained which resembles charcoal and which produces an excellent briquette on binding with pitch and pressing in a roll press. The size of the lignite used is under 1/10-in. mesh, 60 per cent. of the material sizing between 1/10 and 1/20-in. mesh. The binders used are coal-tar and petroleum pitches, with sulphite liquor and various waste flour residues as subsidiary bodies. The quantity of binder required, however, is much higher than that which is necessary for the production of ovoids from anthracite fines, and this would seem to be a critical factor where the cost of pitch is high. It was found that 13 per cent. of coal-tar pitch was necessary to produce a sound fuel.

The strength of fuel is generally measured in an empirical way by the 'cohesion test.' This is carried out by revolving 100 lbs. of the fuel in a horizontal cylindrical drum, 3 ft. long and 2 ft. 6 ins. diameter, provided with three internal ribs of angle iron 6 ins. deep, for two and a half minutes at 28 revolutions per minute. The fuel is sawn out of the solid block into pieces 1 lb. in weight, and after the test is completed the residue in the drum is screened over a sieve of $1\frac{1}{2}$-in. mesh, and the portion remaining on the sieve is weighed. This weight, in lbs., is called the 'cohesion' value of the fuel. The revolution of the drum is supposed to represent the rolling of the ship, the attrition in the drum corresponding to the attrition in the ship due to projecting girders, baulks, etc. The cohesion value should not be less than 55 per cent., and a good fuel will often average over a long period 72–82 per cent.

The chemical tests on fuel are the same as are carried out on coal, and these will be found in any book on the evaluation of coal, such as 'Fuel,' by Coste and Andrews, or 'The Analysis of Coal and its By-Products,' by S. R. Illingworth.

Firebars.

The following proportions are for the usual cast-iron bars :—

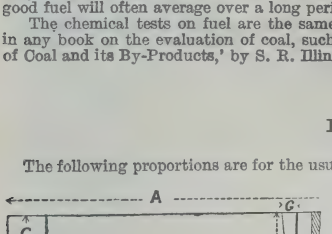


FIG. 10.

A.	B.	C.	D.	E.	F.	G.	H.
Ft. Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
2 0	$3\frac{1}{4}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{3}{8}$	$\frac{3}{8}$
2 6	$3\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{3}{8}$	$\frac{3}{8}$
3 0	$3\frac{5}{8}$	$1\frac{3}{4}$	2	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{3}{8}$	$\frac{3}{8}$
3 6	$3\frac{3}{8}$	2	2	$1\frac{1}{4}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{1}{2}$
4 0	$4\frac{1}{4}$	2	$2\frac{1}{4}$	$1\frac{1}{4}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{1}{2}$
4 3	$4\frac{3}{4}$	2	$2\frac{1}{4}$	$1\frac{1}{4}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{1}{2}$
4 6	5	$2\frac{1}{4}$	$2\frac{1}{4}$	$1\frac{1}{4}$	1	$\frac{3}{8}$	$\frac{1}{2}$
4 9	$5\frac{1}{4}$	$2\frac{1}{4}$	$2\frac{1}{2}$	$1\frac{1}{4}$	1	$\frac{3}{8}$	$\frac{1}{2}$
5 0	$5\frac{1}{4}$	$2\frac{1}{4}$	$2\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$\frac{3}{8}$	$\frac{1}{2}$

Firebars require to be suited to the fuel. Clean riddled coal-nuts or cobbles may be burned on a $\frac{1}{4}$-in. bar with $\frac{1}{4}$ in. to $\frac{1}{2}$ in. spaces, but slack coal requires wider bars and narrow spaces. Bars are often made with projecting fingers or corrugations in plan, thus increasing the length of air space considerably, though reducing it in respect of the increased width of the bar body. Such bars are made to rock for the purpose of shaking the ashes out of the fire and of breaking up clinker. Air, to supply a secondary demand behind the bridge for the combustion of the gases, is often admitted through hollow bars which heat the air and keep themselves cooler thereby, and under suitable conditions good results may be obtained.

Some firegrates are built up of short bars or links pinned together to form a side travelling band or chain; such grates are self-cleaning and are much used in mechanical stokers, and are good, if provided with suitable air control damper plates below. Other firegrates have a somewhat steeply inclined surface, and may be built up either of longitudinal bars or of transverse bars, in which latter case they form the so-called 'step' grate. Such grates may have a creeping or a shaking movement of the bars in order to assist gravity to lower the burning fuel down hill to make up at the foot of the slope the vacant space left as the fuel burns away. The fuel is mechanically fed in, above, or at the head of the sloping grate surface, and gradually descends under the combined influence of gravity and the moving bars. A dumping device is often added at the foot of the slope by means of which the ash and refuse may be cleared from the furnace by movement of a lever.

Grates for anthracite coal, which develops an intense heat in the fire—not sending off much gas by the heat-absorbing process of distillation—are often made in the form of flat plates with a number of perforated nipples or little mounds raised half an inch above the general surface. The space between the air mounds fills level with dust and preserves the iron surface from heat and checks the adherence of clinker.

Firegrate bearers must never be rested upon the actual plates of furnace tubes of internally fired boilers, but must rest on angle bracket pieces riveted to the plates by not less than two rivets each. For convenience in handling it is usual to employ two or three bars in the length of a 6-ft. grate.

Hollow bars are necessarily (or more conveniently) made in one length; this, however, makes them cumbersome, and is one cause of their being sometimes unpopular, for they are heavy to remove and replace when cleaning; but where conditions are favourable, hollow bars with outlets to the surface and a steam-impelled supply of air through the bars will beget economy, by enabling a cheaper grade of fuel to be employed.

Coal Combustion and Smoke.

Bituminous coal cannot be burned smokelessly except under certain proper conditions. These are that the whole of the products of combustion must be allowed to travel over the full length of the fire towards the bridge; that additional air must be admitted in fine streams above the fire after each fresh charge of fuel. The fresh air thus admitted mingles with the gases from the fuel, and the mixture must be hot enough to burn at the bridge, beyond which point there must be a clear space in which combustion may complete itself. The combustion chamber may with advantage be wholly or partially brick-lined. The draught necessary should be not less than 30 ft. per sec. To compel air to enter over the fire, dampers are sometimes fitted to the ash pits.

The most satisfactory results in coal combustion are obtained with the thickest possible fires, but the thickness of a fire depends on the draught and on the size of the fuel. The largest fuel will bear the thickest fire and the strongest draught. The three factors are interdependent.

The combustion space should be at least in a good red heat so as to ensure burning all the combustible into carbon dioxide and water, which would give smokeless operation. Although about 14 lbs. of air is required to burn one pound of soft coal completely, only half this amount should be admitted through the grate; the remainder should be supplied over the fuel bed. If this can be well preheated, smokeless combustion is practically assured.

Leaving the door open a short interval after feeding fresh coal, supplies an extra quantity of this secondary air when most needed. Poor draft always causes smoke and lowered efficiency. Ample combustion space, involving high settings on boilers, is required with such fuels as Indiana bituminous, pulverised coal, or oil.

There are two general ways of firing bituminous coal. One is to place a thick bed of coal on the front grates. Here it will coke, and the gases given off will pass over incandescent fuel and be consumed, the coked fuel then being pushed over the remaining grate area. This is well adapted to return tubular boilers.

The second method is that of spreading the new coal evenly over the grate in small quantities at a time through alternate doors. One-half of the fuel bed is kept incandescent. A uniform size of coal, neither coarse nor fine, gives best results. High-volatile coals require a bed of four to six inches, and those with less volatile somewhat thicker beds.

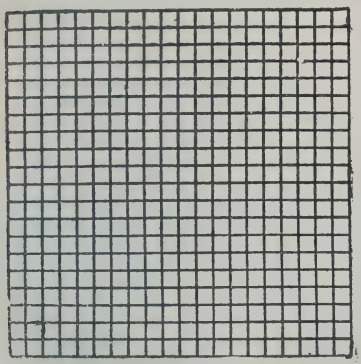
Where gases rise directly upwards from a grate amongst water tubes without the admixture by travel pointed out above, smoke is inevitable.

Fine dust fuel can now be burned at the rate of 40 lbs. per sq. ft. of furnace area per hour. Given furnace room, the cheapest fuel is usually the most economical. In practice the best economy is that due to the use of the cheapest fuel which can be employed to raise the steam necessary.

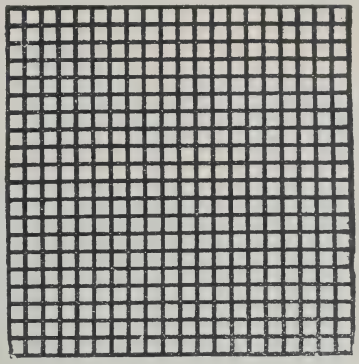
The foregoing, of course, is qualified by the amount of ash in the coal where the extreme of removal is great.

With externally fired boilers which permit of a liberal design of furnace with suitably disposed firebrick, smoke can readily be prevented. It is less easy to secure perfect combustion with internally fired boilers, because of the difficulty of applying durable firebrick linings. But the question of smoke prevention is wholly one of air mixture and temperature. The burning gases must not too soon be brought into contact with cold plates or tubes.

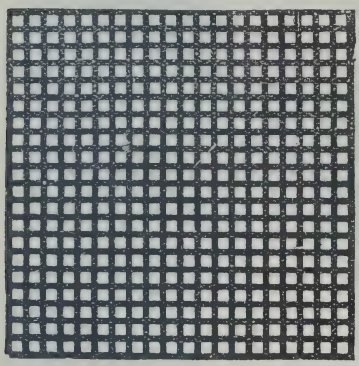
In burning bituminous coal only a part of the total heat is generated at the grate. Part of the heat of the fuel is absorbed in driving off the volatile parts of the coal in gaseous form. This heat becomes latent and very materially reduces the furnace temperature. The gases pass off, containing a third of the total heat capacity of the coal, as well as the heat of gasification. Unless mixed with air and burned later, these gases may be totally lost. To burn bituminous coal not only demands thorough air mixture, but also a sufficient temperature, and the heat of the burning gases must not be abstracted by too early bringing them into contact with the cool surfaces of the boiler. Sufficient length and clear space must be afforded for combustion to complete itself,



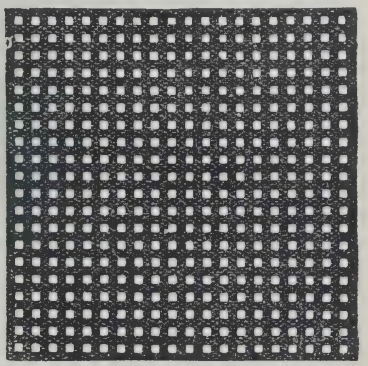
2. Light grey smoke.



3. Darker grey smoke.



4. Very dark grey smoke.



5. Black smoke.

FIG. 11.

or black smoke will be formed. The furnace arrangements of the Lancashire boiler are correct in form, but the water-cooled plates are a hindrance to certain combustion in the case of hand-firing; when high temperature mechanical-stoker conditions are obtained, however, it is much easier to prevent smoke with bituminous coal in a Lancashire boiler than it is in a water-tube boiler.

Water-tube boilers require a special setting to enable them to burn coal smokelessly. No furnace can be smokeless where the gases from the fire rise vertically and pass amongst cold-water tubes. Bituminous fuel can rarely be burned satisfactorily unless there is some extent of firebrick lining to protect the gases from too early chilling. Cross water-pipes in the inner flues of the Lancashire or Cornish boiler are bad, and will seriously hinder perfect combustion and produce black smoke.

Colour of Smoke.

The colour of smoke is best indicated by Ringelmann's smoke charts, which are numbered from No. 1, all white, to No. 6, all black. In fig. 11 intermediate sizes 2, 3, 4, 5, are shown, the ratio of line to space breadth being respectively, 10 : 90 ; 23 : 77 ; 37 : 63 ; and 55 : 45. Large charts drawn to these proportions and hung about 50 feet from the observer where the chimney top can be seen simultaneously, if possible, are very great aids in correctly estimating the relative blackness of smoke.

The four scales of the figure are called light grey, darker grey, very dark grey, and black, the extreme all white or all black being respectively no smoke or dense black.

MECHANICAL STOKERS.

There are several types of mechanical stokers ; for internally fired boilers (such as the Lancashire), the underfeed coking and sprinkler, and, for water-tube boilers, the travelling grate, the single horizontal retort, and the multiple inclined retort.

With regard to the internally fired boilers, by reason of their construction a stoker of the self-cleaning type is not possible, and for this reason the efficiency of such boilers can never be as good as that of the water-tube boilers. In the underfeed stoker the coal is forced onwards and upwards in a central trough by a helix conveyor, the hydro-carbons being distilled while the fuel is rising from the trough, and burnt while passing through the incandescent fuel bed, thus removing the possibility of smoke ; and the coke is burnt while passing over the side grates, so that the ash and clinker collect at the extreme sides of the furnace, from which position they can be conveniently removed. In the coking stokers, the fuel is usually fed on to the front of a moving grate, the volatile portion being expelled and burnt near the front, and the coke burning while travelling towards the back of the fire under the influence of the moving grate surface. In the sprinkler stokers the fuel is thrown on to the grate surface by means of a throwing shovel or similar device. All three types of stoker are capable of burning the fuel efficiently, but the underfeed type has the advantage as regards smokeless combustion.

The single retort machine is quite satisfactory for small boilers, and has the advantage that it is driven by a self-contained steam motor. The multiple retort type of stoker gives satisfactory results with many kinds of coal.

The travelling grate stoker is highly efficient and is capable of dealing with a wide variety of fuels; it has the advantage that each portion of the grate is in the furnace only one-third of each complete revolution, so that it may be adequately cooled and repaired, if necessary, without shutting down the boiler; the ash is discharged automatically over the back end of the grate.

In its original form it had the disadvantage that the fuel bed tapered from full thickness at the front to nothing at the back and thus offered a diminishing resistance to the air supply, the majority of which passed through the thinner fuel layer. In plants subject to varying loads this led to excessive amounts of excess air to secure smokeless combustion and limiting the power of the stoker. It has now been improved by dividing the wind box into a number of compartments provided with suitable regulating dampers so as to secure regular combustion under all load conditions, and this has resulted in greater outputs and considerably increased thermal efficiency.

It is necessary to study carefully the position, size and arrangement of the 'ignition arch' used with such stokers to suit the particular class of fuel to be used if ready ignition and even and smokeless combustion is to be attained.

The Babcock and Wilcox Patent Improved Mechanical Chain Grate Stoker.

The grate of this stoker (*Babcock & Wilcox, Ltd.*), as illustrated on page 1597, consists of an endless chain of short interlocking cast-iron grate bars, linked together, running on angle irons and rollers, and driven by a revolving shaft and sprocket wheels at the front end of the stoker. The necessary power to drive the revolving drum is transmitted through a link chain and sprocket wheels from a shaft arranged either overhead or underground. The starting or stopping of the stoker, or the variation of speed, is effected through the Babcock & Wilcox patent gear box, which can be driven at varying speeds. This box forms an oil bath in which the gear runs, and also contains an efficient clutch for putting the driving gear out of action should any unusual strain arise from an obstruction on the grate. Any small repair, such as replacing a link, can be carried out with the stoker in place under the boiler ; but in case of need the whole stoker can be brought out clear of the boiler, the frames being provided with wheels running on rails placed at the sides of the ashpit. The slack of the chain can be taken up from the front end of the stoker, thus avoiding the necessity for obtaining access to the back end for tightening the grate. The grate bars or links are of a special interlocking pattern, which allow the smallest coals to be used, and, while providing for an ample supply of air for the combustion of the fuel, prevent the influx of cold air at the back of the grate. The coal is fed over the whole width of the grate, and the depth of the fire is regulated by the adjustment of the vertically lifting fire-door.

The feed of the coal is slow, and the gas evolved from the fresh fuel as it drops on to the grate readily perfects its combustion by passing over the incandescent fuel farther on. The depth of the fire being regulated by a vertically lifting fire-door, the stoker can be adjusted in

a moment to suit the class of fuel to be consumed and varying demands for steam, both by the depth of the fire and the speed of the grate. Where the special requirements of the work make it desirable, the stoker can be fitted with an arrangement of vanes for cutting off the air supply to a portion of the grate during periods of light load, thus enabling the general efficiency to be kept up under all conditions of work.

Babcock & Wilcox also manufacture forced draught chain grate stokers of both the compartmental and the boxed-in type, for efficiently burning low-grade and other difficult fuels. When average quality coal is used forced draught is not necessary, and by simply opening the two front ashpit doors—an operation occupying only a few minutes—the stoker can be used as an ordinary natural draught plant.

Bennis Stokers.

(*Ed. Bennis & Co. Ltd., Little Hulton, Bolton.*)

The Bennis Sprinkler Stoker.

This apparatus, fig. 12, affords a perfectly regular output of steam under the most trying conditions; its flexibility permits a rapid response to urgent demands for increased steam. Minimum labour costs, low fuel bills, and smokelessness are a few of the leading advantages

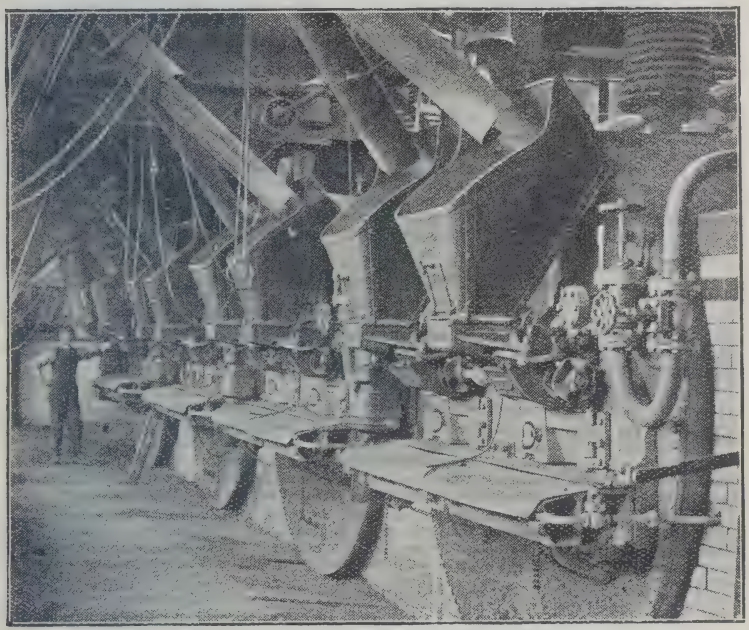


Fig. 12.

that accrue where the sprinkler stoker is in operation. The fuel fed into the hopper of the stoker—preferably mechanically fed—is led to the front of the pusher plate where it is evenly pushed over a ledge formed by the base of the feeding-box. The weight of fuel pushed is regulated, the fuel being deposited on to a flat plate called the shovel box, and then projected into the fire at intervals by an angular shovel actuated by patent pneumatic gear and effectually scattered over different portions of the grate where and as the fire requires it.

The pneumatic gear actuating the shovel, which includes an air cushion to obviate the possibility of shock or jar on the boiler front, is noiseless in operation.

An integral portion of the sprinkler stoker is the self-cleaning compressed air furnace, which prevents clinkering, and ensures perfectly clean fire-bars.

THE BENNIS SELF-CLEANING COMPRESSED AIR FURNACE ARRANGED FOR HAND-FIRING IN COLLIERIES.

This furnace, fig. 13, thus arranged burns any class of fuel, large or small, and burns with special advantage poor quality and refuse fuels, pond settlings, coal pickings, large lumps or small, coke dust, shale, or material which contains a large percentage of ash or clinker. With such refuse it is stated to be able to generate steam as quickly as furnaces fitted with stationary bars burning a higher grade coal. The furnace consists of a strong cast-iron front lined with fire-brick, and fitted with a door of ample size for hand-firing.

THE BENNIS PATENT CHAIN GRATE STOKER.

This arrangement, fig. 14, is applicable to any water-tube boiler or to return-tube boilers—where the fire is under the boiler shell and the flame returns through the tubes—to brewers' coppers and similar furnaces. One of the principal advantages claimed for this particular chain grate is that a larger grate area can be obtained under the boiler than with those of other makes. The Bennis chain grate with the patent halved link was the first chain grate capable of satisfactorily burning fine slack coals. There are many improvements connected with this grate, in addition to the central improvement which gives to the grate its distinction.

The Bennis patent link can be easily fitted to chain grates of other makes, and many successful replacements have been carried out, with the result that considerable economies have been effected by using cheaper coal.

THE BENNIS COKING STOKER.

This stoker, fig. 15, is an improvement on the ordinary type of coker, as it ignites the fuel and cokes it at a more rapid rate than is possible with other types. It is claimed that the immediate result is a higher evaporative boiler duty and greater economical efficiency. It consists of a machine supplied with hoppers into which the fuel is fed. Beneath the hoppers are duplicate feed-boxes with reciprocating feeder rams which feed the coal into the furnace intermittently. A sight hole situated between the feed-boxes makes it easily possible to ascertain the state of the fire at any moment. The feeder rams are worked by means of levers. The rate of the coal feed is easily adjustable. Large air-cooled fire doors are provided beneath the feeder-boxes, their object being to pass cool air into the protection plates fitted immediately round the stoker front on the fireside, thereby preventing damage to the stoker front and also to the boiler.

The Illinois Patent Chain Grate Stokers.

(Vickers Boiler Co., Ltd.)

The Illinois patent Chain Grate Stokers, for forced draught, natural draught, and induced draught, are made under licence by Vickers Boiler Co., Ltd., who hold the sole manufacturing and selling rights in all countries outside the United States.

Broadly, the stoker is of two standard types, viz.:

Type 'A' for natural or induced draught, fitted with air control dampers, and

Type 'G' for forced draught fitted with cast-iron air compartments, hopper bottoms, and device for the discharge of riddlings.

These stokers are of heavy and substantial construction, the frames are of heavy section, well braced and ribbed. The side frames, which, owing to their position inside the casing, are subjected to high temperatures, are independent of the head frames which are outside the setting.

Adjustable ledge plates are arranged at the top of the stoker side frames to prevent leakage of air between these and the chain, and also to prevent the riddling of fine coal. These ledge plates are so set that the distance between them gradually increases towards the rear of the stoker to permit of expansion of the grate and prevent the possibility of the chain binding between the frames.

Drive.

The grates are driven from the front by uniformly spaced driving sprockets which engage on loose rollers on the chain rods, and not on the links or rods, thus minimising frictional loss and reducing the power required, as also the maintenance. The arrangement also ensures a uniform distribution of the driving power over the whole width of the grate.

Idler drums are used on the rear shaft.

The advantages of driving the grate from the front are:

(*a*) The links are in compression, and consequently their strength is greatly increased over that of cast iron in tension.

(*b*) The spaces between the top links are kept closed, and therefore the riddling of fine coal is prevented.

(*c*) The driving sprockets are accessible through inspection doors in the head frame.

(*Continued on p.* 2117.)

BENNIS STOKERS.

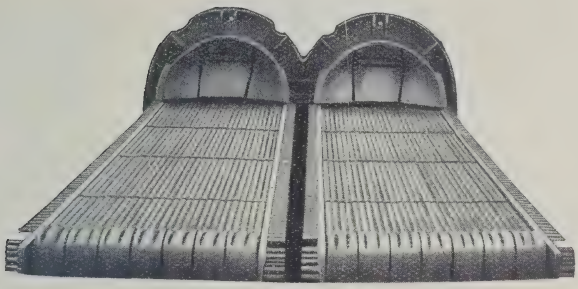


FIG. 13.—BACK VIEW OF THE BENNIS COMPRESSED AIR FURNACE.

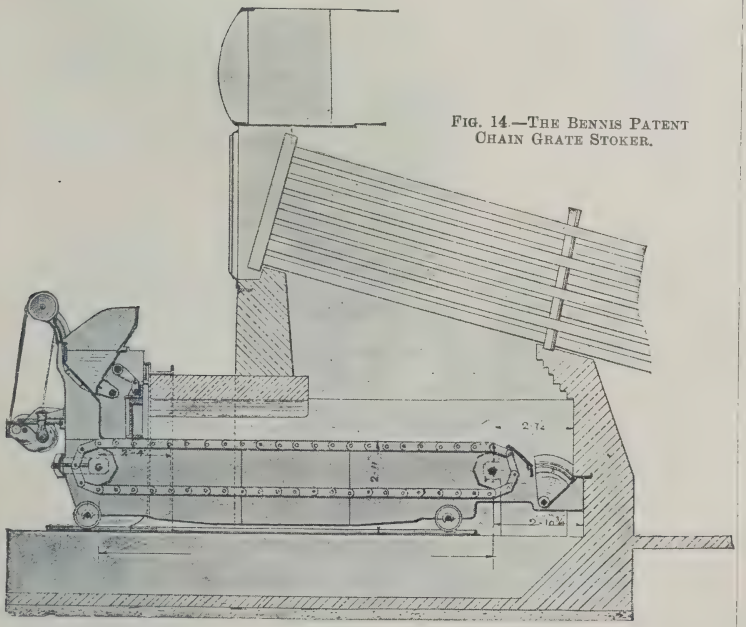


FIG. 14.—THE BENNIS PATENT CHAIN GRATE STOKER.

Bennis Stokers.

The Bennis High Temperature Smokeless and Gritless Coking Stoker.

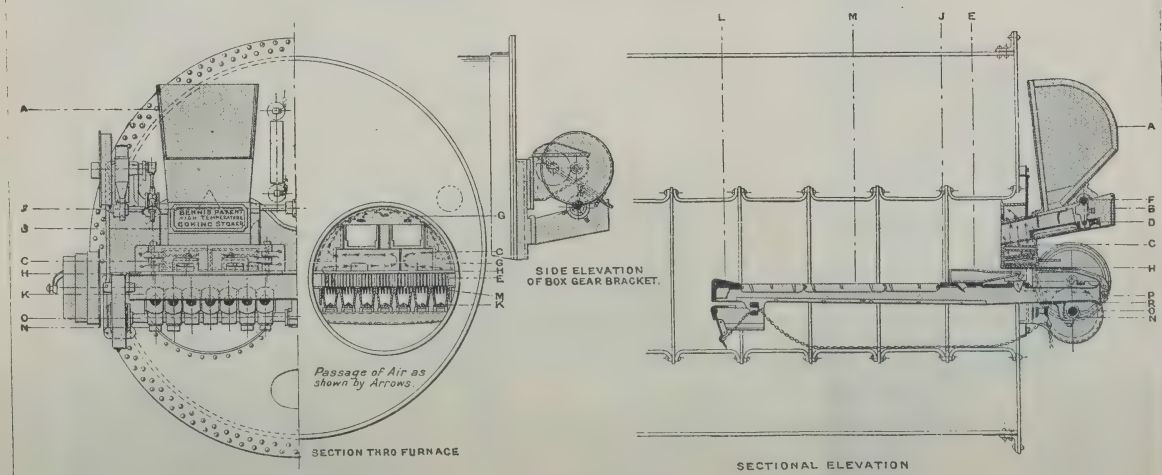


Fig. 15.

THE ILLINOIS PATENT CHAIN GRATE STOKER—*cont.*

The Grate.

The grate is composed of a number of links connected by means of twisted chain rods; the use of these rods prevents the accumulation of dust, rust, and ashes between the rods and links, with a consequent reduction in power consumption and maintenance cost.

The grate tension can be adjusted at any time while the stoker is in operation.

Special repair links are made and supplied for expeditious maintenance under running conditions.

Grate Support.

Chain skids are used instead of the roller construction.

The stokers are made up to 24 ft. in width in a single stoker, which necessitates the use of suspended arches.

Illinois Stokers are in use in this country at the present time up to a width of 18 ft. 6 ins., and stokers up to a width of 24 ft. are in hand.

The forced-draught stoker 'G' is of the compartment type.

The air is delivered from the main air supply duct to a wind box in the side wall extending the full length of the grate.

Stokers of medium width are supplied with air on the one side only; the wider stokers are provided with a wind box on each side.

The whole length of the active grate surface is divided into a number of compartments. Each compartment has its own opening into the wind box controlled by a circular plate damper. These dampers can be locked in any desired position by means of an operating lever and lock placed outside the wind box.

When the stoker has a wind box on both sides, dampers are provided on each wind box. The compartments are constructed of cast iron with hopper bottom and riddling discharging device.

For the burning of anthracite coal, coke breeze, lignite, or other high-moisture or low-volatile fuels, mechanical ignition is provided by means of a mechanical ignition zone at the front end of the stoker, comprising a small compartment running across the entire width of the stoker; in this compartment a suction is maintained whereby a small portion of the hot furnace gases are drawn down through the incoming green coal, causing the fuel to be completely ignited for its entire depth.

The stoker throughout is of robust construction, with a view to ensuring continuity of operation, freedom from trouble, and low maintenance cost.

THE ERITH-ROE STOKERS.

(*Erith's Engineering Co., Ltd.*)

These stokers, for water-tube boilers, figs. 16 and 17, are of the inclined multiple retort underfeed type, in which the fuel is fed from the bottom of the stoker hoppers into a series of retort units, from which it gradually rises to the surface of the fuel-bed and passes down an inclined plane, so that the ash and clinker finally fall into the ash hoppers in a continuous manner, without manual labour.

A heavy fuel-bed is carried, and the *surface* of the fire is maintained in an incandescent condition, so that the heat radiates direct to the boiler tubes; coking arches being entirely dispensed with, their cost and upkeep is eliminated.

The upkeep cost of the stoker is low, due to the metal being kept covered with the incoming fuel that has not risen into the zone of high-temperature furnace combustion.

A regulated supply of air from a fan is furnished to the air chamber below the stoker, and is suitably distributed through the retort tuyeres, etc., at the requisite pressure to secure 'balanced draught' conditions. A special baffle in the air chamber (which forms the subject of an additional patent) has been found of considerable value in improving the air distribution, so that the minimum of excess air, combined with a very high percentage of CO_2, can be maintained in regular service. Under these regulated and controlled conditions, low-grade slack and other fine coal of, say, 10,000 B.Th.U.'s per lb., or less, can now be burned with a thermal efficiency nearly equal to that obtainable with high-grade fuel.

The robust and simple operating mechanism of the stoker consists of a slow-moving crankshaft, which is normally run at about one revolution per minute; it operates, in series, the coal-feeding rams and distributing pushers within each retort, together with the stepped slicing bars reciprocating between each retort (which automatically slice the fires and keep them open and porous) as well as the reciprocating extension grates over which the refuse passes before it finally falls into the ash hoppers at the rear.

In the Erith-Roe stoker provision is made for the immediate and *independent* adjustment of the levers controlling the movement of the fuel-bed, which experience has shown to be essential for

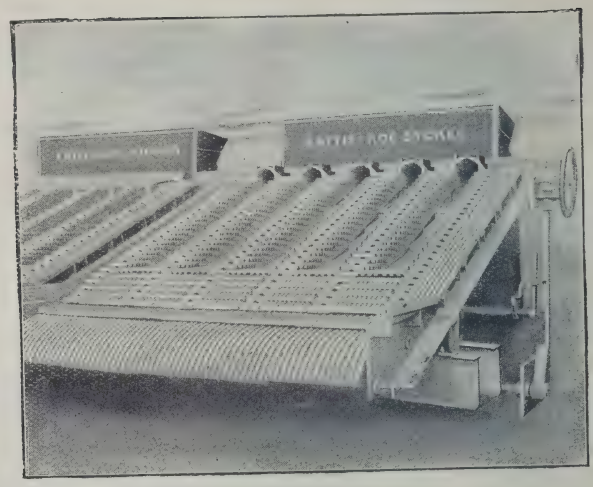


FIG. 16.

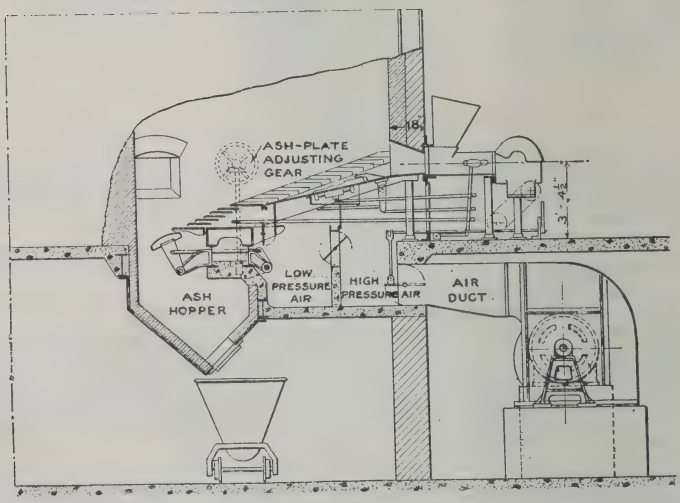
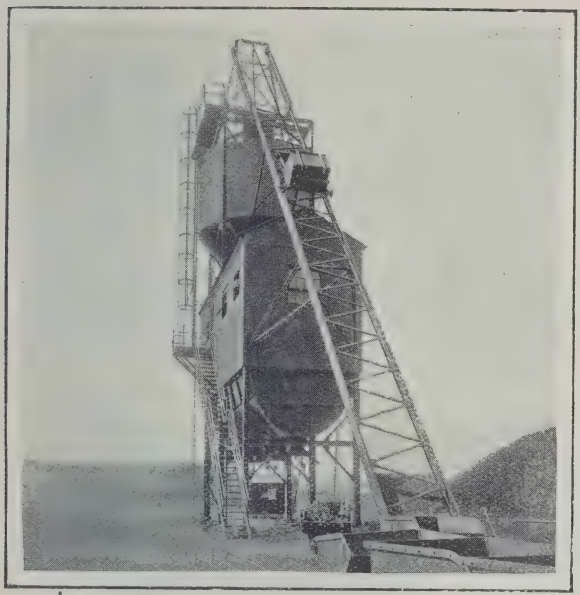


FIG. 17.

burning the widest range of fuels to the best advantage. No alteration from the original adjustment is, however, required, unless the conditions are materially altered through the coal being changed, etc. The Erith-Roe stoker is a British invention, and is being increasingly adopted for central station and industrial service at home and in the Colonies.



USCO

COAL HANDLING PLANT. The Underfeed-Beaumont cable drag scraper and skip-hoist provide the most reliable means of stocking and reclaiming coal in bulk

F. 2118

UNDERFEED STOKER CO. SUPPLY
Air Heaters for boiler plant and process work

Stokers for Lancashire, Cornish, Watertube and Marine Boilers, or Metallurgical Furnaces

Coal and Ash Handling Plants of various types
Flue Gas Washing Plants

Only the best material and methods of manufacture employed

UNDERFEED STOKER CO.
LIMITED
ALDWYCH HOUSE
ALDWYCH, LONDON, W.C. 2
WORKS : DERBY

F. 2119

The Underfeed Stoker Co.'s Stokers.

(The Underfeed Stoker Co., Ltd.)

Type 'L' travelling grate stoker for water-tube boilers.

This arrangement, which is the outcome of over twenty years' experience with the well-known 'A' type, and of years of research, has been developed to meet the demand for a stoker which can be made in such size that only one machine is necessary for each boiler instead of two, three, or four separate machines having division walls between them in the furnace. Type 'L' stoker can be made in single units up to 700 sq. ft. grate area.

The two most important features of any travelling grate stokers are :—

(1) The construction of the grate surface.

(2) The method of air distribution.

The grate surface employed on the type 'L' machine represents a great advance in design as regards durability, tendency to make riddlings, and resistance to adhesion of clinker which, in many forms of grate, tend to clog the air spaces.

Referring to figs. 18 and 19, showing sectional views of the grate, it should be explained that the grate bars are arranged transversely. They are supported at their ends on steel tension chains, which occur at frequent intervals across the width of the stoker, and are sloped downwards in the direction of travel, lying upon each other in the direction of louvres. The bottom edges of the bars are turned upwards to form troughs, into which fall the riddlings, which find their way through the air spaces between the bars.

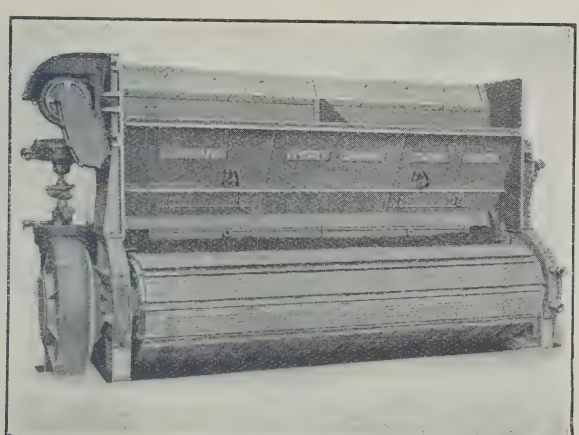


Fig. 18.

The steel tension chains or carrier chains are spaced equally by means of the tie rods having pipe ferrules threaded upon them, and the grate is carried on its journey through the furnace on rollers, which are threaded on to the transverse tie rods and run upon skids of very large area. As the grate passes around the rear drum, the firebars overbalance on the pivots, which are cast on their ends, as shown in fig. 19, and so empty the riddlings from the troughs, at the same time cleaning the grate surface of any clinker adhering thereto.

The following advantages are claimed for the grate surface :—

(*a*) The grate is non-riddling. The actual quantity of dust which is deposited inside the main air chamber, measured over a period of months, averages 1 lb. for each ton of coal burnt on the stoker. This quantity is so small as to eliminate the necessity for any mechanical means for removing the dust from the air chamber. It is quite sufficient if the air chamber is cleaned out once per week, and this can be accomplished either by raking from the access doors or by blowing dust to the rear end and through grids arranged in the bottom of the air chamber.

(*b*) The grate is self-cleaning, due to the movement between individual bars of the grate as they pass around the rear drum. This increases the life of the grate surface, owing to the air space being kept free from clinker.

(*c*) Any part of the grate surface may be renewed without shutting down the boiler and without disconnecting the tension chains. This operation is best accomplished as the grate is passing around the front drum.

(*d*) Owing to the angular disposition of the grate bars, the thickness of each bar is considerably less than the pitch of the bars, measured in the direction of travel, and on this account considerable growth may occur on the surface of the bars without reducing the widths of the air spaces.

(*e*) Due to the 'trailing' progression of the grate bars the possibility of jamming, due to the grate fouling stationary objects, such as clinker dams, etc., is reduced to a minimum.

(*f*) Those parts of the moving chain which are subjected to stress are completely shielded from the heat of the fire.

(*g*) Should one firebar become broken and fall out of position, the space is immediately filled by the preceding bars falling backward and closing the gap.

(*h*) The whole grate is flexible, and therefore unlikely to suffer severe damage in the event of a jam taking place.

Referring to the second important feature, viz. air distribution, this is also quite a novel feature of the type 'L' stoker. Instead of distributing air from side air supply boxes built into the furnace side-walls, and communicating with separate transverse air compartments, the air is admitted into a large air chamber or reservoir which extends the full width and length of the stoker, the height of which is approximately 2 ft. In order to distribute this air to the various combustion zones in the length of the stoker, valves are arranged immediately below the grate. The arrangement of the valve is shown in fig. 16, and the valves in each row are operated simultaneously and independently of all the other rows. There are no divisions in the main air chamber. The advantages of this construction are as follows:—

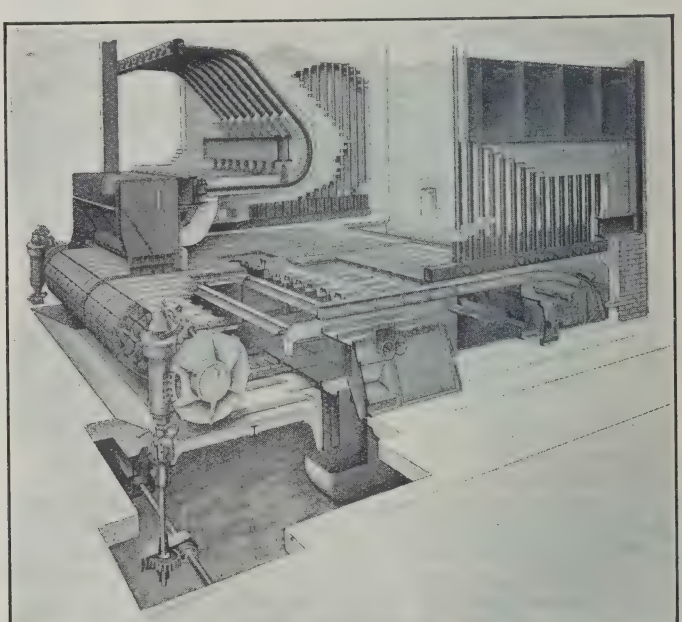


FIG. 19.

(*a*) The volume of the main air chamber is sufficiently large to allow the air to dissipate its velocity, and therefore the pressure over the width of the stoker is uniform.

(*b*) Combustion is under perfect control, and different combinations of air-valve settings may be obtained for the purpose of dealing with different quality of fuel without the need for any change in the stoker itself, as the amount of air passing to each zone of the furnace may be controlled with precision.

(*c*) The whole of the framework of the stoker is kept cool in all circumstances by the movement of air. Even if the rear air valves are completely closed so that no air is passing through that part of the grate, the air is all the time circulating, and thus preventing overheating of the machine.

(*d*) The interior of the stoker is fully accessible. Large doors are arranged through the furnace side walls giving access to the main air chamber, and from the main air chamber to the whole of the return chains and the front sprockets. All the air control valves and their mechanism are immediately accessible from the inside of the main air chamber.

Type 'L' Stoker in widths up to and including 13 ft. 9 ins. are made in the form of single grates, but machines exceeding this width have their grates divided in the middle, the space between the adjacent grates being 5 ins. This space inside the furnace is filled by an air-cooled cast-iron dead plate and no brickwork is necessary. Each half of the stoker is driven and controlled separately, so that in the event of half of the grate going out of commission the boiler may be kept steaming by the other half until a further unit can be brought up on the line. It is claimed for the twin grate that it has all the advantages of the single wide grate and also of the double grate.

The type 'L' Stoker may be driven in a manner similar to the standard 'A' Type Stoker—that is, by means of a line shaft operated from a motor and worm reduction gear, or it may be driven by a motor and gear supported directly on the coal-hopper side-plate. In the case of the Twin Stoker, two self-contained gears are necessary—one for each half of the grate. The self-contained gear is recommended in all cases where electric current is available, as by this means it is possible to avoid all shafting, brackets, bolts, etc., which are objectionable in a boiler-house Apart from this, each driving gear is independent of the other, so that in the event of one going out of commission the other gears are not affected.

THE 'B' TYPE STOKER FOR LANCASHIRE BOILERS.

In this stoker the fresh fuel for the maintenance of the combustion is continuously supplied from *below* the level of the fire and gradually advanced upward to the zone of combustion by the arrangement shown by fig. 20.

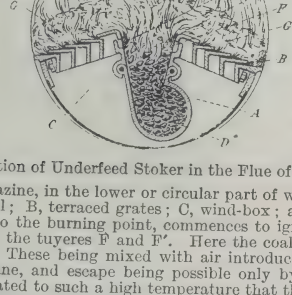


FIG. 20.—Cross-Section of Underfeed Stoker in the Flue of a Lancashire Boiler.

A is a retort or fuel magazine, in the lower or circular part of which revolves a taper-feeding worm which conveys the coal; B, terraced grates; C, wind-box; and D, green coal; the latter being gradually pushed up to the burning point, commences to ignite at the point E, where it meets the incoming air from the tuyeres F and F'. Here the coal is coked, i.e. deprived of its volatile hydrocarbon gases. These being mixed with air introduced at this point through the air inlets of the fuel magazine, and escape being possible only by rising through the glowing coke above, the gases are heated to such a high temperature that they are completely consumed without smoke, while the combustion of the fixed carbon of the coal is completed by the air introduced through the apertures in the sides of the terraced grates B.

The effect of this method of underfeeding is to consume the hydrocarbon gases, so giving a clear fire; it is claimed that at no time, however varying the load, is there the slightest sign of smoke. This stoker caters especially for fluctuating loads, and is much used in works where steam is required for process work.

Another special feature of this stoker is that as it has only three moving parts its upkeep costs are very low.

THE 'UNDERFEED' CLASS 'E' STOKER, FOR WATER-TUBE BOILERS.

In this stoker (*The Underfeed Stoker Co., Ltd.*) the fuel is deposited in a large hopper, and conveyed under the fire by means of a reciprocating sliding bottom which runs the full length of the trough. As the coal rises from the trough it is distributed to the sides of the furnace by means of moving bars. These bars are alternately moving and fixed. The moving bars work transversely to the retort. The extent of the movement is from $\frac{1}{2}$ in. to 1 in., depending upon the size of the furnace and other conditions. On the bottom of each moving bar are cast two lugs which engage with the bulb of the longitudinal rocking bars. These rocking bars in turn receive their movement through the agency of two spirals and nuts, which mechanism is entirely outside the furnace. The nuts are bolted to the crosshead and therefore reciprocate with the bottom. The reciprocation of the nuts causes the spiral to rock to and fro.

The coking and burning coal is constantly carried, by the action of the moving bars, to the dumping trays along each side wall, where the resulting clinker and ash are deposited. These dumping trays are operated by levers on the outside of the furnace, enabling the ash and clinker to be automatically discharged.

One of the important features of this stoker is the distribution of the air, which, upon entering the wind-box, passes upward along each side of the trough, or retort, and is discharged, partly through holes into the retort. The remaining air passes through the bars, which are made hollow. These bars have no openings in their top surfaces, and no air can find its way into the fire above until it has passed through the aperture at the bottom end of the bar, from which aperture it is discharged into the enclosed air chambers. From here the air rises and passes through the small spaces between the bars into the coked fuel. The action of the air in passing through the bars keeps them cool, thus greatly prolonging their life. The heat taken off the bars in this way raises the temperature of the air in the air chambers to from 350° F. to 400° F. The pressure of air in the chamber varies from zero to $\frac{3}{4}$ of an inch of water.

For marine work a variant of this stoker is very successful, and between March 1923 and October 1928 one steamship line alone has thus equipped no fewer than twenty-seven of its ships.

MECHANICAL DRAUGHT.

Various mechanical methods have been introduced for increasing the draught for boilers; this may be accomplished by an exhausting fan drawing the products of combustion through the flues, or by blowing air into a closed stokehold or an enclosed ashpit, and forcing the air through the fire with the necessary pressure, which varies according to rate of combustion desired. There is also a combination of these mechanical means, termed 'balanced draught,' in which air is forced under pressure into the ashpit by one fan and the gases are drawn off by an exhausting fan so arranged that there is neither pressure nor vacuum above the fire. Another method is the 'ejector draught' system in which the flue gases are drawn up the chimney by a high-speed jet of air.

The advantage of mechanical draught produced by a fan is that the first cost is less than half that of a chimney; also the draught is independent of atmospheric conditions, can be varied instantaneously to suit the demand for steam, permits of economisers, and allows the reduction of the temperature of the waste gases to be carried to the lowest practicable point. It increases the capacity of the boiler, and permits of a more perfect combustion with less excess of air. It allows the use of lower grade and less expensive fuel, and so produces steam more cheaply.

Any desired rate of combustion from the normal of 20 lbs., to 30 or 40 lbs. of coal per square foot of grate area, may be maintained with a mechanical draught plant. The air volume required depends upon the style of boiler, condition of boiler setting, skilful stoking, the kind of coal used, and other factors; in round numbers 250 to 300 cu. ft. of air will be required per lb. of coal; these figures represent about 12 per cent. and 10 per cent. CO_2 respectively.

To drive a fan requires about $\frac{1}{2}$ to 1 per cent. of the total power generated. The full benefits of mechanical draught are only obtained where means are employed to utilise to the fullest extent the heat of the flue gases (by means of superheaters or economisers) before the latter are discharged into the chimney. The cost of installing a fan and engine is very much less than the cost of a chimney, so that mechanical draught is economical in first cost.

Fan draught enables a heated air supply to be employed, and it is said that by heating the air supply to 180° the fire temperature will be raised an additional 360°, owing to better combustion, etc.

The Musgrave System.

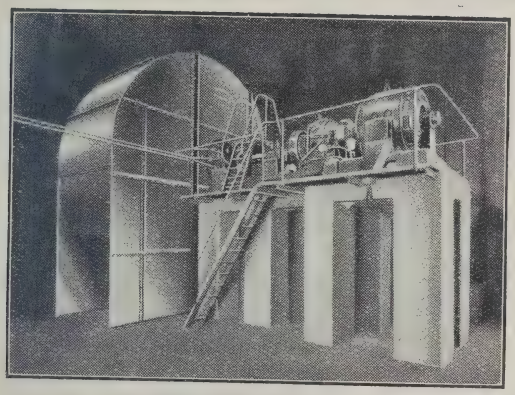
(Musgrave & Co., Ltd., Belfast.)

This is an induced system of mechanical draught. The action of the fan is substantially the same as that of a chimney, the gases being drawn through the furnace and discharged up the stack; this is usually found to be a very convenient and natural method of applying mechanical draught, involving little or no alteration to existing boiler plant, and having a wide range of flexibility. The fact that the draught is set up by suction instead of pressure gives a more uniform flow of air through the fuel, thus reducing the tendency to blowholes through the fire. All leakage being inward, all danger of discharge of flame and smoke at the furnace doors is avoided. In tubular boilers better contact is obtained between the gases and tube surfaces, the tendency with forced draught being for the gases to form a contracted vein at the front end of the tubes, resulting in a consequent loss of heat transmission.

With the induced system the fullest possible use can be made of economisers, either in the form of feed-water or air heaters. There being no necessity to maintain a high gas temperature in the chimney to create draught, as is the case when natural draught is depended on, the whole of the available heat in the gases may theoretically be utilised, and the maximum economy obtained. In actual practice, however, the temperature to which the gases may efficiently be reduced is about 300° F.; to reduce the gas temperature below this figure, the increased cost of economisers or air heaters involved, the additional draught required, and the corrosion due to the condensation of acids from the gases on the cooling surfaces, render further reduction impracticable.

To obtain flexibility and perfect control in a mechanical draught installation, an independent fan drive by steam engine or electric motor is preferable to a drive from a line shaft by means of belt or rope; the speed of the engine or motor can be immediately adjusted to suit the varying draught requirements, and in suitable cases an entirely automatic control can be utilised. The steam

MECHANICAL DRAFT



This Musgrave Induced Draft Fan, one of a pair in an important Power Station, has definitely increased the efficiency of the steam generating plant, and proved a good investment from the economic standpoint.

THE MUSGRAVE SYSTEM of Mechanical Draft gives better combustion, allowing the use of a lower grade fuel; controlled draft securing maintenance of steam pressure. It will pay for itself over and over again in the increased efficiency and economy of your steam-generating plant, while Musgrave Fans are as reliable and trouble free as a chimney.

Write for Catalogue No. C4/13, post free on request.

MUSGRAVE & CO. LTD.

ENGINEERS

BELFAST
St. Ann's Iron Works
Telegrams: CENTIGRAD

GLASGOW
217 St. Vincent St.
Telegrams: CENTIGRAD

MANCHESTER
Arkwright House
Telegrams: CENTIGRAD

BIRMINGHAM
2 Vicarage Rd., Handsworth.

LONDON
Sardinia House, Kingsway, W.C.2
Telegrams: MUSGRAVE HOLB

CARDIFF
24 Queen St.
Telegrams: CENTIGRAD

Codes: Lieber's; A.B.C. 5th Edition; Engineering Telegraph, 2nd Edition.

The Engineer.

Established 1856. Registered as a Newspaper

PROVIDES THE LATEST AND THE MOST RELIABLE INFORMATION FOR ENGINEERS

IN ALL BRANCHES OF THE PROFESSION.

The last issue in each month contains a Supplement, **The Metallurgist,** *a journal devoted to the science of metals, ferrous and non-ferrous.*

Subscriptions

			Per annum
INLAND, YEARLY			£3 5 0
CANADIAN SUBSCRIPTIONS.	Thin paper edition	...	£2 18 6
	Thick paper edition	...	£3 3 0
FOREIGN SUBSCRIPTIONS.	Thin paper edition	...	£3 3 0
	Thick paper edition	...	£3 7 6

'THE ENGINEER' DIRECTORY AND BUYER'S GUIDE

SENT FREE ON APPLICATION.

consumption of the engine will not exceed 1½ or 2 per cent. as a maximum of the total steam generated, while the steam jet type of blower absorbs from 5 to 10 per cent. of this total.

The great loss of heat where chimney draught alone is depended upon puts it out of consideration altogether in modern plants.

To obtain maximum economy the design of the fan must be correctly proportioned to the required duty, so that the air supply necessary for efficient combustion may be obtained with the minimum expenditure of power under all conditions. All flues should be as short and direct as possible, avoiding all unnecessary bends, changes of section, and obstructive dampers.

Musgrave's induced draught fans are constructed both of the multiblade and straight-bladed types, depending upon the duty. The former with their higher rotational speed are more suitable for use with direct-coupled motor or turbine drive, the latter for engine drive.

Where large quantities of dust are entrained in the waste gases, the straight-bladed wheel is better, owing to its freedom from all possibility of clogging.

The casing may be either of the full or three-quarter housing type: in the latter case, the lower portion of the periphery is formed in brickwork or concrete. The shaft carrying the wheel is usually overhung, being adequately supported in two ring oiling bearings of ample size. For normal gas temperatures the bearing adjoining the fan housing is provided with a water-cooling jacket. For particularly high temperatures both bearings are jacketed, and occasionally a forced lubrication system adopted. Reliability is assured by the provision of ample bearing surfaces in engine and fan bearings, and sound scientific and mechanical design throughout.

THE STURTEVANT SYSTEMS.

Figs. 21 to 24 show the various systems installed by the *Sturtevant Engineering Co. Ltd., London*, which are in use in power stations, workshops, factories, warships, passenger and freight steamers, etc.

Induced Draught (fig. 21). This is the most commonly adopted system, as it is generally applicable to an existing boiler installation, and is economical in operation. The products of combustion are drawn through the flues, economisers, etc., by an exhaust fan, and discharged to an existing stack or to a short length of steel chimney built just high enough to carry the gases above surrounding buildings. The fan is fitted with water-cooled bearings.

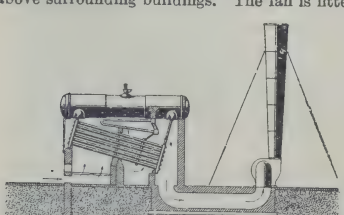


FIG. 21.

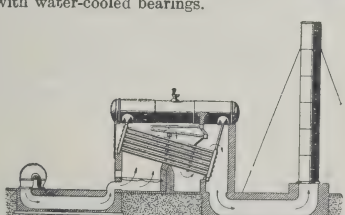


FIG. 22.

Forced Draught (fig 22.) is advantageous for burning small or low-grade fuel. The fan blows air under pressure into the closed ashpit, and the gases are discharged either to existing stack or short steel chimney. For ship use the closed stokehold method is generally used.

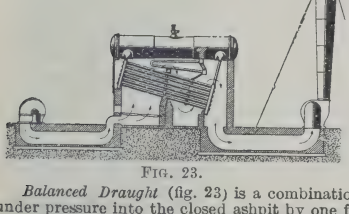


FIG. 23.

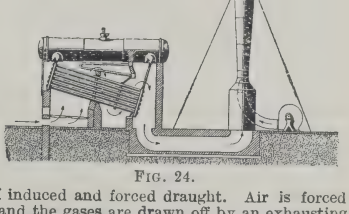


FIG. 24.

Balanced Draught (fig. 23) is a combination of induced and forced draught. Air is forced under pressure into the closed ashpit by one fan, and the gases are drawn off by an exhausting fan so arranged that there is neither pressure nor vacuum above the fire. This is only used in special cases, owing to the cost being much higher than either fig. 21 or fig. 22.

Indirect Induced Draught with Ejector (fig. 24). The combustion gases are drawn up a chimney of special design by the effect of a high speed jet of air, which if desired can be mixed with a portion of hot gases. This system is of special value in cases where the flue gases are corrosive or excessively hot, as they need not pass through the fan. This plant requires very careful designing, otherwise the efficiency is very low indeed.

The Howden Forced Draught System.

The principles on which this system of forced draught (*James Howden & Co., Ltd., Glasgow*) is based are (1) the supplying under pressure of hot air for combustion in quantities suited to the quality of coal used; and (2) the utilisation of the heat in the waste gases for the purpose of raising the temperature of the air for combustion.

Briefly stated, the following are the principal advantages of this method of combustion :—

(1) Great increase in the steaming power of the boilers. (At least 30 per cent. and probably much more over best natural draught conditions. Probably as much as 50 per cent. could be obtained if necessary, though quite such a high rate is not recommended for regular working.)

(2) Unequalled economy in fuel. (At least 10 per cent. and frequently 15 per cent. per lb. of water evaporated over the best natural draught conditions and probably much more.)

(3) Reduced size and weight of boilers for a given power, giving correspondingly additional space available for other purposes.

(4) Complete control of combustion and steam supply under all conditions of the atmosphere. (With this system of draught the volume and pressure of the air for combustion is entirely controlled by the fan and the manipulation of the air valves in the furnace fronts.)

(5) Capability of burning all classes of coal and coke. Low-grade fuel can always be burned satisfactorily where this system of forced draught is applied. Under natural draught conditions this is often difficult and sometimes impossible owing to the inability of supplying the greater volume of air always required where the greater quantity of inferior fuel is used. For the reasons stated in (4) this difficulty is surmounted, the additional quantity and pressure of air necessary being easily obtained by running the fan a little faster.

(6) Preservation of the boilers on account of the more uniform temperature maintained in the furnaces. Boilers fitted with the Howden system of draught are less liable to the leakage and trouble generally present under natural draught conditions; the reasons being that no cold air ever enters the furnace during the period of working, and much greater uniformity of temperature is maintained than under natural draught conditions. Sudden alternations from heat to cold, and the consequent contraction and expansion, have an injurious effect on the internal parts of steam boilers, and such sudden fluctuations in temperature are prevented by the use of this system. When the furnace doors are opened the air pressure in the ashpit immediately ceases, and only the hot air from the upper air supply valve passes into the furnace, first in a direction vertically downwards, thereby forming a hot screen over the furnace mouth and preventing access of cold air. Such conditions when firing or cleaning fires are also conducive to economy. The benefit of the system regarding fluctuations of furnace temperature is most marked and appreciated when it is compared with the working conditions of the closed stokehold system. In the latter a considerable air-pressure is generally used, so that the fluctuation in temperature of furnace is both rapid and violent and puts a severe strain on the boiler structure. This is quite obviated in the Howden system.

(7) Reduction of smoke.

(8) No high chimney required. For land installations a small iron funnel, sufficient to carry off the waste gases clear of the surrounding buildings, is ample for the purpose.

(9) Coolness in boiler-room and ease of working. With this system the air casing, reservoirs, and furnace fronts on the boiler end effectually prevent radiation into the stokehold, which is thereby kept quite cool, making the work easier than with the closed stokehold system, which is very trying for the firemen.

(10) Great reserve power available in case of emergency.

The general arrangement of the Howden system consists of a fan (driven by a steam engine electric motor, or belt), which supplies the air for combustion, discharging it through a conduit into the air-heating chambers. In these air-heating chambers the air circulates round a series of tubes. The waste gases, after leaving the boiler, and before entering the uptake, pass upward through these tubes in the air-heating chambers, or in the Lancashire type of boiler horizontally through tubes in air heaters in the side flues, thus giving up part of their heat to the air, which then passes along passages into air-tight reservoirs or chambers surrounding the furnace fronts. In a recently patented heater developed by this firm, the air is passed through tubes formed by corrugated plates, and the flue gases through spaces between each pair of plates forming the air tubes, and the arrangement is such that a very highly efficient air heater can be accommodated in a moderate amount of space, with a similar arrangement of air reservoirs. From these chambers the air is passed through furnace fronts by valves into the ashpits, and over the fires through air-distributing boxes in proportions exactly suited to the kind of fuel used and the rate of combustion required, thus ensuring a very perfect combustion of fuel.

In the furnace fronts the ashpit air valves are operated by one handle, and a safety locking device is also fitted which prevents the furnace door from being opened until the ashpit air valves are closed. In marine type boilers, retarders are also fitted (one in each boiler tube).

Furnace fronts for powdered fuel have also been developed on the Forced Draught principle, whereby the radiation from the furnace front in general, and the ashpit door in particular, is reduced to a minimum, while simplicity of operation is maintained.

Silley patent smokebox door fastenings form an important adjunct to the forced draught apparatus. These fastenings ensure a thoroughly air-tight smokebox and entirely obviate the trouble usually experienced with buckling, etc., in doors secured by other methods.

At the present time there are about 6,600 installations of the Howden system fitted, and the number is rapidly increasing every year. The aggregate i.h.p. of these is over 24,000,000. They comprise steamers for almost all the principal steamship companies of the world, and a considerable number of installations on land boilers.

PULVERISED COAL.

(Contributed by L. C. Harvey, London.)

General.

Coal in pulverised form was first successfully applied to cement-burning kilns. This method of heating cement kilns has now become universal practice, the steady long flame with its resulting continuous and unvarying heat development inside the rotary kiln being ideal for this work. Moreover, the ash which has always been a source of trouble in other applications forms part of the finished product in this industry.

For cement burning the coal is reduced to a fineness of about 80 per cent. through a 100-mesh screen (10,000 holes to the square inch). It is essential to pulverise to a greater degree of fineness for general applications. A coarser product can be permitted when burning high volatile fuel (30 per cent. V.M. and upwards), or in the case of very large steam-raising units, as now installed in the latest super-power stations in America and elsewhere (see p. 2132), but the American standard of fineness, *i.e.* 85 per cent. through a 200-mesh screen (40,000 holes to the square inch) should be accepted for general boiler firing and for all metallurgical work.

This fine degree of coal-dust, coupled with a low pressure on the air supply, thereby decreasing the velocity of the gases passing through the furnace or boiler, and a wider adaptation of the principle of turbulent mixing of the fuel and air, resulted in the removal of the former troubles due to the formation of ash slag in chequer-work, on metal to be heated, or on boiler tubes.

It is now a recognised and successful practice to apply pulverised coal to all classes of metallurgical furnaces, and in a very marked degree to power-house boilers. (Open-hearth steel furnaces require special consideration.)

The use of pulverised fuel as a means of firing boilers, furnaces, cement kilns, locomotives, public supply, and incidental purposes throughout the world now accounts for a vast quantity of industrial fuel, perhaps 100 million tons per annum.

After the first great reception of this system of burning coal in America the rapid progress which has taken place in the heavy-duty boiler firing has been due to the work of the *International Combustion Engineering Corporation* in bringing the ' Central Mill ' plant to a state of perfection, to the introduction of the water screen in the ash zone, and latterly to the utilisation of Murray Fin Tube and other water-cooled furnace walls and linings, and to ' Turbulent Burners.'

Economy in space, greater flexibility, and reduced first cost of plant have resulted from dispersing the full complement of bunkers, dryers, mills, etc., which constitute the ' Central Mill ' plant into what is now termed by the *International Combustion Engineering Corporation and its Associated Companies*, the ' Direct Firing ' system, *i.e.* one complete and compact plant alongside each boiler.

Advantage has thereby been taken of waste heat from flue gases to dry coal in static dryers instead of fuel utilisation in rotary dryers of ' Central Mill ' plants, and at the same time to eliminate long-distance screw conveyors or pressure transmission systems for the pulverised fuel. ' Self-contained Unit ' mills of small capacity, 200 lbs. per hour, up to relatively large output, 10,000 lbs. per hour, originally established this ' Direct Firing ' system. With these, coal dryers are seldom used though composite types of mills and dryers are in existence. Unit Mills are often designed to draw in heated air or inert gas and thus, in a measure, to dry out moisture in fuel during the process of pulverisation.

Large capacity mills for ' Central Mill ' plants, 5 to 20 ton units, do not operate with coal having more than 5 per cent. free moisture, whereas certain types of ' Self-contained ' mills function quite well with coal containing up to 15 per cent. free moisture.

It is also largely due to the enterprise of makers in bringing the many types of Unit Mills to a stage of relative perfection that pulverised coal firing has been so rapidly extended, the year 1928 crowning their success by the all-important departure in the practical application to marine boilers and the installation of complete ' Unit Mill ' pulverising plant on board ship.

Many important plants are now in operation in France, Belgium, Germany, and other Continental countries. Particular attention is being given to the development of pulverised fuel firing in Europe by *International Combustion Limited*, who have made rapid strides in the development of the ' Lopulco ' system and patents, and have so equipped several power stations in Great Britain (p. 2137).

At the Tennessee Coal, Iron, and Railroad Company's works, and at the Ford Company's works in America, boilers of considerable size are installed for firing either with blast-furnace gas or with pulverised coal. The original practice of using gas and coal together has been discontinued in favour of burning one fuel or the other. It was thought at the outset that the firing of gas in conjunction with pulverised coal prevented the formation of ash clinker, but subsequent improvement in the method of firing pulverised coal has demonstrated that the best results can be obtained when using either one or the other fuel alone, excepting only when pulverised coal must be introduced at times of shortage of blast-furnace gas.

Although chimneys of good height are advisable for large installations, apparatus has been devised to trap much of the dust carried in the flue gases. Tests carried out at Milwaukee show that as little as 15 per cent., and as much as 85 per cent., of the ash in coal passes out at the

chimney top under certain conditions. These conditions are controllable, and whatever may be the amount permitted to escape into the atmosphere, this dust is unlikely to become a nuisance, for it is in such fine particles that the dust rises into space, and its deposition is very widely spread.

Pulverised coal-fired boilers are now guaranteed to give a continuous evaporation of 13 lbs. (from and at 212° F.) of water per sq. ft. of heating surface, at a boiler and furnace efficiency of 81 to 83 per cent. and a peak load evaporation approaching 20 lbs. per sq. ft., with water-tube furnace linings in series with the boiler elements.

As a direct result of high evaporation rates obtained in water-tube protective linings for combustion chambers (the Murray Fin Tubes, etc.), the ' Combustion Steam Generator ' (fig. 25) wherein the main steam generation is effected in a tube-lined combustion chamber has given the first confirmation of the exceptionally high steaming rates, due to radiant transmission of heat, which will be obtainable from these new types of boilers for which refractory linings and external combustion chambers are entirely absent.

In addition to the further progress made in the firing of locomotives with pulverised fuel in Chile, where the ' Fuller ' equipments are in use for the burning of the local lignite, successful trials have been carried out in Austria and Germany and a small number of locomotives so equipped.

Difficulties with the deposit of ash across the tubes at the firebox end have been mainly overcome by methods adopted for the introduction of the fuel to the firebox, turbulent burners, and fine pulverisation.

The use of pulverised fuel for this purpose remains practically dormant, but the statistics relative thereto are retained in view of the possible extension of this method of firing locomotive boilers in various countries, for economies introduced by any successful system will be very important.

For the firing of marine boilers there is every reason to expect that pulverised coal will enter into competition with oil fuel.

Details of Ships in commission and further notes on marine boiler firing and its sequence, ' Lancashire Boiler Firing,' are given below.

The development of low-temperature distillation will be of great influence. The residual fuel is ideal for marine use—soft, friable, readily pulverised, of reasonable volatile and ash content, free from spontaneous combustion tendencies, and therefore safe for storage in ships' bunkers. Semi-coke already pulverised, such as the product of the ' McEwen Runge ' process, offers special advantages. Oil or coal burners, the Peabody, the Buell, etc., are equally suitable for either fuel.

Mill House Plant for Central System.

Fig. 26 shows an outline arrangement of a typical coal-pulverising plant with screen separator mill and rotary coal-fired dryer.

The coal delivered by rail wagon is crushed to ¾-in. cube and passed up the No. 1 elevator to the crushed coal storage bin.

From this bin the crushed coal is fed by gravity into the rotary dryer, wherein the moisture content of the coal is reduced to 1 per cent. This is a necessity for bulk pulverisation, storage, and for conveying the fuel through pipes.

Rotary coal dryers are fitted with furnaces, either hand fired, stoker fired, or arranged for burning powdered coal. The hot gases from the furnace should not pass through the dryer in direct contact with the coal until these gases have been considerably cooled. Some dryers are so designed that the hot gases do not pass over the coal before they have travelled the full length of the dryer through a central tube, when they are diverted into the outer portion of the dryer and pass back to the furnace end in direct contact with the coal. For high volatile coal this type is essential.

The average coal consumption in the dryer is about 20 to 30 lbs. of coal per ton for a reduction of 10 per cent. in the moisture content. When coal is very wet it then becomes necessary to pass the coal twice through the dryer. The final temperature of the coal leaving the dryer should not be above 250° F.

From the dryer the coal passes up No. 2 elevator to the dried coal bin, passing over a magnetic separator pulley in order to remove any pieces of iron from the coal, and is fed by gravity into the pulveriser.

With the static dryer, which has no moving parts, a portion of the waste gases from the boilers, generally about 10 per cent., is forced through the coal, with the result that the coal is thoroughly dried without any loss of the volatile constituents or fine dust.

The stationary type of dryer has been generally adopted for power-house installations. Initial troubles, regarding the flow of coal from the dryer and uneven or over-heating, have been eliminated.

Raw coal can be delivered to the bunker above the dryer, from whence it falls by gravity through the dryer to the pulveriser mill. Thus, the rotary dryer motor, the No. 2 elevator, and, if necessary, the dry coal bunker, shown in fig. 26, are dispensed with, and only in special cases are rotary coal-fired dryers necessary.

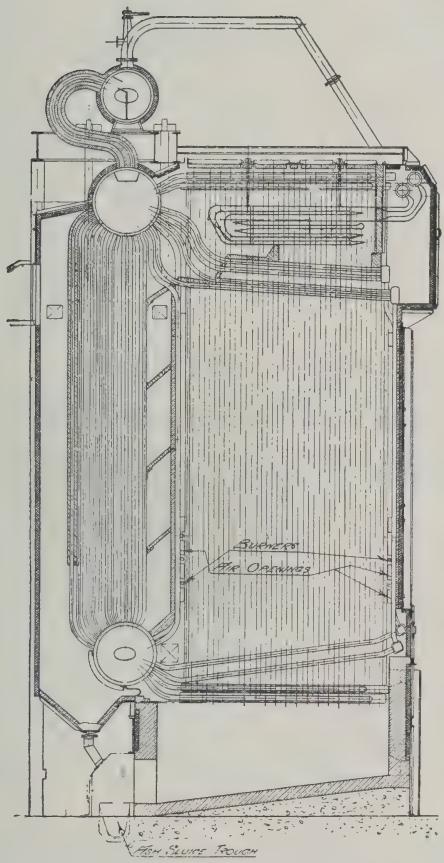


FIG. 25.—Standardised Combustion Steam Generator.
(*International Combustion, Ltd.*)

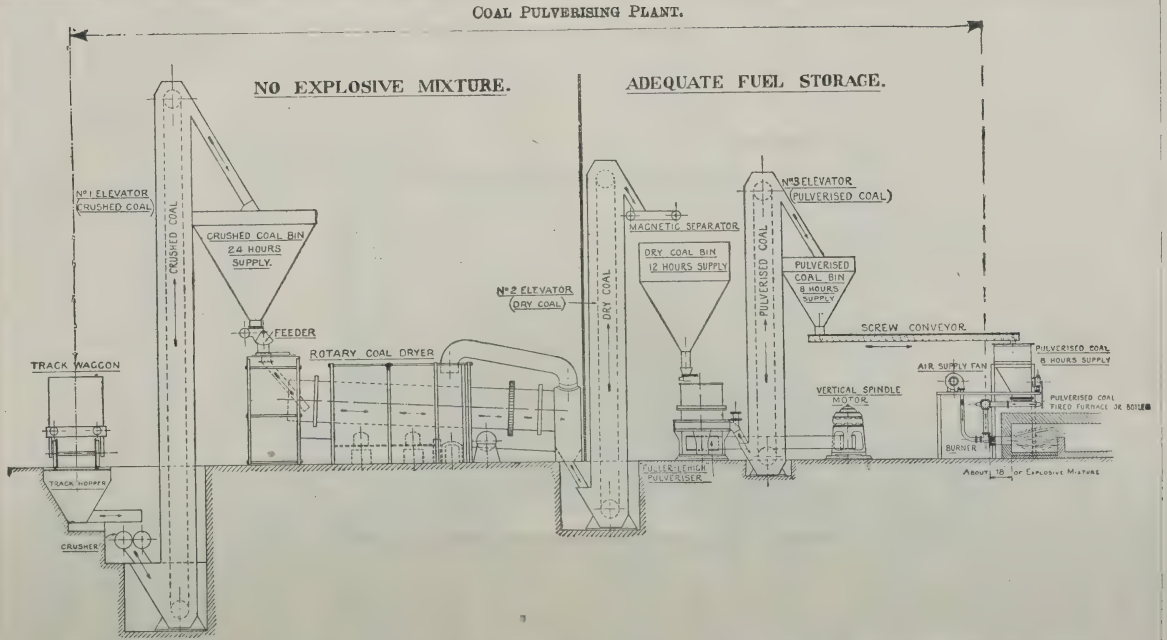


FIG. 26.—General View of 'Central' Pulverised Coa Plant, showing flow of coal from the Track Wagon to the furnaces, and indicating the positions of the Coal Crusher, Elevators, Coal Dryer, Pulveriser Mill, Magnetic Separator, Screw Conveyor, Furnace or Boiler Bin, and Screw Feeder. In place of the Screw Conveyor either the Fuller-Kinyon Pump, Fig. 27 (p. 2130), or the air-pressure system is now generally adopted for long-distance transmission.

The application of this dryer does away with a considerable portion of conveying machinery, for handling both the raw coal and the dry coal at the same time, eliminating expensive plant and fuel used to dry coal. As much as 2 to 3 per cent. of the coal purchased is sometimes employed in drying fuel in revolving dryers, with the static dryer this coal is replaced by waste gases from the boilers. On the Continent the revolving disc, grid, or shelf type of dryer heated by superheated steam is favoured.

Screen separator mills, such as the 'Fuller,' made by the Fuller Lehigh Company, have been used very extensively in the cement industry for pulverising the coal and for grinding the cement clinker. In this type of mill the fine particles of coal are thrown tangentially through the mill screen; the fine product is then raised by means of No. 3 elevator to the powdered coal bin, or it is delivered direct into the screw conveyor or into a Fuller-Kinyon type of pump, for transport to the bins at the furnaces.

In the air-separator type of mill the fine coal-dust is exhausted through a fan and deposited by means of the upward air current into the cyclone collector, whence it is delivered into the powdered coal storage bin, or direct on to the screw conveyor or pump.

The power required for operating a screen separator mill with its pulverised fuel (No. 3) elevator is somewhat less than the power required for operating a screen separator mill and exhaust fan. The wear and tear on the latter may be excessive when abrasive hard coal is pulverised, but the vacuum extraction of the coal-dust from the mill tends towards a cleaner mill-house.

Against the screen separator mill may be placed the occasional damage to the screen by a stray piece of iron. Any perforation of the screen due to this cause permits coarse coal to escape with the pulverised product.

Both types are in general use, and at the present time the advocates of one are as numerous as the advocates of the other.

Hardinge ball mills in conjunction with Hummer screen separation are also in use for pulverised coal production.

Transportation of Pulverised Coal.

The original method of transporting coal-dust was by means of weathertight screw conveyors. This method, fig. 26, although used to some extent for small and compact installations, has given place to air pressure and pump transmission systems for larger plants or greater distance delivery.

The Quigley Furnace Specialities Co. introduced an excellent system of transporting the coal-dust from 'blowing tanks' situate at the mill house. The coal-dust is forced from these tanks by means of compressed air applied to the top of the coal, and the coal is forced through 3-in. or 4-in. pipes to the bins at the furnaces, and over distances up to ½ mile.

Semi-automatic solenoid controlled switch valves are placed at the branches from the main supply pipe. This method of transportation is very compact, convenient, and successful in use.

In the Bonot 'air-transmission system,' the coal-dust is mixed at the mill house with approximately 50 per cent. of the air required for combustion. The mixture is forced through the supply pipes at a velocity of about a mile a minute, and ordinary branch pipes are taken to the furnaces. No furnace bins are therefore needed with this system. Care must be exercised in arranging this system to suit existing conditions, so that danger of accidental explosion taking place in conveyor mains may be realised and necessary precautions taken.

The Fraser & Chalmers air and coal-dust conveying system has been adopted for boilers and ranges of furnaces; its application to Lancashire boilers is illustrated under that heading.

The Fuller-Kinyon system of pumping powdered coal through supply pipes has now been developed by the Fuller Engineering Company, whereby a small quantity of air delivered by a pressure blower is pumped into the powdered coal to prevent 'packing,' so that the fuel can be transported long distances with a minimum of power. Ordinary pipes are used which can be laid underground or placed overhead in the same way as for water pipes.

For any one of these methods of conveying pulverised fuel the product must be dry, otherwise packing troubles will occur.

Feeders.

Whenever coal-dust is held in bulk at furnace bins, feeders are attached thereto fitted with screws or worms to introduce the dust into the air-blast as it enters the furnace, or compressed air is used as a means of syphoning out the fuel.

The screw or worm of the feeder is usually operated by means of a variable speed motor.

One of the best types of variable speed feeder-mixers forms the basis of the 'Lopulco' system of applying pulverised coal to boilers and other furnaces.

The Quigley * feeders have adjustable jaws to govern the supply of dust, and in this way constant-speed motors can be used.

---

* The Quigley, the Bonot, the Buell, and the Fuller patents and designs are now associated with Babcock and Wilcox, Ltd.

THE FULLER-KINYON PULVERISED COAL PUMPING SYSTEM

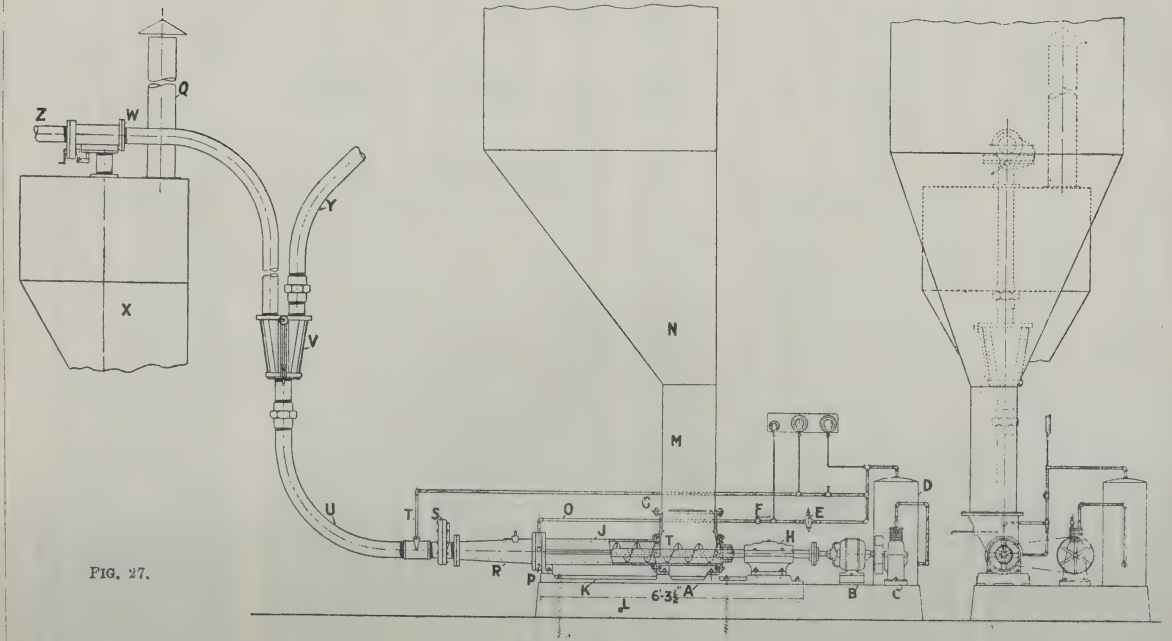


FIG. 27.

A, Fuller-Kinyon 6" Pulverised Material Pump. B, Electric Motor. C, Air Compressor. D, Air Receiver. E, Pressure Reducing Valve. F, Air Valve. G, Pump Hopper. H, Main Bearing. I, Forcing Worm. J, Pump Barrel. K, Pump Base. L, Foundation. M, Supply Spout. N, Material Bin. O, Aerating Line. P, Aerating Ring. Q, Vent Stack to Receiving Bin. R, Reducing Nozzle. S, Single Diverting Valve. T, Conduit Clearing Air Line. U, Convertor Conduit. V, Distributing Valve. W, Double Diverting Valve. X, Receiving Bin. Y, Branch Conveyor. Z, Continuation of Conveyor.

The Simon Carves table feeders have been used with considerable success. This type consists of a horizontal circular table with pockets pierced to take the fuel supply. The rate of rotation is controlled by friction concentric disc drive, and constant speed motors can be employed. On the same principle a single Bailey feeder can be used over a range of 1,000 lbs. to 10,000 lbs. per hour for the supply to any number of burners.

In the 'air-transmission system' no feeders at all are required at the furnaces.

Burners.

The usual type of coal-dust burner met with in general practice has been of very simple construction, merely consisting of an air-carrying supply pipe into which coal dust is fed by the screw feeder. These simple types of burner have given fairly good results, but later practice has established the necessity for a more intimate mixing of the air and powdered coal.

Many special burners have therefore been devised whereby a pre-mixing of coal dust and air is effected before the final stream of air for combustion is introduced.

Exceptionally good results have been recently obtained with the turbulent mixing types of burners, such as the Peabody, the Buell, the Bailey-Tenney, Couch, Fraser & Chalmers, the International Co.'s Type 'R,' and the Clarke Chapman burners, to effect a short diffused flame.

In each type a well-conceived combination of air and coal passages with adjustable or deflection plates or vanes renders it possible to control the direction, length, and intensity of flame.

By reason of the thorough mixture of air and coal, pulverised coal can be applied direct to the fire tubes of Lancashire and Cornish boilers without the use of external combustion chambers, unless high ash coal is used, when it is preferable to collect much of the ash prior to its passage through the boiler tubes. These burners also eliminate the extensive furnace alterations required for long flame burners.

No difficulty is experienced in burning low ash coal (10–15 per cent.) without external combustion chambers.

By reason of wide diffusion of the air and coal-dust mixture, the areas of combustion chambers usually provided for hand or stoker firing are in most cases sufficient.

An actual example of results obtained with the 'Buell' burners, as applied to the stoker settings of a B. & W. boiler of 3,240 sq. ft. of heating surface, without external combustion chambers, is as follows :—

	Stoker Firing.	'Buell' Burner.
Coal—B.Th.U. per lb.	12,562	10,370
„ lbs. per hour	1,792	1,175
Water evaporated from and at 212° F. per hour	13,365	10,552
„ Per lb. of coal	7·4	9·0
Efficiency of boiler and furnace	57 per cent.	83 per cent.

Another method of liberating a very greatly increased number of B.Th.U. in a small combustion area has been developed by the Fuller Engineering Company in their 'Well'-type Burner, which consists of a rectangular chamber with pulverised coal burners at each angle, these producing a high degree of turbulence within the water-cooled burner chamber. (See Combustion Area, p. 2153.)

Power of Motors.

Mill House Plant (for 40 tons per 10 hours).

Preliminary crusher (not required with slack coal)	15 to 20 h.p.
Nos. 1, 2, and 3 elevators	7½ h.p. each.
Dryer	10 to 15 h.p.
Pulveriser mill, screen separation	60 „ 75 h.p.
„ air „	80 „ 100 h.p.

(Mill capacity, 4 to 5 tons per hour.)

Fuel Transportation (say 5 tons per hour, 300 ft. and 40 ft. lift).

*By screw conveyor	15 h.p. (motor).
†By air pressure	75 „ (compressor).
*By air mixture	60 „ (fan mixer)
*By screw pump	20 „ (screw pump and compressor).

* Continuous operation. † Intermittent operation.

(For an approximation of power used, see note on page 2132, 'Progress in America.')

Coal-dust Feeders.

Constant or variable speed motors . . . . . 1 to 3 h.p.
($\frac{1}{2}$ h.p. motors are sometimes used, but are generally unsuitable under mill conditions.)

Air Fans.

No accurate figure can be given for the power required for air fans, as this will depend upon the size of burner used. The total quantity of air per pound of coal allowed is usually 180 cu. ft., 60 to 100 cu. ft. being supplied at a pressure of 3 or 4 ozs. per square inch; the remainder is induced at the burner when furnaces operate under induced or chimney draught.

Pre-Gasification of Coal.

Much of the success obtained in the various applications of pulverised coal has been due to the fine and uniform pulverisation of the coal, and the provision of combustion chambers of considerable cubic area.

Until the advent of the turbulent mixing type of burner, the latter requirement precluded the development of pulverised coal-firing for such purposes as marine boilers, Lancashire boilers, locomotives, and the smaller range of trial furnaces.

A greater measure of success is still to be obtained by subjecting pulverised coal to temperatures n the neighbourhood of 1,000° C. at high velocity, and with an air supply sufficient only to maintain this refractory surface temperature, gas can be generated and delivered direct to furnaces or boiler firing chambers with little loss of the sensible heat.

Developments are now in train for the practical application of pulverised coal in this manner, and a measure of success has already been obtained. Due to the extraction of dust in the pre-gasification chamber, this system lends itself particularly to those applications where dustless conditions on the hearth or in the furnace are essential. Moreover, it will doubtless be possible to utilise such generators or pre-gasifiers with either liquid fuel or pulverised coal.

Progress in America.

The amount of pulverised coal used in the U.S.A. up to 1912, chiefly in the cement industry, was approximately six million tons per annum. In the copper, steel, and iron industries for power-house boilers and all other purposes, including the cement industry, the annual usage of coal in powdered form is now upwards of 50 million tons, following the adoption of pulverised coal-firing at the 200,000 kW. super-power station at Milwaukee. Many other new American power stations have been equipped for pulverised coal-firing, one of the latest stations being the 600,000 kW. station of the Union Electric Supply Co., at 54th Street, New York.

Some super-power stations in America and elsewhere, now operating with, or arranged for, pulverised coal-firing, and equipped with boilers of very considerable size, are tabulated below:

	Approximate Heating Surface of each Boiler. Sq. Ft.
Milwaukee Electric Railway & Light Co., U.S.A.	32,200
Allegheny Steam Heating Co., Pittsburg, U.S.A.	30,800
Ford Company, River Rouge, Detroit, U.S.A.	28,480
Detroit Edison Co. (Marysville), Detroit, U.S.A.	29,800
Detroit Edison Co. (Trenton Tunnel), Detroit (Projector), U.S.A.	28,720
Cleveland Electric Illuminating Co., Cleveland, U.S.A.	30,600
Providence Electric Railway Co., Boston, U.S.A.	12,600
Berlin Electricity Works, Rummelsburg, Germany	17,200
Duquesne Light Co., Brunots Island, U.S.A.	26,500
Western Pennsylvania Power Co., Springdale, U.S.A.	15,300
Metropolitan Edison Co., Middletown, U.S.A.	14,680
Union Electric Light & Power Co., Cahokia, U.S.A.	18,010
Union d'Électricité, Gennevilliers, France	20,200
Union d'Électricité, Vitry, France	16,670
Nippon Electric Co., Tokio, Japan	10,080
Hams Hall, Birmingham, England	20,000
County of London Electric Supply, Barking, England	16,510

At the Cleveland Electric Illuminating Company's plant, the coal, which is of high volatile content (33 per cent.), is pulverised to a fineness of 60 per cent. through a 200 mesh, and 85 per cent. through a 100 mesh screen. Operating power has been determined at 14·5 kW. for production, 1 kW. for screw conveyors, and 4 kW. per ton for feeders and air blowers. Ratio of furnace volume to effective heating surface of boiler is 0·85, or 0·95 to total heating surface, furnace volume being 29,150 cubic ft.

The rating of each boiler is 3,060 h.p., and each boiler operates at 225 per cent. rating on continuous load, and at 300 per cent. rating at peak load.

A special feature of this installation, as at other American power plants, is the automatic control of the fuel and air required for combustion, in direct relation to the load on the boiler, by means of the system perfected by the Bailey Meter Company of Cleveland.

Combustion Area.

The ratio of combustion area to boiler-heating surface at the Cleveland plant is referred to above. The ratio at the Cahokia power station is somewhat less, being 0·635 of the effective heating surface of the boiler.

American practice has established satisfactory results for boiler firing, when burning between 0·6 and 2·0 lbs. of coal (calorific value, 12,500 B.Th.U. per lb.) per hour per cu. ft. of combustion area. At normal steaming this means that in each cu. ft. of combustion area there can be liberated about 18,000 B.Th.U. per hour, and at maximum load about 25,000 B.Th.U.

In Germany, at a special pulverised coal research station, a definite figure for metallurgical high temperature work has been fixed at 33,500 B.Th.U. per hour per cu. ft. of total combustion area. The special Well-type burner recently adopted by the Fuller Engineering Company, referred to under 'Burners' (p. 2131), makes it possible to liberate a much higher number of B.Th.U. per cubic foot of combustion area per hour, and it is claimed that this type of burner, with pulverised coal, has been burned at the rate of upwards of 100,000 B.Th.U. per cu. ft. per hour.

On the above bases of B.Th.U. per cu. ft. of combustion area, calculations can be made for boiler and metallurgical work, for any quality of coal: the higher the ash content, the lower will be the B.Th.U. value per lb., and consequently the larger the combustion area necessary. The greatest obstacle to the general use of pulverised coal for industrial purposes has been the ever-increasing cubic areas of combustion chambers to such a pitch that prices are quoted for the latter which are half the total cost of boilers.

The development of the turbulent mixing types of burners has clearly shown that combustion chambers can be greatly curtailed when using burners giving wide flaring or diffusion effects. (See 'Burners,' p. 2131.) The advent of the 'Wood' boiler may alter the whole design of power-house boiler-house plants, but for ordinary furnaces and boilers the above statistics will still hold good.

Self-contained Pulveriser Units.

Great attention has been given to the perfecting of these small unit mills, which can be placed alongside the boilers or furnaces to be heated. So much progress has been made in design, reduction of power consumption, maintenance costs, etc., that their installation at French and German iron and steel works and at American power houses has been generally accepted. Whereas power consumption was anything up to 30 kW. per ton of coal, these units now operate at 12 to 15 kW. for similar fuel as burned at the furnace, including air supply. These units are also pulverising fuel containing up to 15 per cent. of moisture, but this moist fuel must necessarily be supplied direct to the furnace burners, and not stored in bunkers.

The figures given in the table on p. 2146 represent the capital outlay required for complete mill-house plants of various capacities. The initial cost of same, with transportation system, furnace bins, motors, feeders, etc., necessitates an expenditure which, in many cases, will or can only be contemplated when actual results have proved that the collective advantages and economies thereby introduced render such an investment wholly attractive.

For smaller fuel requirements and for testing pulverised-coal firing upon individual boilers or furnaces, a self-contained pulveriser unit can be conveniently installed. There are now many efficient machines of this type available, and built upon entirely novel principles is the 'Atritor' unit, which embodies the use of eddy currents of air within the grinding zone for the dual purpose of rapid agitation and deflocculation of coal particles and the exposure of maximum surface for driving off moisture. In this unit, therefore, is combined the operation of drying and pulverising.

The French 'Rex' machine is also novel in its provision of carborundum grinding elements, and also in its fuel and air regulation and totalising adjustments. The power consumption with this unit is also exceptionally low; 6 to 10 kW. per ton of bituminous coal.

Coal containing comparatively high percentages of total moisture—20 per cent. or so—can be pulverised in several of the well-known types of units, and through the medium of hot air, up to 500° C.; the resultant fuel powder will be practically bone dry. Coal containing 5 per cent. to 10 per cent. of moisture can be dried and pulverised with air at atmospheric temperature, the resultant degree of fineness depending upon the humidity in the atmosphere at the time of grinding.

By the installation of self-contained units the value of pulverised-coal firing for any specific purpose can be ascertained, so that the question of capital outlay upon a complete mill-house plant and attendant equipment can be based upon the results ascertained in this manner.

The makers of the well-known Bettington sectional water-tube boilers and self-contained pulveriser units have generally improved the construction of both these, so much so that the boiler should find an extended use, and the pulveriser, which is particularly silent and efficient, a useful and general application.

Self-contained pulveriser units have been widely adopted on the Continent, many electric light stations now being fitted with this class of equipment, and some details of the Clarke-Chapman 'Resolutor' Pulveriser unit are given below. This unit is very similar to the French 'Resolutor' unit in general use for boilers and metallurgical furnaces on the Continent. The German B. & W. unit and the Rema Mill are also extensively installed.

The Clarke-Chapman 'Resolutor' Pulveriser.

(Clarke, Chapman & Co., Ltd., Gateshead-on-Tyne.)

This pulveriser, see fig. 28, is a self-contained unit machine and obtains its product by the impact of renewable beaters attached to a rapidly revolving disc. The coal is fed into the machine from a hopper by a belt, the speed of which can be readily controlled. Entering the pulverising chamber the coal is caught up by the high-speed beaters, shattered to an extreme degree of fineness and

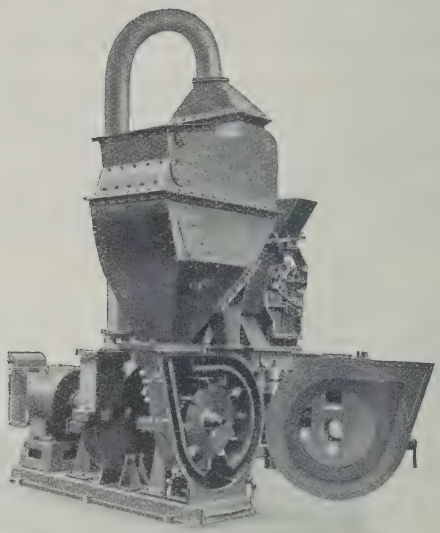


Fig. 28.

drawn up by means of the action of a fan into a separating chamber. In this chamber an adjustable deflector returns any coarse coal for re-pulverisation, whilst the fine material is drawn off into the firing circuit. The 'Resolutor' pulveriser in sizes ranging up to 10 tons per hour is now successfully firing water-tube boilers, Lancashire boilers, marine boilers (seven vessels are now equipped with the 'Resolutor' Pulveriser), and furnaces of all descriptions. This pulveriser is also in use for firing cement kilns and for powdering various materials for industrial purposes.

Marine and Lancashire Boilers.

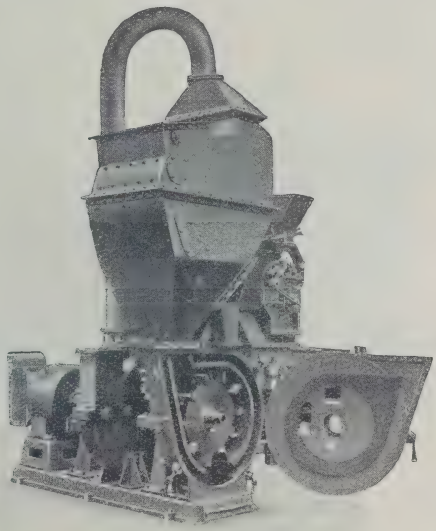
After years of research and improvements in methods of production and application of pulverised fuel to meet the exacting conditions of firing within the confined areas and chilled surfaces of marine and other fire-tube boilers, these great difficulties have been virtually overcome.

The original tests carried out by the United States Shipping Board and the running of the *Mercer* with the Peabody pulverised coal system as an experimental ship has resulted in serious current attention to marine applications.

PROGRESS OF THE RESOLUTOR PULVERISER

The 'Clarke Chapman' System of Powdered Fuel Firing is making rapid strides. During the last twelve months the results obtained in the firing of Lancashire Boilers, Water-tube Boilers, and Marine Boilers, enable us to definitely state that this method of firing is now established as sound engineering practice, and is no longer in the experimental stage

We invite you to send for more detailed information



THE RESOLUTOR FULVERISER

CLARKE CHAPMAN & CO. LTD

VICTORIA WORKS

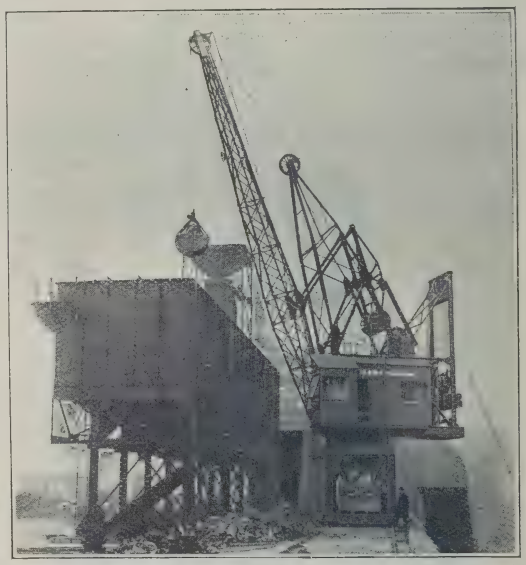
GATESHEAD

TEL. NOS: GATESHEAD 72271 (6 Lines)

TEL. ADD: "CYCLOPS, GATESHEAD"

LEVEL LUFFING WITHOUT ROPES

A Recent Departure in Crane Design



The above photograph shows one of our Level Luffing Cranes which was recently installed at the Deptford Wharf of a well-known Bunkering Firm.

The New Level Luffing Gear dispenses with ropes, ensures perfect balance and control, gives high speed of working and ensures perfect safety.

Write for our Crane Catalogue Number 78.

CLARKE CHAPMAN & CO. LTD

VICTORIA WORKS

GATESHEAD

TEL. NOS: GATESHEAD 72271 (6 Lines)

TEL. ADD: "CYCLOPS, GATESHEAD"

F. 2135

It has been sufficiently proved that ordinary bunker coal carrying up to 12 per cent. free moisture can be dried and pulverised on board by means of ball mills or impact mills and thereafter evenly distributed to the burners at the furnaces. It is probable that the introduction of small capacity storage bunkers for pulverised coal will be eventually adopted in preference to direct supply from the pulverisers.

The outstanding features of pulverised coal firing are, briefly: maintained speed of ship on full pressure steam by reason of 'no cleaning of fires,' small-grade bunker coal at lower cost, easier application of self-trimming bunkers with small coal, stokehold labour and conditions in every way parallel to oil firing, the lower cost of coal as pulverised and burned as compared with oil.

Excepting only for short sea routes, ferry services, and harbour craft, the question of bunkering pulverised fuel (which requires 30 per cent. increased bunker space) in the form of non-spontaneous or carbonised residue coke, is impracticable and attendant economies have as yet to be demonstrated even for favourable applications.

In the following table certain available data are set out concerning four ships already in service. Others are under consideration.

Name of Ship:	s.s. *Hororata.*	s.s. *Metz.*	s.s. *Berwindlea.*	s.s. *Mercer.**
Shipping Line, or Owners:	New Zealand Shipping Co.	Blue Star Line, Ltd.	Berwindlea Steamship Co.	U.S. Shipping Board, leased to Consolidation Navigation Co. (Oriole Line)
Tonnage . .	11,240	—	7,500	6,219
Trade route .	London – New Zealand *via* Panama	Towage on Rhine	U.S.A. and West Indies	New York–England
Name of system .	Howden-Buell	'Resolutor' pulveriser	Clarke Chapman	Peabody
Equipment supplied by	R. & H. Green and Silley, Weir, Ltd., London	Fours & Appareils Stein	Clarke Chapman & Co., Ltd., Gateshead	Peabody Engineering Corporation, New York
First trip on pulverised coal	December 22, 1928	—	August 15, 1929	November 1927
Trips or miles to date †	Over 40,000	—	Over 6,000	Over 60,000
Number and type of boilers in ship	6 Scotch marine 4 furnace 17′ 0″ × 12′ 0″	2 S.M. single-ended	2 S.M. single-ended	3 S.M. 3 furnace
Boilers pulverised coal-fired	3 Scotch marine	2 S.M. single-ended	2 S.M. single-ended	3 S.M. for pulverised fuel and oil
Heating surface to combustion area. Ratio	1 cu. ft. comb. vol. to 5 sq. ft. H.S.	—	—	—
Maximum fuel burned per hr. per cu. ft. Comb. area	2·88 lbs.	—	4·2 lbs.	—
Maximum fuel burned per hr. per boiler	1,870 lbs.	—	1,344 lbs.	1,950 lbs.
Number, type, and rating of pulverisers	3 ball type, 2,000 lbs. per hr.	2 'Resolutor' each 2 tons	2 'Resolutor' (1 per boiler), each 1 ton per hr.	2 ball mills, each 3,600 lbs. per hr.
Method of driving pulverisers	Steam turbines	Steam turbines	D.C. electric motors	Non-condensing steam engines

* This ship has been treated largely as a pioneer experimental ship.
† As at September 1929.

Name of Ship :	s.s. *Hororata.*	s.s. *Metz.*	s.s. *Berwindlea.*	s.s. *Mercer.**
Shipping Line, or Owners :	New Zealand Shipping Co.	Blue Star Line, Ltd.	Berwindlea Steamship Co.	U.S. Shipping Board, leased to Consolidation Navigation Co. (Oriole Line).
Maximum moisture in raw coal	7% New River coal, Panama	12%	12%	—
Method of drying coal	Heated primary air through pulverisers	Heated air through pulverisers	Heated air through pulverisers	Heated primary air through pulverisers
Method of fuel distribution	Howden - Buell patent	Ring main	Ring main	1 Peabody patent mechanical 2 patent pneumatic
Number, rating, and type of burners per boiler	Four 480 lbs. per hr. Howden - Buell patent	1 each furnace Woodeson-Stein	1 each furnace Woodeson patent, 560 lbs. per hr.	3 Peabody patent Combined coal and oil burners
Total power per ton of coal conveyed to burners	18 kW.	15 kW.	15 kW.	—
General results:				
Quantity, economy in fuel	10%	20%	—	—
Cost, economy in bunkers	10%	—	New ship, 1·277 lbs. coal per i.h.p. hr.	—
Labour economy in stokehold and trimming	50% (firemen)	—	Self - trimming bunkers. One man only required in stokehold for 6 furnaces ; no trimmers	A saving of 6 firemen per 24 hrs. as compared with hand-firing
Special consideration applying to the system	Suitable for usual run-of-mine coal as bunkered for hand-firing at coaling ports	—	—	—

* This ship has been treated largely as a pioneer experimental ship.

The Brand powdered fuel system has been installed in the s.s. *Swiftpool*, and she has completed her maiden voyage, but data are not yet forthcoming as to performance.

The Peabody Engineering Corporation has also equipped the s.s. *Lingan*, which has run twenty trips between Sydney, N.S., and Montreal. A vessel of 4,677 tons. Equipment approximately the same type as used in the s.s. *Mercer*.

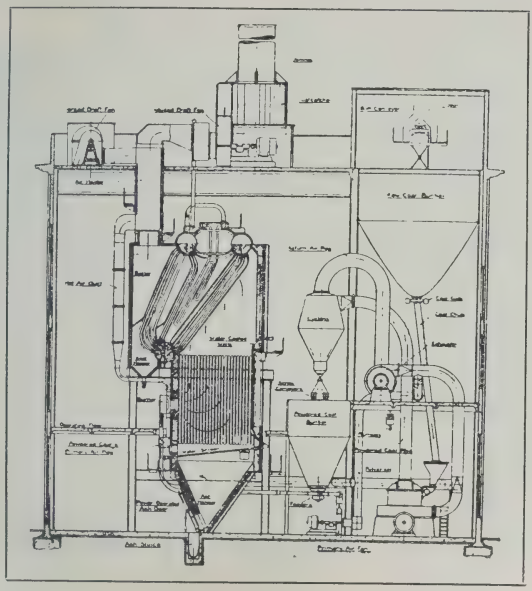
The tug *Tamaqua* was also fitted with this system in April 1929. A vessel of 594 tons. Max. fuel per boiler per hour, 1,000 lbs. 2 S.M., 2 furnace boilers, and 2 impact pulverisers driven by steam turbines.

From now on it will become an increasing practice to fire marine and Lancashire types of boilers in this manner.

The Clarke-Chapman Powdered Fuel system has been installed, after the successful performance of the s.s. *Berwindlea* on two cargo vessels owned by a Liverpool firm. In addition to this the ' Resolutor ' Pulveriser is in successful operation on the Continent on the s.s. *Stassfurt*, the s.s. *Donau*, the tug boat *Metz* and the dredger *Basure de Baas*. An illustration of the *Berwindlea* equipment is given in fig. 29.

The result of the first fourteen months cargo running of the s.s. *Berwindlea* is of importance in that it shows over a period of 57,230 miles an average coal consumption of 1·28 lbs. per i.h.p. per hour and that it reveals an extremely long period in which the powdered fuel plant gave complete satisfaction.

The Clarke-Chapman Ash Ejector Plant ' Woodeson ' patent has been successful in solving the ash problem on several vessels, including the s.s. *Berwindlea*.



THE LOPULCO

System of Pulverised Fuel Firing offers many advantages———

F. 2136

Advantages obtained by installing

THE LOPULCO PULVERISED FUEL SYSTEM

1. Removal of all moving parts from the combustion zone.
2. Increased operating periods without interruption.
3. Greater flexibility of plant.
4. Decreased coal costs per unit of electricity generated.
5. Effective control of combustion conditions.
6. Reduced overall maintenance costs.
7. Higher daily and annual operating efficiencies.
8. Increased evaporative ratings.
9. Rapidity of raising steam to meet sudden demands.

INTERNATIONAL COMBUSTION
LIMITED
ALDWYCH HOUSE, ALDWYCH, LONDON, W.C. 2
WORKS: DERBY

F. 2137

Other types of plants for Lancashire boilers are shown in figs. 34-36, p. 2147. The Fraser & Chalmers systems, air and coal conveyed through a supply main, and direct firing unit mills, are illustrated. As against hand-firing these installations have shown increased steaming 20 per

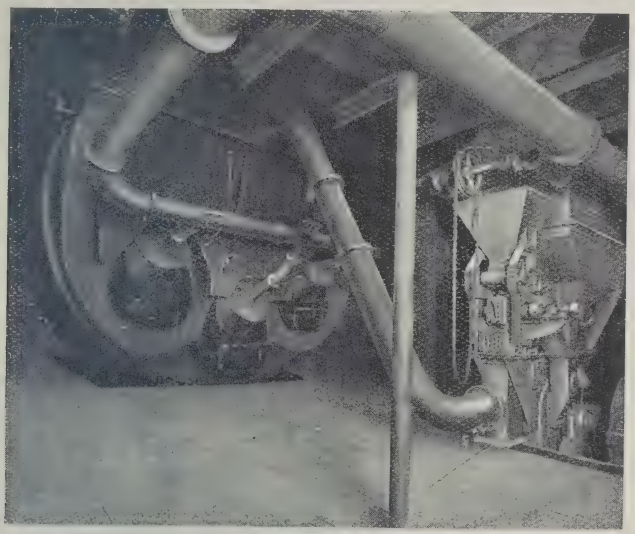


FIG. 29.—View of Stokehold of S.S. *Berwindlea*, showing one of two Marine Boilers fired with Powdered Fuel by 'Resolutor' Pulverisers.

cent., a fuel economy of 20 per cent., steady steaming with inferior coal. Three boilers will supply the steam from four hand-fired boilers. Ash is slagged and run off in the slag pan shown, and ash in the tubes blown out by means of steam jets.

Another method is the pre-gasification system, with which a small external combustion chamber or pre-gasifier is used (see fig. 30); some success has been recorded with this system at a British colliery plant.

THE LOPULCO SYSTEM.

(International Combustion, Ltd., London.)

A typical arrangement of 'Lopulco' Boiler and Pulverised Fuel Plant is shown in fig. 31.

In such a system the raw coal, in pieces of a size which will pass through a 1½-in. diameter ring, is passed through a Raymond Pulveriser which has 4 to 6 pendulum rollers running inside a grinding ring.

The coal is fed by means of a feeder to the grinding chamber, where it is picked up by rotating ploughs and thrown between the rollers and the grinding ring; as soon as it is reduced to the required fineness, it is separated from the coarser particles by a current of air drawn through the pulveriser by an exhauster fan. This air carries the fine particles to a cyclone collector, in which the coal is separated from the air, this latter returning to the pulveriser by way of the exhauster fan.

The degree of fineness is adjusted to suit the amount of volatiles contained in the fuel; thus bituminous coals having a high percentage of volatiles do not require to be ground so fine as semi-anthracite coals, which have low volatiles. A suitable condition of fineness for good bituminous coal is such that 90 per cent. will pass through a 100-mesh sieve, and also that no particles are left on a 40-mesh sieve.

The coal is dried in the mill circuit by means of hot air drawn from the hot air duct of the boiler; the same air is used for extracting the pulverised fuel from the mill. The surplus air from the mill circuit is taken off at a suitable point and supplied (as primary air) for combustion in the boiler by the primary air fan, which delivers it via the feeder to the burner. The burner is of a horizontal type, specially designed to give a turbulent uniform mixture of coal and air to the furnace.

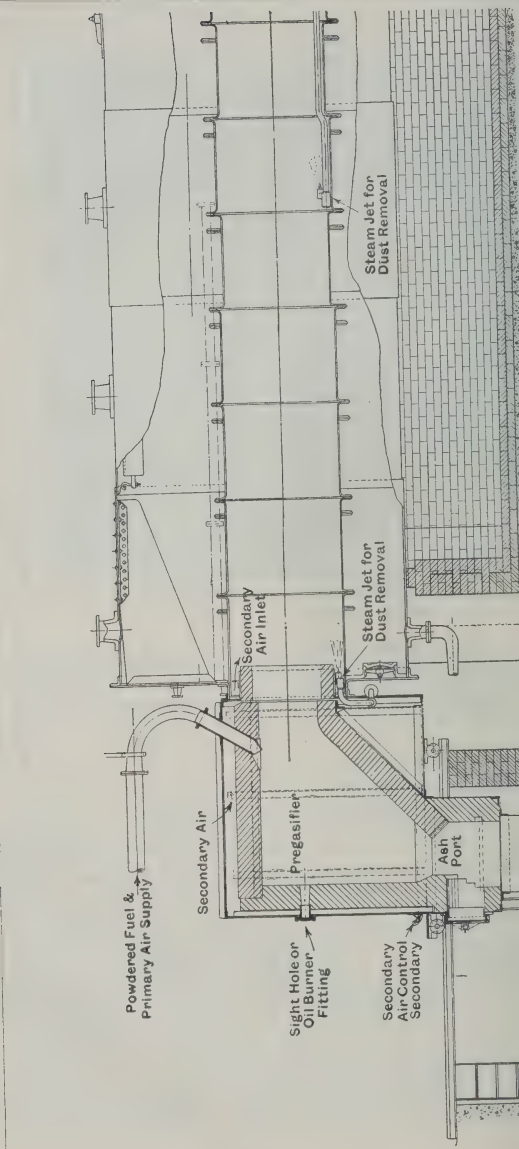


FIG. 30.—Arrangement of Lancashire Boiler with Pre-Gasifying Chamber (Gasified Fuel Co.'s System).

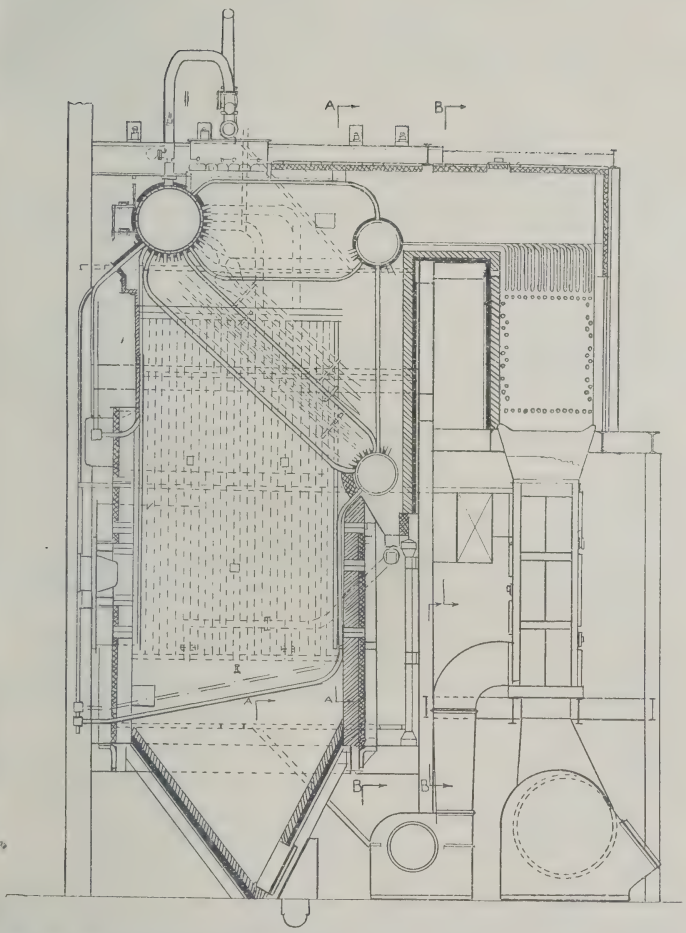


FIG. 31.—Cross-Section of Lopulco Boiler, fired by the Lopulco Pulverised Fuel System, at the Leicester Corporation Electricity Power Station.

POWER-HOUSE CONTRACTS TO UTILISE PULVERISED FUEL ON THE LOPULCO SYSTEM IN GREAT BRITAIN.

(International Combustion, Ltd., London.)

Power House.	Contract Date. (Received.)	Number of Boilers.	Pressure, lbs. per sq. in.	Heating Surface of Boiler, sq. ft.	Heating Surface of Water-Screens, sq. ft.	Total Heating Surface.	Number of Pulverisers.	Number of Rollers per Pulveriser.	Number of Burners per Boiler.
Birmingham Corporation *	1924	6	320	5,400	154	33,324	4	6	4
County Boro. of Derby*	1924	2	300	8,850	148	17,996	2	6	6
Met. Boro. of St. Pancras (1st Order) *	1924	2	220	6,546	159	13,410	2	5	5
Met. Boro. of St. Pancras (2nd Order) *	1924	2	220	6,070	630	13,400	2	5	5
Met. Boro of Poplar *	1924	3	300	6,200	90	18,870	2	6	5
York St. Flax Spinning Co., Ltd.*	1924	1	250	9,270	171	9,441	1	4	4
Synthetic Ammonia and Nitrates Co., Ltd. (1st Order) *	1924	2	275	9,000	500	19,000	2	6	5
Synthetic Ammonia and Nitrates Co., Ltd. (2nd Order) *	1925	4	275	14,500	940	61,760	6	6	9
North Metropolitan Supply Co., Brimsdown †	1925	2	170	1,900	492	4,784	1 (unit)	—	2
North Metropolitan Supply Co., Brimsdown *	1925	5	325	10,100	900	55,000	3	6	8
North Met. Supply Co., Willesden *	1924	1	200	7,200	220	7,420	1	6	6
Ashington Coal Co. Ltd.*	1925	4	200	5,904	139	24,172	2	5	4
Shell Mex, Ltd., Shell Haven, Essex	1926	3	160	7,320	189	22,527	3	6	4
County of London Electric Supply (Barking)	1926	10	375	16,510	1,784‡	182,940	10	6	10
West Midland Joint Electricity Authority, Walsall,	1926	3	305	8,560	530	27,270	6	(unit)	2
Hams Hall, Birmingham (60,000 kW. station)	1927	5	375	20,000	2,398‡	111,990	25	(unit)§	5
Derby Corporation, Electricity Dept.*	1927	1	300	8,851	610	9,461	2	6	6
Taylor Bros. Ltd., Manchester *	1927	2	250	7,400	260	15,320	—	—	1
Brighton Corporation, Electricity Dept.*	1927	2	350	2,230	1,810	8,080	1	6	1
Leeds Corporation, Electricity Dept.	1928	3	490	16,540	1,450	53,970	3	6	3
Leicester Corporation, Electricity Dept.*	1928	2	375	12,960	1,795	29,510	3	6	3
		1	375	16,620‖	1,680	18,300			
Met. Elect. Rly., Neasden *	1298	6	280	9,050	950	60,000	6	6	2
Synthetic Ammonia & Nitrates, Ltd.*	1928	6	800	31,043‖	2,265	33,308	10	—	4
Derby Corporation Elect. Dept.	1928	2	300	Combustion Steam Generator		3,118	1	6	1-vertical 2-'R' 1-'R'
Poplar Borough	1928	3	320	7,530	743	8,273	1	3	
Bolsover Colliery Co. Thoresby Pit	1929	3	300	3,000	304	3,304	3	(unit)	
Nth. Met. Supply Co. Brimsdown	1930	4	325	Combustion Steam Generators		—	8	(unit)	12-corner fired

* New boilers. † Old boilers. ‡ Including water-cooling system for side and rear walls.
§ Unit Pulverisers referred to above are the International Type with the exception of Hams Hall, Birmingham, which are Atritor Unit Mills.

The secondary air for combustion is introduced to the burner box direct from the hot air ducts of the boiler and is controlled by means of suitable adjustable dampers around the outside of the burner casting. With the exception of the front burner wall, which is constructed of high-grade refractory material, the combustion chamber walls are constructed of water-cooled tubes of the Murray fin type; these are 4 in. in diameter fitted with a pair of diametrically opposed fins welded on, grouped together and connected to headers at each end, and so coupled to the boiler drums as to be in direct circulation with the boiler unit.

Across the bottom of the combustion chamber there is a screen of water tubes, also forming part of the boiler circulation, which are of seamless steel expanded into forged steel headers. The particles of dust which drop from the flames pass through the tubes and are thus chilled below melting point so that they fall into the ash hopper and are easily removed as fine, dry particles. The following figures give the power required for a 'Lopulco' equipment to deal with 30,000 lbs. of pulverised fuel per hour:

Motor for Raymond Pulveriser	110 kW.
Motor for fan pulveriser	45 kW.
Duplex feeders	4·5 kW.
Motor for primary air fan	32 kW.

This works out at about 15 kW. per ton of pulverised coal burned per hour, which, at ½d. per unit, is 7·5d. per ton; thus, about 1 per cent. of the total steam production is required to work the pulverising plant.

Pulverised Coal for Locomotives.

The advantage for pulverised coal, lignite, etc., as a locomotive fuel are—

30 to 40 per cent. fuel economy as against hand-firing, running mileage and cylinder power of engines increased, cost of firing tools and repairs to fire grating eliminated, no cinders or sparks from the smoke stack, reduction of fire-box strains, firing conditions with pulverised-coal firing analogous to oil-firing with similar fuel-bunkering facilities. Any local fuel within reason can be used.

The following fuels have been and can be used:

	A	B	C	D
Chicago and North-Western Railway.				
	Per Cent.	Per Cent.	Per Cent.	
Illinois bituminous	3·18–15·36	10·00	34·47	10,720–12,400
Kentucky bituminous	1·9 – 2·8	8·00	36·54	13,964
North Dakota lignite	1·8	9·32	47·25	10,960
New York Central Railway.				
Pennsylvania bituminous	0·51– 0·95	8·94–13·21	21·63–31·35	13,671–14,096
Brazilian bituminous	1·73– 9·15	23·14–29·33	9·5 –28·04	8,820–10,177
The Delaware and Hudson Railway.				
Pennsylvania anthracite culm	0·5	12·22	8·30	12,000 (average)
,, bituminous	0·5	9·0	33·0	13,750

A. Range of moisture. B. Ash. C. Volatile matter.
D. Calorific value. B.Th.U. per lb. (dry).

On the Lehigh Valley Railway a locomotive has been run on a mixture of 55 per cent. raw anthracite silt and 45 per cent. bituminous coal. Full steam pressure is maintained easily at all times, and the anthracite silt contains 38·42 per cent. of ash, and has a B.Th.U. value of 9,675, while the soft coal which is mixed with this has an ash content of 11·6 per cent. and a B.Th.U. value of 12,650. Analyses of these coals are as follows:—

Anthracite Silt.	Per Cent.	Soft Coal.	Per Cent.
Moisture	2·26	Moisture	1·90
Volatile	10·39	Volatile	36·59
F.C.	49·30	F.C.	49·89
Ash	38·42	Ash	11·60

The raw coal and lignite to be burned on locomotives in Italy has the following analysis figures representing the limits in each case are given :—

	Per Cent.		Per Cent.
Moisture .	. 12·52 to 26·80	Volatile .	. 31·5 to 60·94
Ash .	. 11·6 to 32·60	Carbon .	. 35·26 to 68·50
	B.Th.U. values from . . .		3,640 to 10,440

LOCOMOTIVE EQUIPMENT.

A brief specification of the locomotive equipments (Fig. 33) supplied by the Fuller Engineering Company of America is given below:—

Special cast-iron coal hopper base to receive duplex feed screws. Vertical marine-type steam engine for operating same, with intermediate connections, including gears, gear shafts, shafting, etc.

Steam turbine driven fan for conveying the coal dust from the feeder to the tender.

Flexible steam connections between engine and tender, as well as flexible connections for transporting the pulverised coal between engine and tender.

Burner, with necessary dampers and connections for operating same.

3,000° F. pyrometer for indicating fire-box temperatures. Electric revolution counter for indicating speed of engine driving coal-feed screws. Revolution counters for each pair of feed screws, gauges, and valves for operating this equipment.

Mechanism for operating the slag pan or slag pans, as well as mechanism for operating the air dampers in the sides of the combustion chamber beneath the fire-box.

The equipment is started and coal supply commenced with a steam pressure of from 0 to 30 pounds.

Developments in this direction in Germany have been carried out with the 'Bauart' and the 'A.E.G.' patents. The latter embodies a novel burner arrangement in that lateral louvred and water-cooled burners are fitted one on each side of the fire-box. These are clearly shown in fig. 25, p. 623, in 'LOCOMOTIVE' section. Experiments on a freight locomotive have offered sufficient encouragement for the construction of further engines on this principle.

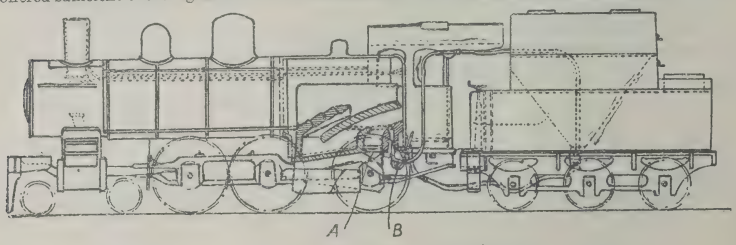


FIG. 32.—Pulverised Peat Fired Locomotive on Swedish Railways.
A. Hand-fired Coal Grate. *B*. Peat and Air Mixture Inlet.

An article by *Bureau Engineer*, Carl Flodin (Centraltryckeriet, publishers, Stockholm, 1916), includes full data regarding tests made with peat-fired eight-coupled freight locomotives: tractive power, 9,000 kg.; weight, 51,000 kg.; tender, 36,000 kg.; water capacity of tender, 14,000 kg.; pulverised peat, 4,000 kg.

It was found on these tests, for equal boiler duty, that 1 kg. of coal was equal to 1·4 kg. of pulverised peat, the coal used having a calorific value of 7,240 cals. and the peat as used 4,400 cals., or approximately 13,000 and 7,900 B.Th U. per lb

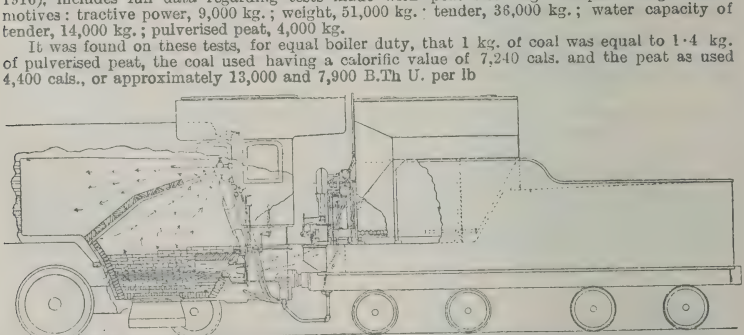


FIG. 33.—Fuller Equipment as fitted to Main Line Freight Locomotive, showing Fuel Hopper, Control Equipment, Burner, and Combustion Chamber.

The arrangement of the firebox setting, fig. 32, allowed for a small grate on which coal was burnt under the incoming mixture of powdered peat and air. On this small hand-fired grate 3 to 4 kgs of coal were consumed per 100 kg. of peat; the analysis of peat and coal was as follows:—

	Powdered Peat. Per Cent.	Coal. Per Cent.
Carbon	47·0	75·5
Hydrogen	4·5	4·4
Oxygen	29·5	8·6
Sulphur	0·5	1·5
Nitrogen	1·1	1·2
Ash	3·2	6·2
Moisture	14·2	4·6
	100·0	100·0

Analogous tests made with coal-fired and powdered peat-fired locomotives gave the following overall boiler efficiencies and fire-box temperature of gases:—

	Efficiency. Per Cent.	Temperature.
Hand coal-fired	65	1,510° F.
Pulverised peat-fired	73	1,670° F.

Recorded and corrected figures for parallel trips with coal- and peat-fired locomotives are given, the following being the more important:—

Parallel Test Run.	Trip Number.	Weight of Loco., Tender and Wagons.	Wagons.	Average Speed, Km. per Hr.	Consumption of Fuel Kg. per 1,000 Kg. Inclusive Loco. and Tender.	Smoke-box Temp.	Kg. Steam per Kg. Fuel.
Service.		Ton per 1000 Kg.	Ton per 1000 Kg.			° F.	
A. Freight	1. Pulverised peat	787	700	29·2	37·5	255	4·47
	2. Coal	782	700	27·5	25·6	290	6·83
B. Freight	3. Coal	782	700	28·7	27·7	312	6·62
	4. Pulverised peat	787	700	27·0	40·7	294	4·56
C. Passenger	5. Pulverised peat	387	300	42·3	56·9	289	4·52
	6. Coal	382	300	41·7	39·0	268	7·0

Many grades of coal (in South Africa, for instance) are considered unsuitable for use on locomotives.

That it is quite possible to eliminate a large quantity, if not the whole, of imported coal, or coal transported long distances by rail, is now being realised in various parts. Brazil has already supplanted imported railway coal to some extent by fuel which hitherto has been considered useless, and continuous extensions in this direction are to be made. The coal used in Brazil is of lower quality than the South African fuels, as the following comparative analyses of Brazilian and South African coals show:—

	Brazilian Coal, as used 'dry' in pulverised form for locomotive fuel.		South African Coal, Undried.		
			Durban.	Kroonstad.	Dundee.
	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.
Moisture	7·90	1·73	1·60	8·29	0·84
Volatile	28·04	9·50	27·10	31·27	25·30
Fixed carbon	34·73	61·50	60·05	45·32	68·94
Ash	29·33	28·27	11·25	15·12	5·76
Sulphur	3·16	9·10	1·51	1·43	1·53
B.Th.U. per lb.	8,820	10,177	12,950	11,020	14,722
Evap. power	9·20	10·40	13·40	11·40	15·23

The Brazilian coal analysis represents the fuel in pulverised form after treatment in a rotary dryer. The fuel is of friable nature, and the ash of such form that when subjected to heat it flows round the lumps of fuel on a hand-fired grate, rendering it impossible to burn the coal. Kroonstad coal contains about the highest degree of moisture recorded—8·29 per cent.—whereas most South African coal contains but 1 or 2 per cent. of water, and for which drying plant would not be required. This is an important consideration.

Refractory Linings for Combustion Chambers.

The construction and composition of the linings of combustion chambers for pulverised coal fired furnaces are questions of considerable importance.

Although the experience gained of recent years has rendered it possible so to design combustion chambers that the refractory linings are not subjected to the severe destructive action from the flame and ash slag as heretofore, it is still highly desirable under all conditions to have a lining that will not only withstand high temperatures, but will at the same time resist the corrosive chemical action of the ash under these conditions.

In the case of high temperature metallurgical furnaces the factor of corrosion is one that must be carefully considered in view of the presence of metallic oxides introduced through the furnace charge.

Most coal ash consists chiefly of alumina, silica, and iron oxide, together with very much smaller quantities of five or six metallic oxides.

Metallic oxides are of primary importance, in view of the fact that many of them have a marked fluxing action on almost any refractory material. Thus, a fuel ash that contains much iron oxide has in itself a destructive action on furnace linings, irrespective of any of its other constituents.

Apart from this, some fuel ash contains a predominating amount of basic material, whilst in other cases there may be an excess of acid material. When a furnace is to be fired generally or entirely with a fuel containing a basic ash it is highly desirable to provide a basic lining, for by so doing much of the trouble and expense for renewal and repair of furnace linings will be avoided.

Preliminary analysis and investigation in connection with the composition of the coal ash and furnace linings will often save expense and prevent lining troubles during the subsequent operation of the plant.

The more common and cheaper grades of refractory bricks usually contain a more or less equal amount of silica and alumina, and the slight excess of one or the other should be taken advantage of. Combustion chambers of suitable design and having the requisite combustion area will then run satisfactorily with linings of ordinary firebricks.

If, however, there are exceptional conditions to contend with, such as high temperatures, confined combustion space, chemically active ash, etc., it is necessary to make use of the best grades of refractory materials.

In many cases carborundum (silicon carbide) can be used as a facing material for linings with good results. Carborundum, however, although generally chemically inert, will not resist the action of iron oxide or ash containing much potassium or sodium compounds.

In some few cases pure silica bricks can be used when the fuel ash composition is suitable and high temperatures have to be withstood, but special provision must be made for expansion of the bricks.

Pure alumina, magnesite, or bauxite are all high-grade refractories that may be used to resist high temperatures when ash or slag is of a basic nature.

Advantages of Pulverised Coal.

The chief advantages of using coal in pulverised form are:—

Utilisation of low grade coals, peats (coals with 30 per cent. and 40 per cent. ash); lignites, anthracite slush and pit washings can be used.

Control of flame temperatures with uniform heat regulation and greatly reduced requirement of attendant labour—in fact, labour required is equivalent to oil-firing practice.

Increased efficiencies of furnaces due to direct firing of the fuel in minute particles, causing instantaneous and complete combustion, without smoke production.

Some approximate savings in fuel consumptions that have been realised in practice are as follows:—

Boilers—80 per cent. efficiency, and savings of 15 to 50 per cent. over hand firing.

Open-Hearth Steel Furnaces—a saving of 300 lbs. per short ton of steel as against producer gas for small plants.

Puddling Furnaces—a saving of 1,500 lbs. per short ton of iron over hand firing, and from 300 to 500 lbs. over mechanical stokers.

Reheating Furnaces—20 to 40 per cent. saving over hand firing.

Heavy Forge Furnaces—25 per cent. saving over hand firing.

Light Forge Furnaces—20 to 40 per cent. saving over hand firing.

Annealing Furnaces—30 per cent. saving over hand firing.
Tinning and Galvanising—20 to 30 per cent. saving over hand firing.
Sheet and Pair Furnaces—30 to 40 per cent. saving over hand firing.
Copper Smelting Furnaces—40 per cent. saving over hand firing
Locomotives—30 to 50 per cent. saving over hand firing.
Cupolas—30 to 50 per cent. saving in coke and hotter iron.
Crucible (Brass Melting) Furnaces—The use of coal in place of metallurgical coke or oil.

The output from many of these furnaces has been greatly increased by reason of coal-dust firing, what are known as 'stickers' in the tin-plate trade having been reduced 60 per cent., whilst the quantity of tin and zinc used has been reduced considerably owing, in a great measure, to the improved surface of the metal, and also to the more even heating of the kettles.

Reduced oxidisation of steel and iron heated in powdered coal-fired furnaces and increased life of annealing boxes have accounted for an annual saving of many thousand pounds in some works.

Very important economies have been made in firing malleable iron melting and annealing furnaces with pulverised coal, so much so that The Grindle Fuel Equipment Co. of America have specialised exclusively on this work.

Mills and Millhouse Plant.

TABLE SHOWING NUMBER OF PULVERISER MILLS REQUIRED FOR INSTALLATIONS OF VARIOUS CAPACITIES, LABOUR HOURS, COAL DRYERS, ETC.

Daily Output in Tons.	Output per Hour in Tons.	Number and Size of Mills.		Number and Size of Dryers.		Number of men per Shift.	Labour. Hours per Day.	No. 8-Hour Shifts per working Day.	Maximum Output per Shift. Tons.	Approx. Output per Labour Hour. Tons.
		Running Plant.	Stand-by Plant.	Running Plant.	Stand-by Plant.					
		Ins.	Ins.	Tons.	Tons.					
5	½	1-24	none	1-4	none	1	16	2	4	0·32
10	2	1-33	"	1-4	"	2	16	1	16	0·62
15	2	1-33	"	1-4	"	2	32	2	16	0·47
20	2	1-33	1-33	1-4	"	2	32	2	16	0·625
30	2	1-33	1-33	1-4	"	2	32	2	16	0·94
40	2	1-33	1-33	1-4	"	2	32	2	16	1·25
50	4	1-42	1-42	1-4	"	2	32	2	32	1·55
60	4	1-42	1-42	1-4	"	2	32	2	32	1·87
70	4	1-42	1-42	1-4	"	2	32	2	32	2·19
80	4	1-42	1-42	1-4	"	2	48	3	32	1·63
90	4	1-42	1-42	1-4	"	2	48	3	32	1·87
100	6	3-33	1-33	1-8	"	2	48	3	48	2·1
150	8	2-42	1-42	1-8	1-8	2	48	3	64	3·1
200	12	3-42	1-42	1-10	1-10	3	48	2	96	4·2
250	12	3-42	1-42	1-14	1-14	3	72	3	96	3·5
300	16	4-42	1-42	1-14	1-14	3	72	3	128	4·2
350	16	4-42	1-42	1-14	1-14	3	72	3	128	4·9
400	20	5-42	1-42	2-10	1-10	4	96	3	160	4·2
450	20	5-42	1-42	2-10	1-10	4	96	3	160	4·7
500	24	6-42	1-42	2-14	1-14	5	120	3	192	4·2
550	24	6-42	1-42	2-14	1-14	5	120	3	192	4·6
600	28	7-42	2-42	2-14	1-14	6	144	3	224	4·2
650	28	7-42	2-42	2-14	1-14	6	144	3	224	4·5
700	32	8-42	2-42	2-14	1-14	6	144	3	256	4·9
750	36	9-42	2-42	4-10	1-10	7	168	3	288	4·5
800	40	10-42	3-42	4-10	1-10	8	192	3	320	4·2
850	40	10-42	3-42	6-10	1-10	8	192	3	320	4·4
900	40	5-57	1-57	4-14	1-14	6	144	3	320	6·25
950	48	6-57	1-57	4-14	1-14	6	144	3	384	6·6
1,000	48	6-57	1-57	4-14	1-14	6	144	3	384	6·9

In a properly designed plant, and with due attention to ordinary common-sense regulations, there is *no fear of explosion* from the coal-dust, and, as has been already stated, the old troubles with the ash slag have been overcome.

There are certainly many instances where the use of fuels in powdered form would show a handsome return for capital invested, and the following table will enable an estimate to be made as to economy that can be expected.

Approximate Cost of Complete Mill House, Plants and Operation Expenses.

If static dryers are installed the cost of plant can be reduced by 5 per cent. up to 350 tons, and by 7½ per cent. thereafter, and the cost for coal for rotary dryers is then omitted.

Daily Output in Tons.	Labour. Cost per Ton.	Power. Cost per Ton. *	Dryer Fuel, Cost per Ton. 2 per cent. of coal dried.	Repairs, Cost per Ton.	Without Stand-by Plant.			With Stand-by Plant.		
					Cost of Mill House Plant, including Building.	Interest per Ton at 6 per Cent.	Depreciation per Ton at 10 per Cent.	Total Preparation. Cost per Ton.	Cost of Mill House Plant, including Building.	Interest per Ton at 6 per Cent.	Depreciation per Ton at 10 per Cent.	Total Preparation. Cost per Ton.
	s. d.	*s. d.*	*d.*	*d.*	£	*s. d.*	*s. d.*	*s. d.*	£	*s. d.*	*s. d.*	*s. d.*
5	5 10	3 0	10½	4	3,500	2 9½	4 8	17 6	—	—	—	—
10	2 11½	2 10	10½	4	6,700	2 8	4 5½	14 1½	—	—	—	—
15	3 11	2 10	10½	4	6,700	1 9½	3 0	12 8½	—	—	—	—
20	2 11	2 10	10	4	9,300	1 9	2 10½	11 6½	10,300	1 11½	3 3	12 1½
30	1 11¾	2 10	10	4	9,300	1 3	2 1	9 3¾	10,300	1 4½	2 3½	9 7½
40	2 2	2 8	10	4	9,300	11	1 7	8 6	10,300	1 0½	1 9	8 9½
50	1 2	2 8	10	4	11,100	10½	1 6	7 4½	12,700	1 0	1 8½	7 8½
60	1 0	2 8	10	4	11,100	9	1 3	6 10	12,700	10	1 5	7 1½
70	10	2 8	10	4	11,100	7½	1 1	6 4½	12,700	8	1 2½	6 7½
80	1 1	2 6	10	3¾	11,100	6½	11	6 2½	12,700	7	1 1	6 5½
90	1 0	2 6	10	3¾	11,100	6	10	5 11½	12,700	6½	11	6 1½
100	10½	2 6	9¾	3¾	14,300	7	11⅛	6 0½	15,350	7½	1 0½	6 2
150	7	2 6	9¾	3¾	15,000	4⅜	8	5 3½	18,100	5¾	9½	5 6
200	5½	2 6	9¾	3¾	20,000	4⅜	8	5 1½	25,500	6	1 0	5 6½
250	6½	2 6	9¾	3¾	20,000	4	6½	5 0½	25,500	5	8½	5 3
300	5½	2 6	9¾	3½	26,100	4½	7	5 0	31,600	5	8½	5 2½
350	4½	2 6	9¾	3¾	28,300	4	6½	4 10½	32,000	4½	7½	4 11¾
400	5½	2 6	9¾	3¾	28,300	3½	5½	4 9¾	32,000	4	6½	4 11
450	4½	2 6	9½	3½	31,000	3½	5½	4 8½	35,500	3½	6½	4 10
500	5½	2 6	9½	3¾	31,000	3	5	4 8½	35,500	3½	5½	4 9½
550	4½	2 4	9½	3¾	32,000	2⅞	4⅞	4 5½	37,000	3½	5½	4 6½
600	5½	2 4	9½	3¾	32,000	2⅞	4½	4 5	37,000	3	5	4 6½
650	5	2 4	9½	3¾	32,000	2¾	4	4 4½	37,000	2⅞	4½	4 5½
700	4½	2 4	9½	3½	35,000	2¾	4	4 3¾	41,000	2⅞	4⅞	4 4¾
750	5	2 4	9	3½	37,000	2¾	4	4 4	42,500	2⅞	4⅞	4 4¾
800	5½	2 2	9	3½	43,000	2¾	4¼	4 2⅞	50,500	3	5	4 4
850	5	2 2	9	3½	45,500	2¾	4¼	4 2½	53,500	3	5	4 1
900	3½	2 2	9	3½	48,000	2¾	4¼	4 0⅝	54,000	2⅞	4¼	4 3½
950	3½	2 2	9	3½	49,500	2¾	4¼	4 0½	55,200	2⅞	4¼	4 1½
1000	3¼	2 2	9	3½	53,500	2½	4¼	4 0⅜	59,500	2¾	4¼	4 1¼

Cost of plant exclusive of foundations and erection.

* Can be reduced to ⅓rd for power-station plants.

Table of costs is based on the following assumptions :—

Labour at £4 per man per 44-hour week.

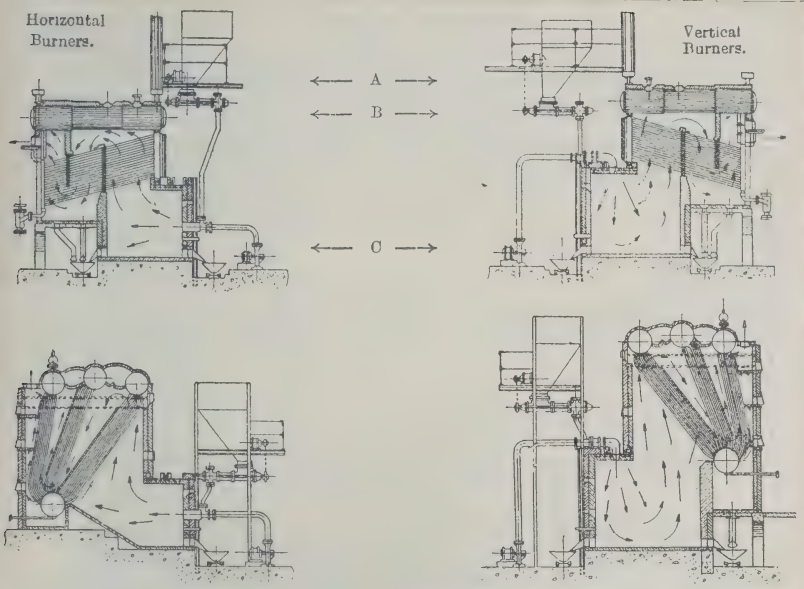
Power at 1½*d.* per kilowatt-hour (a high cost even for local supply), and an average allowance of 20 kw.-hours per ton of coal pulverised and delivered to the exit of mill house, including power for all operations. Dryers to be fired with pulverised coal, which is taken as 40*s.* per ton at the dryer burners ; an average allowance of 2 per cent. of the coal dried is made for dryer fuel, assuming the moisture content of coal to be reduced from 15 per cent. to 1 per cent.

Yearly output is based on 300 working days.

Vertical or Horizontal Flame Projection.

When the straight flow types of burners are used, the burning of bituminous coal and lignite is best effected by means of horizontal burners, and when fuel of less than 20 per cent. volatile matter is to be burned, straight flow burners are placed vertically. The short flame turbulent types of burners are placed horizontally for all classes of fuel.

Illustrations of boilers so equipped with horizontal and vertical burners, and marine, etc., applications, are shown in figs. 34, 35, and 36.



A. Fuel Bunker. B. Fuel Feeder. C. Air Supply Fan.

FIG. 34.

LANCASHIRE BOILERS.

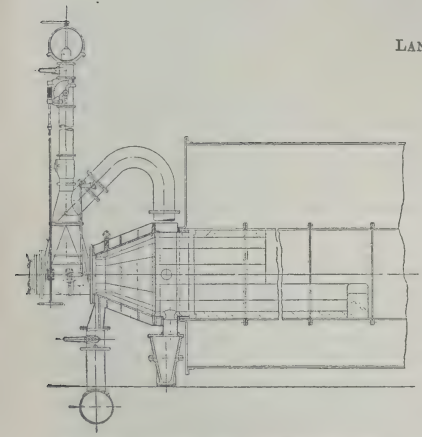


FIG. 35.—Burners (two) connected to Coal and Air Supply Main (top) and Secondary Air Main (bottom); Air Heating Jacket; Cut-off Valves and Hand Regulation Levers. (Fraser & Chalmers' System.)

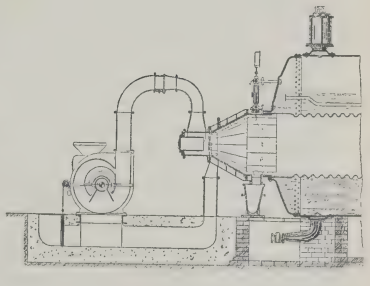


FIG. 36.—'Direct Firing' Self-Contained Unit. (Fraser & Chalmers' System.)

REFUSE DESTRUCTORS.

Calorific Value of Refuse.

The calorific value of refuse is about $\frac{1}{5}$ to $\frac{1}{7}$ that of an equal weight of good dry coal.

Average composition and calorific value of London ashbin refuse, containing the average percentage of water :—

Component parts of Refuse.	Amount in 100 lbs. of Refuse.	Calorific Value.
	Lbs.	B.Th.U.
Coal	·35	3,260
Coke	·15	1,200
Bones and offal	·37	1,970
Breeze and cinder	25·55	102,000
Rags	·40	1,330
Paper, straw, fibrous material, and vegetable refuse	13·15	33,100
Ash	47·00	Nil
Dust and dirt	9·75	,,
Bottles	·48	,,
Metals, including tins	·68	,,
Crockery	1·72	,,
Glass	·37	,,
	99·97	142,860

The above is the average of 1,000 cart-loads.

About 60 per cent. of the refuse is combustible.

The character of the refuse varies a great deal. In factory districts the percentage of unburnt cinder is high, whereas in residential districts with gardens it is much lower.

When it is considered that ordinary green garden refuse when piled in a heap over a nucleus of fire will consume itself, it is obvious that refuse containing cinders will contain enough heat, if properly applied, for its perfect destruction. It is essential that combustion should take place in brick cells, and that no heat should be abstracted until combustion is complete, after which the gases may be used for steam raising. If 40 per cent. only of the refuse is combustible, its calorific value per pound will be about 5,800 B.Th.U. if it were dry. Taking it only as 1,500, this is more than would raise the temperature of the whole gaseous and solid matter to a red heat. If half the weight were water, this would absorb about 600 B.Th.U. to evaporate and about 300 to become red hot, or 900 in all. The remainder of 600 B.Th.U. would raise the gases and solids to nearly 5,000° F. Thus destruction may be perfect with only the equivalent of one-tenth its weight of dry carbon. Inodorous combustion demands high temperature, and hence must occur in brick cells, with suitable movement of the fuel, so that the gases and steam cannot escape until fully incandescent.

The largest and most completely equipped refuse disposal plant in the world has recently been put into service by Heenan & Froude, Ltd., of Worcester, for the city of Glasgow.

The nominal capacity of the plant is 670 tons of crude city refuse per day, but records show that 750 tons can be dealt with if required.

The refuse is mechanically handled throughout and, with the exception of the removal of metals, the whole is cremated. The steam generated is used for driving two 5,000 kw. 3-phase 50-cycle turbo-alternators running condensing.

Over 90 per cent. of the current is transformed from 6,500 V. to 20,000 V. and delivered to the Dalmarnock Power Station for distribution.

The remainder (usually about 7 per cent.) runs the station auxiliaries and charges the batteries of the fleet of refuse collection vehicles after being transformed and converted as required.

Forty cells are provided for the burning of the refuse arranged in 8 groups of 5 cells each, complete with water-tube boiler and superheater. The grates are withdrawn by hydraulic power, the clinker dropping on to a lower platform, where it remains for the time being and gives up its heat to the air for the combustion of the fresh refuse on the grate above. This ensures rapid ignition and an increased thermal efficiency of the plant. When next clinkered, the old clinker is scraped off the lower platform by knuckled scrapers on the underside of the grate and the new clinker drops into its place, in turn igniting the fresh refuse and heating the air for combustion. The old clinker is removed in wagons running on a light railway in the basement.

The actual power delivered to Dalmarnock is shown by the dotted line, and the difference is the power required for driving the station auxiliaries and for charging the refuse collection vehicles which are of 5-ton capacity (15 cub. yds.).

The plant was under steam for 4,060 cell-hours out of a possible total of 5,280, and the high value of 215·6 kw. hours was obtained per cell-hour.

LIQUID FUELS.

Whereas solid fuels are solely used for external combustion it is possible to employ almost all liquid fuels for either external or internal combustion. On comparing the most important liquid fuel petroleum with the most important solid fuel, coal, many remarkable advantages of the former over the latter are found to exist. The heat value of petroleum is some 50 per cent. greater than that of average coal, the stowage space being slightly less. Liquid fuels are more convenient to store (places inaccessible to coal may be employed) and are more easily transported than solid fuels. Liquid fuel is almost entirely free from ash. When used for internal combustion the thermal efficiency of liquid fuels is double that of the solid fuels used for power production. Certain liquid fuels can be employed in high speed engines, whereby a higher power per unit weight of engine is obtainable than by any other type of prime mover.

The most important liquid fuels are petroleum, shale oils, coal tars and their products, and lignite tars and products.

Petroleum.

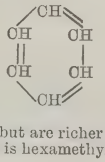
Petroleum occurs in the earth, and is generally found in strata of conglomerate or of oil sands. It is generally accompanied by gaseous and also by solid hydrocarbons. The crude oil is usually brown or black, and the products of different fields vary greatly both in physical and chemical properties.

Petroleum consists almost entirely of carbon and hydrogen. These are present in the form of compounds known as hydrocarbons. Hydrocarbons may be divided into three main groups, known as aliphatic, aromatic and hydroaromatic hydrocarbons. These groups may be subdivided into numerous series, the members of which possess certain definite relationships between the number of carbon and hydrogen atoms present in the molecules. The physical properties of the members of these series generally progress by fairly regular steps. The carbon atoms in the aliphatic hydrocarbons are connected together in the form of chains, and the more important series of this type of hydrocarbons are :—

Paraffin Series. General formula—C_nH_{2n+2}		Olefine Series. General formula—C_nH_{2n}	
Methane	CH_4	Ethylene	C_2H_4
Ethane	C_2H_6	Propylene	C_3H_6
Propane	C_3H_8	Butylene	C_4H_8
Butane	C_4H_{10}	Pentylene	C_5H_{10}
Pentane	C_5H_{12}		

The carbon atoms of the aromatic compounds are connected together in the form of rings, a typical compound being :—

Benzine, formula C_6H_6 . . .



The hydroaromatic compounds also contain carbon rings, but are richer in hydrogen than the aromatic hydrocarbons. A typical hydroaromatic compound is hexamethylene, C_6H_{12}.

Petroleum oils contain members of all these series, but the proportion varies according to the locality from which the oil has been obtained. Pennsylvanian oils consist mainly of aliphatic hydrocarbons, Russian oil mainly of hydroaromatic compounds. The products of the destructive distillation of coal consist mainly of aromatic compounds. Crude petroleum from Mexico and South America generally contains quantities of a viscous sulphur-containing body known as asphaltum, which has great influence on their properties as fuels. Little petroleum is consumed in the crude state, it being usual to separate it into fractions boiling over various ranges by means of repeated distillations. Many petroleum fractions are also treated with sulphuric acid and with caustic soda in order to remove certain undesirable constituents.

The principal fuels obtained from petroleum are :—

Benzine	boiling from 70°–150° C., 158°–302° F.	Specific gravity ·700–·760
Kerosene, illuminating, or paraffin oil . . .	boiling from 150°–300° C., 302°–572° F.	„ „ about ·800
Gas oils, solar oils, etc. .	boiling from 250° C., 482° F. upwards.	„ „ „ ·850
Petroleum residuum, 'fuel oil,' boiling point varies greatly		„ „ ·880–·950

The world's production of petroleum in 1929 was 1,488,604,000 barrels, and the United States production was 1,005,598,000 barrels, or 67·55 per cent. of the total.

The import of motor spirit or petrol into the United Kingdom in 1929 was 826,279,000 galls.

Benzine.

Benzine is the petroleum fraction from which motor spirits are prepared. Motor spirits are generally called petrol in this country. Petrol contains about 14 per cent. hydrogen and 86 per cent. carbon. Heavy fuel oils contain about 12 per cent. hydrogen and 88 per cent. carbon.

Kerosene.

Illuminating oil or kerosene, generally known in this country as 'Paraffin Oil,' possesses a specific gravity in the neighbourhood of ·800. It is slightly lower in calorific value than petrol, but on account of its higher specific gravity it yields more heat per unit volume of fuel. The ultimate composition of kerosene is approximately carbon 87 per cent. and hydrogen 13 per cent. Apart from illuminating purposes, kerosene finds frequent use as a fuel for vaporising engines. A large number of heavy distillates of petroleum are prepared for various technical purposes. Of these gas oil is probably the most important. Gas oil is prepared with a view to its use in the preparation of carburetted water gas or oil gas, and is also used as fuel for semi-Diesel and Diesel engines. It is a brown clear liquid of specific gravity about ·850, and has a net calorific value of about 18,000 B.Th.U. The distillates, whether prepared from asphaltic or non-asphaltic petroleum oils, are generally very similar in quality, though the proportion of each distillate yielded by the different crude oils may vary considerably. The residuum which is run off the still at the end of distillation, and is frequently marketed as 'fuel oils,' varies very greatly in properties according to the nature of the crude oil from which it originates. The non-asphaltic petroleum residuum which is sold as fuel oil is generally dark brown or black, almost opaque, quite mobile, and has a specific gravity of ·870 to ·920. It is largely used as a fuel for semi-Diesel and Diesel engines, also for boiler firing in the Navy. The residua prepared from asphaltic oils are usually black opaque liquids possessing high viscosities and specific gravities from ·920 to ·980, the viscosities and specific gravities being dependent on the asphaltum content.

Shale Oil.

Shale exists in many parts of the British Empire, the shale oil industry being most fully developed in Scotland. The shale is mined in a similar manner to coal, and is then subjected to destructive distillation, the principal products of value being ammonia and crude shale oil. The crude oil is subjected to distillation, whereby a series of products similar to the petroleum distillates is obtained. The shale oils are richer in unsaturated bodies than petroleum. In fuel properties they very closely resemble the petroleum products.

Petrol.

Petrol, known in France as 'Essence' and in America as 'Gasoline,' is a mixture of liquid hydrocarbons principally composed of the chemical 'paraffins,' Hexane C_6H_{14}, Heptane C_7H_{16}, Octane C_8H_{18}, and Nonane C_9H_{20}: these have boiling points 155°F., 208° F., 248° F., and 302° F. respectively. It is obtained by distillation from crude petroleum and, as now used, varies in sp. gr. at 60° F. from about 0·70 to 0·76, the average being about 0·725. The consumption in Great Britain rose from about nineteen million gallons in 1905 to upwards of two hundred million gallons in 1919. Petrol, being a mixture of hydrocarbons, has no fixed boiling point, and its density increases during evaporation; it was largely from this cause that surface carburettors proved unsatisfactory, selective evaporation occurring and a heavy non-volatile residuum being finally left; with the jet-carburettors now almost universally employed no selective action is possible.

As petrol is very expansible, tanks and closed vessels should never be quite filled with the liquid, so as to avoid risk of rupture following a rise in temperature. Coefficient of expansion, 0·00063 per ° F. (0·001138 per ° C.).

The composition of commercial petrol averages about 85¼ per cent. carbon and 14¾ per cent. hydrogen, and the net calorific value per lb. varies but little from 18,900 B.Th.U.

Heavy petrol gives quite satisfactory results with modern carburettors; motor omnibuses employ 0·760 spirit very largely on account of its lower cost.

It was formerly assumed that the suitability of a motor spirit could be determined from its specific gravity, but in recent years the use of benzol in admixture with petroleum spirits has become common, and such a mixture is a very satisfactory fuel, in spite of its having a higher specific gravity.

On the other hand, if the air supply exceed about 15 lbs. per lb. of petro., there is an air excess and no CO appears in the exhaust; instead one gets free oxygen. The practical range is from about 11 air to 1 petrol, to 17 air to 1 petrol; recently the Royal Automobile Club has adopted as the standard for carburettors such a mixture as would give an exhaust containing about 1 per cent. of free oxygen; this prevents the presence of CO, and gives nearly the highest thermal efficiency and also nearly the maximum power. Maximum thermal efficiency usually occurs with a somewhat greater excess of air, and maximum power with a slight defect of air, due in this latter case to the higher mean pressure obtained by reason of the considerable expansion in the products of combustion with 'rich' mixtures. One lb. of petrol, occupying 4 cub. ft., requires for its complete combustion roundly 15 lbs. of air occupying 196 cub. ft. at 60° F.; the 16 lbs. of mixture thus occupies a volume of 200 cub. ft. at 60° F., and contains 2 per cent. by volume of petrol vapour—that is, assuming that the petrol is gasified, and not atomised.

Theoretical Efficiency of Petrol Engines.

The theoretical efficiency, E, of an engine operating on the constant volume cycle is—

where, $$E = 1 - \left(\frac{1}{R}\right)^{n-1}$$

R = expansion ratio.

If no heat is lost, then $n = 1{\cdot}45$, but in practice heat is lost during compression, with a result that the exponential n is reduced to the value of 1·38. Ricardo states that when heat leakage, frictional losses, and losses due to friction of gases have been allowed for, then $n = 1{\cdot}25$, so that the maximum efficiency obtainable in practice may be considered to be

$$E = 1 - \left(\frac{1}{R}\right)^{0{\cdot}25}$$

Cracking Process.

A very large proportion of the world's production of motor spirit is now obtained by cracking from heavier oils. The original Burton process has been largely replaced by the modern liquid-phase cracking plants of Dubbs and Cross. In these the oil is heated while travelling rapidly through tubes arranged in a furnace, and thence passes to a reaction chamber, where the cracking mainly takes place. In the Dubbs plant a pressure of 200 lbs. per sq. in. is maintained, while the Cross plant employs a higher pressure of 800 lbs. per sq. in.

The specific gravities and distilling temperatures of petrols are liable to be varied from time to time as the market conditions warrant. The increased demand for motor spirit during recent years has been to a large extent met by including such higher and lower boiling distillates in the motor spirit fraction. By subjecting the crude oil to high temperatures in the absence of air, frequently accompanied by high pressures, a decomposition of the more complex hydrocarbons is brought about. This action, which is technically known as 'cracking,' results in large yields of motor spirit. The motor spirit prepared in this way is rich in unsaturated compounds, possesses a characteristic odour, and is higher in specific gravity than the direct distilled petrol which comes over between the same temperatures.

Some of the petrol at present being imported into this country is naturally rich in benzene and toluene, and this is particularly the case with the Borneo spirit imported by the Shell group, and it has been found, both experimentally and in practice, that owing to the higher spontaneous ignition temperatures of benzene and toluene, mixtures containing these hydrocarbons do not so readily cause preignition or detonation in petrol engines. In view of this, petrol which does not naturally contain benzene has occasionally been mixed with benzene in order to decrease the liability to preignition.

PROPERTIES OF VARIOUS MOTOR SPIRITS (1931).

Moore and Hobson.

Fuel.	Approximate Analyses.				Sp. Gr. at 60° F.	Engler Distillation.								Doctor Test.	Benzene/ Heptane equivalent.
	Aromatics.	Naphthalenes.	Unsaturateds.	Paraffins.		I.B.P. °C.	50° C.	75° C.	100° C.	125° C.	150° C.	175° C.	F.B. Pt. °C.		
							%	%	%	%	%	%			
Typical	6·0	45·1	3·4	45·5	·734	36	6	22	43	63	82	93	200	O.K.	45–55
Combine No. 1 Spirits .	6·0	34·9	8·1	51·0	·729	38	4	22	39	61	79	93	192	O.K.	—
	10·0	62·7	2·8	23·7	·746	37	5	21	41	60	78	92	200	O.K.	—
Typical	11·6	58·5	5·1	24·8	·753	40	2	15	37	56	75	90	195	O K	47–53
Non-combine No. 1 Spirits	10·2	26·2	1·6	62·0	·738	41	3	19	37	56	71	89	210	O.K.	32–68
	6·5	31·8	1·7	60·0	·736	42	3	16	40	69	86	96	198	O.K.	—
Typical	7·2	30·4	2·2	60·2	·732	43	1	13	37	66	86	95	185	O.K.	38–62
Combine No. 3 Spirits .	10·4	27·4	3·9	58·3	·741	39	2	14	31	54	77	90	197	O.K.	—
	8·9	29·2	4·8	67·2	·742	41	2	13	31	57	77	91	193	O.K.	40–60
Typical	6·5	38·8	0·6	54·1	·746	47	—	7	28	55	76	91	202	O.K.	27–73
Non-Combine No. 3 Spirits	6·2	38·1	7·6	48·1	·739	39	3	17	34	52	70	87	200	O.K.	—
	11·4	30·8	2·2	55·6	·750	44	1	11	30	49	68	85	212	O.K.	—
Typical Combine Benzol Mixture	21·1	58·1	5·6	15·4	·765	45	1	15	47	67	82	93	190	O.K.	—
Non-combine Benzol Mixture . . .	8·6	58·4	2·9	30·1	·751	46	1	16	37	58	73	90	205	O.K.	—

Analyses of Pratt's No. 1 and Shell No. 1 spirit by Mr. Harold Moore, on behalf of *The Motor*,* gave the following results:—

Pratt's No. 1 Motor Spirit:—Specific gravity, 0·729 at 15° C. (59° F.).

ENGLER DISTILLATION TEST: Initial boiling point, 33° C. (91·4° F.).									
Temperature . .	C.°	55	60	70	80	90	100	110	120
" . .	F.°	131	140	158	176	194	212	230	248
Drop per cent. . . .		—	1·0	3·5	11·5	21·5	31·0	45·0	57·5
Temperature . .	C.°	130	130	140	150	160	170	180	190
" . .	F.°	266	266	284	302	320	338	356	374
Drop per cent. . . .		67·5	67·5	77·0	83·0	87·5	91·5	93·0	94·5

Initial boiling point is the temperature at which the first drop falls from the bulb of the thermometer when the latter is placed ½ in. below the outlet tube of flask.

Drop temperature is taken when the first drop leaves the end of the condenser.

Acid Coloration.—Acid has brick-red colour, and petrol separates out slowly.

Aromatic hydrocarbon content = 10·2 per cent. by weight.

Shell No. 1 Motor Spirit: Specific gravity, 0·754 at 15° C.

ENGLER DISTILLATION TEST: Initial boiling point, 34° C.							
Temperature	C.°	57	70	80	90	100	110
"	F.°						
Drop per cent. .		—	1·5	6·0	12·5	24·0	39·0
Temperature	C.°	120	130	140	150	160	170
"	F.°						
Drop per cent. .		53·5	67·0	78·0	86·5	92·0	95·0

Acid Coloration.—Acid only slightly discoloured, light brown, and petrol separates out almost immediately.

Aromatic hydrocarbon content = 21 per cent. by weight.

DETONATION.

High efficiency in petrol engines is best attained by employing very high compression ratios, but in practice it is found that a high compression ratio is accompanied by 'pinking' or 'knocking' when running at slow speeds with heavy loads when ordinary motor spirit is used. For such spirits the limiting compression ratio is about 4·5 to 1. Benzol is very much less prone to give this phenomenon of knocking, or detonation, and it permits of the use of a considerably higher compression ratio. The use of alcohol as a fuel permits of a compression ratio of as much as 7 to 1. Certain organic compounds have been found by Midgley & Boyd to be much more efficient than benzol in preventing detonation; for example, tetraethyl lead is 625 times as powerful as benzene. Fuel containing a minute proportion of this anti-detonating agent has been extensively used in the U.S., and is known commercially as 'ethyl gas.'

* *The Motor*, September 7, 1921.

BRITISH ENGINEERING STANDARDS ASSOCIATION. SPECIFICATION FOR MOTOR SPIRIT.*
(No. 121—1923.) (*Abstract.*)

General.

The liquid shall consist of hydrocarbons and be free from visible impurities.

Distillation.

The range of distillation as determined in the prescribed manner (I.P.T.) † shall conform to the following limits of temperature :—

When the liquid is distilled according to the standard method, the first-drop temperature indicated by the thermometer shall not exceed 55° C.

When 20 per cent. by volume of the distillate has been collected, the temperature indicated by the thermometer shall not exceed 105° C.

The whole of the liquid shall have distilled when the temperature indicated by the thermometer is 225° C.

Acidity.

The motor spirit shall be free from mineral acid.

Benzol.

Benzol had come into prominence before the war as an alternative home-produced fuel for the engines of motor vehicles, partly in consequence of the high price of petrol. Since the withdrawal of the restrictions on the use of benzol its use has become more general than before, though the motoring public do not yet fully appreciate its value as a fuel. Benzol is the commercial name for a mixture of hydrocarbons prepared by the distillation of coal tar, and also obtained by stripping the gas which is produced as a by-product from coke ovens. Commercial benzol consists almost entirely of a mixture of the hydrocarbon benzene (C_6H_6 : not to be confused with the petroleum fraction 'benzine') and the hydrocarbon toluene (C_7H_8). The usual commercial grades are '90 per cent. benzol,' which on distillation yields 90 per cent. of distillate at 100° C., and consists of approximately 70 per cent. benzene and 30 per cent. toluene, together with traces of the xylenes (C_8H_{10}), and '100 per cent. benzol,' which on distillation yields 100 per cent. of distillate at 100° C., and contains a higher proportion of the hydrocarbon bezene than 90 per cent. benzol.

There is little to choose between these two bodies as motor fuels, and provided they have been satisfactorily refined they are both preferable to pure benzene, as the mixtures do not so readily solidify in cold weather. Pure benzene solidifies at a temperature of + 6° C. (43° F.), and boils at 80·4° C. It possesses a specific gravity of ·88 at 15° C., and a gross heat value of about 18,090 B.Th.U.

Both commercial benzol and pure benzene possess very high temperatures of spontaneous ignition, and they will withstand compressions of about 200 lbs. per square inch in internal combustion engines without being affected by preignition. This property, together with the high specific gravity, accounts for the increased mileage per gallon obtained when using benzol as fuel instead of petrol. Benzol gives about 16 per cent. greater mileage than petrol, and also gives greater power to the engine. When changing over from petrol to benzol it is advantageous slightly to reduce the size of jet, though it is not absolutely necessary to make any modification. The colour of benzol for motor use should be perfectly clear and white, as yellowish colour indicates unsatisfactory refining. Certain bodies (mainly phenols, unsaturated hydrocarbons and sulphur containing compounds) are present in benzol when it is first prepared by distillation. These compounds are removed by treatment with caustic soda solution and afterwards with concentrated sulphuric acid. If these bodies are not removed there is a liability for the benzol to form resinous substances after prolonged standing, and also to form gummy deposits in the engine. Petrol undergoes a similar refining process during its preparation, but it is rare to meet with badly refined petrol nowadays, as the quality necessary for the motor spirit market is well understood by the petroleum firms. Petrol and benzol are perfectly soluble in each other, and can be used with satisfaction in the form of mixtures.

* By permission of the British Engineering Standards Association.
† Institute of Petroleum Technologists' method.

BRITISH STANDARD SPECIFICATION FOR BENZOL FOR MOTOR FUEL.*

Report No. 135, 1921 (*Abstract*).

(1) The term 'Benzole' shall denote a liquid consisting essentially of a mixture of benzene and not more than 30 per cent. by volume of toluene and xylenes.

(2) The material shall be a clear 'water white' liquid, free from undissolved water and other visible impurities.

(3) The specific gravity at 15·5° C. (60° F.) shall be not less than 0·870, and not more than 0·885.

(4) When 100 cc. of the material are distilled in the Standard Distillation Apparatus, at a rate of 7 cc. per minute, there shall distil at 760 mm. pressure—

Not less than	60 cc.	at or below		90° C. (194° F.).
„	„	75 cc.	„	100° C. (212° F.).
„	„	90 cc.	„	120° C. (248° F.).

The flask shall be dry at a temperature not exceeding 125° C. (257° F.). The temperature recorded by the thermometer shall be corrected for emergent stem and barometric pressure.

(5) The amount of non-volatile residue remaining when 20 cc. of the material are evaporated, and subsequently heated for two hours on a water bath at 100° C. (212° F.), shall not exceed 0·01 per cent. by weight.

(6) The coloration imparted to pure sulphuric acid (90 per cent H_2SO_4), free from nitric acid, when 90 cc. of the material are shaken with 10 cc. of the acid for five minutes at ordinary temperatures, shall be not more than a light brown.

(7) The total sulphur content shall not exceed 0·4 per cent. by weight. Freedom from acid and alkali.

(8) The material shall give no reaction for free acid or alkali when tested.

(9) The material shall give no reaction for sulphuretted hydrogen when tested.

(10) The freezing point shall not be higher than — 14° C. (7° F.).

Detailed descriptions of the methods employed in testing are given in the original publication.

Alcohol.

Although the heat value of alcohol per lb. is comparatively small, yet, on account of its heaviness as a vapour, and the small volume of air required to burn it compared with benzene and petrol, it yields 89·2 B.Th.U. per cub. ft. of mixture at 60° F. and atmospheric pressure. Moreover, the products of combustion are 6½ per cent. greater in volume than that of the unburnt mixture, with corresponding increase in the pressure developed. There is thus, thermally considered, not very much to choose between alcohol and petrol; alcohol is, in fact, an excellent fuel for suitably designed internal combustion engines, and is largely used in Germany, where its production is encouraged by the Government; in 1904 the price of alcohol in Germany was only 9*d.* per gallon when purchased in large bulk.

Mr. E. Bury of the Skinningrove Iron & Steel Works, Middlesbrough, has made trials of a process for the abstraction of ethylene from coke-oven gas and its recovery as alcohol, by which an average yield of 1·6 gallons per ton of coal carbonised is obtained. The application of this process to the whole of the 14,635,000 tons of coal reduced to coke in this country in 1918 would yield 23 million gallons of alcohol. It is calculated that the recovery of alcohol from the gas-works of the country on similar lines would yield 27 million gallons per annum.

Alcohol has a great affinity for water, and ordinary commercial qualities contain water in various proportions up to about 15 per cent.; in the following table estimated calorific values per lb. are given for alcohol-water mixtures containing from 0 to 50 per cent. of the latter ingredient.

* By permission of the British Engineering Standards Association.

ESTIMATED CALORIFIC VALUE OF ALCOHOL-WATER MIXTURES.

Proportions by Weight.		B.Th.U. per Lb.	
Alcohol.	Water.	Higher Value.	Lower Value.
Column :—	1	2	3
1·00	0·00	12,420	11,285
0·9	0·1	11,178	10,060
0·8	0·2	9,936	8,835
0·7	0·3	8,694	7,609
0·6	0·4	7,452	6,385
0·5	0·5	6,210	5,160

Alcohol engines require larger carburettors and piping and valves than are necessary when petrol is used. As the latent heat of alcohol much exceeds that of petrol, especially when water is present in any considerable proportion, it is also necessary to well jacket the carburettor with hot water or exhaust gas, and to heat the in-going air to about 350°F. in order to ensure complete vaporisation. The cylinder-jacket water should also be kept nearly at 212°F.; too cool a cylinder results in imperfect combustion of the mixture, acetic acid, aldehyde, etc., being formed instead of CO_2 and H_2O only, with loss of efficiency and corrosion of internal parts. Starting up from cold is a special difficulty with alcohol, and engines are usually started with petrol, the alcohol being turned on when everything is well warmed up. Some engines are arranged so as to start and stop on petrol; in this way any traces of water or acid from the alcohol are cleared out of the cylinders and passages during the last few revolutions of the engine, and risk of corrosion of parts is thus avoided. Tests on a 14 horse-power 'Locomobile' alcohol engine of 8·28 ins. bore and 11·8 ins. stroke made by Prof. E. Meyer in 1901 showed a brake thermal efficiency of 24 per cent., using alcohol containing about 13 per cent. of water. The volume ratio of compression was 5·9. The cylinder-jacket water temperature was maintained at 208°F. The engine ran at 280 revs. per minute and developed 13·9 b.h.p. at full load.

COMPOSITE MOTOR SPIRITS.

Various mixtures containing alcohol, etc., have been put on the market as motor spirits. 'Natalite' is said to be a mixture of 40 per cent. of ether and 60 per cent. of alcohol (prepared from molasses), with traces of ammonia and denaturants, and is extensively used in South Africa. 'Discol,' used in this country, is a mixture of alcohol, heavy petrol, and benzol, containing about 50 per cent. of alcohol.

Coal Tars and Coal Tar Distillates.

When bituminous coal is subjected to destructive distillation, gas, coke, tar and ammonia are obtained. The physical and chemical properties of the tars are dependent upon the type of coal from which they originate, and the conditions which exist in the carbonising chamber. The appliances in use for the carbonisation of coal are mainly gas manufacturing plants and coke manufacturing plants. A few special plants exist which do not belong to either of these categories.

Gas manufacturing plants may be divided into three classes—horizontal retort, inclined retort, and vertical retort plants. Retorts of the first type yield tars of high specific gravity, 1·15 to 1·25, rich in free carbon, and viscous. Vertical retorts yield thin brown tars of low gravity (about 1·05), which are comparatively low in free carbon content and rich in aliphatic hydrocarbons. The tars from inclined retorts are intermediate between the products of horizontal and vertical plants in their chemical and physical properties.

Coke-oven tars are very variable, but usually approximate to horizontal retort tars in composition.

Tars from gas producers are generally too high in water-content to be of use as fuels.

Blast furnaces yield tars containing large quantities of ash which render them useless in the raw state.

Several processes for the low temperature carbonisation of coal have been advocated, but on large quantity of the tar, which is exceptionally suitable for fuel purposes, is available.

PROPERTIES OF MISCELLANEOUS FUEL OILS.

(*Moore.*)

Description.	Sp. Gr. 60° F.	H_2O	Chemical Composition.				Ash.	Coke.	Flash Point. ° C.	Viscosity. Redwood Units.		Soft Asph.	Calorific Value.	
			C.	H.	O. & N.	S.				70° F.	100° F.		Gross.	Net.
Coal-Tar Creosote	1·053	Traces	89·7	7·4	2·23	0·66	Traces	1·39	93	10·3	8·5	—	9,342	8,964
Blast-Furnace Oil	·907	Nil	82·6	9·9	7·22	0·29	Nil	1·26	70	5·7	—	—	9,992	9,603
Vertical Retort Tar Oil	1·019	0·7	85·8	7·9	5·1	0·5	Traces	1·65	88	9·7	7·0	—	9,189	8,762
Scottish Shale Oil	·945	1·6	87·0	9·5	1·04	2·35	0·11	0·18	56	23·6	12·5	—	9,917	9,245
Premier Tarless Process Oil	·995	Trace	82·6	8·6	8·8	0·50	0·10	1·52	76	17·1	9·4	—	9,472	9,008
Mexican Fuel Oil (Petroleum)	·952	1·00	83·6	10·9	1·5	2·85	0·17	12·2	93	—	267	36·8	10,350	9,734
Mexican Fuel Oil (Petroleum)	·960	0·5	83·0	10·7	2·54	2·8	0·02	—	—	4,800	822	24·6	10,130	9,548
Californian Fuel Oil (Petroleum)	·952	1·9	83·4	11·5	0·49	2·3	0·4	—	46	387	108	11·5	10,110	9,476
Californian Fuel Oil (Petroleum)	·903	9·5	35·1	12·2	1·0	1·4	0·3	—	138	88·5	34·2	—	10,480	9,832
South American Oil	·887	—	86·9	10·9	1·7	0·6	None	—	27	14	9	—	10,740	10,076
Borneo Oil	·939	0·25	88·3	11·3	0·07	0·1	Trace	—	—	63·2	23·1	1·7	10,360	9,712
Persian Oil	·894	—	86·7	11·7	0·13	1·5	0·01	—	93	46·5	15·7	7·6	10,550	9,902
Mid-Continental Oil	·904	None	83·9	12·1	3·63	0·4	Trace	—	110	43·9	—	—	10,815	10,150
,, ,, ,,	·889	,,	86·7	12·5	0·11	0·6	Trace	—	113	—	—	1·5	10,780	10,108
,, ,, Gas Oil	·850	,,	85·9	12·8	1·09	0·2	Nil	—	84	—	—	0·4	10,890	10,193
Texas	·936	0·5	86·1	11·0	2·0	0·3	0·1	—	121	—	68	—	10,605	9,989
Assam Oil	·890	Traces	86·4	11·7	1·7	0·2	0·03	—	65	14·0	9·1	—	10,312	9,664
Benzole	·875	—	91·1	8·0	0·0	0·9	None	0·27	—	4·5	3·6	—	9,877	9,461
Mineralised Alcohol (Commercial)	·822	—	48·6	12·7	40·56	0·15	—	—	Below 20° C.	5·2	4·7	—	6,115	5,421

The majority of coal tars may be employed for boiler firing purposes. Tars from vertical retorts have been found suitable for use on both two and four cycle Diesel engines, providing that certain special adjustments or fittings are employed, large quantities of such tars being used in Diesel engines on the Continent. Some coke-oven tars are suitable for use in Diesel engines. The compositions of several British coal tars are given in the accompanying table :—

PROPERTIES OF TARS. (*Moore.*)

Description.	Sp.Gr. at 15° C.	Per cent. Chemical Composition.					Ash.	Coke.	Calorific Power (Dry Tar).		Free Carbon.
		H_2O.	C.	H.	O.&N.	S.			Gross.	Net.	
Horizontal retort tar	1·180	1·75	91·5	5·2	2·6	0·5	0·2	24·0	9093	8645	18·2
Inclined retort tar	1·157	1·11	89·9	6·0	3·6	0·5	0·02	18·5	9096	8671	14·0
Vertical retort tar	1·089	2·25	88·0	6·8	3·8	0·6	0·03	6·1	9246	8664	1·7
Otto Hilgenstock coke oven tar	1·208	6·00	90·0	5·4	3·8	0·8	0·02	26·8	8921	86 4	23·9
Simon-Carvé coke oven tar	1·090	0·50	88·1	5·6	6·1	0·2	0·07	6·0	9695	9261	traces
Chamber oven tar	1·082	1·29	88 2	6·9	4·6	0·3	traces	7·3	9229	8737	3·0
Low temperature carbonisation tar	1·058	3·00	85·8	8·1	5·5	0·9	0·11	8·2	9196	8776	2·2
Water gas tar	1·054	0·59	92·2	6·8	0·4	0·6	traces	18·7	8951	8647	6·8
Blast furnace tar	1·172	3·00	89·5	5·7	0·5	0·84	0·36	23·4	8563	8288	9·5

The volatile products obtained by the distillation of coal tar (mainly benzol and toluol) have been extensively used in automobile engines prior to the war. Creosote and anthracene oils are high boiling distillates of coal tars and are extensively employed for fuel purposes, both for external and internal combustion. The 'tar oil' sold for fuel is usually a mixture of both these above-mentioned fractions. Analyses of some tar oils, together with a petroleum residuum, and alcohol, are shown in the table on p. 2157.

Lignite Tars and Lignite Tar Distillates.

Lignite yields valuable liquid products by destructive distillation. The tar is of a butter-like consistency and possesses a specific gravity of ·860 to ·920. Its sole use in the raw state is as fuel for Diesel engines, for which purpose its richness in aliphatic hydrocarbons makes it behave similarly to petroleum. It therefore does not require either raised cylinder compressions or the use of an ignition oil.

Fuels for petrol engines and heavy oil engines can be obtained from lignite tar by distillation. Their properties are very similar to those of the corresponding fractions obtained from petroleum.

Tar oils were used in considerable quantities on Diesel engines in this country during the war period on account of the scarcity of petroleum. The difficulties encountered in burning tar oils are mainly due to their comparatively high spontaneous ignition temperature. The ignition of the fuel charge in a Diesel engine is dependent upon the temperature of compression, no special ignition device being provided. With the fuel oils derived from petroleum or shale it has been found that a compression pressure of 450 to 500 lbs. is sufficient to ensure regular ignition under starting conditions and at all loads. Whereas the spontaneous ignition temperature (in atmosphere of oxygen) of petroleum and shale oils is in the neighbourhood of 270° C., this value for tar oils is generally over 400° C.* As a result of the high ignition point engines will not start, or run at low loads, on tar oil unless special means are taken for raising the temperature of the air in the combustion space at the time of fuel injection. The more important methods adopted for this purpose are—(1) to heat the cycle air, blast air and fuel, (2) to increase the compression ratio, (3) to employ an ignition oil, and (4) to inject a small charge of tar oil prior to the main tar oil charge.

* Moore, *Jour. Soc. Chem. Ind.*, February 15, 1917, p. 109.

Heating of cycle air, blast air and fuel is difficult to effect under starting conditions and has a deleterious action on the fuel valve when running. The increase of compression necessitates a specially heavy design of engine to withstand the higher stresses on liners, cylinder cover, and bearings. To ensure certain starting on tar oils the compression would require to exceed 600 lbs. The starting difficulty may be overcome by running the engine on a petroleum oil before closing down and when starting. If the engines can be always maintained at half load or over, a compression pressure of about 550 lbs. per square inch will give good running.

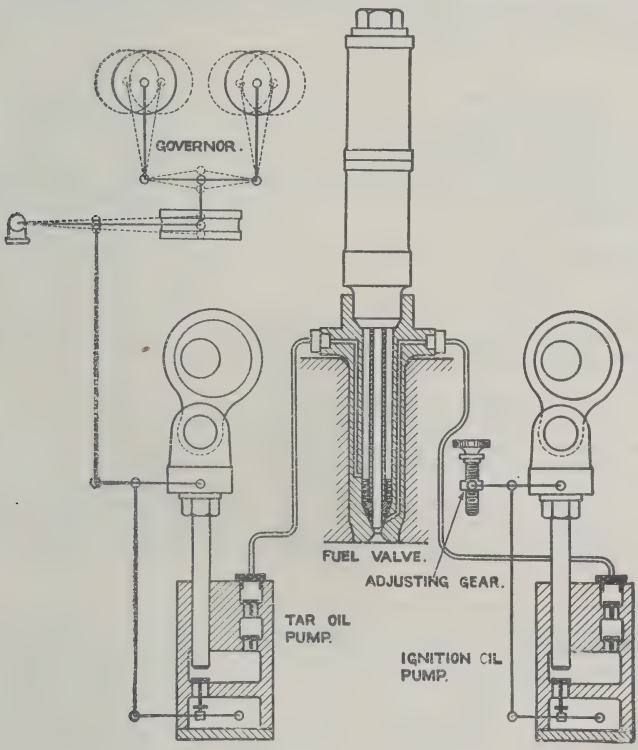


FIG. 37.—Diagrammatic Arrangement of Fuel Pumps for Tar Oil Engine.

The introduction of a small charge of a petroleum oil into the working cylinder prior to the main charge of tar oil has been found the most satisfactory method of burning tar oils. The ignition oil charge is pumped to the atomiser nearer the cylinder than the place where the main tar oil charge enters. Thus when the fuel valve opens the ignition oil is first injected into the cylinder and is immediately followed by the tar oil charge. The combustion of the ignition oil heats the air in the cylinder so that the temperature exceeds the spontaneous ignition temperature of the tar oil. The proportion of ignition oil is 6 to 8 per cent. of the total fuel at full load; at other loads the quantity of ignition oil remains practically constant, the proportion increasing to about 50 per cent. at no load. This method has been found to yield satisfactory results at all loads, and under starting conditions, with cylinder compression pressures of 480 to 500 lbs. per square inch. An arrangement of ignition oil gear as given in a paper by Mr. C. Day,* is shown in fig. 37. Devices have been designed to give a small charge of tar oil prior to the main charge

* Paper read before the Diesel Engine Users' Association, January 19, 1916.

of tar oil. The device is so arranged that the initial tar oil charge is not accompanied by any appreciable blast air, the expansion of which would reduce the temperature of the combustion space. Such devices are said to have given good results on the Continent, but are little used in this country.

In addition to the ignition question tar oils are particularly liable to form deposits in the atomisers of Diesel engines. A résumé of some experiences of burning tar oils on Diesel engines was given by Mr. G. Porter in a paper read before the Diesel Engine Users' Association, May 24, 1917. In this paper several specifications for tar oils for use in Diesel engines are given. An abstract of these specifications is appended below :—

Specifications of Tar Oil Fuel for Diesel Engines.

(1) Specification of the M.A.N. Company :—

The tar oil must be a distillate of coal tar.

The oil must flow freely at 61° F., and on being cooled down to 46·5° F. and resting in a place undisturbed by vibration no separation shall take place at this temperature within the space of half an hour.

The following constituents shall not be present in quantities greater than the stated percentages—

Ash	·05 per cent.
Sulphur	1·0 „

Proportions of water and coke residue not stated.

Per cent. insoluble in xylol ·2.

(2) Continental specification for tar oil quoted by Mr. C. Day in his paper on 'Tars and Tar Oils as Fuel for Diesel Engines' (January 19, 1916) :—

Tar oils must not contain more than a trace of constituents insoluble in xylol.

The water content should not exceed 1 per cent.

The coke residue should not exceed 3 per cent.

At least 60 per cent. of the oil should be distilled on heating up to 300° C. (572° F.).

The net calorific value should not be less than 15,840 B.Th.U. per pound.

The open flash point must not be below 65° C. (143·6° F.).

The oil must be quite fluid at 61° F.

(3) Specification of tar oil proposed by Mr. C. Day (January 19, 1916) :—

The tar oil must be a product of the distillation of coal tar. No product that has not undergone distillation must be present.

Per cent. insoluble in xylol . . .	Not more than		2·0
„ ash	„	„	0·08
„ water	„	„	2·5
„ coking residue	„	„	3·0

The oil must be liquid at 60° F. when maintained at that temperature for half an hour.

(4) Specification of tar oil by Mr. Batho (the Diesel Engine Users' Association, February 23, 1916) :—

Specific gravity—Between 1·0 and 1·1.
Viscosity—Generally 2° Engler.
Flash point—100° F. to 130° F.
Colour—One drop on white paper should show no black residue, as is the case with tar.
Lower calorific value—Between 15,800 and 16,500 B.Th.U. per 1 lb.
Ash—Should not exceed 1 per cent. (unburnt residue of tar oil is mostly harmless).
Water—Should not exceed 1 per cent.
Sulphur—·5 to 1 per cent.

The burning of raw tars on Diesel engines has been accompanied by the same difficulties as the burning of tar oils, but an additional trouble is encountered on account of the free carbon which is always present in raw coal tars.

Furnace Firing with Liquid Fuel.

Liquid fuel possesses many advantages over coal or other solid fuels for furnace firing. The advantages when using heavy petroleum oils may be briefly enumerated as follows :—

(1) Petroleum fuel oil possesses a gross heat value of approximately 19,200 B.Th.U., whilst the gross heat value of coal averages about 14,000 B.Th.U. This fact alone shows two pounds of oil to be about equivalent to three pounds of coal.

(2) In spite of its lower specific gravity, oil can be stored in less space than a corresponding weight of coal.

One ton of coal occupies about 43 cubic eet.
„ oil „ 40

For marine purposes oil is conveniently stored in ballast tanks and inaccessible places, which could not be utilised for coal bunkerage.

(3) Oil rarely contains more than 0·1 per cent. of ash, a negligible quantity, while coal may contain as much as 15 per cent. ash. This ash necessitates the removal of clinker from the fires, and besides involving expense it causes a serious loss of heat.

(4) Oil can be conveyed by pipe line, a cheaper, cleaner and more rapid method than any method of coal conveying.

This is of great importance in naval work.

(5) Oil firing does not involve so much manual labour and attendance charges which accompany furnace firing with solid fuel.

(6) A boiler fired with oil will raise about 30 per cent. more steam than the same boiler coal fired.

(7) The furnace doors can be kept closed when oil firing, thereby increasing the efficiency of the boiler and lessening the wear and tear on the boiler plates caused by frequently being chilled with cold air, as is the case when burning solid fuel.

(8) Boilers can raise steam much more quickly when oil fired, thus lessening stand-by charges.

(9) Boilers fitted for burning both solid and liquid fuel may be run on coal under normal conditions, while oil can be burned for peak loads, thus minimising the number of boilers in use.

1 lb. coal under favourable conditions raises 9 lbs. steam.
1 lb. oil under similar conditions raises 14½ lbs. steam.

It is essential that firing devices for liquid fuel should break the oil into an exceedingly fine state of division, so that the air can reach the oil readily and ensure speedy combustion.

The methods of breaking up the oil are :—

(1) By means of steam.
(2) By means of compressed air.
(3) By a high pressure on the oil, which is pulverised on passing through a fine jet.
(4) By means of mechanical breaking up in a pulverising chamber.

Steam Jets.

The majority of oil-burning installations used on land are operated by steam. Steam is always available at the face of the boiler, and at the same time steam burners, though not giving quite so high an efficiency as air burners, give a soft flame, which is less liable to injure the plates or tubes with which it comes into contact. For marine purposes there is a distinct disadvantage in the use of steam on the oil burners, as it causes the loss of a considerable quantity of water, which has to be carried in storage tanks or else has to be specially prepared by a distillation plant whilst at sea. This drawback does not apply to land installations, where plenty of water is usually available. Steam burners operate on a variety of principles, the majority of constructions being somewhat in the form of an injector.

The Kermode steam jet burners, which have met with much success on land service, are designed to impart a rotary motion to the gases by passing the oil through a helical groove. The steam is passed through an annular space surrounding the oil duct, and in addition to causing atomisation, it is used to induce an air current which assists in the combustion of the fuel.

With the Holden steam jet burner, fuel steam and air are passed through annular spaces, the centre of the burner being left open to facilitate cleaning.

Another principle used for pulverising oil by a steam jet is to allow the oil to trickle from a weir over a thin slit through which the steam is passing. This system is used in the common type of burner known as the 'Mexican trough.' A modification of this principle is also used in the Ajax burner, made by Meldrums Ltd.

Compressed Air Burners

The use of compressed air for pulverising purposes allows a higher efficiency on boiler-firing installations. At the same time the saving is not so great as might be anticipated, as the steam

or power used for driving the compressors is almost as great as the amount lost through steam burners. The flame has a higher temperature than that of the steam jet, and greater care must be taken to prevent it impinging on the plates. Some of the most recent types of compressed air burners will operate on air at low pressure (about 10 lbs. per square inch), and therefore do not necessitate costly air compressors, as they can be operated with complete satisfaction by low-pressure air from a rotary blower. Such burners are the 'White' burner, the 'J. Samuel White,' and one of the burners by Alldays & Onions.

The Oil Pressure System.

This is mainly used for marine purposes, and complete oil-burning installations on this principle are comparatively expensive, on account of the necessity for well-constructed and effective preheaters, filters, and the specially constructed fuel pumps.

In this type of burner the oil is forced through fine jets under a comparatively high pressure. Many burners are so constructed as to impart a rotary motion to the pulverised fuel (*e.g.* Korting burner). The pressure system, although expensive, is probably the most efficient of oil-burning appliances. The power consumed in driving the fuel pumps is almost negligible, and the combustion is excellent. The pressure system is meeting with much success in marine work at the present time. Good examples of this type of apparatus are made by the Wallsend Slipway Co., Kermode, Korting, and J. Samuel White & Co.

Mechanical Pulverisers.

It is possible also to pulverise oil by mechanical means, such as breaking the oil up in a squirrel cage. The rich oil-fog thus produced can then be treated as a gas and burned together with secondary air in the furnace. Installations of this type, however, have not met with much success, probably on account of their high initial cost.

Atomisers.

Holden's steam atomiser is shown by fig. 38; fig. 39 shows the Guyot atomiser used in the French Navy.*

Where boilers have the mixed coal and oil system, the oil is best atomised into the front of the furnace, parallel with the grate and about 10 ins. above it.

Where liquid fuel is used alone, the draught required is very small, because the greater part of the draught is always spent in overcoming the resistance of the fuel bed on the grate.

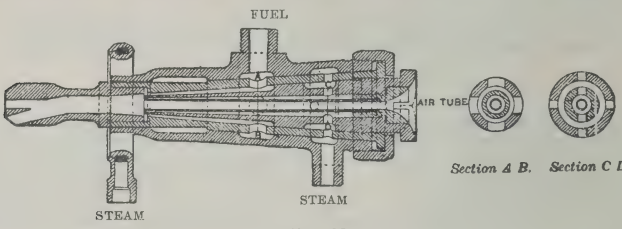


Fig. 38.

Whereas the calorific capacity of liquid fuel is roundly 20,000 B.Th.U. per pound, as compared with 15,000 for best coal, the commercial value is more nearly as 2 : 1, because liquid fuel can be burned perfectly with very little excess of air, and the temperature in the furnace is higher, the boiler efficiency is better, the boiler being also cleaner, and there are no expenses for removal of ashes and for flue cleaning. About 15½ pounds of air are required chemically to burn one pound of liquid fuel, or say 200 cubic feet of air. In the mixed system of coal and oil firing, a large excess of air may be admitted through the coal fire, and will be used by the oil, which is atomised above the fire.

In locomotives the blast-pipe may be much enlarged for liquid fuel so as to soften the draught. The Macallan variable blast-cap is used on the G. E. Section of the L. & N.E. Railway for this purpose. The flow of oil requires special regulation. Exactness is secured by such devices as a stop-cock, in which the hole through the plug is square and placed diagonally, thus admitting of fine regulation.

* From 'Liquid Fuel and its Combustion,' by W. N. Booth.

To be successful, the burning of oil fuel must follow the lines of successful bituminous coal combustion, viz., air mixture, space, and temperature; hence the need for refractory brick linings to keep off the too early chilling effect of cold plates. In steam raising, liquid fuel has a higher percentage efficiency than solid fuel, because less excess of air need be employed; furnace temperature is higher, causing higher efficiency of heating surface, which is also cleaner. There is a large saving in labour of stoking, cleaning, ash removal, etc., but perhaps greater expense in firebrick renewals.

Liquid fuels can be burned by means of mechanical burners, without steam or air spraying, if heated to not much more than 140° F., and forced through a whirling jet nozzle at a pressure of not less than 40 lbs. per sq. in. and, with benefit, up to 150 lbs.

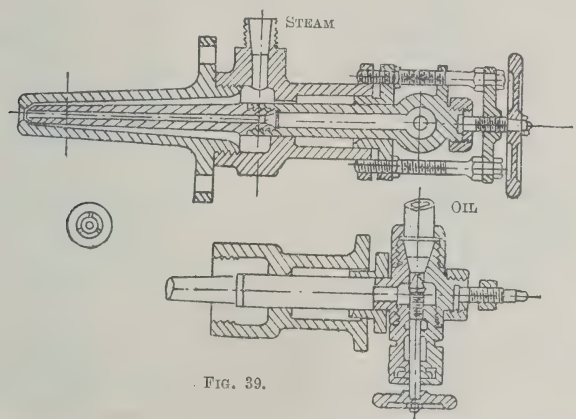


Fig. 39.

For general purposes ashore, high-pressure steam is a more satisfactory spraying agent than air. At sea, for forced draught conditions, the United States Navy Liquid Fuel Board consider air to be the best medium, especially as it avoids the otherwise inevitable loss of fresh water.

In the ss. 'Nebraskan,' oil heated to 175° F. was atomised by air of 1½ lb. pressure. The oil had a pressure itself of 15 lbs., and a very fine spray resulted; the air was heated. Retarders were necessary in the 3-in. tubes of the Scotch boilers to enable the heat of the gases to be utilised by retarding the flow, or probably by mixing up the gas.

Wallsend-Howden Patent Liquid Fuel Burning System.

In this system the liquid fuel is injected into the furnaces under pressure by means of special of fuel pumps, these pumps being fitted in duplicate so that one can act as a stand-by in cases of breakdown to the working pump. On its way to the burners the liquid fuel is passed through strainers and heaters in which it is thoroughly filtered and then heated to the correct temperature.

The oil is forced into the furnace in the shape of a conical spray of exceeding fine particles, which burst into flame at a distance of six to eight inches from the nozzle. The flame being conical, and there being no fire-bars fitted in the furnace, the whole circumference of the furnace is available for heating surface. This is a great advantage in the ordinary multitubular boiler, as the lower portion of the boiler becomes heated sooner than would be the case with coal, and consequently the circulation of the water in the boiler is improved. Moreover, as the lower portion of the boiler is heated up uniformly with the upper portions, and as there is no inrush of cold air, since there are no furnace doors to be opened, there is practically none of that straining action which results from unequal expansion and contraction, due to rapid changes consequent on the opening of the door in coal stoking.

In the improved construction of the furnace front arrangement (fig. 40), which constitutes the principal feature of the arrangement, each of the oil fuel spraying nozzles projects through a baffle plate or the front plate of the casing into an air trunk having lateral openings at its outer end. This air trunk projects concentrically with a second air trunk carried by the furnace front, and the annular space between the inner and outer air trunks is fitted with deflectors constructed in such a manner as to give the air passing through the annulus a spiral motion. One result of

this arrangement is, that the openings for the supply of air are so disposed that direct radiation of heat from the furnace is prevented. The admission of air through the second or outer air trunk keeps the outer casing cool, and hence radiation from this part is reduced to a minimum. A feature of the arrangement is the small amount of brickwork required.

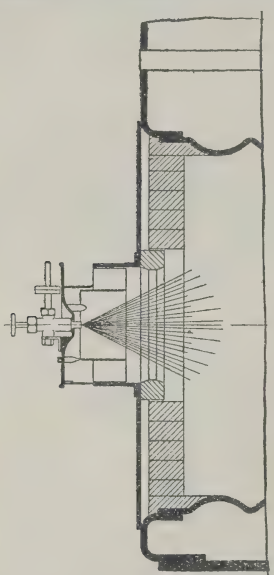


FIG. 40.—Arrangement for Working under Natural Draught or Closed Stokehold Draught.

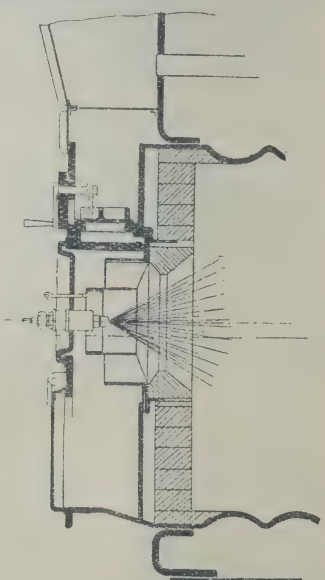


FIG. 41.—Arrangement for Working under Howden's Forced Draught.

With this system steam can be raised on any boiler from the start by means of oil fuel. This advantage is obtained by means of a special design of auxiliary heater and pump.

The Wallsend System is specially adapted for burning oil in conjunction with coal.

When burning liquid fuel on this system with Howden's Forced Draught, fig. 41, 16·22 pounds of water were evaporated (from and at 212° F.) per pound of oil burnt. The calorific value of the liquid fuel was 18,770 British Thermal Units, giving a boiler efficiency of 83·9 per cent.

THORNYCROFT OIL FUEL SYSTEM.

In this system, fig. 42, the oil is sprayed under pressure into the furnace by means of a special patent sprayer having both edges of supply grooves tangential to the whirling chamber, the spray being produced within a steel cone of special construction which introduces and mixes the air and the oil-spray in such a manner as to produce perfect combustion. The system can be applied to either water-tube or marine type boilers. Water-tube boilers can be arranged for either oil only or coal combined with oil.

The oil, which may be of any crude description such as Texas, Borneo, Mazout, or Shale, is stored in tanks in any part of the ship, and is drawn to the stokehold by a pump which forces it through the necessary strainers and the tubular steam-heater to the sprayers.

Lighting up is effected with crude oil by means of a U tube and hand-pump. Steam can be raised in from half to three-quarters of an hour.

When fitted in conjunction with Thornycroft boilers of latest design and fairly large size the makers are prepared to guarantee that the equivalent evaporation will be at least 16 lbs. of

water from and at 212° F. per lb. of oil per hour, if the rate of evaporation does not exceed 4 to 5 lbs. per sq. ft. from and at 212°, and they can guarantee a maximum output of 18½ lbs. per sq. ft. from and at with an evaporation of 14 lbs. per lb. of oil from and at. This rapid rate of evaporation has no detrimental effect on the boiler if the arrangement is properly designed,

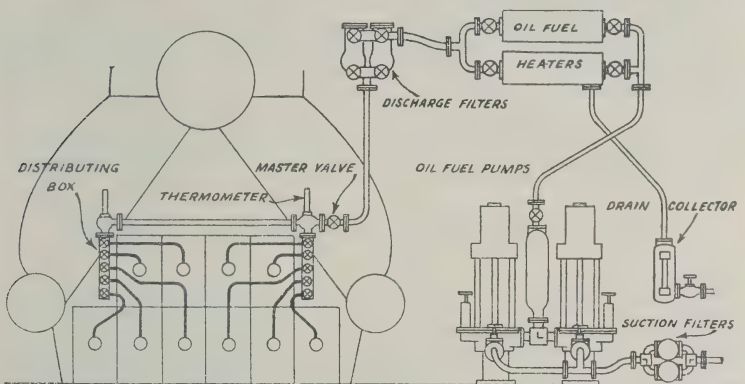


FIG. 42.—The Thornycroft Oil Fuel System.

while the combustion is smokeless up to moderate powers, and smoke is only just visible at the highest powers. These results enable very large powers to be developed on small weight, and with the highest efficiency, and at small cost. The number of men necessary for a stokehold developing 16,000 h.p. is only 3.

THE PRIOR OIL FUEL SYSTEM.

The burner in this system (which has been introduced by the Akt. Ges. Prior, of Lugano) has a trough-like form and the oil flows into it by gravity. Air for burning the fuel is supplied through rows of holes drilled in the hollow walls. By varying the positions of these air apertures and the shape of the trough the burner can be adapted, it is claimed, for widely different services.

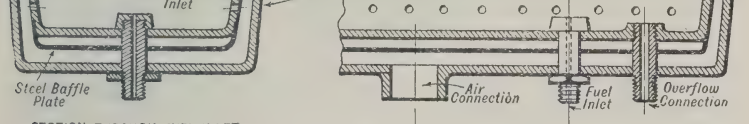


FIG. 43.

The oil to be burned is fed by gravity, and an air pressure of about 8 in. water column is provided by a fan. The burner itself (fig. 43) consists of an open cast-iron trough with hollow walls which serve as air-heating spaces. The oil is delivered at the bottom of the trough through one or more inlet connections with approximately a $\frac{3}{16}$ in. aperture, terminating about $\frac{1}{2}$ in. above the bottom of the trough. There is also an oil overflow connection, which is lower. Air to support the combustion of the oil is supplied through two rows of staggered holes which are arranged in the sides of the trough, and are given such an angle that the air jets tend to pick up the gasified fuel from the bottom of the burner. Below the bottom of the trough a steel baffle plate is welded on to the inner wall for the purpose of preventing undue heat from penetrating down to the air preheating spaces.

The burner is lighted by placing a piece of oily waste in the trough and turning on the fuel supply with a reduced quantity of air. Starting from cold the time taken may be only four to five minutes before the flame is brought up to a normal working height. If the burner has been

previously in use the lighting-up period will be still shorter, while with a thoroughly hot burner the flame can be shut off and instantaneously re-ignited. A feature of the burner is the way in which the height of the flame can be regulated from only a few inches or about the height of the trough, to its full height of about 3 ft. Under varying loads a bright and smokeless flame can be maintained. An indexed fuel cock enables a constant flame height to be maintained without any attention on the part of the attendant.

A wide variety of oil fuels has been tested with the arrangement, and it has been found possible to burn heavy Mexican oils without trouble. With such oils there is a slight asphaltic deposit on the inner surfaces of the burners, which, however, can be easily removed. When using ordinary furnace oils the deposit is of a sooty nature and is dispersed automatically by the action of the flame.

Air Required for Oil Fuel Combustion.

The amount of air required in practice at normal atmospheric temperature for the combustion of 1 lb. of oil fuel, is from 250 to 450 cub. ft.; this is 1·6 to 2·5 times the theoretical amount.

British Admiralty Specification for Oil Fuel for the Navy.

(*Abstract.*)

'The oil fuel supplied shall consist of liquid hydrocarbons, and may be either (*a*) shale oil or (*b*) petroleum, as may be required, or (*c*) a distillate or a residual product of petroleum, and shall comply with the Admiralty requirements as regards flash-point, fluidity at low temperatures, percentage of sulphur, presence of water, acidity, and freedom from impurities.

'The flash-point shall not be lower than 175° Fahr., close test (Abel or Pensky-Martens). The proportion of sulphur contained in the oil shall not exceed 3·00 per cent. The oil fuel supplied shall be as free as possible from acid, and in any case the quantity of acid must not exceed 0·05 per cent., calculated as oleic acid when tested by shaking up the oil with distilled water, and determining by titration with decinormal alkali the amount of acid extracted by the water, methyl orange being used as indicator. The quantity of water delivered with the oil shall not exceed 0·5 per cent.

'The viscosity of the oil supplied shall not exceed 2,000 secs. for an outflow of 50 cubic centimetres at a temperature of 32° Fahr., as determined by Sir Boverton Redwood's standard viscometer (Admiralty type for testing oil fuel). The oil supplied shall be free from earthy, carbonaceous, or fibrous matter, or other impurities which are likely to choke the burners.'

British Admiralty Specification for Oil Fuel for use in Diesel Engines.

'Quality:—The oil fuel supplied shall consist of liquid hydrocarbons, and may be either (*a*) shale oil, or (*b*) petroleum, as may be required, or (*c*) a distillate, or a residual product of petroleum, and shall comply with the Admiralty requirements as regards flash-point, fluidity at low temperatures, percentage of sulphur, presence of water, acidity and freedom from impurities.

'The flash-point shall not be lower than 175° Fahr., close test (Abel or Pensky-Martens). (This compares with a flash-point of 200° Fahr. in 1910.)

'The proportion of sulphur contained in the oil shall not exceed 3·00 per cent. (as against 0·75 in 1910).

'The oil fuel supplied shall be as free as possible from acid, and in any case the quantity of acid must not exceed 0·05 per cent., calculated as oleic acid when tested by shaking up the oil with distilled water, and determining by titration with decinormal alkali the amount of acid extracted by the water, methyl orange being used as indicator. (In 1910 it was required that the oil should be free from acidity.)

'The quantity of water delivered with the oil shall not exceed 0·5 per cent.

'The viscosity of the oil supplied shall not exceed 2,000 secs. for an outflow of 50 cubic centimetres at a temperature of 32° Fahr., as determined by Sir Boverton Redwood's standard viscometer (Admiralty type for testing oil fuel).

'The oil supplied shall be free from earthy, carbonaceous, or fibrous matter, or other impurities which are likely to choke the burners.

'The oil shall, if required by the inspecting officer, be strained by being pumped on discharge from the tanks, or tank steamer, through filters of wire gauze having 16 meshes to the inch.

'The quality and kind of oil supplied shall be fully described. The original source from which the oil has been obtained shall be stated in detail, as well as the treatment to which it has been subjected and the place at which it has been treated. The ratio which the oil supplied bears to the original crude oil should also be stated as a percentage.'

SPECIFICATIONS FOR BRITISH STANDARD FUEL OILS FOR HEAVY-OIL ENGINES.*
No. 209, 1924 (*Abstract*).

Item.	Specification A. B.S. Fuel Oil No. 1 for Heavy-Oil Engines.	Specification B. B.S. Fuel Oil No. 2 for Heavy-Oil Engines.	Specification C. B.S. Fuel Oil No. 3 for Heavy-Oil Engines.	Specification D. B.S. Fuel Oil No. 4 for Heavy-Oil Engines.
Flash point (closed) .	Not to be less than 150° F.	Not to be less than 150° F.	Not to be less than 150° F.	Not to be less than 150° F.
Hard asphalt content .	Not to exceed 0·5 per cent.	Not to exceed 2 per cent.	Not to exceed 4 per cent.	Not to exceed 12 per cent.
Ash content . .	Not to exceed 0·01 per cent.	Not to exceed 0·05 per cent.	Not to exceed 0·05 per cent.	Not to exceed 0·05 per cent.
Viscosity (Redwood No. 1 at 100° F.) . .	Not to exceed 75 secs.	Not to exceed 250 secs.	Not to exceed 750 secs.	Not to exceed 1,500 secs.
Water content . .	Not to exceed 0·5 per cent.	Not to exceed 1 per cent.	Not to exceed 1 per cent.	Not to exceed 1 per cent.
Cold test	Oil to remain liquid at 20°F.	Oil to remain liquid at 35°F.	Oil to remain liquid at 40°F.	Oil to remain liquid at 45°F.

NOTE 1.—It is desired to call attention to the fact that the Admiralty's Specification for Fuel Oils demands a flash point of not less than 175° F. for all fuel oils. The Admiralty's requirements as regards the other conditions specified for fuels for heavy-oil engines are met by the Specification A given above, and to a less extent by Specification B, but not by Specifications C and D.

NOTE 2.—Fuel Oils purchased to these specifications will have a gross calorific value of not less than the following :—

B.S. Fuel Oil No. 1, 19,000 B.Th.U.'s per lb. B.S. Fuel Oil No. 3, 18,750 B.Th.U.'s per lb.
" " No. 2, 19,000 " " " " No. 4, 18,500 " "

Special Oil Mixtures used for External Combustion.

PITCH-CREOSOTE MIXTURES.

Recently the use of mixtures of pitch and coal tar creosote has become common for furnace firing. The pitch creosote oils are marketed in three grades containing 25 per cent., 50 per cent., and 75 per cent. pitch respectively. As pitch is but little lower than creosote in heat value, and generally commands about half the price, there is a pronounced economic advantage in burning pitch mixtures wherever possible.

PETROLEUM-TAR MIXTURES.

The addition of tar or tar products to heavy petroleum oils causes a thick semi-solid deposit of pitch to separate, and this has till recently prevented the use of such mixtures. This difficulty has recently been overcome by the use of emulsifying agents which are capable of maintaining the constituents in a state of emulsion for prolonged periods.

COAL DUST-PETROLEUM MIXTURES.

The U.S. Navy authorities have met with success in preparing fuel mixtures from very fine coal dust mixed with petroleum fuel oil. The coal is prevented from settling by the action of emulsifying agents. Such oils are claimed to have given satisfaction for boiler firing on the U.S. warships.

Safety in Use of Oil Burners.

Serious accidents to engineers and firemen have frequently occurred from the practice of improper methods of lighting and handling oil fires.

In starting a new fire the steam should be turned on and the bypass opened in order that the oil and steam passage may be cleaned out and warmed up. The bypass should then be closed and the steam shut off. Then the damper must be opened ; on no account try to start a fire with a damper closed. Next light a torch or bunch of oily waste and place about 10 ins. to 12 ins. in front of the burner, turn on a small amount of steam (just enough to carry the oil from the burner tip to the torch) and crack the oil control valve a little. As it generally takes some little time for the oil to fill the burner passage, do not turn on more fuel but wait until the passage is filled, when the fire will light easily.

* By permission of the British Engineering Standards Association.

The moment the fire is out, either intentionally or accidentally, the oil should be shut off and the door opened. After a few minutes the gases (which are violently explosive) will have passed off and the fire may be rekindled.

Many authorities insist on the installation of valves which shut off the oil automatically if the flame goes out for any reason, and such valves should be so arranged that they cannot re-open automatically, but must be reset by hand. *The Drayton Regulator & Instrument Co., Ltd.*, supply patented valves of this description.

In regulating a fire the attendant should hold the door open just enough to enable him to watch the flame and be in a position to duck his head in case of a flare back. To increase the fire the oil valve should be operated first, then the steam turned on until the flame is clear. If not enough steam is used, the flame will appear red and smoky ; too much steam will cause the fire to look ragged and will have a tendency to extinguish it. In reducing the flame the steam is throttled down first, then the oil is reduced.

Once the fires have been set, they may be regulated over quite a range by increasing or decreasing the pressure applied to the oil supply without opening the fire-door or changing the steam supply. This can be done by manipulating the pump throttle or a bypass valve between the suction and discharge lines.

When extinguishing a fire, the flame should be cut down as low as possible ; then by opening the bypass immediately after shutting the oil valve, the oil passage will be blown out without danger, as the flame will not be extinguished until the passage is clear.—*A. C. McHugh.*

Automatic Regulation of Oil Burners.

The operation of oil burners can be made completely automatic by using in steam boilers the steam pressure, in water boilers the water temperature, and in furnaces the furnace temperature, for controlling automatic oil and steam or air valves. When air from an independently driven compressor or blower is used for atomising, it is important to provide automatic means for shutting off oil in case of failure of air supply, in order to avoid explosions.

For automatic control, oil must be kept at a constant temperature by automatically controlling the steam coil or electric heater in an oil-heating tank, otherwise control on account of viscosity changing with temperature is rendered difficult.

The Drayton Regulator and Instrument Co., Ltd., West Drayton, supply complete automatic regulating outfits.

Notes on Oil Fuel.

At a large London factory at which some Lancashire boilers were converted from coal to fuel oil firing, the water evaporation per lb. of coal, having a calorific value of 11,451 B.Th.U., was 7·22, whereas when working with the pressure system of oil burning, using an oil having a calorific value of 18,750, the evaporation per lb. of oil reached 14·44 lbs. The quantity of water evaporated per sq. ft. of heating surface with coal was 3·3 lbs., whereas, according to Mr. A. F. Baillie in a paper read before the Manchester Association of Engineers, the quantity evaporated with oil was over 7 lbs., thereby increasing the boiler rating by over 100 per cent.

In most engines of the hot-bulb type, oil fuel commonly known as gas oil or residuum is employed, this having a specific gravity of between 0·86 and 0·9. In some motors Tarakan crude oil has been used with success, whilst the other Mexican fuel oil, in spite of its relatively high percentage of sulphur, has operated very successfully. Paraffin is, of course, also suitable for these motors, but owing to its high price it is not often made use of.

Oil fuel heaters should be opened up and cleaned out periodically. In the operation of the average plant this should be done at intervals of not less than three months. Deposits of carbonised oil and other impurities quickly lower the efficiency of the heater by preventing the passage of the heat from the steam through the heater walls to the oil.

The fuel oil of oil-fired boilers is often maintained at too high a temperature. The best results are obtained where the oil is at a temperature just sufficient to insure proper and complete atomization. If too hot, the oil tends to turn into vapour and the burners puff and feed irregularly.

GASEOUS FUELS.

The more important gaseous fuels used commercially are generally mixtures. The combustible gases found in such mixtures are hydrogen, carbon monoxide together with the three hydrocarbons—methane, ethylene, and acetylene—also higher hydrocarbons of the series of which these are examples, together with benzene vapour, and occasionally other hydrocarbons.

Calorific Values, etc., of Gaseous Fuels.

In the table, p. 2169, the calorific values per lb. of the usual fuels are given, together with the weight of the products of combustion, both when burnt in pure oxygen and also when burnt in dry air. The ' higher value ' is that obtained when the water produced in the combustion is considered as having been condensed to the liquid condition, thus yielding up heat at the rate of about 970 B.Th.U. per lb. produced.

Usually in calculations relating to internal combustion engines it is found more convenient to deal with the volume rather than the mass of the gaseous mixture ; accordingly in the

Fuel. 1 lb. of :—	Sp. Gr. (H = 1).	Burnt to :—	Combustion per Lb. of Fuel.					Heat of Combustion in B.Th.U. per Lb.	
			Lbs. of Oxygen needed.	Lbs. of Air needed.	Weight of Products in Lbs.				
					Burnt in Oxygen.	Burnt in Air.	Lbs. of Nitrogen.	Higher Value.	Lower Value.
Column :—	1.	2.	3.	4.	5.	6.	7.	8.	9.
Hydrogen . . .	1	H_2O	8·0	34·80	9·0	35·80	26·80	61,500	52,800
Marsh gas . . .	8	H_2O & CO_2	4·0	17·40	2·75 CO_2 2·25 H_2O 5·00 Total	18·40	13·40	24,000	21,820
Carbon	—	CO	1·334	5·80	2·334	6·80	4·466	4,400	4,400
Carbon	—	CO_2	2·667	11·60	3·667	12·60	8·933	14,600	14,600
Acetylene . . .	13	H_2O & CO_2	3·077	13·385	3·3846 CO_2 0·6924 H_2O 4·077 Total	14·385	10·308	21,850	21,180
Carbon monoxide . .	14	CO_2	0·571	2·484	1·571	3·484	1·913	4,370	4,370
Olefiant gas . . .	14	H_2O & CO_2	3·4285	14·914	3·143 CO_2 1·285 H_2O 4·428 Total	15·914	11·486	21,900	20,660
Alcohol (abs.) (vapour)	23	H_2O & CO_2	2·087	9·078	1·913 CO_2 1·174 H_2O 3·087 Total	10·078	6·991	12,420	11,285
Normal benzene (vapour)	39	H_2O & CO_2	3·077	13·385	3·385 CO_2 0·692 H_2O 4·077 Total	14·385	10·308	18,000	17,330

following table the calorific value and weight per cubic foot at N.T.P. and also at 60° F. and atmospheric pressure are given for the usual fuels :—

CALORIFIC VALUES, AND WEIGHTS PER CUBIC FOOT.

1 Cubic Foot of Gas.	Weight in Lbs. per Cub. Ft.		Calorific Value ; B.Th.U. per Cub. Ft.			
			At N.T.P.		At 60°F. & 1 Atm.	
	At N.T.P.	At 60°F. & 1 Atm.	Higher.	Lower.	Higher.	Lower.
Column :—	1.	2.	3.	4.	5.	6.
Hydrogen	·00559	·00529	344	295	325	279
Methane	·04472	·04231	1,073	976	1,015	924
Acetylene	·07267	·06875	1,588	1,539	1,502	1,456
Carbon monoxide	·07826	·07403	342	342	323·6	323·6
Ethylene	·07826	·07403	1,714	1,617	1,621	1,529

One cubic foot of CO_2 at N.T.P. weighs ·12298 lb. ; in its formation from free carbon and oxygen 490 B.Th.U. were evolved.

At 60°F. (15·5° C.) and 1 atm. one cubic foot of CO_2 weighs ·11634 lb., and in its formation 463 B.Th.U. were evolved.

In the following table the volumes of oxygen and of air just necessary for the complete combustion of one cubic foot of gas are given, together with the change of volume of the products of combustion and the heat evolved per cubic foot of the mixture of gas and air before combustion :—

VOLUMES OF OXYGEN AND OF AIR NECESSARY PER CUBIC FOOT OF GAS, AND HEAT PER CUBIC FOOT.

One Cubic Foot of Gas.	Cubic Feet for Complete Combustion.		Vol. of Mixture before Combustion in Cub. Ft.		Cubic Feet of Nitrogen in Mixture.	Ratio of Vol. of Nitrogen to whole Volume.	Vol. of Mixture after Combustion in Cub. Ft.		Ratio of Final to Initial Volume.	B.Th.U. evolved per Cub. Ft. of Initial Volume with Air at 60°F. & 1 Atm.
	Of Oxygen.	Of Air.	With Oxygen.	With Air.			With Oxygen.	With Air.		
Column :—	1.	2.	3.	4.	5.	6.	7.	8.	9.†	10.*
Hydrogen	0·5	2·4	1·5	3·4	1·9	·56	1·0	2·9	·853	82
Methane	2·0	9·6	3·0	10·6	7·6	·716	3·0	10·6	1·000	87·2
Acetylene	2·5	12·0	3·5	13·0	9·5	·731	3·0	12·5	·962	112·0
Carbon monoxide	0·5	2·4	1·5	3·4	1·9	·56	1·0	2·9	·853	95·2
Ethylene	3·0	14·4	4·0	15·4	11·4	·740	4·0	10·6	1·00	99·3

† Of mixture with air. * Lower values.

Hydrogen.

From the above table it will be noted that a hydrogen-air mixture contracts after combustion to only ·853 of its initial volume, the pressure of the explosion being correspondingly reduced. Further, notwithstanding the very high calorific value of hydrogen *per pound*, its excessive bulkiness results in a heat evolution of only 82 B.Th.U. *per cubic foot* of its admixture with air at 60°F. and atmospheric pressure. Hydrogen-air mixtures are also very sharply explosive and trouble is experienced in engines from pre-ignition at the compression pressures usual in practice when hydrogen is present in large proportion in a gaseous fuel. Thus it is clear that hydrogen alone would be a comparatively poor and very troublesome fuel to employ for power purposes, though its presence is often beneficial in conferring sufficiently prompt inflammability upon dilute mixtures of other gases. For heating purposes a large hydrogen content is advantageous on account of the high temperature obtained in its combustion. Hydrogen constitutes about 45 per cent. by volume of modern coal gas, on the average.

Marsh Gas.

Marsh Gas (Methane) is an excellent fuel for power purposes ; it has a greater heat value per cubic foot of air mixture than hydrogen, and the volume of the products of combustion is equal to that of the mixture before combustion.

The 'natural gas' so largely used in the United States consists principally of marsh gas, that of Olean (N.Y.), for example, containing no less than 96½ per cent. of this constituent. In 1908 more than 100,000 horse-power was developed by engines operated by natural gas in the United States; the gas usually issues from the wells at high pressure, occasionally approaching 1,000 lbs. per sq. in. The Snow engine of the N.Y. & Penn. Traction Co. at Olean is supplied by natural gas issuing at a pressure of 200 lbs. per sq. in., which is reduced down to 1·8 lb. per sq. in. before being admitted, mixed with air, to the engine. With natural gas compression pressures of 150 lbs. per sq. in. are common, and mean effective pressures of 80 lbs. per sq. in. are often attained.

Acetylene.

Acetylene has so far been but little used as a fuel for internal combustion engines. It is very violently explosive when mixed with air, and owing to its endothermic nature the pressure developed is unusually high.

The range of explosive mixtures with air is also considerable; Grover found that at atmospheric pressure explosive mixtures were produced with one volume of acetylene to from 4 to 18 volumes of air; at a compression of three atmospheres even a 30 : 1 air-acetylene mixture was found to be explosive.

Acetylene is soluble in water to the extent of 1·1 volume; in alcohol to about 6 volumes; while acetone at 60°F. and atmospheric pressure absorbs about twenty-five times its volume of the gas, and at a pressure of about 180 lbs. per sq. in., about 300 volumes. A solution of acetylene in acetone contained in a porous matrix under a pressure of 150 lbs. per sq. in., is sold commercially as dissolved acetylene ('D. A.'); the 'D. A.' cylinders contain 100 times their own volume of acetylene at 60°F. and ten atmospheres. Commercial calcium carbide on addition of water usually yields about 4½ cubic feet of acetylene per lb.

If bubbled through petrol, acetylene gas becomes heavily 'carburetted'; 100 cubic feet of acetylene thus treated will produce about 150 cubic feet of 'carburetted' acetylene. According to Dr. Caro, as a heating agent this exceeds pure acetylene about in the ratio of 3 : 2.

Used as a fuel for internal combustion engines some remarkable results have been obtained with acetylene. Some early tests on a 2 h.p. engine showed that only 8¾ cub. ft. of the gas were required per b.h.p. hour, as against 35 cub. ft. of coal gas. In another case 6·35 cub. ft. of acetylene yielded 1 b.h.p. hour. And with a 6 h.p. engine using a 20 : 1 mixture, and a compression pressure of 115 lbs. per sq. in., Cuinat found that only 6·26 cub. ft. of acetylene were consumed per b.h.p. hour, the consumption by volume of coal gas being about three times as great. Specially designed acetylene engines of from 1 to 10 b.h.p. are built; these are mainly employed in pumping water for the generators of large village installations. The average consumption is about 7 cub. ft. of acetylene per b.h.p. hour, costing about 2¼*d.*; it is thus an expensive fuel for power purposes, as at present produced.

Several types of acetylene self-starting devices for the engines of motor vehicles have appeared during recent years, but unless caution be exercised in their use the extreme suddenness and violence of the starting explosion may result in damage to piston or cylinder.

See also Index.

Carbon Monoxide.

Carbon Monoxide yields the large return of 95·2 B.Th.U. per cubic foot of mixture with air, notwithstanding its low calorific value per lb.; this results from its weight being fourteen times that of hydrogen. The products of combustion occupy, however, only 0·835 of the volume of the original mixture; it is the principal combustible constituent of producer and blast furnace gas. Carbon monoxide is extremely poisonous, forming with the red corpuscles of the blood a stable chemical compound of a bright red colour termed carboxyhæmoglobin; care must always be taken to prevent any leakage of the gas into the atmosphere, and all engine and producer houses should be freely ventilated.

Olefiant Gas.

Olefiant Gas (Ethylene) is usually present to a small extent (up to about 4 per cent. by volume) in coal gas, and is a valuable producer of heat.

Owing to Government restrictions on the use of petrol during the war, many trials were made in the use of coal gas as a fuel for motors designed for the use of liquid fuels. Large gas-proof bags have been attached to the roofs of taxi-cabs, or placed underneath motor buses, and good running obtained by filling up with coal gas at frequent stopping stations. Needless to say, the mileage of such an arrangement is very limited; otherwise its performance is quite satisfactory, and, at present prices, considerably more economical than the use of petrol.

GASEOUS FUELS EMPLOYED IN GAS ENGINES.

1. Natural Gas.
2. Coal Gas.
3. Coke-oven Gas.
4. Acetylene Gas.
5. Producer Gas.
6. Blast Furnace Gas.

1. NATURAL GAS.

Natural Gas.—Large quantities of natural gas are evolved in most petroleum oilfields. The composition of the gas varies in different localities, but it always contains a high percentage of methane. The net calorific power is usually 750 to 950 B.Th.U. per cubic foot. It is used in gas engines and for heating and illuminating purposes.

2. COAL GAS.

Coal Gas is still almost exclusively used in the smaller sized stationary gas engines, especially in towns ; it is an admirable fuel for internal combustion engines of any size, however large, but its use for considerable powers is usually precluded by its relatively high cost. The introduction of the Welsbach mantle, and the largely increased use of gas for domestic heating and cooking, has enabled the gas companies within recent years to supply a gas of lower intrinsic calorific power than formerly at less cost to the consumer, and this reduction in cost has done much to encourage its use for pow.. purposes.

Calorific Power of Coal Gas.

Since the introduction of the Gas Regulation Act in 1920 the gas undertakings in Great Britain ceased to operate on official standards of candle power and are now subject to penalties in default of maintaining their respective standards of calorific power. The choice of calorific power of the gas which is distributed is left in the hands of individual undertakings, thus a wide range of quality is found throughout the country. When utilised in gas engines the factor of calorific power plays an important part, so that the operating data and adjustment of the engine must depend largely on the quality of the gas supply in any particular district. In general, it may be said that the calorific power of the gas supplied by undertakings in the country varies between 470 and 500 B.Th.U. per cubic foot, but there are a few outstanding exceptions, as, for example, the South Metropolitan Gas Company of London which distributes a gas of 560 B.Th.U. per cubic foot, and the Nuneaton Gas Company whose standard is 280 B.Th.U.

CALORIFIC STANDARDS.

	Calorific value of gas.
London (Gaslight & Coke Company)	500 B.Th.U. per cu. ft.
,, (South Metropolitan Gas Company)	560 ,, ,, ,,
,, (Commercial Gas Company)	500 ,, ,, ,,
,, (Wandsworth Gas Company)	470 ,, ,, ,,
,, (South Suburban Gas Company)	500 ,, ,, ,,
,, (Tottenham Gas Company)	500 ,, ,, ,,
Manchester	450 ,, ,, ,,
Liverpool	475 ,, ,, ,,
Birmingham	500 ,, ,, ,,
Bristol	480 ,, ,, ,,

So far as the composition of towns' gas is concerned, it is not possible to lay down any arbitrary figures for the reason that the ultimate analysis varies not only in different localities, but it varies to some degree even in the same locality in accordance with the proportion of water-gas which is added to the mixture of coal-gas and water-gas distributed. As a general guide, however, the analysis given below represents the composition of a normal straight coal-gas having a calorific power of 520 B.Th.U. per cubic foot.

Composition of 520 B.Th.U. Coal-Gas.

	Per cent. by Volume.
Hydrogen	47·0
Methane	27·5
Unsaturated hydrocarbons	3·5
Carbon monoxide	7·75
Carbon dioxide	3·5
Nitrogen	10·5
Oxygen	0·25
	100·00

The gas supplied to-day by nearly all the principal undertakings consists, however, of a mixture of from 75 to 80 per cent. of straight coal-gas (of above analysis) and 20 to 25 per cent. of either blue or carburetted water-gas. In this case the typical composition will be represented by the following analysis :—

Composition of Mixed Gas consisting of 80 per cent. Straight Coal-Gas and 20 per cent. Blue Water-Gas. (480 B.Th.U.)

	Per cent. by Volume.
Hydrogen	48·0
Methane	22·2
Unsaturated hydrocarbons	2·8
Carbon monoxide	13·8
Carbon dioxide	3·7
Nitrogen	9·3
Oxygen	0·2
	100·0

It is important for the gas-engine user to bear in mind that in recent years it has been established that one B.Th.U. will always give the same service irrespective of the quality or concentration of the mixture. In other words, from the point of view of efficiency a poorer mixture (say, 400 B.Th.U.) can be utilised as efficiently as a rich one (say, 550 B.Th.U.) so long, of course, that combustion is always complete. Proper adjustment depends, therefore, on the variation of the air supply in accordance with the quality of the gas in any particular locality. The volume of air theoretically just necessary for complete combustion may be estimated by aid of the second table on page 2170.

Increased economy in consumption is obtained by using mixtures of gas with such excess of air, or other gases, that only about 50 B.Th.U. are introduced per cub. ft. of piston displacement for cylinders up to about 20 ins. diameter, while for larger cylinders (30 ins. dia. and above) the figure must be reduced to 40 or even 35 B.Th.U. in order to avoid overheating troubles.

When excess of oxygen is present in the mixture, practically no carbon monoxide is found in the exhaust; with deficiency of oxygen, or irregularity in the mixture, the contrary is to be expected.

Usually in practice the volume ratio of mixture employed is from 7 to 10 of air to 1 of coal gas; the richer mixture gives a greater power output, the more dilute a greater economy in consumption of gas per horse-power hour.

3. COKE-OVEN GAS.

Coke Oven Gas is produced in large quantities at many collieries in the process of manufacture of coke for metallurgical purposes; its composition varies somewhat widely, but it is usually specially characterised by its large content of hydrogen; this renders it peculiarly liable to pre-ignite on compression, and lower compression pressures than usual are generally adopted (90 to 100 lbs. per sq. in. only) on this account when coke-oven gas is used for power purposes.

Recently, however, the pre-ignition difficulty has been entirely overcome by Sir Dugald Clerk's patented process of introducing from 10 per cent. to 20 per cent. by volume of cooled *exhaust gas* into the inlet pipe; the addition of this inert gas reduces inflammability by diminishing the quantity of oxygen present, and diluting the charge without reduction of its mass. The application is very simple and easily made, and the maximum practicable power of the engine is materially increased.

An average characteristic composition by volume of coke-oven gas may be taken as :—

H, 50 per cent.; CH_4, 25 per cent.; CO, 5 per cent.; $CO_2 + N + O$, 20 per cent.

Such a gas has a lower calorific value per cub. ft. at 60° F. (15·5° C.) and 1 atm. of about 390 B.Th.U., and requires, roundly, 3¾ cub. ft. of air for complete combustion; the heat evolved per cubic foot of this maximum mixture is accordingly $\frac{390}{4{\cdot}75} = 82$ B.Th.U. Actually, the calorific value varies from 350 to 450 B.Th.U. per cub. ft. at 60°F. and 1 atm., and requires from 3 to 4 cub. ft. of air for complete combustion.

In a test by Prof. E. Meyer of a 500 h.p. two-stroke Borsig-Oechelhauser engine of 26½ ins. dia. bore and 37½ ins. stroke (of each piston), running at about 106 revolutions per minute, and driven by coke-oven gas of 395 B.Th.U. (lower) value per cub. ft., the heat consumption per b.h.p. hour deduced from the blower driven by the engine amounted to 9,070 B.Th.U. corresponding to $\frac{9070}{395} = 23$ cub. ft. of coke-oven gas per b.h.p. hour only; the mean effective pressure averaged 70 lbs. per sq. in.

4. ACETYLENE GAS.

Experiments carried out in Germany with acetylene gas for internal combustion engines showed that the proportion of air to gas should be in the ratio of 12·02 to 1 to avoid the formation of soot. The maximum compression which may be used for a mixture of acetylene and air is about three atmospheres, compared with 4–5 atmospheres for petrol and 10 atmospheres for coal gas. One ton of carbide is sufficient to produce 10,000 cubic feet of acetylene.

5. PRODUCER GAS.

In normal combustion, when an excess of air is present in the combustion space, the carbon of the fuel is burnt to form carbon dioxide according to the reaction

$$C + O_2 \rightarrow CO_2.$$

Nitrogen from the air passes through the combustion space into the resulting gas without entering into the reaction. By this reaction one pound of carbon generates 14,647 B.Th.U.

By restricting the air supply and arranging to maintain a comparatively great depth of fuel upon the grate the carbon may be burnt into carbon monoxide according to the reaction

$$2C + O_2 \rightarrow 2CO,$$

nitrogen again passing through the grate without being affected.

By this latter reaction, 4,400 B.Th.U. are generated for each pound of carbon gasified, the residual heat value obtainable from the carbon by complete combustion, *i.e.* 14,647 − 4,400 = 10,247 B.Th.U., remaining in the resultant gas.

The greatest quantity of heat obtainable in the form of cool carbon monoxide from the air-carbon reaction is therefore only $\frac{10,247}{14,647} = 70$ per cent. of the heat originally available in the carbon.

Of the 30 per cent. of heat given out during the reaction, a large portion leaves the producer in the form of sensible heat.

The heat generated in the producer is sufficient to cause a temperature too high for satisfactory working.

In order to cool the producer and to utilise this excess of heat, steam is introduced into the combustion space, when, if conditions are favourable, the following reaction takes place:

$$C + H_2O \rightarrow CO + H_2$$

For each pound of fuel entering into the reaction 4,320 B.Th.U. *are absorbed* from the heated fuel.

By the use of air and steam in certain proportions, it is possible to absorb the excess heat generated by the air reaction and maintain a practically constant temperature in the producer.

By alternately blowing the producer with an excess of air and raising its temperature by the combustion of carbon, either to the monoxide or to the dioxide, and then cutting off the air supply and passing steam into the hot fuel until the producer temperature has again fallen, it is possible to generate water gas (the result of the steam-carbon reaction) which may be collected separately.

Water gas in conjunction with oil gas (generated by the cracking of heavy petroleum distillates) is largely used for the preparation of illuminating gas in normal times.

As the fuel used is not pure carbon, other considerations than those mentioned in the above theoretical outline affect producer design and working. As coals undergo a destructive distillation in the producers it is necessary to make use of special devices for the removal of tar from the producer gas. The recovery of ammonia generated from the nitrogen of the coal is also of the greatest importance in the economics of producer gas plants. These particulars are treated in greater detail in the following descriptions of various types of plant.

(*a*) Pressure producer gas from anthracite (Dowson).
(*b*) „ „ „ coke („)
(*c*) Suction and suction-pressure producer gas from anthracite (Dowson).
(*d*) „ „ „ „ wood, sawdust, chips (Dowson).
(*e*) „ „ „ „ charcoal (Dowson).
(*f*) „ „ „ „ coke and anthracite (Dowson).
(*g*) „ „ „ „ coke (Dowson).
(*h*) „ „ „ „ bituminous coal (Dowson).
(*i*) Pressure producer gas from bituminous coal (Mond).
(*j*) Suction and suction-pressure producer gas from anthracite (Mond).
(*k*) „ „ „ „ coke („)
(*l*) „ „ „ „ charcoal („)
(*m*) Suction-pressure producer gas from wood, semi-bituminous („)
(*n*) Suction producer gas from bituminous coal (Farnham).
(*o*) Producer gas from other substances, as colliery refuse, sawdust, wood chips, cocoanut shells, belt pickings, lignite, and peat.

In the table on page 2175 some results of analyses are given of producer gas from anthracite, coke, and bituminous coal. The number of cu. ft. of air necessary for just complete combustion is obtained by aid of table on page 2170 from the formula :—

$$\text{Cu. ft. of air} = \frac{4{\cdot}8}{100}\left\{\tfrac{1}{2}(H + CO) + 2\ CH_4 - O\right\}$$

Gas Producers and Producer Gas.

Some account of producers and producer gases from material supplied by manufacturers of gas plant is given hereunder :—

Dowson Gas.

The Dowson gas is made by passing a mixture of superheated steam and air through incandescent fuel. There is no external fire, as in ordinary gas retorts, and no skilled labour is required to work the apparatus.

The original type of Dowson plant included a small steam boiler, to give the steam required and to work an air injector. Excellent results have been obtained in this way, and for some purposes it is still the best plant to use. It is now known as the 'Dowson Steam-jet Pressure Plant,' and is used when gas is required from one apparatus to serve several gas engines or one or more engines and heating appliances.

COMPOSITION OF PRODUCER GASES WITH HEAT VALUES, ETC.

Description of Producer Gas.	Kind of Fuel used.	Per Cent. Composition by Volume.						B.Th.U. per Cub. Ft. (lower) at 60°F. (15·5° C.) and 1 Atm.	Cub. Ft. of Gas made per Ton of Fuel.	Cub. Ft. of Air required per Cub.Ft. of Gas.	B.Th.U. evolved per Cub.Ft. of Mixture.	Per Cent. Thermal Efficiency of Producer on lower value of Gas.
		H.	CH_4.	CO.	N.	CO_2.	O.					
Column :—	1.	2.	3.	4.	5.	6.	7.	8.	9.	10.	11.	12.
Dowson Pressure	Anthracite	19·8	1·3	23·8	48·8	6·3	0·0	150·7	171,500	1·41	60·0	82
Dowson Suction	Anthracite	15·7	1·8	22·0	54·5	6·0	0·0	137·5	180,000	1·02	61·8	80
Dowson Suction	Gas Coke	13·0	0·55	25·4	55·65	5·4	0·0	130·0	168,000	0·95	63·3	75
Mond 1. Pressure	Non-Caking Bituminous Coal	22·65	3·5	16·05	44·55	13·25	0·0	147·3	147,500	1·26	65·2	60·4*
Mond 2. Pressure	Ditto	16·60	3·35	27·3	47·5	5·25	0·0	165·4	138,250	1·38	69·6	71·5*
Mond 3. Pressure	Ditto	17·36	1·20	25·55	49·82	5·77	0·30	142	190,000	1·13	66·7	—

* Including steam for blowing engine and washers.

1. With ammonia recovery and steam saturation temp. 175°F. (79·4° C.). 2. Ditto, and 140°F. (60·0° C.). 3. Without ammonia recovery.

Another type of Dowson gas-making plant is the 'Dowson Suction Plant,' which has been installed in large numbers in this and many other countries. This type of apparatus requires no boiler or gas-holder, and the engine itself draws the air required through the fire in the gas generator: the engine, therefore, governs the rate of producing the gas to suit its varying consumption, and there can be no waste of gas. Moreover, the dry steam required is produced by waste heat, and the fuel which is to be converted into gas is also heated by waste heat before it reaches the combustion zone. Throughout the apparatus the loss of heat is therefore reduced to a minimum, and the efficiency is exceptionally high. Sir Dugald Clerk's 'James Forrest Lecture,' delivered in 1904 to the Institution of Civil Engineers, refers to exhaustive tests made by him with this suction plant, the heat efficiency being proved to be about 90 per cent.

In both types of the Dowson apparatus anthracite peas or nuts, charcoal, or coke can be used. The consumption depends somewhat on the type of plant and the kind of fuel used. With the suction plant the Dowson Company guarantee that the consumption of small coke of good quality will not exceed 1¼ lb. per b.h.p. per hour, at full load, with any good gas engine suitably adjusted for working with the plant.

If anthracite peas (from ⅜ in. to ¾ in. cube) are used, the consumption is guaranteed not to exceed 0·9 to 1 lb. per b.h.p. per hour.

As regards stand-by losses careful independent investigations have shown that at four different places Dowson gas generators, ranging from 100 to 375 b.h.p. each, during long stand-by hours, consumed an average of only 3 lbs. per generator per hour, while gas of good quality could be ready for use in about ten minutes. With steam power the stand-by loss is far greater.

A further important advantage of the gas is that it does not condense or lose power on its way to the engine; on the contrary, the more the gas is cooled the better it is for the engine, as each charge of gas which enters the cylinder is more dense, and has consequently greater energy. With steam the condensation in the pipes, etc., is considerable; with gas there is no such loss. It is easy to prevent leaks of gas from the piping, owing to the low pressure of the gas, whereas with steam there is often much loss and inconvenience on this account.

With the suction plant there cannot be a leak of gas when the engine is working, but piping, scrubbers, etc., must be absolutely tight to avoid air leaks.

The wages of firemen are about the same for the gas plant as for a steam boiler of equal power.

The quantity of water per b.h.p. per hour in the gas plant is about 1 gall.; with steam power it is usually two or three times as great. The cost of repairs (pre-war period) is approximately as follows:—

For 100 b.h.p.	£4 per annum.	For 400 b.h.p.	£12 per annum.
„ 200 „	6 „ „	„ 500 „	15 „ „
„ 300 „	9 „ „		

As regards labour, with the suction plant the usual allowance is one man for about an hour each day to start the fire, clean, etc., and about one hour per day to put fuel in the generator. When the plant is worked with a boiler the labour required is greater, but not more than for steam power of equal size.

Although about four volumes of Dowson gas are required to do the same work in a gas engine as one volume of town gas, yet Dowson gas is produced so cheaply that the equivalent of 1,000 cubic feet of ordinary town gas costs only seven to twelve pence, according to the scale of working and price of fuel.

With Dowson gas 1 lb. of coal will produce ·70 cubic feet of gas, and 1 cubic foot of this gas requires only 1·1 cubic foot of air for complete combustion, against 6 to 8 cubic feet for ordinary gas.

A complete Dowson apparatus does not occupy more space than that required for the boiler of a steam engine of the same power.

Dowson gas is used for heating and furnace work of all kinds, as it is a cheap and cleanly fuel, under perfect control.

Their latest plant for heating is known as the high pressure type wherein the gas can be boosted to any pressure up to 100 ins. w.g., delivering the gas clean and steady to the burners. The producer is of the open-hearth type similar to the producer used on their latest suction plant. Space will not permit giving varied examples of the performance of plant and the splendid results with the gas obtained, even for delicate work such as singeing.

The following comparative estimates (pre-war period), for 25 and 50 b.h.p., show that it is much cheaper to use gas power than electricity, even where the latter is sold at 2*d.* per unit:—

25 B.H.P.	Per week. £ *s.* *d.*	Per annum. £ *s.* *d.*
Electricity—25 B.H.P.=18¾ B.Th. units × 50 hours = 937½ units at 2*d.*	7 16 3	390 12 6
Town Gas Power.—25 B.H.P. × 20 cubic ft. × 50 hours = 22,500 cubic ft. at 2*s.* 6*d.*	2 16 3	140 12 6
Dowson Gas Power.—25 B.H.P. × 85 cubic ft. × 50 hours = 106,000 cubic ft. at 3*d.*	1 6 6	66 5 0

50 B.H.P.	Per week. £ s. d.	Per annum. £ s. d.
Electricity.—50 B.H.P.=37½ B.Th.units×50 hours =1,875 units at 2*d.*	15 12 6	781 5 0
Town Gas Power.—50 B.H.P. × 20 cubic ft. × 50 hours=45,000 cubic ft. at 2*s.* 6*d.*	5 12 6	281 5 0
Dowson Gas Power.—50 B.H.P.×85 cubic ft.×50 hours=212,000 cubic ft. at 3*d.*	2 13 0	132 10 0

More recently Mr. Dowson has introduced a new type of producer, to work with bituminous coal, which is a new departure in several respects. In producers which are worked with bituminous coal the tarry vapours from the hydrocarbons in the fuel mix with the gas, and the tar is usually afterwards extracted in separate apparatus. Mr. Dowson has always held the opinion that the tar should be got rid of in the producer itself, and he has succeeded in doing this.

The producer is double-acting—*i.e.* air is drawn in through the top and through the bottom of the fuel column. The coal is put in at the top, and the upper part of the fire burns downwards. The hydrocarbons are distilled off and various reactions occur, the coke which remains sinks downwards into the lower part of the producer. The coke is there converted into ordinary producer gas by the upward current of air (accompanied by steam), which is drawn in at the bottom. The mixture of gases leaves the producer through an outlet about halfway between the top and bottom. There is no external fire, and the gas is made as quickly as it can be consumed, as in the other types of Dowson gas plant.

When working with ordinary bituminous coal, or with lignite and several other varieties of fuel, there is no tar; it is removed altogether in the producer, and no mechanical or other tar extractor is required. After leaving the producer the hot gas passes through a vaporiser to cool the gas, and to raise the steam required for making the gas. It then passes through suitable scrubbers and is ready for use in the engine to be worked with it.

This bituminous plant is worked by suction, and consequently there is no escape of gas; it is also easy to work, and the cost of labour is about the same as with an ordinary plant working with anthracite or coke.

When the plant is to be worked by the suction of an engine as in the smaller powers, viz., 25 up to 100 h.p., there is a fan to start it, as in ordinary suction plants, and this is stopped when the engine is running. For powers 100 h.p. and upwards there is a fan or blower to draw the gas from the plant and deliver it to the various engines or heating apparatus requiring gas. Each fan or blower produces the suction on the plant and governs the make of gas. Of course with all heating work and where there is more than one engine requiring gas, even for the small powers, a fan or blower is always installed to deliver the gas under pressure.

The quality of the gas made in this plant is about the same as that of gas made with coke in ordinary suction plants. The heat efficiency of the new plant is exceptionally high, and with a good engine under full load the consumption of fuel is little over 1 lb. per b.h.p. hour. The stand-by losses are small, and after standing over-night or during meal hours, good gas can be ready for the engines in a few minutes.

Many plants of this description have been installed both at home and abroad for industrial and municipal undertakings with splendid results, and in the case of the foreign installations the coal in some cases being much inferior to our British standard. These plants can be made in any size from 25 to 500 b.h.p.

The Kerpely Revolving Grate Gas Producer.

The Kerpely Gas Producer, with its mechanical ash discharge and mechanical feed and agitators, is entirely mechanical in operation. It effectively gasifies good or bad fuels, even waste fuels containing nearly 50 per cent. ash. As an example of the latter fuel, mention is made of an installation of Kerpely gas producers at a colliery in Scotland for gas-firing colliery boilers. The fuel gasified is unsaleable from the colliery and consists of 'parrot' and 'belt pickings' of the following approximate analysis:—

Parrot and Belt Pickings Mixed.

Fixed carbon (by difference)	24·50
Volatile matter	23·22
Ash	48·10
Moisture	4·18
	100·00
Calorific value	6,580 B.Th.U. per lb.

It will be noted that the fuel contains *nearly 50 per cent. of ash.*

The following is an average analysis of the gas produced from this fuel by the Kerpely gas producers, viz.:

CO_2	5·80
O	0·58
CO	23·21
H	11·50
CH_4	2·85
N	56·06
	100·00

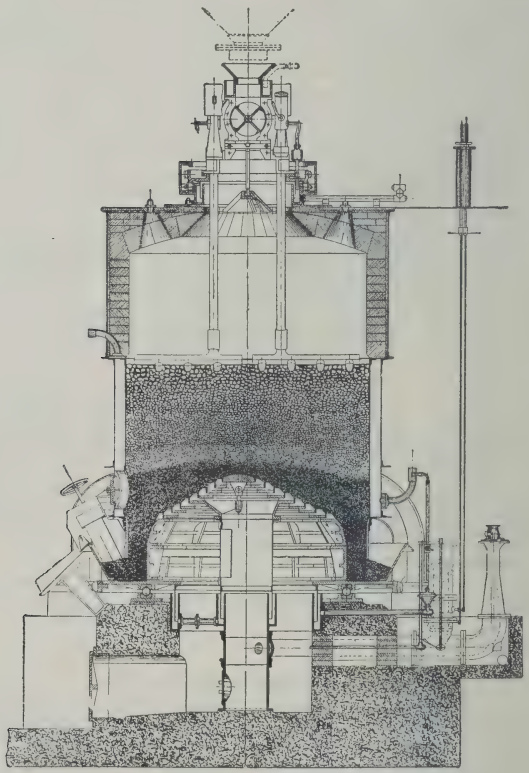


FIG. 44.—The Kerpely Revolving Grate Gas Producer.

The Kerpely grate, a sectional view of which (dimensioned for the standard 3-metre diameter producer) is shown by fig. 44, is of polygonal shape and is placed eccentrically on the water trough. It has a continuous and not an intermittent action in crushing any formation of clinker that may occur. The water-cooled jacket in the fire zone prevents the formation of clinker on the walls of the producer. The clinker, when entering the ash zone, is caught by the numerous edges of the revolving grate and pressed horizontally sideways against the seal apron, and is consequently broken or crushed into small fragments, which, dropping into the water trough, are removed by the ash scraper. The latter automatically discharges the ash and clinker accumulating in

the bottom of the producer, and, being constructed to suit the shape of the ash trough, the ash is not only pushed aside, but is piled up and continuously discharged into an ash shoot. The automatic discharge of the ash and clinker from the Kerpely producer results in considerable saving of labour in the removal of ash, and ensures continuous working of the producer and gas of uniform quality. Labour is saved in poking, a small amount only of steam is required, and a dry gas is obtained, which ensures a prolonged life of the furnace linings. The supply of air and steam to the Kerpely producer is separately conveyed to the inner and outer sections of the grate, and, being transmitted through spaces between the plates of the revolving grate, ensures uniform combustion over the whole grate area, the maximum gasification capacity per square foot of grate area, and effective control of the fuel bed.

The Kerpely producer can be fitted with the Appleby-Bentley Patent Mechanical Rotary Fuel Feed and Distributor, or this mechanical feed combined with patent balanced and adjustable fuel agitators and levellers, as the fuel conditions may determine.

These mechanical feeds and agitators are claimed to ensure greatly increased gasification (50 to 100 per cent., according to type of producer), constant high-value gas, saving of labour and fuel, and greater production from furnaces. The following table gives daily working results taken over a period of one month with a Kerpely gas producer fitted with the Appleby-Bentley mechanical feed and agitator:—

KERPELY PRODUCER WITH MECHANICAL FEED AND AGITATORS (BENTLEY'S SYSTEM).

Gas Analysis Monthly Report, June 1922.

Day.	Saturation °C.	Steam Press at Blowers.		Gas Analysis.					Total Combusts. per Cent.	B.Th.U.	Kind of Coal.
		Outer.	Inner.	CO_2 per Cent.	CO per Cent.	CH_4 per Cent.	H_2 per Cent.	N_2 per Cent.			
1	59	25	45	3·4	28·0	3·43	13·10	52·07	44·53	167	A
2*	61	20	40	5·4	24·6	3·74	12·46	53·86	40·86	157	A
3	61	20	40	3·2	28·2	3·66	14·18	50·76	46·04	174	A
6	56	25	50	3·2	28·4	4·11	13·39	50·90	45·90	177	A
7	57	25	50	2·8	29·6	3·63	13·30	50·67	46·53	176	A
8	57	25	50	2·6	29·0	3·86	13·93	50·61	46·79	178	A
9	57	25	50	3·2	29·0	3·63	13·01	51·16	45·64	173	A
10	57	25	50	3·6	28·0	3·45	12·28	52·67	43·73	165	A
12	60	20	45	3·8	28·0	3·41	13·50	51·29	44·91	165	A
13	62	27	50	2·4	29·2	3·88	13·09	51·43	46·17	176	A
14	63	25	50	3·2	28·4	3·42	14·29	50·69	46·11	173	A
19	62	25	50	4·0	27·2	3·21	13·46	52·13	43·87	165	A
20†	62	33	60	4·4	25·2	3·70	13·88	51·82	42·78	154	A
21	62	30	40	2·4	29·0	3·66	12·81	52·13	45·47	172	A
22	62	30	40	2·8	29·0	4·38	12·73	51·19	46·11	179	A, B
23	60	30	40	3·2	29·0	3·63	12·69	51·46	45·32	173	C
26	61	30	50	3·8	27·0	3·45	15·28	50·47	45·73	172	D
27	61	23	45	3·2	27·8	3·71	13·80	51·49	45·31	172	D
28	60	30	50	3·6	28·6	3·18	13·01	51·61	44·79	167	E
29	60	30	50	3·4	27·6	3·45	14·12	51·42	45·17	170	E
30	63	35	60	4·0	27·4	4·14	14·92	49·94	46·06	173	E

Comments.—Carbon in ash (sample taken 9th inst.), 2·77 per cent.

* Residue from bunkers, mostly dust.

† Day of stoppage for burning out.

A = Mansfield. B = Cannock. C = Great Wyrley. D = Cresswell. E = Langwith.

APPLEBY-BENTLEY PATENT MECHANICAL GAS PRODUCER.

This producer is mechanical in its action, as regards fuel feeding, agitating and levelling, ash levelling and discharge. The fuel is carbonised, then gasified, producing high or low temperature gas as desired from the same producer. The apparatus has an improved blast distribution, effecting perfect combustion with high gasification. Cheap fuels can be effectively gasified by its means.

THE 'CROSSLEY' SUCTION GAS PLANTS.

Messrs. Crossley Brothers, Ltd., build several types of suction gas plants, each specially designed to suit the various classes of mineral and refuse fuels.

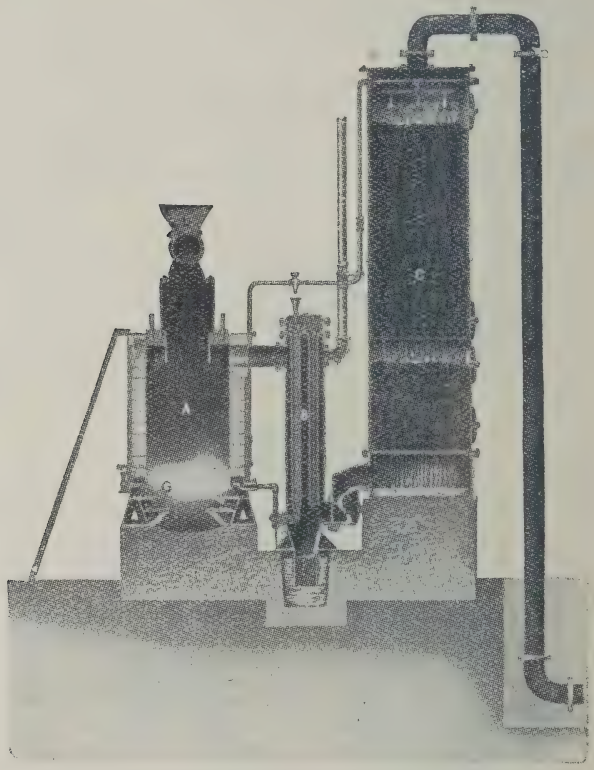


FIG. 45.—'Crossley' Suction Gas Plant.

The producer manufactured by them to work on anthracite, coke, or charcoal is of the open-hearth type shown by fig. 45.

With this type of producer the fire is always in full view of the operator, the firegrate consisting of a number of annular cast iron plates arranged in the form of steps and so disposed that the outside edge of each succeeding lower plate lies beyond the angle of repose of the fuel which falls through the opening in the plate above. These grate plates are supported on cast iron legs which also support the producer itself, comprising a simple steel shell internally lined with firebricks.

The grates are cooled by water which is supplied by small drip pipes to the top plate, from whence it overflows on to each succeeding plate in turn; a certain amount of steam is thus generated which assists in the gas production.

As will be seen from the fig. the lower part of the fuel bed can at all times be readily cleaned of ash and clinker without interfering with the process of gas making or necessitating shutting down of the plant, indeed, numbers of these plants have worked continuously for over 12 months without the fire having to be withdrawn for any purpose whatever.

The absence of a vaporiser in the top of the producer enables the operator to use a poker in a direction parallel to the firebrick lining, the clinkers being thus very easily broken away, without injury to the brickwork, and allowed to drop on to the grate plates, from whence they can be removed at will.

From the producer the hot gas passes into an external vaporiser consisting of a casing supporting a cast iron box divided into compartments, to the under side of which a number of water tubes are fixed.

With the exception of the first and the last in the series. the tubes are provided with internal circulating tubes of the ' Field ' type.

The water to be vaporised is carried to the bottom of the first tube through an internal feed pipe and then rises into the first compartment, from which it passes in through each succeeding compartment and tube in turn, until finally it is flashed into steam in the last tube, which also forms the steam connections between the vaporiser and the producer. The steam is taken from the vaporiser into an annular box-shaped casting situated at the base of the producer, from which it is distributed to the fuel bed through a number of small holes drilled around the inside edge.

In this design of plant the gas is external to the water to be heated, so that both the cooling of the gas and heating of the water are effectually provided for.

Leaving the producer, the partially cooled gas passes through two cascades of water produced by the overflow from the cooler and scrubber ; these remove the heavier impurities from the gas. It is next passed through a cooler and scrubber consisting of two coke charged compartments, each fitted with separate cleaning and charging doors, so arranged that the coke in the lower compartment (which is the first to become fouled) can be removed without interfering with that in the upper compartment. During its passage through the coke the gas is thoroughly cooled by streams of water supplied by the spray situated above the top layer.

Leaving the scrubber the gas passes into a combined drier and expansion box, and from thence to the engine.

' Crossley ' Refuse Fuel Plants.

' Crossley ' refuse fuel plants are designed to work on a variety of fuels, including waste wood of all kinds, spent tan, rice husks, olive refuse, sugar cane refuse, cocoanut shells, coir dust, cocoa and coffee refuse, etc. Particular attention has been given by the designers to the removal of impurities from the gas and to allow of continuous working of the plant by unskilled labour. A sectional view of a ' Crossley ' Refuse Plant is shown by fig. 46.

The lower part of the producer is similar in design to that of the anthracite plant previously described, with the exception that the grate is shielded by carrying the external shell to the foundation block.

Access to each section of the grate for removal of ash is provided for by large cleaning doors bolted to the producer shell ; these doors can be opened at any time whilst the plant is at work without disturbing the process of gas production.

Owing to the comparatively high moisture content of nearly all classes of waste fuels, little or no steam is required for gasification purposes, but when necessary, provision is made for generating a supply of steam by feeding water on to the stepped grate plates.

One of the features of this plant is the provision of an Airlock fuel feeding hopper of large capacity and special design to allow of the plant being used for all classes of fuel without the slightest fear of explosion or interruption of gas supply when charging.

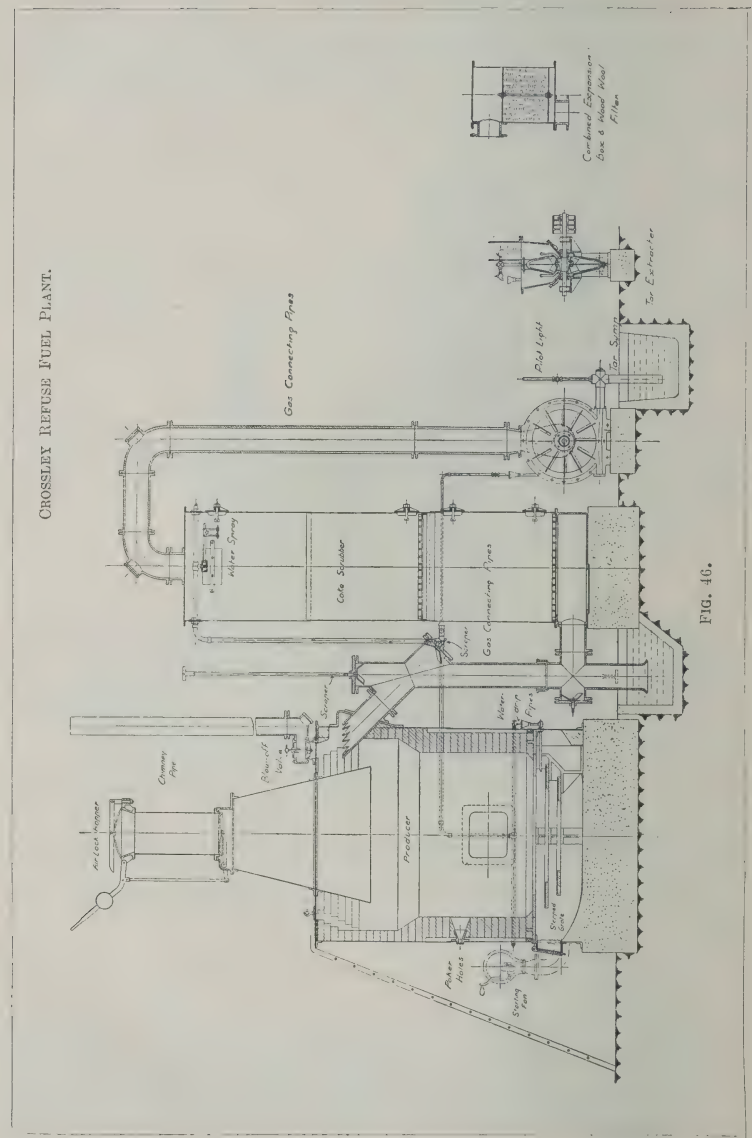
Other features of this producer are as follows :

(1) Provision for storing a large quantity of fuel in the producer itself, thus obviating the necessity of frequent charging.

(2) Exceptional facilities for spreading the fuel, thus ensuring a constant fuel depth and gas of constant calorific value.

(3) An effective design of mouthpiece for the gas outlet pipe, to reduce to a minimum any tendency for fine fuels, such as sawdust, to be drawn over with the gas.

(4) Gas outlet pipes from producer to scrubber are provided with scrapers for removing tarry deposit and dust whilst the plant is in operation. This device prevents the pipes from becoming choked and thus enables the installation to work continuously for long periods.



CROSSLEY REFUSE FUEL PLANT.

FIG. 46.

For cooling and washing the gas (preparatory to the removal of tar) a coke scrubber (of similar design and construction to that previously described) is provided.

Leaving the top of the scrubber and cooler, the gas is passed through a mechanically driven centrifugal tar extractor, where it is subjected to a combined heating and centrifugal action which throws out the tar on to the inside of the casing, from whence it is washed into the tar collecting sump by a continuous spray of water.

Before the gas reaches the engine it is passed through a combined expansion box and wood wool filter which acts as a final drier.

Messrs. Crossley Brothers also manufacture suction plants for bituminous coal, lignite or peat; these being of different design to suit the characteristics of the fuel to be gasified. The bituminous fuel plant is provided with a central cone plate grate which allows of a continuous water lute around the base of the producer for the removal of ash and clinker.

All steam required for gasification purposes is generated in an annular steel boiler, situated near the base of the producer, the smooth surface of which prevents clinker from adhering. The steam and air supply to the producer is controlled by a single valve, and the saturation temperature of the air can be regulated to a degree.

This design admits of free use of pokers through both the top and side poking holes.

For feeding the fuel a rotary barrel hopper is provided. The gas cleaning apparatus is very similar to that supplied with the 'Crossley' Refuse Plant.

Mond Gas.

The Mond Gas-Producer Plant was originally developed by the late Dr. Ludwig Mond for the conversion of bituminous fuels into power and heating gas, along with the simultaneous recovery of the nitrogen in the form of sulphate of ammonia.

Plants of both the recovery and non-recovery type are constructed for power and for heating purposes, but considerations of economy rarely justify the installation of the recovery type in units of less than 2,000 h.p. or equivalent capacity. The non-recovery type, however, is installed in much smaller units.

Both types of plant can be adapted for utilising inferior fuel, such as the waste products from collieries and coke ovens, and the gas used for gas-engine driving, industrial furnace heating or boiler firing, and in such cases with the recovery plant the return from the sulphate of ammonia will go a good way towards meeting the running expense of the plant.

The Mond Non-recovery Water-cooled Gas-Producer Plant.

(*Mond and Lymn System.*)

The modern type of Non-recovery Water-cooled Gas-Producer Plant, as installed for generating gas from bituminous producer coal of suitable grading, quality, and condition for use in gas engines or for other purposes where cleanliness equal to gas-engine practice is required, may be considered as composed of two sections: the 'Gas Generating Section'—comprising the producer and its accessories—and the 'Gas Cooling and Cleaning Section'—comprising vertical static cooler and washer, centrifugal cleaners, and sawdust scrubber.

The producer is of the central blast type, with steel casing designed to sit on concrete foundations (provided by purchasers) and dip into a water-lute, formed in the foundations, and from which the ashes can be removed evenly all round without interrupting the working of the producer. The grate is conical in form, built up of two or more C.I. sections, and provided with air slots. It is located concentrically with the producer shell, and is connected up to the blast pipe which passes through the foundations to the outside of the producer. The lining of the producer is built of special quality firebricks and fireclay. The producer is fitted with a series of horizontal poke holes about the level of the air slots in the grate, and also with poke holes on the top giving access to the whole of the fuel bed, these top poke holes being fitted with spherical closers.

A fuel hopper is fitted on the top of the producer, with sliding cover and inner cone valve operated by lever and balance weight. The fuel in the hopper is discharged into a fuel container or inner bell, the top of which is provided with suitable poke holes, and from this container, automatically on to the fuel bed; this method of charging ensuring a more constant quality of gas. The gas outlet branch is fitted on the side of the producer casing close to the top.

A steel chequer plate operating platform is arranged at the level of the top of the producer shell with suitable protecting handrailing and ladder giving access from ground level.

Fuel handling plant is frequently included, and the type usually recommended comprises a power-driven enclosed bucket type elevator, lifting the fuel from a boot at ground level into a bunker, which is mounted on substantial supports over the producer and fitted with a shute furnished with lever-operated discharge valve leading down to the producer hopper.

The air-blast for the producer is provided by a blower of the steam-jet type, arranged so as to ensure a thorough intermixture of air and steam before the blast reaches the grate, but in some installations a steam-driven Roots or turbo-type blower is included, the exhaust steam from the blower engine or turbine being led into the blast pipe.

Leaving the producer, the gas enters the cooler and washer, which is of patented design, and provides an effective means of gas washing by the employment of a novel and efficient method of utilising the momentum of upward flowing gases to atomise liquid falling in thin sheets, with a resultant intimate contact between gases and liquid without the use of any mechanical spraying device.

The apparatus consists of a cylindrical casing of mild steel, fitted with top and bottom plates, gas inlet and liquid outlet at the bottom, and with gas outlet and liquid inlet at the top. When desirable, the bottom of the casing can be arranged for water-luting.

A vertical stationary shaft is located in the centre of the apparatus, extending the height of the casing. At suitable distances from each other, a series of discs or distributing devices for the liquid are fixed on this shaft, and attached to the casing are a corresponding series of inverted truncated cones which serve as collecting devices, the design of these internal parts being varied to suit particular requirements.

Falling by gravity alternately from the distributors to the collectors and *vice versa*, the thin sheets of liquid are atomised, and in the form of fine spray actually driven by the gas—itself flowing in zigzag path in the opposite direction—on to the collectors and distributors, where the spray again forms into thin sheets of descending liquid. Each particle of gas is thus brought into contact with the liquid many times before it leaves the apparatus.

The distributors are constructed with diameters considerably smaller than the lower diameters of the collectors, and thus the gas passage is not restricted, and the loss of pressure occasioned is very small.

Packing or filling materials, such as coke, earthenware tiles, or wooden boards, which baffle the passage of the gas, are dispensed with, and the troubles of stoppage and renewals attendant upon such materials getting choked with tar and dust are thus eliminated.

This type of washer is of established efficiency, and the absence of moving parts, together with the freedom from possibilities of blockage, which is a feature of the design, make it a thoroughly reliable apparatus requiring a minimum of attention, with no upkeep charges for labour and material for packing or filling renewals such as have to be met with in other types of washer.

From the washer the gas passes to the first of two centrifugal cleaners. These two cleaners are normally arranged to work in series, but where desired, the requisite cross connections and valves can be added to enable either cleaner to be cut out of operation and by-passed for purposes of overhauling should such be necessary. Each cleaner has a tar drain water-sealed in a lute in the foundations, and the gas delivery branch of the second cleaner is joined up to a sawdust scrubber, to the outlet of which the gas main leading to the gas engines, etc., is connected.

The scrubber is a rectangular cast-iron box fitted with wooden grids, on which are spread layers of wood chips, shavings, and sawdust. The gas enters at the bottom, and passing up through these layers receives its final drying and purification.

The scrubber requires very little attention, only needing to be cleaned out and refilled a few times per year, according to the load on the plant. A special lifting arrangement is provided to facilitate the removal of the cover, and all the spent material taken out can be burnt under the boiler which provides the steam for the plant.

The drains from the various parts of the plant are run to the settling tank, which takes the form of a concrete pit with iron or reinforced concrete partition plates, so arranged as to keep back the tar and allow only the water to run to the drains. The tar is then scooped or run off and disposed of in the most convenient manner, either being sold, used for boiler-firing purposes, dehydrated, or distilled, according to the quality and quantity available and local conditions.

The auxiliaries of the plant—elevator and centrifugal cleaners—can generally be most advantageously and economically driven by electric motors if electric supply is available, a direct or continuous current being preferable to an alternating current supply.

Where requirements call for a minimum fluctuation in the gas supply pressure, an automatic governor can be added to plants in which a power-driven blower is installed. Such a governor would be of the gas-holder type, but of small capacity, and hence of negligible storage value. The bell of the holder would be so arranged that as it approaches its top position it actuates a relief valve connected to the air-blast main, reducing the blast supply to the producer when the gas make is in excess of the requirements, and thus automatically controlling the gas production.

In installations where the gas generated is used to supply gas engines and also to fire furnaces, it is advisable that a non-return valve should be provided in the gas main to the furnaces.

A suitable and sufficient supply of steam is required for blast saturation purposes, also a suitable and sufficient supply of cold water to the washer, cleaners, and producer lute. Where cold water is scarce, water-cooling and circulating plant may be installed, and a considerable portion of the water required can be used over and over again. Suitable motive power for the auxiliaries is also required, in the form of electricity or steam.

This description is typical of units up to about 1,100 b.h.p. capacity, larger units being generally similar but having two or more producers, gas main connections between producers and washer, to suit, and additional scrubber units.

Low Temperature By-product Recovery Gas-Producer Plant.

(*Mond and Lymn System.*)

Under normal circumstances, the installation of this type of plant is not, generally speaking, justified in units smaller than 2,000 b.h.p. in capacity, and local conditions and values may even require a larger unit, to afford an economical proposition. Thus, the usual plant of this type comprises more than one producer unit with corresponding superheaters, etc., but, except for plants of unusually large capacity, the washing, etc., vessels are installed in single units, with the scrubbers in multiple units, according to requirements.

This plant is made up to a certain extent of similar parts to the non-recovery type, but the design of several of these parts is modified to suit the different working conditions, and additional apparatus is included.

The plant may be considered as composed of four sections :—

(1) The Gas Generating Section, comprising producers.
(2) The Gas Washing and Cooling Section, comprising dust washer, ammonia absorber, gas cooler, and in particular cases air saturator.
(3) The Gas Cleaning Section, comprising gas cleaners and scrubbers.
(4) The By-Product Recovery Section, comprising :—
  (*a*) Sulphate of ammonia, evaporating and drying plant.
  (*b*) Tar dehydration and/or distillation plant.

If the gas from the plant is required only for heating purposes, then Section (3), or, in some cases, the scrubber portion of this section, is not required.

The producers are generally similar to the non-recovery type, but with increased depth and other constructional modifications.

A steel chequer-plate operating platform common to the battery of producers is arranged at the level of the top of the producer shells, with suitable protecting handrailing and ladder giving access from ground level.

Fuel-handling plant is generally included, and the type usually recommended comprises a power-driven enclosed bucket type elevator, lifting the fuel from a boot at ground level into an overhead bunker, together with push-plate conveyor. The bunker, constructed of steel, is generally of parabolic section, extending over the whole of the battery and suspended from substantial supports, which also carry the producer platform. The bunker is provided with discharge shutes leading down to the fuel hoppers of the producers, each shute being provided with a lever-operated discharge valve. When required, a light iron framing and corrugated sheet-iron covering is supplied to cover in the coal conveyor and bunker top. For large installations, fuel-handling plant of the travelling overhead crane and grab type offers advantages which merit its adoption.

The air blast for the producers is provided by blower or blowers of either the turbo or Roots type, and after the necessary exhaust steam has been added to it to supply the vapour for blast saturation, it is passed into the producers. In certain cases some of the vapour required for blast saturation purposes is provided by means of an air saturator working in conjunction with the gas-washing plant.

The gas from each producer passes into a gas collecting main which joins up the various producer units and from which it enters the dust washer, where intimate contact with falling sprays of water wash out a large proportion of the dust. From this vessel the gas passes on to the ammonia absorber, which is the same type of patented vessel as used for the washer in the non-recovery gas producer plant.

From the absorber the gas passes to the final cooler, which again is a vessel of the patented washer type.

For certain heating purposes the gas may be taken off for use from the exit of this cooler, but if further cleaning is required, it next passes forward to the first of two centrifugal cleaners, which in detail and arrangement are exactly similar to those described as part of the non-recovery plant.

Where the gas is required to be of gas-engine cleanliness, the delivery branch of the second cleaner is connected to a bus-pipe joining up to the inlets of the scrubbers, the outlets of these vessels being connected with a bus-pipe, from which the main for the gas engines may be taken.

Where for special purposes the final cleaning in scrubbers is not required, then the gas may be taken off from the exit of the second centrifugal cleaner.

The scrubbers are formed in separate sections in a rectangular cast-iron box, each section being provided with wooden grids on which are spread layers of wood chips, shavings, and saw-dust. The gas enters at the bottom, the various sections working in parallel, and the operation, etc., being generally similar to this portion of the non-recovery plant.

The fourth section of the plant, in which the sulphate of ammonia is made in crystal form from the liquor obtained from the absorber, is generally housed in a building to itself.

The plant comprises stock liquor tanks, into which the liquor at suitable saturation is withdrawn from the circulating system of the absorber and stored. From these tanks the liquor is pumped up to overhead feed tanks, from which it is fed into the vacuum evaporator, which operates in conjunction with a condenser of the ejector-jet type. Here the liquor is evaporated and the crystals discharged from the base into a specially designed aspirator, and from thence passed on for final drying in power-driven hydro-extractor. The dry salt is then delivered into the store, usually arranged in the basement of the building, where it is packed in bags ready for distribution or shipment.

A steel circulating tank is provided for use in connection with the absorber, and located closely adjacent thereto, to contain the weak sulphuric acid solution which is used as the circulating liquor in this vessel, and a pump, constructed of acid-resisting metal, is provided for circulating this liquor, with the requisite connections between the tank, pump, and absorber, and from the tank to the stock liquor tanks.

It is necessary for purchasers to provide acid and water supply pipes to this tank, to supply the make-up for the liquor withdrawn from circulation.

Suitable pumps are also provided for supplying the circulating water to the dust washer, but it is usually left for purchasers to supply the cold water supply and pipe-line to the top of the final cooler, to the centrifugal gas cleaners, and to the jet condenser, as also the requisite drain connections for the disposal of the waste water from condenser and settling tanks.

The tar produced is collected from the various circulating tanks into a common settling tank. From this tank it is pumped to an overhead storage tank and dehydrated either by steam or by mechanical means. The dehydrated tar can either be sold as such or pumped forward to a distillation plant to yield oils suitable for fuel, disinfectants, wood preservatives, etc., and pitch.

The auxiliaries of the plant—elevator and conveyor, centrifugal cleaners, pumps, and hydro-extractor—can generally be most advantageously and economically driven by electric motors, if electric supply is available, a direct or continuous current being preferable to an alternating current supply, but it is the usual practice to couple up the blowers to steam prime movers, the exhaust from which, as before mentioned, is utilised for blast saturation.

Automatic governor and non-return valve are provided when required to meet the same conditions as in the non-recovery type of plant, while steam, water, and motive power supplies will similarly be required.

Bituminous Suction Producers.

The production of a satisfactory suction plant using bituminous fuel has proved a problem of very great difficulty, but several types of such plant have recently appeared, and they are now built commercially by Messrs. Dowson, Messrs. Crossley, and others. In these the leading idea is to destroy any tar produced by distillation of the fuel *in the producer* before the crude gas enters the coolers and scrubbers. This is usually accomplished by taking the gas from the

hot zone of the producer, and arranging the supply of air and steam from both above and below; the fuel enters from a hopper on top, and the gases coming off from the fresh coal pass down together with some air and steam to the hot zone where reactions take place which destroy the tar by converting it into useful gases. The fuel, now largely carbonised, passes slowly downwards towards the grate supporting the whole mass, and being traversed by a current of mixed air and steam passing upwards yields carbonic oxide and hydrogen as usual.

Engines driven by these plants frequently require only about 1·2 lb. of slack to the producer per b.h.p. hour.

The 'Hornsby-Stockport' Suction Gas Plant for Anthracite, Coke, Charcoal, etc.

The outstanding features of this producer, fig. 47, are simplicity of design, ease in handling, and reliability under working conditions. Shaker grates to prevent the formation of clinker are fitted on all producers of 37 b.h.p. and upwards. The blow-off pipe of the larger sized producers

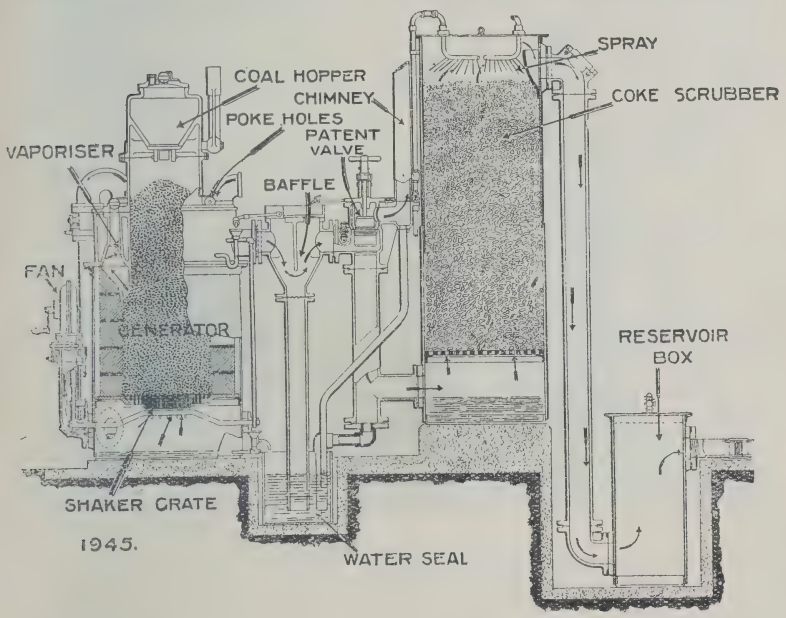


Fig. 47.—Hornsby-Stockport Improved Suction Gas Plant.

is fitted with a patented valve, closing the coke scrubber to the atmosphere when the generator is open to the blow-off pipe, thus keeping air from entering the coke scrubber.

Gas can be raised from cold in about 20 minutes.

The 'Ruston' Wood-Refuse Suction Gas Producer.

This gas producer (*Ruston and Hornsby, Ltd.*), fig. 48, is for the purpose of dealing with wood refuse, non-caking bituminous coal, lignite, brown coal, anthracite, etc.

The chief endeavour has been to obviate the stoppages for cleaning usually necessary when working on such fuels. The little cleaning that is required is at the gas outlets from the generator, and this can be easily performed in a few minutes while the plant is in operation by means of special scrapers, so that the usual week-end cleaning is done away with, and the plant can be worked continuously if required.

A special feature of the producer is the multiple gas outlet pipes from the top of the generator, which ensure an evenly burning fire.

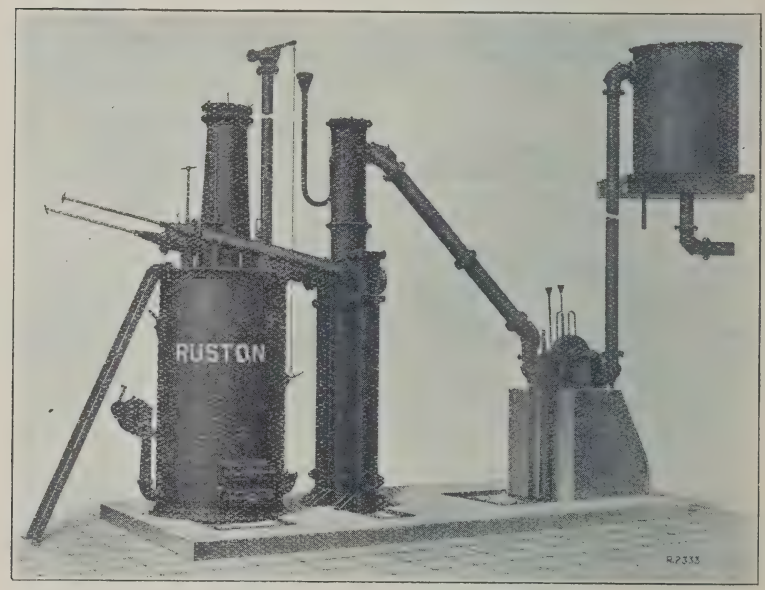


FIG. 48.—'Ruston' Wood-Refuse Gas Producer.

Other advantages claimed are as follows:—small space occupied, very little attention required, no coke scrubber to become clogged with tar, simple in design and construction, and gas free from grit, dust, or tar. The consumption of wood refuse is from $2\frac{1}{2}$ to $3\frac{1}{2}$ lbs. per b.h.p., according to the amount of moisture in the fuel, and the plant will produce four or five times the power as when the fuel is burned for direct steam raising.

When bituminous coal, lignite, anthracite, or coke is required to be used, solely or alternatively with refuse fuel, the details of the producer are somewhat modified. An annular steel vaporiser is arranged around the upper part of the generator, and an air lock feeding hopper is fitted. Poking holes are provided to give access to all parts of the fuel bed, and special facilities are given for the quick removal of ash and clinker.

A large number of these producers have been supplied at home and abroad, for a variety of refuse fuels, and for bituminous coals.

Water Gas.*

In ' Water Gas ' producers the action is intermittent, the fuel being alternately raised to incandescence by an air blast which is then shut off and steam alone passed through the glowing mass; the steam is dissociated into free hydrogen and oxygen, and the oxygen combines with the carbon present in excess and forms CO; thus :—

$$H_2O + C = H_2 + CO$$
$$2 \text{ vols.} + 1 \text{ vol.} = 2 \text{ vols.} + 2 \text{ vols.}$$

In theory, therefore, water gas should contain 50 per cent. by volume each of hydrogen and carbon monoxide; practically a small quantity of CO_2, N and O generally appears, as shown in the analysis given below. The passage of the steam through the mass of incandescent coke soon cools it down, when the steam is cut off and the air blast again turned on; the cycle of operations is then repeated.

Water gas plants are frequently installed in gas works, and the water gas is often enriched or ' carburetted ' by admixture of heavy hydrocarbon vapours and in this condition used either alone or mixed with coal gas for lighting, heating, and power purposes.

The following results of analyses suffice to give a general idea of the characteristic constitution of water gas, carburetted water gas, mixed gas (i.e. mixed water gas and coal gas), and normal coal gas :—

	H.	CH_4.	CO.	Heavy Hydrocarbons.	CO_2.	O.	N.
Water gas	48·6	0·4	44·0	—	3·3	—	3·7
Carburetted water gas	29·35	20·48	33·19	11·32	—	—	5·66
Mixed gas	46·73	24·57	14·46	4·91	1·45	0·73	7·15
Normal coal gas	45	33	10	5·3	0·5	0·3	5·9

6. BLAST FURNACE GAS.

In 1895 the late Mr. B. H. Thwaite showed that the hitherto waste gases from blast furnaces could be used in gas engines, and his discovery has stimulated the construction of the very large gas engines now becoming common.

The gas is taken from the gas main and forced by a blower through a scrubber into a holder regulator, and then to the engine. About 180,000 cubic feet of gas are produced per ton of fuel consumed in the furnace, and about 124 cubic feet of gas are used in the engine for each horse-power hour.

From tests made on Mr. Thwaite's first plant at Wishaw by the late Mr. W. H. Booth, it is estimated that for each ton of fuel used per week in a blast furnace, 10 horse-power may be continuously obtained by gas engines.

It is proposed to thus utilise the gases of all blast furnaces and so produce several millions of horse-power for distribution for railway, lighting, canal, traction, and general power purposes.

Table I. gives the analyses of blast furnace gas in two cases where the Thwaite system is in use, and a comparison with producer gas, while Table II. gives results of a volumetric analysis of the Wishaw gas.

* See also Section XXXVII.

TABLE I.

	Coal Fuel. *Wishaw.*	Coke Fuel. *Frodingham.*	A Dowson Gas Producer.
Carbonic oxide	24·75	27·3	23·8
„ acid	5·75	6·0	6·3
Hydrogen	2·33	1·5	19·8
Marsh gas	0·75	—	1·3
Nitrogen	66·42	65·2	48·8
	100·00	100·00	100·00
Heat units per cubic foot	97·8	96·7	164·0
Calculated cubic feet per H.P.	79·44	80·34	70·0
Percentage combustible	27·83	28·8	44·9

TABLE II.

Gas.	Per Cent. Vols.	Weight per cub. ft.	Lbs. Wt. per 100 cub. ft. of Mixed Gas.	Actual Weight of Elements per 100 cubic feet of Gas.			
				C	O	H	N
Carbonic oxide, CO	24·75	·0784	1·9404	·8316	1·1088	—	—
Hydrogen, H	2·33	·0056	0·0130	—	—	·0130	—
Marsh gas, CH_4	0·75	·0448	0·0336	·0252	—	·0084	—
Nitrogen, N	66·42	·0784	5·2073	—	—	—	5·2073
Carbonic acid, CO_2	5·75	·1234	0·7095	·1935	·5160	—	—
	100·00	·079038	7·9038	1·0503	1·6248	0·0214	5·2073
					7·9038		

From 80 to 100 cubic feet of gas are obtained from each pound of fuel used. The foregoing table shows about 98 cubic feet of gas per pound of carbon, but some of the carbon comes from the limestone flux and from the ore, and even from the blast itself. In the gas engine, with the necessary enlargement of the gas inlet as compared with that necessary for town gas, and with a fair amount of initial compression, the action of blast-furnace gas is perfect, and no misfires are found to occur. The poverty of the gas is, in fact, no bar to its employment for power purposes. In fact the blast furnace is described in Mr. W. N. Booth's report as an ideal gas-producer. The employment of blast-furnace gas for power purposes enables cold-blast iron to be made, for with a sale for gas the economy of fuel per ton of iron ceases to be of importance.

Approximately each pound of carbon in the fuel used will produce about 98 cubic feet of gas, and each 100 cubic feet of gas produced weighs nearly 8 lbs. The tests on the small engine at Wishaw showed only about 1½ lb. of coal per electrical horse-power, or 1·12 per i.h.p., assuming an efficiency of 75 per cent. between the cylinder and the terminals of the dynamo and allowing for temperature. Approximately each ton of fuel used per week in a furnace would give continuously 10 horse-power.

There has been a very considerable extension of the use of blast-furnace gas in Germany, where full advantage was promptly taken of Thwaite's teachings, and after a period of attempting to work with uncleaned gas Thwaite's method of cleaning was practically adopted. It is very important that the gas should be cleaned of its dust. Failure has occurred at an American iron-works where the engine valves became heavily loaded with dust. Fans may be employed through which the gas is circulated, jets of water being entered at the side of the fan casing. Rotating gas washers also are employed, and washer boxes in which the gas is passed under diaphragms which project an inch or two below water and compel the gas to bubble up through the water.

Large scrubbers with cellular work of light lathwork with water trickling over them will arrest much dust. Thwaite employed a sawdust filter as a final catcher, arranged so that the sawdust could be rapidly changed. Thwaite relied much on sawdust for removal of the tar, of which it is difficult to get rid. Though blast-furnace gas may contain as little as 27 per cent. of combustible gas it presents no ignition difficulties when electrically fired in a state of compression in the cylinder. A. Witz has shown that the average calorific value of the gas from twenty-seven Belgian furnaces was 111 B.Th.U. per cubic foot, while nineteen Luxembourg samples showed 114 B.Th.U., twenty-three German furnaces showed 108 B.Th.U., and nine French and Spanish furnaces 107 B.Th.U. He takes a general average of 110, but the value may fall to 97 with an acid flux. Such poor gas will bear high compression, and will safely bear 7 to 9 atmospheres. No matter how poor a gas it will ignite if sufficiently compressed, and at Seraing as high as 11 atmospheres were employed in 1901. Hence the excellent efficiency that is obtained with blast-furnace gas.

Blast-furnace gas contains but little hydrogen, as no steam is used in the blast furnace ; it approximates, in fact, closely to the gas that would be given by an ordinary producer supplied with air only. The small hydrogen content enables very high compression pressures to be used without danger of pre-ignition ; when well cleaned, this gas is an excellent fuel for large engines, the heat evolved per cubic foot of maximum mixture at 60° F. and 1 atm. being about 57 B.Th.U. as an average lower value ; as already pointed out in an earlier part of this section, with cylinders of 30 ins. in diameter and over, in order to avoid overheating troubles it is found desirable in practice to introduce only from 35 to 40 B.Th.U. per cubic foot of piston displacement ; thus, although low in comparison with most other gaseous fuels, the heat evolution per cubic foot of blast-furnace gas mixture is fully adequate to the needs of large gas engines.

Some average results for the principal gaseous fuels used at the present day are given in the following table :—

HEAT VALUE AND COMBUSTION OF USUAL GASEOUS FUELS.

(*From Sir Dugald Clerk.*)

Nature of Gaseous Fuel.	B.Th.U. per Cub. Ft. (Lower Value) at 60°F. and 1 Atm.	Cub. Ft. of Air Necessary for Complete Combustion.	B.Th.U.(Lower) per Cub. Ft. of Max. Mixture at 60°F. and 1 Atm.	B.Th.U. as in Col. 3 × a Volumetric Eff. of 0·85.
Column :—	1	2	3	4
Coal gas	500	5·0	83·3	71·0
Coke oven gas . .	400	4·0	80·0	68·0
Mond gas	150	1·1	71·0	60·3
Pressure producer gas from anthracite . .	150	1·1	71·0	60·3
Pressure producer gas from coke . . .	130	0·95	67·0	57·0
Suction producer gas from anthracite . .	130	0·93	67·0	57·0
Suction producer gas from coke . . .	130	0·92	68·0	57·8
Blast furnace gas . .	100	0·75	57·0	48·4

60° F. = 15·5° C.

THE KIRKE PATENT GAS-FIRED BOILER.

The Kirke Patent Gas-fired Boiler, invented by Mr. P. St. G. Kirke, M.A., Assoc.M.Inst.C.E., A.M.I.M.E., etc., is manufactured by *Spencer-Bonecourt, Ltd.*, 32-33 *Farringdon Street, London, E.C.* 4.

Fig. 49 shows a Kirke Patent Gas-fired Steam Boiler. The design is based on the discovery that, if such gases as coal gas, coke-oven gas, blue water gas, producer gas, and natural gas are induced into empty tubes of great length relative to their bore, together with the air theoretically required for combustion, or some 10 per cent. in excess thereof, they can be completely burnt in the tubes without being previously mixed, and the resulting products of combustion reduced to 70° C., or less, above the temperature of the water in the boiler. Further, that this can be done either with natural draft or with high induced draft of 4 ins. of water and over, provided that the length of the tubes is greater relative to their bore the higher the draft adopted.

The Kirke Patent Gas-fired Boiler in its simplest form consists of a drum traversed by a number of long tubes with nothing in them, which function as combustion chambers, as mixing chambers, and as extractors of the sensible heat remaining in the resulting products of combustion.

KIRKE PATENT GAS-FIRED BOILER. (Patents No. 134,938, etc.)

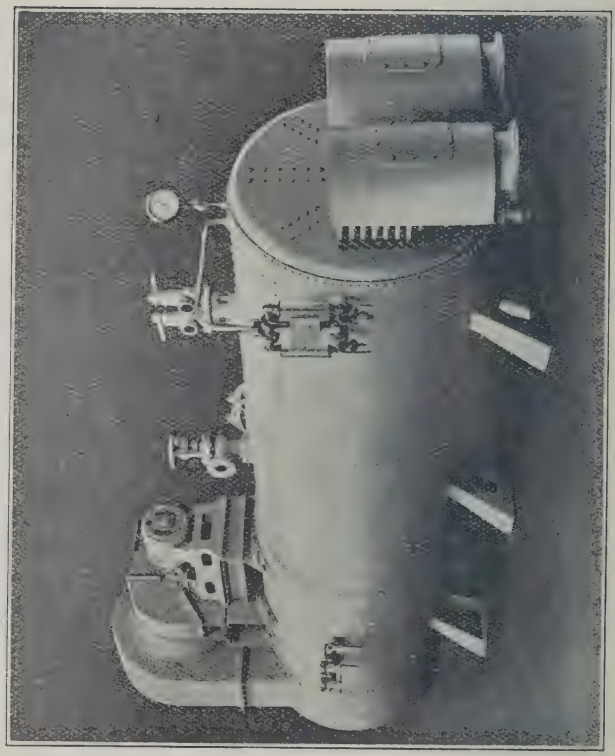


FIG. 49.

At the inlet end of the tube is a patent gas burner or distributor which discharges a jet of gas into each tube, the amount of which is determined by the size of the holes from which the gas issues and the pressure of the gas in the burner, the latter being controlled by the large gas inlet valve at the bottom of the burner. The requisite air for combustion enters each tube from the atmosphere, the quantity being controlled by the amount of draft which is varied to suit the load.

The following are figures obtained from a single tube evaporating water in an open wood tank lined with zinc :—

Gas burnt in cubic feet per hour	230
Steam raised from and at 212° F. in lbs. per hour	104
Cubic feet of gas per lb. of steam from and at 212° F.	2·21
Temperature of gases leaving the boiler	188° C.
Suction in inches of water	8
Gas pressure in inches of water	4
Evaporation per sq. ft. of heating surface in lbs. from and at 212° F. per hour	18
Net calorific value of the gas in B.Th.U.'s.	480
Thermal efficiency without economiser	91 per cent.

A single drum boiler of this type 7 ft. diameter by 21 ft. long, fired with coke-oven gas, will evaporate 21,000 lbs. of steam per hour with a suction of 4 ins. ; 16,000 lbs. with a suction of 2½ ins. ; and 10,500 lbs. with a suction of 1 in. The radiation loss from the shell when completely lagged (except for the front tube plate) is only ½ per cent. at full load and under 3 per cent. when unlagged.

The following table gives the power required by the fan, assuming 15½ lbs. of steam from and at 212° F. are required to generate 1 b.h.p.

Nature of Fuel burnt in Kirke Patent Fuel-fired Boilers.	Draft.	Percentage of Boiler Output required to create 4 in. suction.	Approximate B.H.P. required by the Exhauster for an Evaporation of 10,000 lbs. of steam from and at 212° F. per hour.
	Ins.	Per Cent.	
Water gas	4	0·93	6
Coke oven gas	4	1·00	6½
Producer gas	4	1·25	8
Crude oil	4	1·35	8¾
Coal	4	1·71	11
Blast-furnace gas	4	2·00	13

When the floor space is very restricted, it is easy to put one drum on the top of another. In this way, for a given evaporation only one-fifteenth of the floor space required by Lancashire boilers for the same evaporation is required.

The cleaning of the tubes on the water side is made easy by division of the tubes into two nests. A man can enter the boiler through the manhole and walk along the whole length of the boiler between the two nests, so that he can scrape the surface of all the tubes for their entire length, the distance between the tube centres being sufficiently great to enable him to get a suitably shaped scraper to all the tubes. This, however, is better avoided by softening the water or using a suitable boiler fluid. An internal scale-separating device is also fitted.

For superheating the steam a separate chamber containing fire tubes fired in the same way, but with natural draft, is used, the temperature being kept constant by a thermostat operating the gas valve.

The following are the approximate fuel consumptions per lb. of steam raised from and at 212° F. for various fuels burnt in Kirke patent boilers without economisers :—

Producer gas	8 cubic ft.	Poor coal-gas	2·7 cubic ft.
Coke oven gas	2½ ,,	Natural gas	1·2 ,,
Water gas	3⅝ ,,	Crude oil	$\frac{1}{17}$th lb.
Rich coal gas	2·2 ,,		

Kirke Patent Waste Heat Boiler.

This boiler, fig. 50 (Patents numbered 140,851, 209,878, etc.), consists of one or more drums, having inlet products chambers at one end and outlet products chambers at the other, connected together by a number of empty and straight fire tubes of great length relative to their bore through which the hot gases are drawn at a high vocity by means of an induced draught fan.

The high velocity enables large volumes to be handled in a boiler of comparatively small dimensions, and prevents most of the dust from lodging on, and insulating, the heating surfaces. It also reduces the cost of the boiler, and increases the return on the capital expenditure.

Notwithstanding the high velocity at which the gases flow over the heating surfaces, the great length of the tubes relative to their bore enables the temperature of the hot products to be reduced to about 30° C. above the temperature of the water in the boiler, before they enter the fan impeller. Consequently an economiser is not required, and higher thermal efficiencies can be obtained than have ever been obtained before.

Cleaning.—The fire tubes are arranged in two or more nests, between which it is possible for a man to enter, so that the tubes are exposed for cleaning on the water side over the whole of their length.

As the interior of the fire tubes is smooth, round, and straight, and the induced draught causes the hot products to travel through them at a high velocity, without changing their direction, there is nothing to catch the dust, which consequently is carried straight through to the outlet products chamber, from which it can be easily removed from time to time. To deal with dirty gases containing very large quantities of dust—such as are obtained from cement kilns—an arrangement (Patent No. 186,801) for enabling these boilers to be cleaned while they are running is now being fitted to all these boilers. The fan is arranged at the top or at the side of the boiler so as to leave the cover plate of the outlet products chamber unobstructed. In this cover plate small holes are drilled large enough to pass a tube brush or steam lance through for pushing through the whole length of tubes. One of these little holes will allow some nine tubes to be cleaned, and, while leakage of air into several of the holes is not sufficient to reduce the draught, it is not necessary to open more than one at a time for cleaning purposes.

Circulation.—The circulation in the Kirke boilers is particularly good. About 70 per cent. of the evaporation occurs in the first third of the tubes, and only about 10 per cent. in the last third. Consequently there are strong convection currents upwards at the front end and downward at the rear end.

Dry Steam.—By taking the steam off at the back end of the boiler, where the circulation is downward, and by the use of an efficient anti-primer, the steam is rendered remarkably dry. In an independent test on a boiler of this type unlagged, the steam was found to be 99·3 per cent. dry. In a second independent test carried out on a lagged boiler of this type, the moisture in the steam was so small that it could not be measured and was reported to be 'practically quite dry.' In a third case, where water-tube boilers were giving trouble with priming, it was found impossible to make a boiler of this type prime even under conditions which no boilers would be put to in practice—namely, when supplying steam to a number of colliery winding engines, all of which were purposely started up at once.

Loss due to Radiation.—In Kirke boilers the radiation loss is extremely small, owing to the fact that, except for the front tube plate, the whole of the shell may be lagged. With air at 72° F., the heat loss from uncovered steam pipes carrying steam at 100 pounds per sq. in. is 1,170 B.Th.U.'s per sq. ft., and this is reduced to 100·2 B.Th.U.'s when 2 ins. of magnesia lagging is used to cover them (see 'Mechanical Engineers' Handbook,' by S. Marks). In the case of a large standard pattern Kirke gas-fired boiler, 6½ ft. diameter, evaporating 18,000 lbs. of steam from and at 212° F. per hour with 4 ins. induced draught, the loss of steam due to radiation is, however, only 2¾ per cent. before the boiler is lagged and about ½ per cent. after the boiler is lagged.

KIRKE PATENT WASTE HEAT BOILER FOR WASTE HEAT, COAL, BLAST-FURNACE GAS, OR PULVERISED FUEL. (Patents No. 140,851, etc.)

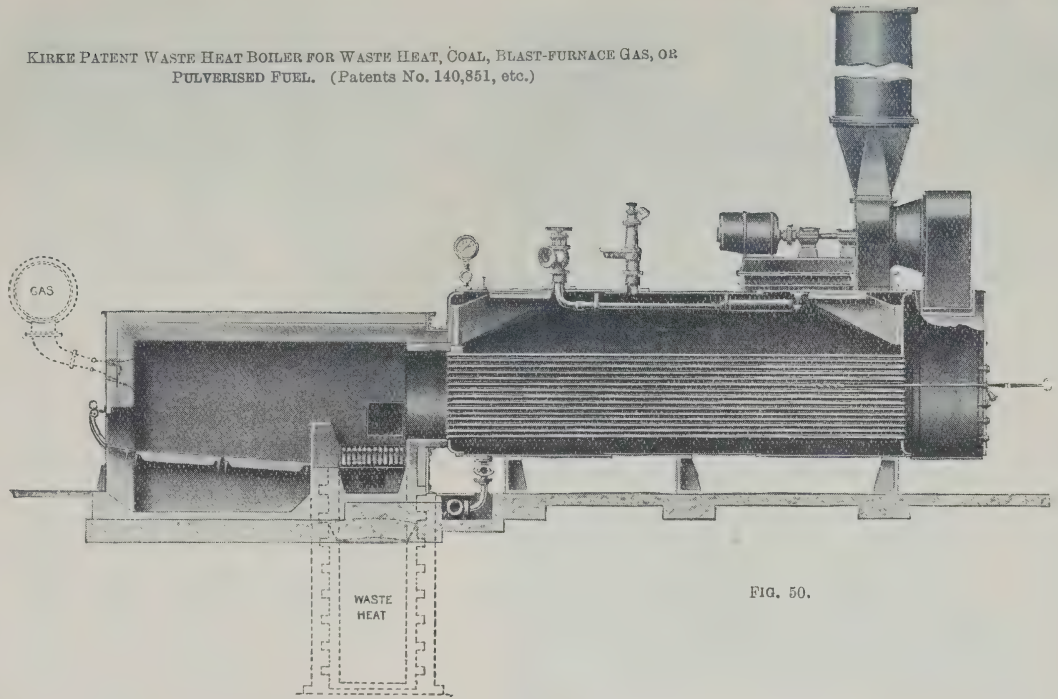


FIG. 50.

Efficiency.—Two Spencer-Bonecourt Kirke patent waste heat boilers which are generating steam from the waste heat discharged from 10 settings of Glover-West retorts at the Stratford works of the Gas Light and Coke Co., gave the following figures when tested by the staff of the Gas Light and Coke Co. :—

Date of test, November 19, 1923.

Duration of test, 24 hours.

Temperature of the products entering the boilers	689° C.
Temperature of the products leaving the boilers	225° C.
Steam pressure in lbs. per sq. in.	115
* *Mean tube efficiency of the boilers*	90·3 per cent.
Suction in the outlet products chamber of the boiler in inches of water	4·4 ins.
Suction in the inlet products chamber available for the settings and collecting flues	1·4 ins.
B.H.P. consumed by the motor for the retorts	11½
B.H.P. consumed by the motor for the boilers	24¼
Water evaporated per hour in lbs. actual	16,783
Water evaporated from and at 212° F. per hour in lbs.	18,500
Percentage of power absorbed by the draught for boiler (assuming 15 lbs. of steam from and at 212° F.) per b.h.p. hour	2 per cent.
Percentage of power developed absorbed by the draught for collecting flues and settings (assuming 15 lbs. of steam from and at 212° F. per b.h.p. hour)	0·93 per cent.
Estimated saving in small coke and breeze per annum of 300 days freed for market in tons (assuming 6 lbs. of steam from and at 212° F. per lb. of this fuel)	10,000 tons
Steam produced from and at 212° F. per lb. of coke used in the producers	5·8
Steam produced from and at 212° F. per lb. of coal carbonised	0·9 lb.
Heat loss between the outlet of the settings and the inlet of the boiler due to radiation from the steel flues	4·9 per cent.

One of these boilers fitted to a Talbot steel furnace at another works brought down the consumption of coal per ton of steel to 3½ cwts. Before the Kirke boiler was installed the coal consumption was 5 cwts. per ton of steel.

These waste heat boilers have been successfully installed on horizontal retort benches, inclined retort benches, vertical retort benches, open hearth steel furnaces, soaking pits, cement kilns, water gas plants, puddling furnaces, etc.

The boilers have also been installed in Europe, Asia, Africa, America, and Australia.

The Clarkson Boiler.

(*The Clarkson Thimble Tube Boiler Co., Ltd.*, 613 *Australia House, Strand, London, W.C.* 2.)

The 'Clarkson Thimble Tube Boiler' has for years met the needs of hundreds of steam users who require all the advantages that an ordinary quick steaming water-tube boiler provides, and for whom other considerations render inadvisable the use of such a boiler.

The boiler can be used for many duties and with various fuels, although the most suitable use for it is the extraction of the waste heat from engine exhaust gases. The makers claim that for this duty the boiler is the most serviceable on the market from almost every point of view. Installations on engines of all sizes from 7 b.h.p. to 10,000 b.h.p. have been successfully carried out.

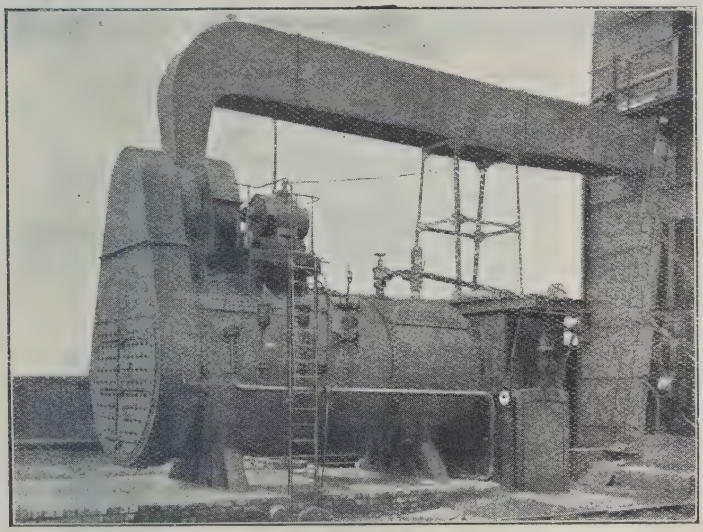
The Clarkson Thimble Tube Boiler is of the vertical water-tube type, the general construction of which is shown in fig. 51, page 2197. The tubes, being thimble shaped, are expanded into a cylindrical shell plate at one end only, so that the troubles caused by the expansion and contraction of water-tubes in ordinary boilers do not exist for the Clarkson Boiler, whilst at the same time quick steaming properties and other advantages of water-tube boilers are retained.

The boiler unit consists of two cylindrical shell plates, through the centre of which the flue gases pass. Several hundred thimble tubes are expanded into holes in the inner plate, each successive ring being staggered. A centre core fills up the cylindrical space between the tube ends and compels the exhaust gases to travel tortuously through the nest of tubes which comprise the heating surface. The water space is thus annular in shape with projections inside each thimble.

The usual boiler or heater fittings are mounted on the shell. A combustion chamber and oil burners are added to the boiler unit, and make a highly efficient steam generator.

* To obtain tube efficiency, divide difference between the temperature of the gases entering and leaving the tubes by the difference between the temperature of gases entering the tubes and that of the water in the boiler.

WASTE-HEAT RECOVERY



WITH A PROPERLY LAID OUT WASTE-HEAT RECOVERY SYSTEM, THE USE OF FUEL-FIRED BOILERS FOR POWER REQUIREMENTS IS GREATLY REDUCED AND IN CERTAIN INDUSTRIES ENTIRELY ELIMINATED . . . . FURTHER INFORMATION WILL BE SENT ON APPLICATION

SPENCER-BONECOURT

PATENT WASTE-HEAT

BOILERS

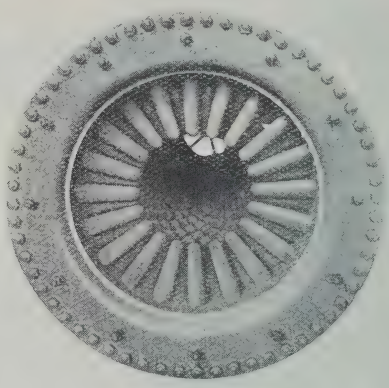
SPENCER-BONECOURT LTD. 32-33 FARRINGDON-STREET LONDON, E.C.4

F. 2196

Clarkson-Silencer-Boilers

for

2 or 4 Stroke



Diesel Engines

See Editorial Pages, Nos. 2196–2198.

100% Accessibility *is the Outstanding feature of the latest Clarkson Silencer Boilers*

The Predominant Marine Waste Heat Boiler for 2 or 4 stroke Diesel Engines

Standard Boiler units adaptable for Waste Heat Recovery

Diesel Exhaust-Gas and Oil-Fired Boilers

Silencer - Electric - Boilers

THE

Clarkson Thimble Tube Boiler Co., Ltd.

613 Australia House, Strand, W.C.2.

Telegrams: CLARSTEAMO, PHONE, LONDON *Telephone:* TEMPLE BAR 2112

Non-Scaling.—It will be realised that steam is generated inside the thimbles, and therefore when a certain amount of steam is generated it blows the water back out of the mouth of the tube. The water then returns into the tube when the steam has escaped. The pulsating action thus set up has a frequency of the order of 90 to 160 per minute, and effectively scours the tube of scale. The scale-forming matter is then directed to the foot of the boiler by the cylindrical baffle plate, which also prevents the cold feed water from impinging direct on to the tube ends, and helps an effective circulation.

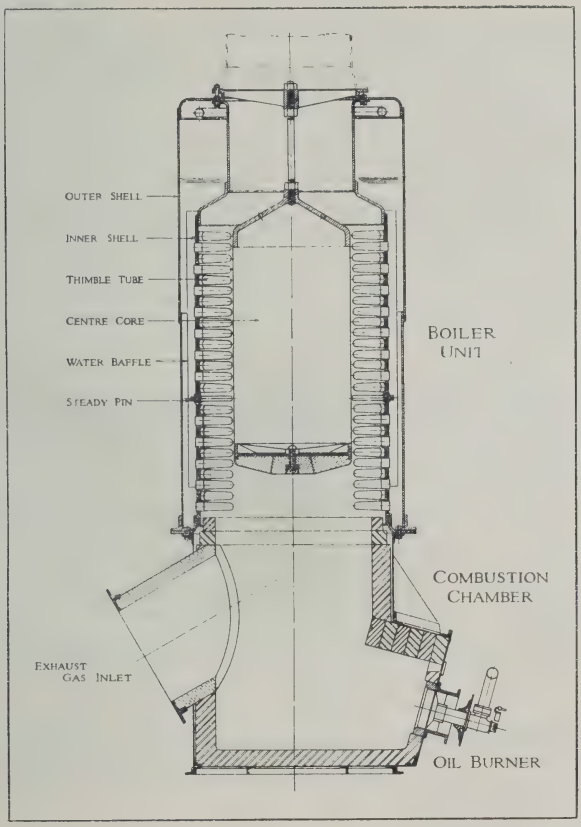


FIG. 51.

Soot Cleaning.—Should the outside of the thimble tubes foul, as may happen when running off a dirty engine exhaust, the boiler can be emptied, a wad of oil-soaked waste ignited and put at the foot of the tube nest, and the impurities burnt off with no danger of damage occurring.

Accessibility.—The tubes are easily accessible on the steam side of jointed designs, similar to that shown in fig. 51, by raising the outer shell. Larger sized boilers are now being designed with a large annular space around the inner shell, so that complete accessibility is obtainable without raising the outer shell. The whole boiler is then of riveted up construction, and sufficient space is allowed for a man to get inside the boiler and inspect all parts of the water space, and for the renewal of tubes, if necessary. The gas side of the tubes is accessible in the jointed

types by removing the centre core, whilst in some Clarkson designs the same accessibility is obtained by making the centre core sectional or telescopic, so that no head room is required.

Centre Core.—The centre core or gas baffle, of the best fire-resisting material, forces the exhaust gases to travel tortuously through the nest of tubes, and is suspended from the uptake, being steadied by three to six steady pins, screwed into the inner shell.

Exhaust Heat Extraction.

Silencing.—The Clarkson Boiler is an efficient exhaust silencer, and can be designed so that it may be run dry off exhaust gas with no ill effects, and put on to steaming load at short notice.

Back Pressure.—In estimating the size of boiler required for extracting heat from exhaust gases, the main consideration is the avoidance of any extra back pressure on the engine. The contraction of the gases as they cool in the boiler assists the exhaust from the engine cylinders, tending to reduce back pressure.

Supercharged Engines.—Where an exhaust gas boiler is required to operate on the outlet side of an exhaust gas turbine, then very little back pressure is allowable, and for this reason the Clarkson Boiler has been used on a considerable number of large marine supercharged installations. It is the only waste heat boiler which has been used on installations of this nature.

Installations.—Clarkson Boilers have now been supplied for a multitude of uses as coal and oil-fired donkey boilers for Admiralty work, for dredgers, steam cranes, steam wagons, etc., and for waste heat recovery from the exhaust gases of hundreds of internal combustion engines of all types, particularly for marine purposes. Amongst these can be numbered boilers supplied to the British Admiralty for submarines; boilers supplied for four-cycle engines on the White Star liner *Britannic* (which is incidentally the largest waste heat installation afloat); boilers supplied for most of the largest motor vessels constructed during 1930, including the *Reina del Pacifico*, *Otaio*, *Orari*, *Opawa*, *Winchester Castle*, and *Warwick Castle*, which list is by no means exhaustive.

Recent developments include simultaneous oil and exhaust gas-fired boilers, also a combined electric and exhaust gas boiler.

Fuel Firing.

Owing to the quick steaming properties and strong design of the Clarkson Boiler, it was found necessary to adapt the unit for coal or oil firing, and a range of thimble tube boilers with water legs has been built and have given satisfactory service as donkey boilers, galley boilers, and dredging duties. Types of these boilers, used on steam wagons, have withstood successfully for many years the rough treatment that is often met with in road work.

The makers claim the following advantages for this boiler:

1. A smaller size of thimble tube boiler will do the work of steam generation more efficiently and at less cost than is possible with any other type of boiler, especially other vertical boilers.
2. In consequence of low size and weight, it can be fitted in the place of silencers, in the exhaust pipe run above the engine room.
3. Silencer boilers may be fitted with alternative oil firing for use in port, and may also be designed for simultaneous oil firing, or for additional steam raising by means of electrical heaters.
4. Ample accessibility for cleaning and inspection of all internal parts, particularly in jointless designs, where the whole boiler can be made accessible in a few minutes.

Summary of Test Results on Oil Firing.

Clarkson Silencer Boiler: Type Begatoo-600. Date of Test: May 30, 1928.

Boiler.—Clarkson Patent Vertical Thimble Tube, 6 ft. diam. by 18 ft. 10 in. overall.

Working Pressure.—100 lbs. per sq. in.

Heating Surface.—600 sq. ft.

Weight complete.—12 tons.

Oil Burners.—Clyde Oil Fuel System, Low-Pressure Air.

Utility.—Silencer—Boiler passing 70% of exhaust gases of 10,000 B.H.P. twin engines, or Oil-Fired Donkey Boiler.

Guaranteed Evaporation.—4,800 lbs. of water from and at 212° F.

Evaporation per hour, actual	5,071 lbs.
Evaporation per hour, f. & a., 212° F.	6,085 ,,
Oil consumed per hour	491·7 ,,
Water evaporated, f. & a., 212° F. per lb. of oil	12·38 ,,
Steam per sq. ft. of heating surface per hour, f. & a., 212° F.	10·2 ,,
Thermal efficiency of boiler—unlagged	64·5%

WASTE HEAT RECOVERY.

(Contributed by Chas. F. Wade, A.M.I.Mech.E., A.M.I.E.E.)

The directions in which profitable recovery of waste heat must be looked for are

(i) Waste heat from various metallurgical and other furnaces ;

(ii) Waste gases from internal combustion engines ;

(iii) Exhaust steam from non-condensing engines, steam hammers, steam-hydraulic intensifiers, etc.

Waste heat from high temperature metallurgical operations, such as open-hearth melting furnaces, consists of the sensible heat of the waste gases and the heat carried away by radiation and convection from the furnace walls. The latter is permitted deliberately in order to prevent unnecessarily rapid burning of the walls. The heat of the waste gases is generally utilised for the purpose of preheating the incoming air and gas by means of regenerators and reversing valves. By jacketing the furnace walls and utilising the heat formerly wasted in radiation for air preheating, thus releasing approximately 50 per cent. of the heat of the waste gases for steam-raising in a waste heat boiler, a very substantial saving of fuel will result. The loss of heat by radiation and convection from unjacketed furnaces is certainly not less than 25 per cent. of the total heat of the fuel used. Even in a reheating furnace working at 1000° C., jacketing of the furnaces instead of regeneration in chequer chambers will often give an air temperature of over 300° C.

Small reheating furnaces for light forgings and similar work air-jacketed at the sides only will often give a temperature of over 150° C. in the air supplied for combustion. In all cases the result will be that the final temperature of the gases will be sufficient to make a waste heat boiler a profitable investment.

In metallurgical furnaces wherein the process requires a densely smoky atmosphere and the air supply is deliberately restricted, the use of a waste heat boiler for the products of combustion together with a supplementary coal fire, through which an excess of air is passed, will not only lead to large recovery of heat in the form of steam, but will also furnish a solution to the smoke-abatement problem. Without the supplementary fire-grate a supply of highly preheated air brought through ducts on the furnace settings will usually ensure reignition of the combustible matter in the smoky gases. Cold air will not do this owing to the low temperature of the gases to begin with, thus keeping the products below ignition point.

For ordinary waste heat boiler work without supplementary firing a temperature of at least 800° C. is needed to make the proposition profitable. With tubular boilers of the Cochran type the usual evaporation per pound of coal consumed in the furnace is :—

Temperature of Waste Gases.	1500° F.	1800° F.	2000° F.
Lbs. water per lb. of coal from water at 62° F.	3·08	4·25	5·00

Using boilers of the ' Bonecourt ' type, in its more modern developments, wherein gas at a high velocity, aided by mechanical draught, is in use, gases with an initial temperature of 350° C. may be used profitably.

Gases from internal combustion engines usually allowed to expand in a silencer, with corresponding fall of temperature, are maintained at smaller volume and part with their heat to the water in the boiler with a final temperature of about 75° C.

This system is of considerable value also in dealing with the waste heat from gas retort furnaces.

Where low-temperature furnaces are in use only the waste gases may be used for heating of water for shop and office heating by means of economisers, and where both high- and low-temperature furnaces are worked, the hot water from the latter should be used for feed to the high-temperature waste heat boilers. An interchange of heat and power from waste sources between adjoining establishments is often a practicable proposition where the means of power and heat generation and utilisation are collectively equal.

The heat wasted in exhaust steam can be applied to the drive of low or mixed pressure turbines, which latter will have the additional economical effect of levelling out the load on the boiler plant, in that a falling-off of the supply of exhaust steam will cause the turbine to draw on the boilers directly for a high pressure supply. Other important advantages are the returning of soft water to the boilers for feeding purposes, either with or without preliminary electrolytic treatment for oil elimination,

Other applications of the heat of exhaust steam are for heating processes in manufacture, heating of buildings, etc., all of which permit of the return of the condensate for boiler feed purposes.

All proposals for the utilisation of waste heat of any kind require to be dealt with individually on their merits, the conditions being so variable that no hard and fast rules can be laid down which will be universally applicable.

Generation of Electricity in Great Britain by Waste Heat.

In the Annual Report of the Electricity Commission * it is stated that the total units generated by waste heat during the year ending March 31, 1928, was 217,987,219 (from 21 stations), as against 78,332,571 (from 17 stations) in 1927.

* Generation of Electricity in Great Britain, year ending March 31, 1928. H.M. Stationery Office, Adastral House, Kingsway, London, W.C. 2. Price 1*s.* 6*d.* net.

SECTION XXXV

ELECTRICAL ENGINEERING

COPPER AND ALUMINIUM CONDUCTORS — FUSES — PORCELAIN INSULATORS — MINISTRY OF TRANSPORT REGULATIONS — ELECTRICAL EQUIPMENT OF BUILDINGS AND SHIPS — WIRING SYSTEMS — PRIMARY AND SECONDARY BATTERIES — GENERATORS — CONVERTERS — MOTORS — CONTROL GEAR — TRANSFORMERS — STATIC BALANCERS — POWER FACTOR IMPROVEMENT — INSTRUMENTS — METERS — LAMPS — LIGHTNING CONDUCTORS — ELECTRIC HEATING AND COOKING — ELECTRICITY IN AGRICULTURE — TELEPHONY — RADIO - COMMUNICATION — INSULATING MATERIALS AND OILS — X-RAYS IN INDUSTRY.

(Revised and amplified by Prof. S. Parker Smith, D.Sc., Assoc.M.Inst.C.E., M.I.E.E.)

(Pp. 2203–2350.)

ADDENDA.

CABLES FOR HIGH VOLTAGE—TRANSFORMERS—ELECTRIC CLOCKS AND TIME SERVICE—PORCELAIN INSULATORS FOR OVERHEAD LINES - ELECTRIC MEASURING INSTRUMENTS—PHOTO-ELECTRIC CELLS AND THEIR USES.

(Pp. 2350-2350B.)

SECTION XXXV

ELECTRICAL ENGINEERING

COPPER AND ALUMINIUM CONDUCTORS — FUSES — PORCELAIN INSULATORS — MINISTRY OF TRANSPORT REGULATIONS—ELECTRICAL EQUIPMENT OF BUILDINGS AND SHIPS — WIRING SYSTEMS — PRIMARY AND SECONDARY BATTERIES —GENERATORS — CONVERTERS — MOTORS—CONTROL GEAR—TRANSFORMERS—STATIC BALANCERS — POWER FACTOR IMPROVEMENT — INSTRUMENTS — METERS — LAMPS — LIGHTNING CONDUCTORS — ELECTRIC HEATING AND COOKING — ELECTRICITY IN AGRICULTURE — TELEPHONY — RADIO - COMMUNICATION — INSULATING MATERIALS AND OILS — X-RAYS IN INDUSTRY.

(Revised and amplified by Prof. S. Parker Smith, D.Sc., Assoc.M.Inst.C.E., M.I.E.E.)

Electrical Units.

(For Glossary of Terms used in Electrical Engineering—see B.E.S.A. Report No. 205—1926. For Graphical Symbols for Electrical Purposes—see B.E.S.A. Report No. 108—1922.)

Unit.	Symbol.	Name.	Value in C.G.S. Units.	Dimensions in terms of	
				Length Time Mass.	Length Time Force Velocity.
Electromotive force . . .	E	Volt	10^8	$L^{\frac{3}{2}}M^{\frac{1}{2}}T^{-2}$	$V\sqrt{F}$
Current	I	Ampere	10^{-1}	$L^{\frac{1}{2}}M^{\frac{1}{2}}T^{-1}$	$\sqrt{F}$
Quantity	Q	Coulomb	10^{-1}	$L^{\frac{1}{2}}M^{\frac{1}{2}}$	$T\sqrt{F}$
Resistance	R	Ohm	10^9	$L\,T^{-1}$	V
Capacitance	C	Farad	10^{-9}	$L^{-1}T^2$	TV^{-1}
Inductance	L	Henry	10^9	L	L
Power or rate of working . .	P	Watt	10^7	$L^2M\,T^{-3}$	VF
Energy	W	Joule	10^7	$L^2M\,T^{-2}$	L F

The values of these units are as follows:—

One Ampere (*international*) is the unvarying electric current which, when passed through a solution of nitrate of silver in water, deposits silver at the rate of 0·001118 gramme per Second.

One Coulomb is the quantity of electricity represented by 1 Ampere passing for 1 Second.

One Ohm (*international*) is the resistance to an unvarying electric current of a column of mercury of uniform cross-section and length equal to 106·3 cm. whose mass is 14·4521 grammes at the temperature of melting ice.

One Volt (*international*) is the electrical pressure which, when steadily applied to a conductor whose resistance is one international Ohm, will produce a current of one international Ampere.

One Farad is that capacity which is charged to a potential of 1 Volt by a current of 1 Ampere flowing for 1 Second.

One Henry is the inductance of a circuit in which an E.M.F. of 1 Volt is induced by a current variation of 1 Ampere per Second.

One Watt is the power, or the rate at which work is being done, when a current of 1 Ampere passes through a resistance of 1 Ohm.

One Joule is the amount of work done, or heat generated, by 1 Ampere flowing for 1 Second through a resistance of 1 Ohm.

The following multiples and sub-multiples of these units are in common use :—

One millivolt	= one-thousandth of a volt.	One microhm	= one-millionth of an ohm.
One milliampere	= one-thousandth of an ampere.	One microfarad	= one-millionth of a farad.
		One millihenry	= one-thousandth of a henry.
One megohm	= one million ohms.	One kilowatt	= one thousand watts.

The following relations will be found useful :—

One horse-power is equal to 746 watts.

One foot-pound is equal to 1·356 joules = 0·323 calorie.

One joule is equal to 0·238 calorie.

One (gm.) calorie is the work done in raising 1 gramme of water 1° C., = 4·2 joules.

One (kg.) calorie = 1000 (gm.) calorie.

One joule is equal to 0·7373 foot-pound.

One dyne is the force which, acting on a mass of 1 gramme for 1 second, will give it a velocity of 1 cm. per second.

One erg is the work done by 1 dyne acting through 1 centimetre.

One watt = 0·00134 horse-power = 0·7373 ft.-lb. per second = 44·24 ft. lb. per minute = 0·0573 British thermal unit per minute = 1 joule per second.

One kilowatt-hour = 1,000 watt-hours = one Board of Trade Unit = 1·34 horse-power-hours = 2,656,400 ft.-lb. = 3,415* British thermal units = 860,600 calories.

The practical standard of electromotive force is the Weston Normal cell, which has an E.M.F. of 1·0185 volts at 15° C., decreasing by 0·00004 volt per degree C. rise of temperature.

Laws of Currents in Conductors.

Ohm's law states that: Voltage = Current × Resistance, or $V = I \times R$.

Kirchhoff's first law states that : The algebraic sum of the currents meeting at any point of a network of conductors is zero.

Kirchhoff's second law states that: The algebraic sum of all the products of resistances and currents in all the wires forming a closed network equals the algebraic sum of all the electromotive forces in the network.

RESISTIVITY OF PURE ANNEALED METALS, AND ALLOYS, AT 0° C. (32° F.).

(*Fleming & Dewar.*)

Metal.	Microhms per cm. cube.	Microhms per inch cube.	Ratio to Copper.	Temp. Coefft. F. per cent.	Temp. Coefft. C. per cent.
Aluminium	2·665	1·048	1·68	0·342	0·435
Antimony	36·0	14·17	22·7	0·217	0·39
Arsenic	33·3	13·1	21	—	—
Bismuth	111·0	43·7	69·6	0·195	0·35
Copper	**1·588**	**0·6252**	**1**	**0·237**	**0·426**
Gold	2·20	0·866	1·38	0·209	0·377
Iron	9·07	3·57	5·71	0·347	0·625
Lead	20·4	8·03	12·8	0·228	0·411
Mercury	94·07	37·04	59·2	0·040	0·072
Nickel	12·3	4·84	7·75	0·345	0·62
Platinum	10·92	4·299	6·88	0·204	0·367
Silver	1·47	0·579	0·926	0·222	0·40
Tin	13·05	5·138	8·22	0·244	0·44
Zinc	5·75	2·26	3·62	0·226	0·406
Constantan	51	20	32	±0·0005	±0·001
German Silver	30	11·8	19	0·0152	0·0273
Manganin	46·7	8·4	29·4	0·0008	0·0015
Nichrome	95	37·5	60	0·0239	0·043
Platinoid	38	15	24	0·0122	0·022

* British Engineering Standards Association, 'Glossary of Terms used in Electrical Engineering' (Publication No. 205—1926). The Electricity Commissioners have adopted the value 3412, which takes the *mean* British thermal unit to be $\frac{1}{180}$ of the quantity of heat required to raise the temperature of 1 lb. of water from 32° F. to 212° F.

Lenz's law states that : The current induced in a conductor flows in such a direction that its reaction opposes the change of position or state by which it is produced.

The power represented by a current of I amperes flowing under a difference of potential of V volts through a conductor of resistance R ohms $= I \times V = I^2R = V^2/R$ watts.

The heat energy generated by a current of I amperes flowing through a conductor of resistance R ohms for t seconds $= I^2Rt$ in joules.

Effect of Temperature on Resistance.

The resistance of metals increases with the temperature in accordance with the approximate formula

$$R_t = R_0 (1 + kt)$$

where k is the coefficient given in col. 5 or 6 on p. 2298 (according to the temperature scale employed) divided by 100.

If the resistance R_1 at temperature t_1 in ° C. be known, then for copper the resistance R_2 at temperature t_2 in ° C. will be :—

$$R_2 = R_1 \left(\frac{234{\cdot}5 + t_2}{234{\cdot}5 + t_1} \right)$$

Metallic Resistance Materials.

Metallic resistance materials are classified by the British Engineering Standards Association for maximum temperatures of 60, 200, 300, 700, and 1,000° C. respectively. According to Report No. 115—1924, the resistivity at 20° C. is not to differ more than 5 per cent. from that declared by the supplier, and the material must be uniform in resistance within 5 per cent. ; the temperature coefficents in the two low-temperature classes must not exceed 0·00002 and 0·00004 respectively for ten hours without the resistance changing more than 0·01 per cent. Lists of standard sizes of wires, sheets, and tapes are given in the Report.

Copper Conductors.

British Standards for Insulated Annealed Copper Conductors for Electric Power and Light.* No. 7—1926. (*Abstract.*)

The International Standards of Resistance for Copper are as follows :—At 20° C. (68° F.), the resistance of a wire of standard annealed copper 1 metre long and of 1 sq. millimetre cross-section is $\frac{1}{58}$ = (0·017241) ohm. The density of such copper at 20° C. is 8·89 grams per cub. centimetre, and the temperature coefficient of resistance is 0·00393 per degree C. At 60° F. (15·56° C.) the resistance of a solid conductor of standard annealed copper 1,000 yards long and of 1 sq. in. cross-section is 0·0240079 ohm ; the density is 8·892015, and the temperature coefficient of resistance is 0·0022221 per degree F.

At 15·5° C. (60° F.), the resistance of a copper conductor is accurately given by the expression :

Resistance in microhms = 8 × length in feet/cross section in square inches.

For the purposes of the specification an increase of 2 per cent. in the length of each wire in the stranded conductor has been assumed, and the area of the stranded conductor has been taken to be that of the solid wire which has the same resistance. In the case of multicore cables, an increase of 2 per cent. has been taken, to allow for the laying-up of the cores.

Tolerances allowed on solid conductors are, in weight, 3 per cent. ; in resistance, 3 per cent. plain, 4 per cent. tinned (0·036 in. diameter upwards), 5 per cent. (below 0·036 in.). On stranded wires, in weight 2 per cent. ; in resistance, 2 per cent. plain, 3 per cent. tinned (0·036 in. diameter upwards), 4 per cent. (below 0·036 in.).

The resistance of hard-drawn copper is about 3 per cent. higher than that of annealed copper conductors.

This specification also gives tables of sizes, resistances and weights of standard solid and stranded circular copper conductors, flexible cords and cables. Details of standard thickness of dielectric for paper, rubber and vulcanised bitumen insulated cables, the sizes and thickness of lead and tough rubber coverings, galvanised steel wire and tape for armouring are tabulated for cables up to 11,000 volts. Pressure tests are specified together with the method of procedure. An appendix contains diagrams showing sections of single and multicore cables and an extract from the British Standard Glossary of Electrical Terms and Definitions.

---

* For ' Bare Annealed Copper Wire for Electrical Machinery and Apparatus,' see p. 2261. For ' Enamelled Plain Copper Wire for Electrical Instruments and Apparatus,' see p. 2309.

BRITISH STANDARDS FOR DIMENSIONS, WEIGHT, AND RESISTANCE OF SOLID AND STRANDED CIRCULAR CONDUCTORS.*
(Standard Annealed Copper.)

Standard Sizes.		Number and Diameter of Wires.	Weight per 1,000 yards.	Resistance per 1,000 yards at 60° F.			Overall Diameter of Conductor.
Nominal Area.	Calculated Area.			Standard.	Maximum for Plain Wires.	Maximum for Tinned Wires.	
Sq. in.	Sq. in.	Inch.	Lbs.	Ohms.	Ohms.	Ohms.	Inch.
0·0010	0·001018	1/·036	11·77	23·59	24·29	24·53	0·036
0·0015	0·001521	1/·044	17·58	15·79	16·26	16·42	0·044
0·0020	0·001943	3/·029	23·37	12·36	12·61	12·85	0·062
0·0030	0·002994	3/·036	36·02	8·019	3·180	8·260	0·078
0·0030	0·003217	1/·064	37·20	7·463	7·687	7·761	0·064
0·0045	0·004546	7/·029	54·39	5·281	5·387	5·493	0·087
0·0070	0·007005	7/·036	83·81	3·427	3·496	3·530	0·108
0·0100	0·01046	7/·044	125·2	2·294	2·340	2·363	0·132
0·0145	0·01462	7/·052	174·9	1·643	1·675	1·692	0·156
0·0225	0·02214	7/·064	264·9	1·084	1·106	1·117	0·192
0·0300	0·02840	19/·044	340·4	0·8468	0·8637	0·8721	0·220
0·0400	0·03960	19/·052	475·5	0·6063	0·6184	0·6244	0·260
0·0600	0·05999	19/·064	720·3	0·4002	0·4082	0·4122	0·320
0·0750	0·07592	19/·072	911·6	0·3162	0·3225	0·3257	0·360
0·1000	0·1009	19/·083	1211	0·2380	0·2427	0·2451	0·415
0·1200	0·1168	37/·064	1403	0·2056	0·2097	0·2118	0·448
0·1500	0·1478	37/·072	1776	0·1625	0·1657	0·1673	0·504
0·2000	0·1964	37/·083	2360	0·1223	0·1247	0·1259	0·581
0·2500	0·2465	37/·093	2963	0·09738	0·09933	0·1003	0·651
0·3000	0·3024	37/·103	3635	0·07939	0·08098	0·08177	0·721
0·4000	0·4064	61/·093	4886	0·05908	0·06026	0·06085	0·837
0·5000	0·4985	61/·103	5994	0·04816	0·04913	0·04961	0·927
0·6000	0·6062	91/·093	7290	0·03961	0·04040	0·04079	1·023
0·7500	0·7435	91/·103	8942	0·03229	0·03294	0·03326	1·133
0·8500	0·8459	127/·093	10175	0·02838	0·02895	0·02923	1·209
1·0000	1·0376	127/·103	12481	0·02314	0·02360	0·02383	1·339

* For comparative Tables of Old and New Standard Sizes, see p. 2207.

COMPARATIVE TABLE OF OLD AND NEW STANDARD SIZES.

New Standard.		Old Standard.		New Standard.		Old Standard.	
New Nominal Area in Sq. Inches.	Number and Diameter in inches of Wires.	Number and Gauge of Diameter in inches of Wires.	Old Nominal Area in Sq. Inches.	New Nominal Area in Sq. Inches.	Number and Diameter in inches of Wires.	Number and Gauge or Diameter in inches of Wires.	Old Nominal Area in Sq. Inches.
0·001	1/·036	1/20 S.W.G.	0·0010	—	—	7/14 S.W.G.	0·0346
0·0015	1/·044	—	—	0·04	19/·052	—	—
—	—	1/18 ,,	0·0018	—	—	19/17 ,,	0·0459
—	—	3/22 ,,	0·0018	0·06	19/·064	19/16 ,,	0·0600
0·002	3/·029	—	—	0·075	19/·072	19/15 ,,	0·0750
—	—	7/25 ,,	0·0022	—	—	19/14 ,,	0·0937
0·003	3/·036	3/20 ,,	0·0030	0·1	19/·083	—	—
—	—	7/23 ,,	0·0031	0·12	37/·064	37/16 ,,	0·1168
0·003	1/·064	1/16 ,,	0·0032	—	—	19/13 ,,	0·1250
—	—	7/22 ,,	0·0042	0·15	37/·072	37/15 ,,	0·1500
0·0045	7/·029	—	—	—	—	37/14 ,,	0·1824
—	—	7/21½ ,,	0·0049	0·2	37/·083	37/·083″	0·2000
0·007	7/·036	7/20 ,,	0·0070	0·25	37/·093	37/·092″	0·2500
—	—	7/19 ,,	0·0086	0·3	37/·103	37/·104″	0·3000
0·01	7/·044	—	—	0·4	61/·093	61/·092″	0·4000
—	—	7/18 ,,	0·0125	0·5	61/·103	61/·104″	0·5000
0·0145	7/·052	—	—	0·6	91/·093	61/·112″	0·6000
—	—	7/17 ,,	0·0170	0·75	91/·103	91/·101″	0·7500
0·0225	7/·064	7/16 ,,	0·0220	0·85	127/·093	—	—
0·03	19/·044	—	—	1·0	127/·103	127/·101″	1·0000
—	—	19/18 ,,	0·0338				

FLEXIBLE CABLES.

A flexible cable is one in which the conductor (or conductors) exceeds 0·007 sq. in. in cross-section, and comprises a number of wires, the diameter of the wires and the material of the dielectric being such as to ensure flexibility.

DIMENSIONS AND RESISTANCE OF FLEXIBLE CABLES.*

Number and Diameter of Wires comprising Conductor.					Resistance in Ohms per 1000 yards at 60° F. (15·6° C.).		
Nominal Area in sq. in.	Diameter.				Standard.	Maximum for Plain Wires.	Maximum for Tinned Wires.
	0·010 in.	0·012 in.	0·018 in.	0·029 in.			
0·01	140/·010	97/·012†	—	—	2·29	2·34	2·39
0·0145	195/·010	—	60/·018†	—	1·64	1·68	1·71
0·0225	296/·010	—	91/·018†	—	1·08	1·11	1·13
0·03	—	266/·012	117/·018†	—	0·847	0·864	0·881
0·04	—	368/·012	163/·018†	—	0·606	0·618	0·631
0·06	—	557/·012	248/·018†	—	0·400	0·408	0·416
0·075	—	705/·012	313/·018	121/·029†	0·316	0·323	0·329
0·1	—	—	416/·018	160/·029†	0·238	0·243	0·247
0·12	—	—	482/·018	186/·029†	0·206	0·210	0·214
0·15	—	—	610/·018	235/·029†	0·163	0·166	0·169
0·2	—	—	810/·018	312/·029†	0·122	0·125	0·127
0·25	—	—	1017/·018	392/·029†	0·0974	0·0993	0·101
0·3	—	—	—	481/·029	0·0794	0·0810	0·0826
0·4	—	—	—	646/·029	0·0591	0·0603	0·0614
0·5	—	—	—	792/·029	0·0482	0·0491	0·0501

* For current-carrying capacity, see p. 2209. † For trailing cables and similar purposes.

Flexible Cords.

A flexible cord is a flexible cable of cross-section not exceeding 0·007 sq. in.

Dimensions and Resistance of Flexible Cords.

Nominal Area.	Ordinary Flexible Cords.				Flexible Cords with Tough Rubber Sheathing.		
	Number of 0·0076-in. Diameter Wires comprising Conductor.	Resistance in Ohms per 1,000 Yards at 60° F. (15·6° C.)			Number of 0·012-in. Diameter Wires comprising Conductor.	Resistance in Ohms per 1,000 Yards at 60° F. (15·6° C.)	
		Standard.	Maximum for			Standard.	Maximum.
			Plain Wires.	Tinned Wires.			
Sq. in.	Inch.	Ohms.	Ohms.	Ohms.	In.	Ohms.	Ohms.
0·0006	14/·0076	39·7	40·5	41·3	7/·012	40·5	41·3
0·0010	23/·0076	24·2	24·6	25·1	11/·012	24·6	25·1
0·0017	40/·0076	13·9	14·2	14·4	16/·012	14·2	14·4
0·0030	70/·0076	7·94	8·10	8·26	28/·012	8·10	8·26
0·0048	110/·0076	5·05	5·15	5·25	44/·012	5·15	5·25
0·0070	162/·0076	3·43	3·50	3·57	65/·012	3·50	3·57

Thickness of Insulation and Current-carrying Capacity of Flexible Cords.*

Nominal Area of Conductor. Sq. In.	Minimum Thickness of Dielectric in Inches (Ordinary Flexible Cords).				Flexible Cords with Tough Rubber Sheathing.			Maximum Currents in Amps. (subject to Voltage Drop).
	High Insulation, Kind 1.		Medium Insulation, Kind 2.		Thickness of Tough Rubber.	Overall Diameter in Inches.		
	Pure Rubber.	Pure and Vulcanising Rubber.	Pure Rubber.	Vulcanising Rubber.		250 Volts.	600 Volts.	
0·0006	0·02	0·033	0·015	0·028	0·05	0·200	0·244	1·8
0·001	0·02	0·034	0·015	0·029	0·05	0·214	0·255	3·0
0·0017	0·02	0·035	0·015	0·030	0·05	0·227	0·267	5·0
0·003	0·02	0·036	0·015	0·031	0·05	0·248	0·290	8·5
0·0048	0·02	0·038	0·015	0·032	0·05	0·268	0·308	13·0
0·007	0·02	0·039	—	—	0·05	0·288	0·328	17·0

* A flexible earth wire is not regarded as a conductor for the purpose of this table.

VULCANISED RUBBER CABLES.

CURRENT-CARRYING CAPACITY OF COPPER CONDUCTORS.

Nominal Area of Conductor.	Number and Diameter (In.) of Wires.	Single Cables Run in Pairs.	Concentric or Twin Cable.	Three-core Cable.	Length of Lead and Return for 1-Volt Drop.
Sq. In.		Amps.	Amps.	Amps.	Feet.
0·001	1/·036	4·1	3·5	—	30
0·0015	1/·044	6·1	5·2	—	30
0·002	3/·029	7·8	6·7	—	30
0·003	3/·036	12·0	10·3	—	29
0·003	1/·064	12·9	11·1	—	29
0·0045	7/·029	18·2	15·7	13·6	28
0·007	7/·036	24·0	20·6	18·0	33
0·01	7/·044	31·0	26·6	23·2	39
0·0145	7/·052	37·0	32·0	27·8	45
0·0225	7/·064	46·0	39·0	34·0	55
0·03	19/·044	53·0	46·0	40·0	61
0·04	19/·052	64·0	55·0	47·0	71
0·06	19/·064	83·0	71·0	59·0	83
0·075	19/·072	97·0	83·0	69·0	90
0·1	19/·083	118·0	100·0	83·0	98
0·12	37/·064	130·0	118·0	90·0	103
0·15	37/·072	152·0	126·0	105·0	112
0·2	37/·083	184·0	149·0	126·0	123
0·25	37/·093	214·0	170·0	146·0	132
0·3	37/·103	240·0	188·0	—	145
0·4	61/·093	288·0	220·0	—	162
0·5	61/·103	332·0	249·0	—	172
0·6	91/·093	384·0	—	—	181
0·75	91/·103	461·0	—	—	185
0·85	107/·093	512·0	—	—	190
1·0	127/·103	595·0	—	—	200

The figures for vulcanised rubber cables apply to single cables run in pairs in iron conduits or in wood casing, and to single cables sheathed with tough rubber compound, and to concentric, twin, and three-core cables of any finish run singly.

The maximum permissible currents (subject to voltage drop) given in the table for conductors up to 1 square inch in area allow for a rise in temperature of 20° F. (11·1° C.). For sizes below 0·007 square inch the table is based on a current density of 4,000 amperes per square inch. The length of lead and return given in the last column is based on the figures in the third column.

Where the temperature of the air exceeds 80° F. (26·6° C.), the current should be reduced so that the normal maximum running temperature does not exceed 100° F. (37·7° C.). Rubber-insulated cables should not be allowed to attain a temperature higher than 120° F. (48·8° C.) for long periods.

IMPREGNATED-PAPER AND LEAD-COVERED CABLES.

CURRENT-CARRYING CAPACITY OF COPPER CONDUCTORS.

Nominal Area of Conductor.	Number and Diameter (In.) of Wires.	Single Cables run in Pairs.	Concentric or Twin Cable.	Three-core Cable.	Length of Lead and Return for 1-volt. drop.
Sq. In.		Amps.	Amps.	Amps.	Ft.
0·001	1/·036	4·1	3·5	—	30
0·0015	1/·044	6·1	5·2	—	30
0·002	3/·029	7·8	6·7	—	30
0·003	3/·036	12·0	10·3	—	29
0·003	1/·064	12·9	11·1	—	29
0·0045	7/·029	18·2	15·7	13·6	28
0·007	7/·036	28·0	24·0	21·0	27
0·01	7/·044	42·0	36·0	31·0	27
0·0145	7/·052	57·0	49·0	43·0	28
0·0225	7/·064	75·0	65·0	56·0	32
0·03	19/·044	87·0	76·0	66·0	35
0·04	19/·052	104·0	89·0	76·0	41
0·06	19/·064	135·0	116·0	97·0	48
0·075	19/·072	157·0	135·0	111·0	52
0·1	19/·083	191·0	162·0	134·0	57
0·12	37/·064	210·0	177·0	146·0	60
0·15	37/·072	246·0	204·0	170·0	65
0·2	37/·083	296·0	240·0	203·0	72
0·25	37/·093	343·0	265·0	233·0	78
0·3	37/·103	385·0	302·0	258·0	85
0·4	61/·093	464·0	354·0	—	95
0·5	61/·103	540·0	405·0	—	100
0·6	91/·093	624·0	—	—	105
0·75	91/·103	738·0	—	—	109
0·85	127/·093	815·0	—	—	116
1·0	127/·103	932·0	—	—	121

The figures for impregnated-paper and rubber-insulated cables apply to single cables run in pairs, and to concentric, twin, and three-core cables run singly.

The maximum permissible currents (subject to voltage drop) given in the table for conductor up to 1 square inch in area allow for a rise in temperature of 50° F. (27·7° C.). For sizes below 0·007 square inch, the table is based on a current density of 4,000 amperes per square inch. The length of lead and return given in the last column is based on the figures in the third column.

Where the temperature of the air exceeds 80° F. (26·6° C.), the current should be reduced so that the normal maximum running temperature does not exceed 130° F. (54·4° C.). Impregnated-paper lead-covered cables, for pressures not exceeding 660 volts, should not be allowed to attain a permanent temperature higher than 176° F. (80° C.).

"LIVERPOOL CABLES"

For Dependable Service in every Situation

HOUSE WIRING
FACTORY WIRING
OVERHEAD LINES
UNDERGROUND MAINS
ON SHIPS AND DEEP DOWN IN MINES

Stocks held at:
LONDON
LIVERPOOL
LEEDS
MANCHESTER
BIRMINGHAM
NEWCASTLE
NOTTINGHAM
CARDIFF
GLASGOW

Members of the Cable Makers Association

THE LIVERPOOL ELECTRIC CABLE COMPANY LIMITED,
LINACRE LANE, BOOTLE, LIVERPOOL

F. 2210

CRAIGPARK ELECTRIC CABLES

INSULATED WITH

IMPREGNATED PAPER

VULCANISED INDIA RUBBER

VULCANISED BITUMEN

GUTTA-PERCHA

TO ALL SPECIFICATIONS

Sole Manufacturers of

"BRUMITE" AND "BRUMAC" CABLES

As supplied to leading Railway Companies, Chemical Works, &c., for special conditions.

The Craigpark Electric Cable COMPANY LIMITED

Head Office and Works :

SPRINGBURN, GLASGOW

Branches :

LONDON:
17 Victoria Street, Westminster.

MANCHESTER:
49 Deansgate.

F. 2211

Rubber-Insulated Flexible Cables.*

Current-carrying Capacity of Rubber-Insulated Flexible Cables for use with Portable Appliances.

Nominal Area of Conductor.	Maximum Current in Amps. (subject to Voltage Drop).	
	Two Conductor.	Three Conductor.
Sq. In.		
0·01	20	17
0·0145	24	20
0·0225	30	25
0·03	35	30
0·04	42	35

SIZES, WEIGHT, AND INSULATION RESISTANCE OF CABLES INSULATED WITH VULCANISED RUBBER.

Association Cables, 600-Megohm Grade.

(Abstract from Silvertown List.)

Nominal Area of Conductor.	Number and Diameter of Wires.	Minimum Thickness of Dielectric.	Minimum Insulation Resistance for a Mile.	Grade S. Taped, Braided and covered preservative Compound.		Grade RL. Taped and Lead covered.	
				Approximate Overall Diameter.	Approximate Weight per 1,000 yds.	Approximate Overall Diameter.	Approximate Weight per 1,000 yds.
Sq. Inch.	Inch.	Inch.	Megohms.	Inch.	Cwt.	Inch.	Cwt.
0·0010	1/·036	0·034	2000	0·165	0·43	0·200	3·01
0·0015	1/·044	0·034	,,	0·173	0·5	0·208	3·2
0·0020	3/·029	0·036	1250	0·195	0·64	0·230	3·64
0·0030	1/·064	0·036	2000	0·197	0·75	0·232	3·76
0·0030	3/·036	0·038	1250	0·215	0·83	0·250	4·13
0·0045	7/·029	0·039	,,	0·226	1·02	0·261	4·49
0·0070	7/·036	0·041	900	0·259	1·44	0·314	6·49
0·0100	7/·044	0·043	,,	0·287	1·92	0·342	7·53
0·0145	7/·052	0·046	900	0·317	2·53	0·392	10·28
0·0225	7/·064	0·049	,,	0·359	3·56	0·434	12·30
0·0300	19/·044	0·052	750	0·393	4·48	0·468	13·82
0·0400	19/·052	0·056	,,	0·441	6·11	0·516	16·53
0·0600	19/·064	0·062	,,	0·513	8·82	0·608	23·33

* A flexible wire is not regarded as a conductor for the purpose of this table.

SIZES, WEIGHT, AND INSULATION RESISTANCE OF CABLES INSULATED WITH VULCANISED RUBBER—(*cont.*).

Nominal Area of Conductor.	Number and Diameter of Wires.	Minimum Thickness of Dielectric.	Minimum Insulation Resistance for a Mile.	Grade S. Taped, Braided and covered preservative Compound.		Grade RL. Taped and Lead covered.	
				Approximate Overall Diameter.	Approximate Weight per 1,000 yds.	Approximate Overall Diameter.	Approximate Weight per 1,000 yds.
Sq. Inch.	Inch.	Inch.	Megohms.	Inch.	Cwt.	Inch.	Cwt.
0·0750	19/·072	0·066	600	0·596	10·96	0·656	26·73
0·1000	19/·083	0·072	,,	0·663	14·25	0·723	31·79
0·1200	37/·064	0·075	,,	0·702	16·25	0·762	34·82
0·1500	37/·072	0·080	,,	0·768	20·16	0·848	43·94
0·2000	37/·083	0·088	600	0·889	26·17	0·949	52·94
0·2500	37/·093	0·095	,,	0·973	33·12	1·053	66·02
0·3000	37/·103	0·102	,,	1·057	40·13	1·137	75·96
0·4000	61/·093	0·114	,,	1·197	53·44	1·297	99·96
0·5000	61/·103	0·121	600	1·301	64·72	1·421	121·13
0·6000	91/·093	0·125	,,	1·425	77·42	1·525	138·3
0·7500	91/·103	0·131	,,	1·547	93·78	1·667	166·77
0·8500	127/·093	0·135	,,	1·631	106·09	1·751	183·05
1·0000	127/·103	0·141	,,	1·773	128·63	1·893	212·29

ASSOCIATION CABLES, 2,500-MEGOHM GRADE.

Nominal Area of Conductor.	Number and Diameter of Wires.	Minimum Thickness of Dielectric.	Minimum Insulation Resistance for a Mile.	Grade C. Taped, Braided and covered preservative Compound.		Grade CL. Taped and Lead covered.	
				Approximate Overall Diameter.	Approximate Weight per 1,000 yds.	Approximate Overall Diameter.	Approximate Weight per 1,000 yds.
Sq. Inch.	Inch.	Inch.	Megohms.	Inch.	Cwt.	Inch.	Cwt.
0·0010	1/·036	0·034	5000	0·165	0·43	0·200	3·01
0·0015	1/·044	0·034	,,	0·173	0·50	0·208	3·2
0·0020	3/·029	0·036	4500	0·195	0·64	0·230	3·64
0·0030	1/·064	0·036	5000	0·197	0·75	0·232	3·76
0·0030	3/·036	0·038	4500	0·215	0·83	0·250	4·13
0·0045	7/·029	0·039	,,	0·226	1·02	0·261	4·49
0·0070	7/·036	0·041	4000	0·259	1·44	0·314	6·49
0·0100	7/·044	0·043	,,	0·287	1·92	0·342	7·53
0·0145	7/·052	0·046	4000	0·317	2·53	0·392	10·28
0·0225	7/·064	0·049	3500	0·359	3·56	0·434	12·30
0·0300	19/·044	0·052	,,	0·393	4·48	0·468	13·82
0·0400	19/·052	0·056	3000	0·441	6·11	0·516	16·53
0·0600	19/·064	0·062	,,	0·513	8·82	0·608	23·33
0·0750	19/·072	0·066	3000	0·596	10·96	0·656	26·73
0·1000	19/·083	0·072	,,	0·663	14·25	0·723	31·79
0·1200	37/·064	0·075	,,	0·702	16·25	0·762	34·82
0·1500	37/·072	0·080	,,	0·768	20·16	0·848	43·94
0·2000	37/·083	0·088	2500	0·889	26·17	0·949	52·94
0·2500	37/·093	0·095	,,	0·973	33·12	1·053	66·02
0·3000	37/·103	0·102	,,	1·057	40·13	1·137	75·96
0·4000	61/·093	0·114	,,	1·197	53·44	1·297	99·96
0·5000	61/·103	0·121	2500	1·301	64·72	1·421	121·13
0·6000	91/·093	0·125	,,	1·425	77·42	1·525	138·3
0·7500	91/·103	0·131	,,	1·547	93·78	1·667	166·77
0·8500	127/·093	0·135	,,	1·631	106·09	1·751	183·05
1·0000	127/·103	0·141	,,	1·773	128·63	1·893	212·29

ALUMINIUM CONDUCTORS.*

Conductivity.

The conductivity of aluminium depends upon the purity, and, to a less extent, upon the amount of work put upon the metal in its evolution from billet to the finished wire or bar. The British standard of resistance for hard copper is 2 per cent. higher than that of soft, and this difference also applies to hard and soft aluminium. The conductivity of all materials is normally compared with that of the International Standard for annealed copper, and on this basis hard-drawn aluminium will have a minimum conductivity of 60 per cent. Hard-drawn copper will have a conductivity of about 98 per cent. and hence compared with hard-drawn copper, hard-drawn aluminium will have a conductivity of 61·3 per cent.

Weight.

The specific gravity of aluminium wire is about 2·703. As the specific gravity of copper wire is about 8·89, the relative weight of a given volume of each metal is in the ratio of 8·89 to 2·71 = 3·3. In other words, a copper wire of any given size and length will weigh 3·3 times as much as an aluminium wire of the same size and length. Aluminium, therefore, having a conductivity of 61 per cent. of that of copper, will have a sectional area 1·642 times as great as a copper conductor of the same resistance and length, and the ratio of the weights will be 3·3 to 1·642 = 2·01; that is to say, a copper conductor of any given resistance and length will weigh approximately twice as much as an aluminium conductor of the same resistance and length.

The ratios are therefore as follows:—

	Copper.	Aluminium.
Conductance for equal section	1	0·61
Section for equal conductance	1	1·642
Diameter for equal conductance	1	1·28
Weight for equal bulk	3·3	1
Weight for equal conductance	2·0	1

Tensile Strength.

Although the tensile strength of aluminium in the annealed state is low, the strength of aluminium in the form of wire is greatly increased owing to the work put upon it during rolling and drawing. The strength can be increased more than 100 per cent. without making the elongation too small for engineering purposes.

Durability.

Aluminium has withstood the test of varying and severe climatic conditions. Lines have been in operation for over twenty years in exposed positions on the sea coast which show little signs of deterioration. The metal is less affected by sulphurous and sulphuric fumes than copper, and shows less deterioration in the presence of any of the commoner acids. On the other hand, aluminium is attacked by strong alkalies, and should not be used in such circumstances without efficient painting.

Increase of Resistance with Temperature.

The coefficient of increase of resistance with temperature varies somewhat with the extent and nature of the impurities present and the temperature range over which it is taken. At 20° C. the value for aluminium may be taken at 0·00407 per °C., at 15·6 °C. in accordance with *B.E.S.A. Specification*, No. 215–1930.

Coefficient of Linear Expansion.

The coefficient of linear expansion being somewhat greater with aluminium than with copper, due allowance should be made for this when installing long lines of solid feeders or bars.

Action of Acids

Aluminium is not acted upon by any of the common acids in a cold state, with the exception of hydrochloric acid, and withstands the effects of sea-water better than either copper or iron; but galvanic action is set up between aluminium and any of the common metals in the presence of moisture, and for this reason, if iron and copper joints must be used, they should be thoroughly protected from moisture by taping or painting with bituminous paint. It is found that galvanised iron parts can be used with aluminium owing to their closer proximity in the electro-chemical series, and the hardware employed with aluminium overhead lines is usually of galvanised malleable iron.

* See also B.E.S.A. Report No. 215—1930, 'Hard-Drawn Aluminium and Steel-Cored Aluminium Conductors for Overhead Power Transmission Purposes.'

STANDARD STRANDED ALUMINIUM

1	2	3	4	5	6	7	8	9	10	11
Standard Nominal Copper Area.	Resistance of Standard Hard-drawn Copper Conductor at 60° F.		Stranding and Wire Diameter of Aluminium Conductor (inches).	Calculated Area of Aluminium.	Weight per 1,000 yds. lb.			Weight per mile. lb.		
	Ohms per 1,000 yds.	Ohms per mile.			Standard.	Max.	Min.	Standard.	Max.	Min.
Sq. in.				Sq. in.						
0·02	1·236	2·175	3/·118	0·03217	117·6	120·0	115·3	207·0	211·2	202·9
0·025	0·9887	1·740	3/·132	0·04025	147·2	150·2	144·3	259·1	264·3	253·9
0·03	0·8237	1·450	3/·144	0·04790	175·2	178·7	171·7	308·3	314·5	302·2
0·04	0·6177	1·087	7/·110	0·06540	237·9	242·6	233·1	418·6	427·0	410·3
0·05	0·4943	0·8700	7/·122	0·08045	292·6	298·4	286·7	515·0	525·3	504·7
0·06	0·4118	0·7248	7/·134	0·09706	353·0	360·0	345·9	621·3	633·7	608·8
0·07	0·3530	0·6213	7/·144	0·1121	407·6	415·8	399·5	714·4	731·8	703·1
0·075	0·3294	0·5798	7/·149	0·1200	436·4	445·2	427·7	768·1	783·5	752·8
0·08	0·3089	0·5437	7/·154	0·1282	466·2	475·5	456·9	820·6	837·0	804·1
0·09	0·2745	0·4831	7/·164	0·1454	528·7	539·3	518·2	930·6	949·2	912·0
0·10	0·2469	0·4345	7/·173	0·1618	588·4	600·1	576·6	1036	1056	1015
0·125	0·1976	0·3477	7/·193	0·2013	732·3	746·9	717·6	1289	1315	1263
0·15	0·1656	0·2914	7/·211	0·2407	875·2	892·7	857·7	1540	1571	1510
0·175	0·1408	0·2478	19/·139	0·2830	1033	1053	1012	1818	1854	1781
0·20	0·1224	0·2155	19/·149	0·3251	1187	1210	1163	2089	2130	2047
0·225	0·1096	0·1928	19/·157	0·3610	1318	1344	1291	2319	2365	2273
0·25	0·09861	0·1736	19/·166	0·4036	1473	1502	1444	2592	2644	2541
0·30	0·08126	0·1430	19/·183	0·4905	1790	1826	1754	3151	3214	3088
0·35	0·06978	0·1228	19/·197	0·5684	2074	2116	2033	3651	3724	3578
0·40	0·06111	0·1076	19/·211	0·6520	2380	2427	2332	4188	4272	4105
0·45	0·05428	0·09553	37/·161	0·7389	2700	2754	2645	4751	4846	4656
0·50	0·04919	0·08657	37/·168	0·8045	2939	2998	2881	5173	5277	5070
0·60	0·04175	0·07348	37/·183	0·9546	3488	3558	3418	6138	6261	6016
0·75	0·03297	0·05803	37/·206	1·2096	4420	4508	4331	7778	7934	7623

CONDUCTORS (B.E.S.A. STANDARDS).

12	13	14	15	16	17	18	19	20	1
Resistance at 60° F. When corrected to Standard Weight.				Resistance at 20° C. When corrected to Standard Weight.				Approximate Ultimate Strength of Conductor.	Standard Nominal Copper Area.
Ohms per 1,000 yds.		Ohms per mile.		Ohms per 1,000 yds.		Ohms per mile.			
Standard.	Max.	Standard.	Max.	Standard.	Max.	Standard.	Max.		
								lb.	
1·231	1·243	2·166	2·188	1·253	1·266	2·206	2·228	860	0·02
0·9836	0·9934	1·731	1·748	1·002	1·012	1·763	1·781	995	0·025
0·8265	0·8347	1·455	1·469	0·8417	0·8501	1·481	1·496	1185	0·03
0·6053	0·6113	1·065	1·076	0·6164	0·6226	1·085	1·096	1725	0·04
0·4920	0·4970	0·8660	0·8747	0·5011	0·5061	0·8820	0·8908	2040	0·05
0·4079	0·4119	0·7178	0·7250	0·4154	0·4195	0·7311	0·7384	2370	0·06
0·3532	0·3567	0·6216	0·6278	0·3597	0·3633	0·6331	0·6394	2735	0·07
0·3299	0·3332	0·5806	0·5864	0·3360	0·3393	0·5913	0·5971	2930	0·075
0·3088	0·3119	0·5435	0·5489	0·3145	0·3176	0·5535	0·5590	3065	0·08
0·2723	0·2750	0·4792	0·4840	0·2773	0·2801	0·4881	0·4929	3480	0·09
0·2447	0·2471	0·4307	0·4350	0·2492	0·2517	0·4386	0·4430	3790	0·10
0·1966	0·1986	0·3460	0·3495	2·2002	0·2022	0·3524	0·3559	4620	0·125
0·1645	0·1661	0·2895	0·2924	0·1675	0·1692	0·2949	0·2978	5405	0·15
0·1399	0·1413	0·2462	0·2487	0·1425	0·1439	0·2508	0·2533	6775	0·175
0·1218	0·1230	0·2143	0·2164	0·1240	0·1252	0·2182	0·2204	7785	0·20
0·1098	0·1108	0·1930	0·1949	0·1117	0·1128	0·1966	0·1985	8470	0·225
0·09809	0·09907	0·1726	0·1744	0·09990	0·1009	0·1758	0·1776	9470	0·25
0·08071	0·08152	0·1421	0·1435	0·08220	0·08302	0·1447	0·1461	11160	0·30
0·06965	0·07035	0·1226	0·1238	0·07093	0·07164	0·1248	0·1261	12790	0·35
0·06071	0·06132	0·1069	0·1079	0·06183	0·06245	0·1088	0·1099	14360	0·40
0·05358	0·05411	0·09430	0·09524	0·05456	0·05511	0·09603	0·09699	16610	0·45
0·04921	0·04970	0·08660	0·08747	0·05011	0·05061	0·08820	0·08908	18090	0·50
0·04147	0·04188	0·07299	0·07372	0·04223	0·04266	0·07433	0·07507	20800	0·60
0·03273	0·03305	0·05760	0·05817	0·03333	0·03366	0·05866	0·05925	25800	0·75

STANDARD STEEL-CORED ALUMINIUM

1	2	3	4	5	6	7	8	9	10	11
Standard Nominal Copper Area.	Resistance of Standard Hard-drawn Copper Conductor at 60° F.		Stranding and Wire Diameter (inches).		Calculated Area of Aluminium.	Total Weight per 1,000 yards. lb.			Total Weight per mile. lb.	
	Ohms per 1,000 yds.	Ohms per mile.	Alum-inium.	Steel.		Stand-ard.	Max.	Min.	Stand-ard.	Max.
Sq.in.					Sq. in.					
0·025	0·9887	1·740	6/·0930	1/·0930	0·04000	215·4	221·1	209·7	379·1	389·2
0·03	0·8237	1·450	6/·102	1/·102	0·04807	259·1	266·0	252·3	456·1	468·2
0·04	0·6177	1·087	6/·118	1/·118	0·06433	346·8	356·0	337·7	610·3	626·6
0·05	0·4943	0·8700	6/·132	1/·132	0·08050	434·0	445·5	422·5	763·8	784·1
0·06	0·4118	0·7248	6/·144	1/·144	0·09580	516·4	530·2	502·9	908·9	933·1
0·07	0·3530	0·6213	6/·157	1/·157	0·1139	613·9	630·2	597·8	1080	1109
0·075	0·3294	0·5798	6/·161	1/·161	0·1198	645·6	662·7	628·6	1136	1166
0·08	0·3089	0·5437	6/·166	1/·166	0·1273	686·3	704·5	668·3	1208	1240
0·09	0·2745	0·4831	6/·177	1/·177	0·1447	780·3	801·0	759·8	1373	1410
0·10	0·2469	0·4345	6/·186	7/·0620	0·1598	803·8	824·3	783·4	1415	1451
			7/·173	7/·0760	0·1613	919·4	944·5	894·5	1618	1662
0·125	0·1976	0·3477	*26/·100	7/·0750	0·1995	1053	1081	1026	1853	1902
			30/·0930	7/·0930	0·1998	1224	1258	1190	2154	2215
0·15	0·1656	0·2914	30/·102	7/·102	0·2403	1472	1514	1431	2591	2664
0·175	0·1408	0·2478	30/·110	7/·110	0·2795	1712	1761	1664	3014	3099
0·20	0·1224	0·2155	30/·118	7/·118	0·3217	1970	2026	1915	3468	3566
0·225	0·1096	0·1928	30/·125	7/·125	0·3609	2211	2273	2149	3891	4001
0·25	0·09861	0·1736	30/·132	7/·132	0·4025	2466	2535	2397	4339	4462
0·30	0·08126	0·1430	30/·146	7/·146	0·4924	3016	3101	2932	5309	5459
0·35	0·06978	0·1228	30/·157	7/·157	0·5694	3488	3586	3391	6139	6312
0·40	0·06111	0·1076	30/·168	7/·168	0·6520	3994	4107	3882	7029	7228
0·45	0·05428	0·09553	30/·177	7/·177	0·7237	4433	4558	4310	7803	8023
0·50	0·04919	0·08657	54/·139	7/·139	0·8034	4054	4158	3952	7136	7318
0·60	0·04175	0·07348	54/·152	7/·152	0·9607	4848	4972	4725	8533	8751
0·75	0·03297	0·05803	54/·170	7/·170	1·2017	6064	6219	5911	10673	10946

* The 26 aluminium/7 steel is the

CONDUCTORS (B.E.S.A. STANDARDS).

12	13	14	15	16	17	18	19	20	21	1
Total Weight per mile. lb.	Resistance at 60° F. When corrected to Standard Weight.				Resistance at 20° C. When corrected to Standard Weight.				Approximate Ultimate Strength of Conductor.	Standard Nominal Copper Area.
	Ohms per 1,000 yds.		Ohms per mile.		Ohms per 1,000 yds.		Ohms per mile.			
Min.	Standard.	Max.	Standard.	Max.	Standard.	Max.	Standard.	Max.	lb.	Sq. in.
369·2	0·9907	1·001	1·744	1·761	1·009	1·019	1·776	1·794	2160	0·025
444·1	0·8236	0·8318	1·450	1·464	0·8388	0·8472	1·476	1·491	2545	0·03
594·3	0·6154	0·6215	1·083	1·094	0·6267	0·6330	1·103	1·114	3410	0·04
743·7	0·4918	0·4967	0·8655	0·8742	0·5008	0·5058	0·8815	0·8903	4105	0·05
885·0	0·4132	0·4174	0·7273	0·7346	0·4208	0·4250	0·7407	0·7481	4885	0·06
1052	0·3476	0·3511	0·6118	0·6179	0·3540	0·3576	0·6231	0·6293	5735	0·07
1106	0·3306	0·3339	0·5818	0·5876	0·3367	0·3400	0·5925	0·5984	6035	0·075
1176	0·3110	0·3141	0·5473	0·5528	0·3167	0·3199	0·5574	0·5629	6415	0·08
1337	0·2735	0·2762	0·4814	0·4862	0·2785	0·2813	0·4902	0·4951	7220	0·09
1379	0·2477	0·2502	0·4359	0·4403	0·2522	0·2548	0·4439	0·4484	7385	0·10
1574	0·2454	0·2479	0·4319	0·4362	0·2499	0·2524	0·4399	0·4443	9270	
1805	0·1977	0·1997	0·3480	0·3515	0·2014	0·2034	0·3544	0·3580	10660	0·125
2094	0·1981	0·2001	0·3487	0·3522	0·2018	0·2038	0·3552	0·3587	12870	
2519	0·1647	0·1664	0·2899	0·2928	0·1678	0·1694	0·2952	0·2982	15240	0·15
2929	0·1416	0·1430	0·2493	0·2518	0·1442	0·1457	0·2539	0·2564	17720	0·175
3371	0·1231	0·1243	0·2166	0·2188	0·1253	0·1266	0·2206	0·2228	20390	0·20
3783	0·1097	0·1108	0·1930	0·1950	0·1117	0·1128	0·1966	0·1986	22520	0·225
4218	0·09836	0·09934	0·1731	0·1748	0·1002	0·1012¾	0·1763	0·1781	24710	0·25
5161	0·08040	0·08120	0·1415	0·1429	0·08188	0·08270	0·1441	0·1455	30230	0·30
5968	0·06953	0·07022	0·1224	0·1236	0·07081	0·07151	0·1246	0·1259	34590	0·35
6833	0·06072	0·06133	0·1069	0·1079	0·06184	0·06246	0·1088	0·1099	39600	0·40
7585	0·05470	0·05525	0·09628	0·09724	0·05571	0·05627	0·09805	0·09903	43600	0·045
6955	0·04952	0·05001	0·08715	0·08802	0·05043	0·05093	0·08876	0·08965	36300	0·50
8317	0·04141	0·04182	0·07288	0·07361	0·04217	0·04259	0·07422	0·07497	42860	0·60
10403	0·03311	0·03344	0·05827	0·05885	0·03372	0·03405	0·05934	0·05993	53630	0·75

recommended construction for the 0·125 sq. in. conductor.

Physical Properties of Aluminium and Copper.

Atomic weight of aluminium (oxygen = 16) = 26·97 (*Int. Atomic Wt. Comm.*, 1925).

Coefficient of linear expansion per °C. = 23 × 10^{-6} (*B.E.S.A. Specification*,

,, ,, ,, ,, °F. = 12·78 × 10^{-6} *No.* 215–1930).

Specific gravity, rolled or drawn = 2·703 (*B.E.S.A. Specification*, *No.* 215-1930).

With regard to the tensile strength of H.D. wires, the minimum wire diameter given should be 0·093 ins. (not 0·0935 ins.). See *B.E.S.A. Specification*.

The modulus of elasticity of H.D. aluminium conductors = 9·9 × 10^6 lbs. per sq. in.

The modulus of elasticity of H.D. aluminium stranded conductor = 9·6 × 10^6 lbs. per sq. in.

Coefficient of increase of resistance with temperature at 15·6° C. (60° F.) = ·00407 per ° C.

Coefficient of increase of resistance with temperature at 60° F. (15·6° C.) % 0·00226 per ° F.

Weight per 1,000 yds. per sq. in. of cross section = 3515 lbs.

Standard specific resistance in microhms per in.^{3} at 20° C. of H.D. aluminium = 1·1199.

All the above specific resistance figures are standard values.

Sulphur has no effect upon aluminium at ordinary temperatures, and consequently aluminium withstands the action of smoke and fog much better than either silver or copper. A thin transparent film of insoluble oxide forms immediately on the surface of aluminium when it is exposed to the atmosphere, protecting it from further oxidisation or corrosion. This film of oxide is a good insulator and will withstand the application of considerable electrical pressure before breaking down. Advantage is being taken of this feature in the manufacture of magnet windings, etc., which are otherwise uninsulated.

Capacitance and Inductance.

Aluminium being a non-magnetic material, the inductance or capacitance of the line can only be affected by the increased diameter. This increased diameter may decrease the inductance and increase the capacitance by some 5 per cent., but as all transmission lines possess more inductance than capacitance, this difference is in favour of aluminium, tending to improve the power factor of the system and to reduce the pressure drop due to inductance.

Heating Effect.

Aluminium, having a larger surface than copper of equal conductance, is more easily able to get rid of its heat, and when transmitting a given current with a given I^2R loss, aluminium will remain the cooler. This is a considerable advantage when installing electric apparatus in a hot atmosphere, or where current-carrying capacity is of more importance than the permissible pressure drop. In any case aluminium will have a higher overload capacity, from a heating point of view, than copper of equal resistance. In consequence of this aluminium is being largely used for busbars and solid rod battery connections. Erection on site is facilitated by the ease with which bends and joints can be made.

Corona Losses.—With a high voltage overhead line a serious loss of energy may occur in the form of a luminous discharge due to the breakdown of the air surrounding the conductors. The point at which corona begins is a function of the conductor diameter, and the use of aluminium or, better still, steel-cored aluminium, both of which involve a larger diameter than that of the equivalent copper, will safely enable a higher working voltage to be employed.

Aluminium Cable, Steel-Cored.

After the introduction of aluminium for electrical conductors it became apparent that if a conductor could be obtained, combining the advantages of lightness of aluminium with the strength of high-grade steel now obtainable on the market, the highest efficiency would result. Cables of this character were made up and have now been in service in such a variety of localities and conditions that their efficiency has been thoroughly demonstrated.

The cables consist of a steel core about which the aluminium is stranded. The steel for the smaller size cables consists of a single strand, while for the larger sizes several strands are used. These are composed of patent steel, having an ultimate strength of 160,000 lbs. per square inch to 200,000 lbs. per square inch, according to the diameter, an elastic limit of 70 per cent. of the ultimate strength, and an elongation of 4 to 6 per cent. on 10 inches. The wires are heavily

galvanised, and after the aluminium strands are in place over the steel, around which they fit closely by reason of the stress resulting from the load on the cable, the steel is to a large extent protected from corrosive action. This protection is due to the almost complete exclusion of moisture from the steel core by the aluminium envelope, and further on account of the insulating oxide film on the aluminium, which tends to prevent the flow of small local electric currents. Another feature which adds to this desirable condition is the fact that zinc and aluminium are closer together in the electro-chemical series than any other two metals in common use.

The following table shows at a glance the superiority in mechanical characteristics of steel-cored conductors over all-aluminium, or copper of equivalent conductance.

Size.	—	Aluminium.	Steel-cored Aluminium.	Copper.
0·025 sq. ins. copper area	Weight . .	1·0	1·48	2·04
	Diameter . .	1·0	0·98	0·79
	Ultimate load .	1·0	2·19	1·52
0·10 sq. ins. copper area	Weight . .	1·0	1·37	2·04
	Diameter . .	1·0	1·07	0·79
	Ultimate load .	1·0	1·95	1·55
0·40 sq. ins. copper area	Weight . .	1·0	1·68	2·04
	Diameter . .	1·0	1·11	0·79
	Ultimate load .	1·0	2·58	1·58

Physical Properties of Steel-Cored Aluminium Conductors.

The strength of a steel-cored aluminium conductor is somewhat less than the sum of the strengths of the component wires, owing to the fact that the aluminium wires will break before the steel wire reaches its ultimate stress. The strength of the composite conductor may be taken as (*a*) 98 per cent. of the sum of the strengths of the aluminium wires, plus 89 per cent. of the sum of the strengths of the steel wires when taken from the stranded conductor and tested, or (*b*) 98 per cent. of the sum of the strengths of the aluminium wires, plus 85 per cent. of the sum of the strengths of the steel wires based on the strengths of the component wires before stranding, *i.e.* in the coil (*B.E.S.A. Specification*, No. 215–1930).

The stress is not distributed uniformly over the different wires, and the stress in the steel will normally be about three times the stress in the aluminium.

The coefficient of expansion and modulus of elasticity of the composite conductor are calculated upon the assumption that there will be no relative movement between the aluminium and steel wires. This is found to be the case in practice, and calculations based on 'virtual' values of the constants as so determined are found to conform with sensible accuracy to observed values. The constants depend upon the ratio of steel section to aluminium section in the construction, and the following values apply to standard forms:

No. of Wires.	Coefficient of Linear Expansion per °F.	Modulus of Elasticity. Lbs. per sq. in.
7	$10{\cdot}55 \times 10^{-6}$	$12{\cdot}49 \times 10^{6}$
37	$10{\cdot}02 \times 10^{-6}$	$13{\cdot}31 \times 10^{6}$
61	$10{\cdot}90 \times 10^{-6}$	$11{\cdot}98 \times 10^{6}$

Splicing.

Small sizes of aluminium and steel-cored aluminium conductors are joined by aluminium McIntyre Joints of the same form as commonly employed with copper. For larger sizes of steel-cored aluminium conductors different systems are employed, in all of which the aluminium and the steel are joined separately. Among the methods employed may be mentioned the com-

pression joint in which the steel wires are joined by a steel McIntyre joint, which is enclosed in an aluminium sleeve pressed on to the aluminium wires, and the 'Pairard' joint, which employs the principle of wedging cones, the steel core and the aluminium wires being gripped by separate cones. With the latter type of joint a guaranteed mechanical strength of 95 per cent. of that of the conductor is obtained, together with a conductivity equal to that of the cable.

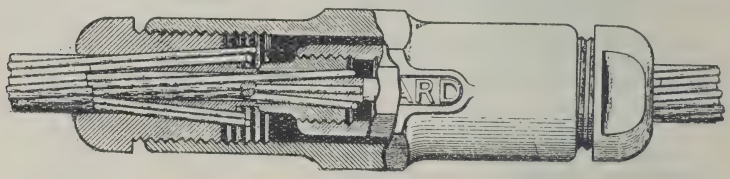


FIG. 1.

Conductivity.

The conductivity of a steel-cored aluminium conductor is taken as being that of the aluminium portion.

National Grid Transmission Lines.

The standard line conductor of the National 'Grid' Transmission System in Great Britain consists of a central core of 7 strands of galvanised steel wire, each 0·11 in. diam., surrounded by 30 strands of aluminium wire of the same diameter. The overall diameter is 0·77 in., equivalent in conductivity to a copper conductor of 0·175 sq. in. The carrying capacity of each 3-phase circuit is 50,000 kVA., and the current at 132 kV. is 219 amps. The critical corona voltage for this diameter conductor under normal atmospheric conditions is 184 kV. The earth conductor has 7 steel and 12 aluminium strands, all 0·11 in. diam. The normal span length is 900 ft. At the Firth of Forth crossing the span is 3,050 ft., and the conductors consist of 19 strands of galvanised steel each 0·125 in. diam., and 18 aluminium strands of the same diameter.

Aluminium Overhead Lines.

Span Length.

With short spans where a large cross section is required, all-aluminium conductors have a sag differing but little from that of copper, while in the case of a small cross section steel-cored aluminium may be installed with a smaller sag and lower poles. In the case of long spans steel-cored aluminium is also employed, but in this case the span length may be increased. This leads to a reduction of capital outlay and upkeep expenses owing to the reduced number of insulators.

SAG IN STILL AIR AT 120° F. OF VARIOUS CONDUCTORS ERECTED IN ACCORDANCE WITH THE BRITISH H.T. REGULATIONS.

Span Length (Ft.).		100	200	300	400	500	600	700	800	900	1000
Conductors having the conductivity of 0·05 sq. in. copper	Steel-cored Aluminium	0·25	1·31	3·8	7·62	—	—	—	—	—	—
	Copper	0·46	2·4	6·1	11·37	—	—	—	—	—	—
0·1 copper	Steel-cored Aluminium	—	—	2·72	5·18	8·46	12·55	—	—	—	—
	Copper	—	—	3·94	7·16	11·35	16·5	—	—	—	—
0·15 copper	Steel-cored Aluminium	—	—	—	—	—	7·10	9·91	13·2	17·0	21·3
	Copper	—	—	—	—	—	14·0	19·0	24·7	31·3	38·5

Loading conditions.— { 8 lbs. per ft. wind. / ⅜ in. radial thickness ice. / Factor of safety = 2. }

Hard-Drawn Copper Conductors for Overhead Lines.

BRITISH ENGINEERING STANDARDS ASSOCIATION.

(No. 125—1930.) (*Abbreviated.*)

TABLE I.—PRIMARY STANDARD SOLID CONDUCTORS.

Standard Nominal Area. Sq. in.	Calculated Area. Sq. In.	Diameter, Inches.			Weight per 1,000 Yds. Lbs.			Weight per Mile. Lbs.			Resistance per 1,000 yds. at 60° F.		Resistance per mile at 60° F.		Ultimate Tensile.		Minimum Elongation on 10 in. %
		Standard	Max.	Min.	Standard	Max.	Min.	Standard	Max.	Min.	Standard	Max.	Standard	Max.	Strength. Lbs.	Stress. Tons per Sq. In.	
·00350†	·00849487	·104	·1050	·1030	98·22	100·2	96·25	172·9	176·3	169·4	2·908	2·937	5·118	5·169	550	28·9	—
·01629	·01628602	·144	·1454	·1426	188·3	192·1	184·5	331·4	338·0	324·8	1·515	1·530	2·667	2·693	1016	27·9	—
·01697	·01697167	·147	·1485	·1455	196·2	200·2	192·3	345·4	352·3	338·5	1·454	1·468	2·559	2·584	1056	27·8	—
·02061	·02061199	·162	·1636	·1604	238·3	243·1	233·6	419·4	427·8	411·1	1·197	1·209	2·106	2·127	1266	27·4	—
·02164	·02164243	·166	·1677	·1643	250·2	255·2	245·2	440·4	449·2	431·6	1·140	1·151	2·006	2·026	1324	27·3	—
·02545	·02544690	·180	·1818	·1782	294·2	300·1	288·3	517·8	528·2	507·5	0·9689	0·9786	1·705	1·722	1540	27·0	—
·02688	·02688025	·185	·1868	·1831	310·8	317·0	304·6	547·0	557·9	536·1	0·9172	0·9264	1·614	1·630	1621	26·9	—
·03269	·03268513	·204	·2060	·2019	377·9	385·5	370·4	665·1	678·4	651·8	0·7540	0·7616	1·327	1·340	1943	26·5	1·81
·03631	·03630503	·215	·2171	·2128	419·8	428·2	411·4	738·8	753·6	724·0	0·6787	0·6855	1·195	1·206	2142	26·3	1·85
·0825	·08244796	·324	·3272	·3207	953·3	972·3	934·2	1678	1711	1644	0·2983	0·3012	0·5249	0·5302	4488	24·3	2·27
·095	·09511486	·348	·3515	·3445	1100	1122	1078	1936	1975	1897	0·2584	0·2610	0·4548	0·4594	5078	23·8	2·37
·126	·1256637	·400	·4040	·3960	1453	1482	1424	2557	2608	2506	0·1954	0·1974	0·3440	0·3474	6461	22·9	2·55

† Single conductors of a cross-sectional area of less than 0·0098 square inch are not permitted by the Electricity Commissioners under their Regulations for Overhead Lines. The conductor marked with an obelus in the above table is, therefore, to be used in the make-up of stranded conductors only.

TABLE II.—PRIMARY STANDARD STRANDED CONDUCTORS.
(*Abbreviated.*)

Standard Nominal Area. Sq. In.	Calculated Area Sq. In.	Strands and Diameter.	Approximate over-all Diameter. Ins.	Weight per 1,000 Yds. Lbs.			Weight per Mile. Lbs.			Resistance per 1,000 Yds. at 60° F.		Resistance per Mile at 60° F.		Ultimate Strength of Wire before Stranding. Lbs.
				Standard	Max.	Min.	Standard	Max.	Min.	Standard	Max.	Standard	Max.	
·025	·0249849	3/·104	·224	300·6	306·6	294·5	529·0	539·5	518·4	0·9887	0·9985	1·740	1·757	550
·075	·0748439	3/·180	·388	900·3	918·3	882·3	1585	1616	1553	0·3294	0·3327	0·5798	0·5856	1540
·100	·0999780	7/·136	·408	1196	1220	1172	2105	2147	2063	0·2469	0·2493	0·4345	0·4388	912
·15	·148951	7/·166	·498	1782	1817	1746	3136	3198	3073	0·1656	0·1672	0·2914	0·2943	1324
·175	·175134	7/·180	·540	2095	2137	2053	3687	3761	3613	0·1408	0·1422	0·2478	0·2503	1540
·225	·224951	7/·204	·612	2691	2745	2637	4736	4830	4641	0·1096	0·1107	0·1928	0·1947	1943
·25	·249864	7/·215	·645	2989	3049	2929	5260	5365	5155	0·09861	0·09960	0·1736	0·1753	2142
·4	·403569	19/·166	·830	4845	4941	4748	8526	8697	8356	0·06111	0·06173	0·1076	0·1086	1324
·5	·501239	19/·185	·925	6017	6137	5897	10590	10802	10378	0·04919	0·04968	0·08657	0·08743	1621
·75	·748094	37/·162	1·134	8989	9169	8810	15821	16138	15505	0·03297	0·03330	0·05803	0·05861	1266

TABLE III.—SECONDARY STANDARD SOLID CONDUCTORS. (*Abbreviated.*)

Standard Nominal Area. Sq. In.	Calculated Area. Sq. In.	Diameter, Inches.			Weight per 1,000 Yards. Lbs.			Weight per Mile. Lbs.			Resistance per 1,000 Yds. at 60° F.		Resistance per Mile at 60° F.		Ultimate Tensile.	
		Standard.	Max.	Min.	Standard.	Max.	Min.	Standard.	Max.	Min.	Standard.	Max.	Standard.	Max.	Strength Lbs.	Stress. Tons per Sq. In.
·003217†	·00321699	·064	·06464	·06336	37·20	37·94	36·45	65·46	66·77	64·15	7·683	7·760	13·52	13·66	213	29·5
·006648†	·00664761	·092	·09292	·09108	76·86	78·40	75·32	135·3	138·0	132·6	3·717	3·754	6·541	6·607	434	29·1
·008012†	·00801185	·101	·1020	·09998	92·63	94·49	90·78	163·0	166·3	159·8	3·083	3·114	5·427	5·481	520	29·0
·01057	·01056832	·116	·1172	·1148	122·2	124·6	119·7	215·1	219·4	210·8	2·337	2·360	4·112	4·154	676	28·6
·01208	·01207628	·124	·1252	·1228	139·6	142·4	136·8	245·7	250·7	240·8	2·045	2·065	3·598	3·634	769	28·4
·01348	·01347822	·131	·1323	·1297	155·8	159·0	152·7	274·3	279·8	268·8	1·832	1·850	3·223	3·256	852	28·2

† See note to Table I., p. 2221.

TABLE IV.—SECONDARY STANDARD STRANDED CONDUCTORS. (*Abbreviated.*)

Standard Nominal Area. Sq. In.	Calculated Area. Sq. In.	Strands and Diameter In.	Approximate over-all Diameter. In.	Weight per 1,000 Yards. Lbs.			Weight per Mile. Lbs.			Resistance per 1,000 Yds. at 60° F.		Resistance per Mile at 60° F.		Ultimate Strength of Wire before Stranding. Lbs.
				Standard	Max.	Min.	Standard	Max.	Min.	Standard	Max.	Standard	Max.	
·022	·0221405	7/·064	·192	264·8	270·1	259·5	466·1	475·4	456·8	1·116	1·128	1·965	1·984	213
·046	·0457512	7/·092	·276	547·2	558·2	536·3	963·2	982·4	943·9	0·5400	0·5454	0·9505	0·9600	434
·058	·0584647	7/·104	·312	699·3	713·3	685·3	1231	1255	1206	0·4225	0·4267	0·7436	0·7510	550
·094	·0937306	19/·080	·400	1125	1148	1103	1980	2020	1941	0·2637	0·2663	0·4640	0·4687	330
·125	·123959	19/·092	·460	1488	1518	1458	2619	2671	2567	0·1993	0·2013	0·3508	0·3543	434
·175	·174002	19/·109	·545	2089	2131	2047	3676	3750	3603	0·1419	0·1434	0·2498	0·2523	602
·20	·197069	19/·116	·580	2366	2413	2318	4164	4247	4080	0·1253	0·1266	0·2205	0·2227	676
·25	·251330	19/·131	·655	3017	3077	2957	5310	5416	5204	0·09822	0·09920	0·1729	0·1746	852

Fuses

APPROXIMATE FUSING CURRENTS OF COPPER WIRES IN FREE AIR.

Diameter of Wire.	Equivalent S.W.G. Size.	Fusing Current.	Maximum Safe Working Current.
In.		Amps.	Amps.
0·0092	34	8·6	4·3
0·01	33	9·8	4·9
0·0108	32	11·0	5·5
0·012	—	12·8	6·4
0·0124	30	13·5	6·8
0·0148	28	17	8·6
0·018	26	22	11
0·022	24	30	15
0·028	22	41	21
0·029	—	43	22
0·036	20	62	31
0·040	19	73	37
0·044	—	86	43
0·048	18	98	49
0·052	—	111	56
0·056	17	125	63
0·064	16	156	78
0·072	15	191	96
0·080	14	229	115

APPROXIMATE FUSING CURRENTS OF LEAD-TIN ALLOY (LEAD 75 PER CENT, TIN 25 PER CENT.) WIRES IN FREE AIR.

Diameter of Wire.	Equivalent S.W.G. Size.	Fusing Current.	Maximum Safe Working Current.
In.		Amps.	Amps.
0·02	25	3	2·0
0·022	24	3·5	2·3
0·024	23	4	2·6
0·028	22	5	3·3
0·032	21	6	4·1
0·036	20	7	4·8
0·048	18	10	7·0
0·064	16	16	11·0

NOTE ON FUSING CURRENTS.

The fusing currents in the tables may be taken to be correct where the fuse wire passes through an asbestos tube and does not closely touch the tube, but they do not apply where a substantial length of the wire is in contact with a porcelain holder. With the latter, more current is needed to melt the fuse.

For copper wires, the values of the current given are those necessary to fuse the wire in one minute, and are not appreciably different for other periods. For the lead-tin alloys, the fusing time is two minutes. Under normal running conditions, the actual temperature-rise at the hottest part of the fuse wire will be 100° to 150° C. for copper and 50° to 75° C. for the lead-tin alloy.

Porcelain Insulators for Overhead Lines.

British Standard Specification for Porcelain Insulators for Overhead Power Lines (3,000–150,000 Volts). No. 137—1930.*

The porcelain must be white, sound, completely vitrified, with insulation independent of the glazing, which must cover all exposed parts. The electrical design must conform to requirements in Table I., and when of more than one part the voltage drop across the assembled parts shall be proportional to their thickness. Mechanical factor of safety to be not less than 2·5. The porcelain must not engage directly with a metal screw, and must be unaffected by stresses set up by temperature changes. Every insulator shall be impressed with the manufacturer's mark before fixing, and, when conforming to this specification, bear the Association's brand. Type tests to be carried out on not less than three samples completely fitted in the following order. In tests 1 to 4, failure shall not occur below the appropriate values in Table I.

1. *Inherent Stresses.*—Insulator and glaze to withstand test voltages in columns 2, 3, or 4 of Table I. (applied as for Test 2) after being heated in water at 50° C. for one hour and immediately immersed in ice and water for one hour, the test being repeated three times.

2. *Dry Spark-over Tests.*—Testing pressure to be applied between the support and a wire of 0·4 inch diameter attached to insulator. Test to begin at one-third full voltage, and increased at 1 k.v. per second until spark-over occurs.

3. *Rain Tests.*—Insulator to be tested as in Test 2, but subjected to artificial rain at 15° C. of 1 inch in 5 minutes directed downwards at 45° from vertical. Specific resistance of the water to be not more than 20,000 ohms per cm. cube.

4. *Puncture Test.*—Insulator to be tested as in Test 2 while immersed in oil.

Table I.—Test Voltages. (R.M.S. Values.)

1	2	3	4	5	6	7	8	9	10
	Dry Spark-over Test.			Rain Test.			Puncture Test.		
A. C. Delivered or Declared Voltage.	Supporting Insulator.		Tensioning Insulator	Supporting Insulator.		Tensioning Insulator	Supporting Insulator.		Tensioning Insulator
	Pin Type.	Suspension Type.		Pin Type.	Suspension Type.		Pin Type.	Suspension Type.	
Kilovolts.	Kilovolts.	Kilovolts.	Kilovolts.	Kilovolts.	Kilovolts.	Kilovolts.	Kilovolts.	Kilovolts.	Kilovolts.
3	40	—	30	20	—	10	70	—	52
6	50	—	40	30	—	20	87	—	70
10	62	—	54	39	—	30	108	—	95
†20	95	90	90	62	57	57	166	158	158
30	125	125	125	84	84	84	220	220	220
†40	145	150	150	105	105	105	250	260	260
†50	158	175	175	125	125	125	275	310	310
60	165	200	200	140	142	142	290	350	350
†80	—	250	250	—	180	180	—	440	440
100	—	300	300	—	220	220	—	525	525
120	—	335	335	—	260	260	—	585	585
150	—	365	365	—	315	315	—	610	610

Note I.—The term 'Delivered Voltage' denotes the normal voltage between conductors at the delivery end.

For the purposes of this Specification the Working Voltage is assumed not to exceed the Delivered Voltage by more than 10 per cent.

Note II.—Test voltages for special insulators required for intermediate voltages may be ascertained from a curve prepared from the above figures.

Note III.—In cases where testing apparatus is not available for puncture tests at pressures exceeding 310 kV., arrangements may be made to test the insulator units separately and to estimate the equivalent puncture test voltage of the complete insulator from these individual tests.

* By permission of the British Engineering Standards Association.

† These four voltages are not included in B.S. Specification No. 77—1925, Voltages for New Systems and Installations.

5. *Mechanical Test.*—Complete insulator with metal fittings must withstand loads in accordance with Table II. For a pin insulator the tension is applied by a flexible wire rope placed round neck, and for tensioning and suspension types the load is applied between metal connecting parts on either side. Any permanent deformation constitutes failure.

6. *Porosity Test.*—The finished porcelain shall show no signs of impregnation when broken pieces free from external glaze are immersed for twenty-four hours in a 0·5 per cent. alcoholic solution of fuchsin under a pressure of 2,000 lbs. per sq. in.

7. *Routine Tests.*—Every complete pin insulator shall be inverted in water, and withstand, without puncture, for five minutes the spark-over voltage between testing terminal and wire attached to neck groove. Every unit of a tension or suspension insulator must withstand for five minutes the spark-over voltage of that unit without puncture, and the complete insulator a load 20 per cent. above maximum working load applied vertically for one minute.

Appendices to specification give experimental methods of carrying out mechanical and electrical tests.

TABLE II.

Maximum Working Load.	Test Loads.					
	Supporting Insulators.				Tensioning Insulators.	
	Pin Type.		Suspension Type.			
Lbs.	Approx. Kgs.	Lbs.	Approx. Kgs.	Lbs.	Approx. Kgs.	Lbs.	Approx. Kgs.
Up to 400	182	1,000	454	1,000	454	1,000	454
,, 800	363	2,000	907	2,000	907	2,000	907
,, 1,400	635	—	—	3,500	1,588	3,500	1,588
,, 2,800	1,270	—	—	7,000	3,175	7,000	3,175

Electricity Supply.

The standard voltages of supply laid down by the British Engineering Standards Association (Report No. 77—1925) for new systems and installations are as follows (at consumers' terminals):—

Direct current, 230 and 460 volts; alternating current, 230 volts between neutral wire and each of the principal conductors, 400 volts between any two of the principal conductors.

The standard high voltages for new systems and installations are: 3,000, 6,000, 10,000, 30,000, 60,000, 100,000, and 120,000 volts.

In each case the approximate station voltage is assumed to be 10 per cent. higher at full load.

Periods allowed by the Electricity Commissioners for Repayment of Loans.

Land—	Years.		Years.
Freehold	60*a*	Mains and services	25
Leasehold (subject to duration of lease)	30	Meters	10
Substantial buildings	30	Wiring of premises	10
Plant and machinery	20	Apparatus to be let on hire	7–10
H.P. trunk transmission mains	40*a*		

(*a*) Except in Scottish cases, for which the present maximum period is 30 years.

Generation of Electricity in Great Britain.*

The total number of units sold by undertakers in Great Britain during the year ending March 31, 1928, was 9,927,850,630, an increase of 18·7 per cent. on the figures for the previous year. The total number of stations (from which returns were received) was 570. The total coal, coke, and fuel oil consumed was 9,224,151 tons, or an increase of 4·1 per cent.

The steam stations generated 95·70 per cent. of the total; gas-producer stations, 0·19 per cent.; oil-engine stations, 0·63 per cent.; waste heat, 2·19 per cent.; refuse destructors, 0·21 per cent.; gas engines using town's gas, 0·03 per cent.; water power, 0·05 per cent. In 1927 the steam percentage was slightly more, viz. 97·16 per cent., water power less, viz. 0·59 per cent.

No figures are available for the consumption of electricity generated by privately owned stations, but figures published in 1924 show 2,564,000 kW. of such plant installed as compared with 3,943,000 kW. of plant belonging to undertakings supplying mainly to the public. Regional schemes prepared by the Electricity Commissioners to date show that 256 undertakers are affected, 2,883,870 kW. of plant installed or sanctioned (4,668,295 kW. estimated by 1940–41), 1,108 route miles of primary transmission line, and 3,725,000 kVA. of associated transformer plant are involved, with a total capital expenditure, to the end of 1934–35, by the Central Board and owners of selected stations of 27,425,121*l.*, including 4,982,000*l.* on frequency standardisation. During 1927–28, the extension of generating plant sanctioned by the Commissioners amounted to 668,357 kW.

* Eighth Annual Report of the Electricity Commissioners, 1927–28. H.M. Stationery Office, Adastral House, Kingsway, London, W.C. 2. Price 3*s.* net.

In the First Annual Report of the Central Electricity Board * an account is given of the procedure of the Board under the Electricity Act of 1926, of the schemes considered and adopted 1927–28, and a full technical description of the design, construction, and operation of the Grid System.

Ministry of Transport Regulations in reference to Overhead Electric Lines.

Every application for the consent of the Electricity Commissioners to the placing of electric lines above ground should be accompanied by the following particulars :—

1. Where the undertakers are a company, or a local authority supplying outside their own area, evidence of consent of the local authority for the district.

2. A statement showing commercial or other considerations why underground cables should not be used.

3. A brief description of the proposed system, whether by continuous or by alternating current ; the working voltage ; the kind of wire, whether copper or aluminium ; whether solid or stranded ; the total sectional area ; tensile strength and elongation ; average and maximum length of span ; minimum height of wires from the ground ; name or description of automatic protective device, if any.

4. A statement whether the supply is to form (1) an extension of an existing system of underground cables, or (2) of an existing traction system, or (3) an independent system.

5. An ordnance map on a scale of 6 ins. to the mile, showing the proposed route of the overhead lines and any existing overhead lines. The sheets of these maps must be fastened together.

6. In the case of high and extra high pressure, plans of construction of poles, etc., on a scale of about 1 in. to the ft., or a reference to previously deposited plans where these are identical with the proposed work.

Regulations of the Electricity Commissioners. (El.C. 38—1926.)

Regulations for Securing the Safety of the Public.

1. Supply pressure shall not exceed the low-pressure limit of 250 volts, except for power or arc lamps in series, when the medium pressure limit of 650 volts is permissible under the following conditions. Machine frames must be earthed, motor leads enclosed in earthed metal casing with enclosed fireproof and shockproof switches and fuses, switch when opened must completely cut off motor and all auxiliary devices. Similar conditions apply to arc lamps and incandescent lamps in series. Regulations 1 and 2 do not apply to a factory, workshop, mine, or quarry.

2. Where the voltage between the outer conductors of a 3-wire system exceeds 250 volts and is taken into premises, two pairs of terminals shall be provided and the wiring from these kept distinct.

3. The Commissioners reserve their consent to the supply of extra high pressure to consumers' premises.

4. Minimum area of conductor installed in street line to be 7/20 gauge except for lamp-post wiring.

5 Supply mains to be tested at at least 200 volts, and results of tests to be recorded before the supply is begun.

6. Insulation of every complete circuit used for supply of energy to be maintained so that leakage current never exceeds $\frac{1}{1000}$ part of maximum supply current. Measurements must be made weekly, records kept, leakage remedied without delay, and means provided for indication and localisation of leakage.

7. High-pressure circuits (650 to 3,000 volts) must not be brought into use until successfully tested for ½ hour at twice the maximum working pressure in the case of lines and 50 per cent. excess of normal for machines and apparatus. Results to be recorded.

8. Every high-pressure main, etc., to be fitted with fuse or automatic circuit breaker, except the outer earth conductor of a concentric main.

9. Automatic quick-acting device must be fitted on transformer circuits to prevent contact or leakage from high-pressure circuit to consumers' wires.

10. All metallic portions (except conductors) of high-pressure transformers to be earthed.

11. Exposed lines and their supports to be protected from injury by lightning.

12. The undertakers must inform Commissioners of all accidents.

Overhead Lines.

13. Overhead lines erected after this date (1926) to be in accordance with Commissioners' Regulations.

Electric Lines other than Overhead.

14. Conduits, street boxes, etc., to be constructed of durable material, and where laid underground to be unaffected by heavy traffic. Accumulation of gas must be prevented.

15. Precautions must be taken to prevent the charging of any metallic substance directly or indirectly from any electric line.

* Whitehead, Morris, Ltd., London. Price 2*s.* 6*d.* net.

16. All high-pressure conductors carrying conduits to be jointed, connected across street boxes, and efficiently earthed.

17. Bare conductors in conduit must not exceed 300 volts to earth, and must be secured in position. Such conduits must be moisture proof and insulators so placed that they can be inspected.

18. High-pressure lines on ground surface or public subway must either be enclosed in insulating tube embedded in brickwork, masonry or concrete, or in an earthed metal casing.

19. Means must be taken to prevent the ground surface or other conductors becoming charged by leakage from a high-pressure line.

20. A high-pressure line must be properly laid, jointed, examined, and tested and in the sole charge of the undertakers before being used for supply of energy.

Sub-stations and Street Boxes.

21. To be in sole occupation and charge of the undertakers and placed above ground wherever possible; if underground, provision must be made for ventilation and drainage.

22. In addition to regulation 14, (*a*) covers of street boxes only to be opened by special appliance; (*b*) covers of street boxes containing high-pressure apparatus other than cables must be in connection with metal strips laid immediately underneath the street, and means taken to prevent all exposed material, including the street surface, becoming charged by any means; (*c*) street boxes used as transformer chambers must not be subject to influx of water, and when of greater size than one cubic yard must be adequately ventilated; (*d*) street boxes must be regularly inspected for the presence of gas, and notice of gas influx must be immediately given to the gas authority concerned; (*e*) mains at different pressures in the same street box must be readily distinguishable.

23. Maximum power supplied to underground sub-station or street box containing one transformer must not exceed 30 kW., or 75 kW., if with two or more transformers, without consent of Commissioners.

Consumers' Premises.

24. The undertakers shall be responsible for maintenance of their property on consumers' premises.

25. The undertakers must guard against fire risks in delivering energy to premises.

26. Fuse or automatic circuit breaker in fireproof, sealed receptacle to be fitted near entry of each service main if latter is unprotected by fuse in the street box. No fuse or circuit breaker must be inserted in intermediate conductor of a 3-wire system.

27. Service lines and apparatus on consumers' premises must be highly insulated, moisture-proof and all exposed bare metal forming part of a circuit, efficiently earthed. Lines must be fixed and protected to prevent electrical discharge to nearby metal.

28. Consumers' wires must not be connected to undertaker's mains unless the resulting leakage is not in excess of one ten-thousandth of maximum current supplied to the premises. Notice of objection must be served upon the consumer.

29. Undertakers have the right, after giving notice to consumer, to test wires and fittings for leakage and to discontinue supply if excessive leakage is found (see 28), or facilities for testing and inspection are refused.

30. Consumer may appeal for an independent test to an electrical inspector on payment of prescribed fee. This regulation must be endorsed on every notice served under regulations 28 and 29.

Arc Lighting.

31 and 32 respectively fix the minimum height of any part of an arc lamp 10 feet above ground, and provide for the provision of arc lamp guards, forbidding the use of arc lamps where explosive dust or gas may be present.

Connection of Circuits with Earth.

33. Where supply pressure between adjacent conductors of a 3-wire system of mains exceeds 125 volts, the intermediate conductor for each distinct circuit shall be earthed at one point only, namely, at generating station, sub-station or transformer, the insulation being maintained elsewhere. The earth current from intermediate conductor must be continually recorded and must not exceed at any time one-thousandth of maximum supply current.

34. Penalties are provided for default of regulations 1 to 34.

Regulations for Ensuring a Proper and Sufficient Supply of Electrical Energy.

1. Supply must be constantly maintained, and in case of D.C. without change of polarity. Notice of any interruption for testing purposes or polarity change must be given to consumer.

2. Distribution mains must be sectionalised on their respective feeders and connected only through circuit breakers or fuses.

3. Variation from declared pressure of supply shall not exceed $\pm$ 4 per cent. Public notice with consent of Commissioners is required for any alteration of conditions.

4. The system and frequency of supply (in A.C. systems) must be declared. Variation of frequency from declared value must not exceed $\pm$ 2½ per cent. Notice and consent required for alteration (see 3).

5. Penalties are prescribed for default of regulations 1 to 5.

Regulations of Electricity Commissioners for Overhead Lines. (El.C. 53—1928.)

General Regulations.

1. Conductors to be of copper, aluminium, or other approved material.

2. Conductors to conform to B.E.S.A. specification for elongation, breaking load, and elasticity.

3. Minimum permissible size for copper and other line conductors (other than service lines) must have an actual breaking load of not less than 1,237 lbs. Minimum area and weight per mile for copper, No. 8 S.W.G., 0·0201 sq. ins., 409 lbs. For service lines, minimum breaking load 816 lbs.; minimum area and weight, No. 10 S.W.G., 0·0129 sq. ins., 262 lbs. per mile.

4. Conductors must be rendered inaccessible to persons not having the use of appliances. Regard must be paid to the normal use of his land or premises by the occupier. See 18.

5. Except in the case of contact between broken conductors and earth wire on the same support, precautions must be taken by the undertakers against contact between line conductor and any other overhead wire.

6. Line insulators to be carried on supports of wood, iron, steel, or reinforced concrete. Corrosion of metal work on or below ground surface to be prevented. Wooden supports must be of red fir, except oak or hardwood cross arms, unless otherwise approved.

7. Supports, in conjunction with any struts or stays, shall take all loads without damage or movement in the ground. Strength in direction of line must not be less than one quarter of transverse strength. Factor of safety for iron and steel 2·5, for wood and reinforced concrete 3·5. These factors to be calculated for 22° F. with ice covering specified in 12 or 15, and with a wind pressure of 8 lbs. per sq. ft. on whole of the projected surface, corresponding to a velocity of 50 m.p.h. at right angles to the line. Wind pressure on lee side members of lattice or other compound structures, A and H poles, to be taken as one-half that on windward side, and factor of safety calculated on crippling load of struts and upon elastic limit of tension members.

8. Service lines to be connected only at a point of support, and fixed to insulators on consumer's premises. Service lines (other than an earthed neutral conductor) which are accessible from a building must be protected by insulating material or otherwise.

Service lines over carriage ways must not be lower than 20 ft. from the ground except with consent of Commissioners.

9. Adequate protection to be made against danger to linesmen, and from a lower voltage system becoming charged above normal from a higher voltage system when conductors of the different systems are carried on same poles or supports.

10. The whole structure with all appliances must be regularly inspected and maintained.

11. Materials must conform at time of erection to B.E.S.A. specifications and to the Post Office Technical Instructions of Aerial Lines.

Specific Regulations.

A. For pressures not exceeding 650 volts D.C. and 325 volts A.C.

12. Factor of safety of line conductors, 2, based on breaking load at 22° F., with $\frac{3}{16}$ in. radial thickness of ice and simultaneous wind pressure as in 7.

13. Minimum height of any wire (except service) at 122° F., not to be less than 19 ft. across a public road or 17 ft. in other positions. In places inaccessible to vehicular traffic, 15 ft. Minimum height of a service line across or along a carriage-way, except with permission, 19 and 17 ft. respectively.

14. Where pressure to earth exceeds 250 volts D.C. or 125 volts A.C., precautions shall be taken to prevent danger

(*a*) from a broken conductor; by a neutral or earthed conductor suitably disposed or other approved method:

(*b*) from leakage; in cases of metal poles, by provision of earthed wire connecting the poles, or by supporting the insulators on insulated metallic frame connected to neutral conductor. With wooden poles by a bonding wire connected to supporting metal of all insulators and terminating at lowest part of supporting metal work. Other means may be approved.

Where lighting or uninsulated conductors are run down wooden poles to within 10 ft. of the ground, precautions against leakage must be as for metallic poles. Unearthed stay wires must be insulated by insulator placed not less than 10 ft. above ground.

B. For pressures exceeding 650 volts D.C. and 325 volts A.C.

15. Factor of safety of conductors as in 12, except that for $\frac{3}{16}$ in. ice, read $\frac{3}{8}$ in.

16. Minimum heights of conductors above ground at temperatures below 122° F. are 20 ft. for pressures not exceeding 66,000 volts, 21 ft. exceeding 66,000 and below 110,000 volts, 22 ft. 110,000 to 165,000 volts, and 23 ft. above 165,000 volts. Height from the ground of any earth wire must not be less than the minimum prescribed in Regulation 13.

17. Means must be taken to render broken conductors dead. All other metal work to be earthed by continuous earth wire earthed at four equidistant points per mile; alternatively metal shall be effectively earthed at each support. When earth contact is made by a line conductor, leakage current must be at least twice that required to operate safety devices.

18. At road, canal, or railway crossings, in addition to 16, there shall be provided (*a*) duplicate line insulators with automatic earthing of a fallen conductor or (*b*) duplicate insulators tied at intervals not exceeding 5 ft.

Where line is erected along or within 50 ft. of road, canal, or railway, (*a*) duplicate insulators or (*b*) automatic earthing for fallen conductor shall be provided. Other means may be approved.

19. Supports must be consecutively numbered, carry permanent danger notice, and guarded from unauthorised climbing.

An explanatory memorandum El.C. 53A is issued.

Regulations Relating to Extra High Pressure. (El.C. 13—1920.)

The following *additional* regulations relating to *extra high pressure* have been issued by the Commissioners.

A. *Regulations for Securing the Safety of the Public.*

3. An extra-high-pressure main before being brought into use shall be tested in position to withstand for half an hour continuously a pressure of 10,000 volts in excess of the working voltage, when this exceeds 10,000 volts, or twice the working pressure when this is less than 10,000 volts. Results of tests must be recorded.

4. Suitable fuse or circuit breaker must be provided for every main, but not in earthed outer conductor of concentric mains.

8. Where leakage from 3-phase mains is prevented by a copper strip under the lead sheath of cable, this shall not be less that 0·016 in. thick, and where steel wires are thus employed outside a lead sheath, each shall not be less than 0·1 in. in diameter. A lead sheath must be efficiently earthed and not less than 0·1 in. thick.

9. Single-phase supply mains to consist of two concentric conductors or separate conductors. Outers of concentrics to be earthed at one point: with separate conductors, leakage to be prevented as with 3-phase mains.

Street Mains.

11. These mains must be enclosed in strong metal casing where passing through street box with other mains; boxes must not contain any other undertaker's electric mains, nor water or gas pipes. Telephone wires belonging to the undertakers are allowed.

Sub-stations.

12. Must be in sole or joint occupation of the undertakers or authorised distributor.

13. When constructed below streets, must contain no switches or apparatus other than transformers.

14. The transforming apparatus must be arranged to avoid danger of excessive voltage on connected mains.

15. Precautions must be taken against fire risks.

17. Gives right of entry and examination to the Commissioners.

Connection of Circuits with Earth.

18. Earth connection to be made at only one point on a circuit, viz. generating station, sub-station, or transformer. Insulation must be maintained elsewhere.

19. When neutral star point of each distinct 3-phase circuit is earthed through a resistance, this must be low enough to ensure the action of the cut-out in the mains.

Separate electrostatic voltmeters must be connected in the generating station between each distinct circuit and earth to act as leakage indicators. Faulty circuit insulation must be immediately restored.

20. Penalties are prescribed against default which are not to affect liability of undertakers in respect of damages caused by default.

B. *Regulations for Ensuring a Proper and Sufficient Supply of Electrical Energy.*

1. Undertakers must maintain sufficient and constant supply from the main to authorised distributors and consumers.

2. Maximum load on a main not to exceed 1,000 kW. without consent of Commissioners, unless provision is made for alternative supply in the event of breakdown.

3. Minimum pressure to be declared. Pressure at authorised distributors' terminals must not be less than this minimum, and not exceed it by more than 12½ per cent. Alteration of minimum to be allowed only by consent of Commissioners after one month's public notice of such application.

4. Frequency to be declared 50 or 25 cycles per second; 2½ per cent. variation allowed. Consent and notice of alteration as in 3.

5. Penalties are prescribed for default.

These regulations are in addition to any regulations relating to mines and factories made by any Act, and also include regulations 9, 16, 18, 20, and 27 of El.C. 38, summarised above.

The Electrical Equipment of Buildings.

REGULATIONS ISSUED BY THE INSTITUTION OF ELECTRICAL ENGINEERS.

(Ninth Edition, 1927.)

(*Abbreviated.*)

SUPPLY PRESSURE.

These regulations provide for wiring based on the following pressures of supply :—

Medium pressure, i.e. above 250 volts and below 650 volts.
Low pressure, i.e. above 30 volts A.C. or 100 volts D.C., but not exceeding 250 volts.
Extra low pressure, i.e. below 30 volts A.C. or 100 volts D.C.

GENERATING PLANT.

Generators must comply with the following :—

(*a*) They must be placed in well-ventilated rooms where inflammable dust or gases cannot accumulate.

(*b*) They must be protected from damp, dust, or mechanical injury.

(*c*) No unprotected woodwork or other combustible material shall be within a distance of 12 ins. from them measured horizontally, or within 4 ft. measured vertically above them.

(*d*) When the supply is at medium pressure, the generating plant shall have its frame and bedplate earthed.

SECONDARY BATTERIES.

(*a*) A p.d. exceeding 50 volts must not exist between adjacent cells without adequate protection. Each cell shall be readily accessible from the top and at least from one side.

(*b*) In a lead-sulphuric acid battery with more than 33 cells, and in a nickel-iron alkaline battery with more than 53 cells, the cells shall be supported on glass or vitreous porcelain insulators. The stands shall also be insulated if there are more than 56 and 88 cells respectively.

(*c*) When acid is used as an electrolyte for the cells, the battery-connecting bolts, unless of non-corrosive type, shall be kept covered with petroleum jelly.

(*d*) Cells having containers not sealed nor provided with screw-down covers shall have spray arrestors.

(*e*) Celluloid shall not be employed for non-portable batteries.

(*f*) The battery-room shall be well ventilated.

(*g*) As a minimum, the gear controlling the current shall comprise an automatic cut-in and cut-out switch and fusible cut-out, or a circuit breaker with overload and reverse-current trips.

SWITCHBOARDS.

(*a*) Must be placed in dry and well-ventilated situations away from explosive dust or acid fumes.

(*b*) Must be constructed wholly of durable, non-ignitable, non-absorbent materials.

(*c*) If marble or slate be used, all conducting parts shall be insulated with mica or other non-hygroscopic substance.

(*d*) Terminals must be provided for earth connection.

(*e*) Live parts must be spaced or shielded so that an arc cannot be maintained.

(*f*) All connections shall be readily traced.

(*g*) All parts shall be readily accessible, and no fuse shall be fixed at the back of the board.

(*h*) All nuts at back of board must be effectively locked.

(*i*) All circuits, instruments, etc., shall be clearly labelled.

(*j*) Connecting wires to every voltmeter and earth lamp must be protected by a fuse on each insulated pole.

(*k*) The moving parts of switch, in the 'off' position, must be disconnected from the supply.

(*l*) With fuses and switches on same pole, the fuses must be disconnected from supply when their respective switches are in the 'off' position.

MAIN SWITCHGEAR.

Every main switchboard must be fitted with the following switchgear as a minimum :—

1. *Two-wire Systems of Wiring.*

(*a*) When supply is obtained from one generator or a single service main, for these and for each outgoing circuit either a double pole overload circuit breaker or a linked switch with a fuse on each pole. (For earthed system, single pole on the insulated main.)

(*b*) With several generators not running in parallel or with duplicate service mains, change-over switches must be used, in addition, on each outgoing circuit.

(*c*) With generators in parallel, a double-pole circuit breaker (single-pole for earthed system) fitted with overload and reverse-current trips. For compound-wound machines an additional single-pole equaliser switch interlocked so that equaliser must be closed before and opened after the main circuit breaker.

2. Three-wire Systems of Wiring.

Each generator or each side of service mains and each two- or three-wire outgoing circuit double-pole apparatus as for two-wire system. No fuse or unlinked switch on neutral. When neutral is known to be directly earthed, a two-conductor earthed circuit may have single-pole overload circuit breaker or switch and fuse in outer conductor.

3. Two-phase Three-wire Systems of Wiring.

Each generator or service main and each outgoing three-conductor circuit a triple-pole circuit breaker with overload trips on each phase or a triple-pole linked switch with a fuse on each conductor except the neutral. Each outgoing two-wire circuit, double-pole circuit breaker with overload trip on at least one conductor or double-pole switch and fuse.

4. Three-phase Three-wire Systems of Wiring.

For each generator or service main and each outgoing three-conductor circuit, either a triple-pole circuit breaker with overload trips on at least two phases, or a triple-pole linked switch with a fuse on each conductor. For each two-wire outgoing circuit these to be double-pole.

5. Two-phase Four-wire Systems of Wiring.

Each generator or service main and each outgoing four-conductor circuit, either a four-pole circuit breaker with overload trip coil on each phase or a four-pole linked switch with a fuse on each conductor. For each two-wire outgoing circuit double-pole circuit breaker with overload trip on at least one conductor or double-pole switch and fuse.

6. Three-phase Four-wire Systems of Wiring.

For each generator or service main, a triple-pole circuit breaker with overload trips on each phase (at least on two phases for outgoing three-conductor circuit), or a triple-pole linked switch with a fuse on each phase. For each outgoing two-wire circuit supplied from one phase and directly earthed neutral, single-pole gear as above connected in the phase conductor. (Double-pole with unearthed neutral.) For four-wire outgoing circuits supplying a distribution board, triple-pole gear as above, but no fuse or unlinked switch in neutral conductor, except for testing.

INSTRUMENTS.

Every main switchboard not supplied from an external source must have the following minimum :—

Two-wire Systems of Wiring.

For a single generator an ammeter and voltmeter. For generators not to be run in parallel, one ammeter for each machine and a voltmeter to be fitted with multiple-way switch or plug. For parallel running, an ammeter for each machine and two voltmeters, one being connected to bus-bars and the other arranged for use on any generator. Alternators to have synchronising device. The ammeter in the circuit of compound wound machines to be connected in opposite pole to equaliser connection.

Three-wire Systems of Wiring.

An additional voltmeter to be connected between neutral and each outer bus-bar and a central-zero ammeter for D.C. or central-zero wattmeter for A.C. in main neutral conductor.

Two-phase and Three-phase Systems of Wiring.

For a single generator an ammeter on each phase and one voltmeter. For each generator running in parallel, an ammeter in each phase; one voltmeter on bus-bar and one on multiple-way switch, for use on one phase of any generator. A synchronising device.

Instruments should conform to British Standard Specification No. 89–1929.

Earth Testing.

Except for extra low pressure plant of less than 5 KW. capacity, the switchboards of private installations must have suitable means for indication insulation.

Switches and Circuit Breakers.

(*a*) Must be able to carry working current continuously with a temperature rise above the air of not more than 36° F. (20° C.), if rated at less than 100 amperes, and 54° F. (30° C.), for ratings of 100 amperes or more.

(*b*) Must not arc over when breaking 50 per cent. excess current at a pressure 50 per cent. above that of the supply. Fuse switches and circuit breakers must fulfil regulations for fusible cut-outs. (See British Standard Specifications 109—1923 and 110—1923.)

(*c*) Accidental closing impossible.

(*d*) Provision for renewal of main arcing contacts.

(*e*) Handles must be robust and must not be operated through unprotected slots. Operator must be protected from live metal or any injury from adjacent apparatus.

(*f*) Bases to be of durable, fireproof, non-absorbent insulating material. Slates to be planed all over, free from metallic veins, and if of slate must be treated (including holes) with damp-proof medium after drying. All bolts fixing slate or marble slabs must be insulated from slab, and the latter similarly from metallic frame by non-hygroscopic insulating bushes and washers.

(*g*) The live parts of switches not mounted on a switchboard must be enclosed by metal or fireproof insulating covers, guarded from mechanical injury, and giving proper clearance from live parts and protection against arcing.

(*h*) When placed out of doors or in damp places, must have weather-proof cases and proper provision for leading-in cables. Suitable glands are given in British Standard Specification No. 94—1920.

(*j*) Operating current must be adjustable on electromagnetic circuit breakers. and latter must be incapable of being held in against this current.

(*k*) Arrangements and position must avoid all risk of fire.

Fusible Cut-outs.

A. *For Extra Low Pressure.*

(*a*) Temperature rise of any part of the fitting above air not to be more than 54° F. (30° C.), with normal current continuously.

(*b*) Fusing current to be double the current capacity of the smallest cable controlled by the fuse when fusing time is one minute (two minutes for a lead-tin alloy fuse). No fuse to blow at less than 7 amperes need be inserted in final sub-circuits.

(*c*) The base to be of durable, fireproof, non-absorbent insulating material, and all construction must avoid arcing and scattering of metal.

(*d*) No fuse to be placed in ceiling roses, wall plugs, or sockets, or in switches other than fuse-switches of regulation patterns with metal covers. A fuse rated at not more than 5 amperes may be used in projecting wall socket adapters taking the pins of a smaller plug.

(*e*) Cut-outs not fixed on switchboard must be grouped on distribution boards or contained in regulation cases unless completely enclosed.

(*f*) Removable carriers to be used except where the covers provide against scattering of metal, and the maximum power in the building does not exceed 1,000 watts.

B. *Low-pressure Cut-outs must in addition :*

(*a*) Be provided with safe insulating carrier for the fuse, with convenient contacts for the wire.

(*b*) Base contact to be unaffected by vibration.

(*c*) Risk of contact with any live metal to be avoided when handling a carrier.

(*d*) Cut-outs and carriers to be graded ' Ordinary Duty ' and ' Heavy Duty.'

Maximum working current.	Maximum short-circuit current for ' Ordinary Duty ' cut-outs.
10 amperes	1,000 amperes
30 ,,	2,000 ,,
60 ,,	4,000 ,,
100 ,,	6,500 ,,

' Heavy Duty ' cut-outs to be used where ' Ordinary Duty ' cut-outs are inadmissible.

C. *Medium-pressure cut-outs must fulfil conditions* A (*a*) *to* (*f*) *and* B (*a*) *to* (*e*).

(See British Standard Specification No. 88—1919 for Low-pressure Electric Cut-outs.)

Section and Distribution Boards.

A. Must generally comply with all regulations for switchboard, circuit breakers, and fusible cut-outs, except for final sub-circuits carrying less than 20 amperes.

B. Unless current-carrying parts of all accessories are permanently protected, they must have a protecting case, which with earthed concentric wiring systems must provide, if of metal, for attachment to case of all external conductors; if of wood, an interposing metal sheet between it and the support not less than $\frac{1}{16}$ in. thick, to which all external conductors are connected.

C. Arcing under short-circuit conditions must be impossible.

D. Cases of soft wood to have fire-proof insulating lining, and all boxes not provided with backs shall be so shielded from the supports. A glass front must have 1 in. (2·5 cm.) clearance from live parts.

E. When placed out of doors or in damp places, must be weatherproof with proper provision for leading-in.

F. Dimensions must allow of easy access for connecting.

G. Cut-outs must be grouped vertically with fire-proof insulating material between opposite poles.

Conductors of Cables.

(*a*) All conductors, other than the outer conductors of earthed concentric systems, shall be of annealed copper.

(*b*) The sizes of conductors set out in the table on p. 2206 are recognised as standard. The standard sizes and resistances of conductors for flexible cables and flexible cords are set out in tables on pp. 2207 and 2208.

(*c*) No cable having a conductor of nominal sectional area less than 0·0015 sq. in. (1/·044 in.) shall be used except for wiring fittings.

(*d*) All conductors having a nominal sectional area exceeding 0·003 sq. in. (1/·064 in.) shall be stranded.

(*e*) For lighting, the drop from the consumer's terminals to any and every point in the installation shall not exceed 1 volt plus 3 per cent. of the pressure at the consumer's terminals when the conductors are carrying the maximum demand under the practical conditions of service. The current shall not exceed that in tables on pp. 2209–2210. For flexible cables the permissible currents are given in tables on pp. 2208 and 2211.

Insulation and Protective Covering of Conductors.

The following types have been approved :—

A. *Vulcanised Rubber-insulated Cables*—(*a*) insulated with a layer of pure rubber, an intermediate layer of vulcanising rubber, and an outer jacket of vulcanising rubber and compounded or, alternatively, a homogeneous dielectric of vulcanised rubber free from corrosion of tinning—if the latter is used as protective covering it must conform to the requirements of the tough rubber compound (see p. 2244); (*b*) taped as (*a*) and braided; (*c*) sheathed with a closely fitting seamless covering of commercially pure lead, having a smooth exterior surface and of uniform radial thickness; (*d*) lead-covered as above and bedded and armoured, with or without serving or braiding over the armour; (*e*) bedded and armoured, with or without serving or braiding over the armour; (*f*) encased in a closely fitting covering of brass, copper, or equally hard incorrodible metal; (*g*) sheathed with a closely fitting seamless covering containing not less than 95 per cent. of commercially pure lead; (*h*) covered with tough rubber compound.

B. *Paper-insulated Cables*: as (*c*), (*d*), (*f*), and (*g*).

C. Of the types in A, the following shall not be used for a.c. except in connection with an earthed concentric system in which the sheathing forms one conductor; single cables armoured with steel wire or tape or encased in a ferrous sheath, or single, unarmoured, lead-covered cables having a conductor of area exceeding 0·25 sq. in.; or brass, copper or equally hard incorrodible metal with conductor above 0·1 sq. in.

Twin Flexible Cords.

(*a*) Twin twisted flexible cords shall be used only for fixed wiring, fixed fittings and portable lamp standards; in all other positions and for all other purposes, flexible cords made up to a circular or oval section shall be employed—the use of circular or oval section flexible cords is recommended for all portable fittings.

(*b*) The maximum weight carried by a twin-twisted flexible cord shall be as follows :—

Number and diameter of wires in conductor.	Maximum weight.
14/·0076 ins.	3 lbs.
23/·0076 ,,	5 ,,
40/·0076 ,,	10 ,,

Installing and Fixing of Conductors and Cables.

Bare conductors may be used as collectors on trolley wires for cranes, etc., and for battery connections, supported on insulators and spaced so that risk of accidental contact is reduced to a minimum. At the ends of each conductor, straining gear with double insulation to be used. The supply circuit must be protected as for insulated conductors. Lightning arresters on each pole or phase to be used where necessary. Wall supports for span wires must not be within 12 ins. of any gas pipe. Except as above, bare conductors shall only be used in positions not ordinarily accessible to unauthorised persons and under conditions sanctioned by fire offices.

Taped and Braided Cables (Class L) may be used without conduit or casing provided that they are:—

(i) Open to view throughout their length and not installed under floors or buried in plaster.
(ii) Kept away from structural metal work.
(iii) They are protected where within 6 ft. above the floor or when liable to mechanical damage.
(iv) Kept away from gas and water pipes.
(v) Secured by porcelain cleats, clips, saddles, or clamps to prevent the cables coming into contact.
(vi) In damp situations, not less than ½ in. away from wall.
(vii) Protected by non-absorbent, non-ignitable conduits when passing through floors, walls, or ceilings.
(viii) When passing through party walls or fireproof floors, the conduits to be close-fitting and holes plugged with non-ignitable material.

Metal-sheathed Cables (Class M1) may be used without conduit or casing provided they are in accordance with condition (iv) above and
(ix) Secured by clips, saddles, or clamps to prevent appreciable sagging or damage to cable.
(x) Vertical cables are fixed in the same way as horizontal. When inaccessible, 10 ft. allowed between supports.
(xi) All metallic envelopes of cables to be effectively earthed and made electrically continuous throughout by means of soldered joints or bonding clamps or joint boxes.
(xii) The resistance of the metallic envelope in a complete installation, between the main switch and any other point in the installation must not exceed 2 ohms.
(xiii) If liable to mechanical damage they are to be adequately protected.
(xiv) In damp and exposed situations, saddles and fixings to be of non-rusting material.
(xv) Bushed holes to be used when passing through steel or iron structures.
(xvi) Holes through party walls or fire-resisting floors to be plugged with non-ignitable material.
(xvii) When run under floors or behind partitions, all connections to be made in boxes of ample capacity and of non-absorbent, non-ignitable material.

Hard Metal-sheathed Cables (Class M2) may be used without conduit or casing provided they are in accordance with conditions (iv) and (ix) to (xvii) inclusive.

Armoured Cables (Class R) may be used without conduit or casing provided they are in accordance with conditions (iv), (ix), (x), (xiii), (xiv), (xvi), and (xvii). When lead sheathed, such sheathing shall comply with (xi) and (xii), and when not lead sheathed the armouring shall so comply.

Cables covered with Tough Rubber Compound (Class S) may be used without conduit or casing provided they are in accordance with conditions (iv), (ix), (x), and (xiii) to (xvii) inclusive.

Conduits Screwed (Class T1). All approved classes of cable may be enclosed in steel conduits provided that the conduits are in accordance with conditions (iv), (xiii), (xiv), (xvi), and (xvii), and in addition:

(xviii) The conduits and fittings are made in accordance with British Standard Specification No. 31–1923 and if used on circuits of medium pressure are of heavy gauge. Metal bosses must be secured on both sides to conduit by screwing or nuts.
(xix) The conduits are mechanically and electrically continuous across all joints and are earthed.
(xx) The resistance of the conduit in a complete installation, between the main switch and any other point in the installation, does not exceed 2 ohms.
(xxi) Ventilating outlets are provided at the highest and lowest points of each circuit.
(xxii) The conduits of each circuit are erected complete before the cables are drawn in (see table on p. 2246).
(xxiii) Abrasion of covering of cables emerging from conduits must be prevented.
(xxiv) The end of conduits where terminating at accessories and fittings are screwed thereto or provided with lock-nuts, or are led into separate blocks, preferably of non-ignitable material.
(xxv) No elbows or tees, unless of the inspection type, are used except at the ends of conduits immediately behind fittings or accessories; and no bend has a radius less than 2½ times the outside diameter of the conduit.
(xxvi) In damp or exposed situations, the conduits are welded, brazed, or solid drawn.

Metal Conduits not Screwed (Class T2). All approved cables may be enclosed in conduits of steel or other metal provided that the conduits are in accordance with conditions (iv), (xiii), (xiv), (xvi), (xvii), (xix) to (xxiii), (xxv) and (xxvi), and (xxvii). The ends of the conduits where terminated at accessories and fittings are adequately clamped thereto, or are led into separate blocks, preferably of non-ignitable material.

NOTE.—Plain slip sockets do not comply with the above regulations—some form of screwed or grip joint giving ample and permanent conductivity and rigidity throughout being necessary.

Non-metallic Conduits (Class T3). All approved cables may be enclosed in conduits of non-ignitable, non-absorbent damp-proof material, provided that such conduits are mechanically continuous and strong, and are in accordance with conditions (iv), (xiii), (xiv), (xvi), (xvii), (xxi) to (xxiii), (xxv) and (xxvii).

Wood Casing (Class W). All approved cables may be enclosed in wood casing provided that

(xxviii) It is used in dry situations only.

(xxix) It is not buried in plaster or cement nor fixed in contact with gas or water pipes, nor immediately below water pipes.

(xxx) The capping is secured by screws.

(xxxi) If the casing forms part of ornamental woodwork, ready access to the cables is provided.

Flexible Cords (Class Z). High insulation twin or multicore flexible cords may, in addition to being used for pendants and portable appliances, be used provided that conditions (iii) and (vi) are complied with, and in addition:

(xxxii) They are for low-pressure sub-circuits for currents not exceeding 6 amperes.

(xxxiii) If insulated with pure rubber, they are not used in damp situations.

(xxxiv) Insulating cleats not more than 3 ft. apart are used and placed so that the cords do not come into contact with walls, ceilings, or structural metal work.

(xxxv) They are open to view.

(xxxvi) The premises do not come under the Factory and Workshop Acts and the Coal Mines Regulation Acts.

(xxxvii) They are not used in shops, warehouses, or places of public resort.

(xxxviii) Where passing through walls or floors, they are protected by non-ignitable, damp-proof conduits.

(xxxix) Where issuing from fittings and unavoidably passing into ceilings they are enclosed in non-ignitable tubes terminating in non-ignitable junction boxes.

Bunching of Cables.—When installed in wooden casing, cables carrying direct or alternating current may, if desired, be bunched whatever their polarity, provided that the number of cables bunched is not more than:

10	if the sectional area of each does not exceed				0·007	sq. in.
6	,,	,,	,,	,,	0·0025	,,
4	,,	,,	,,	,,	0·1	,,
3	,,	,,	,,	exceeds 0·1 sq. in.		

The size of the casing must not exceed that of the maximum number of cables permissible.

When installed in metal conduits, cables carrying direct current may, if desired, be bunched whatever their polarity; but if carrying alternating current, they shall always be bunched.

NOTE.—Where large cables are bunched, the rating should be reduced to prevent overheating.

Bell and Telephone Circuits in a building must not be run in casing or conduits used for the distribution of electricity supply throughout the building.

Cable Sockets and other Connections.—(*a*) The ends of all conductors exceeding 0·04 sq. in. shall be provided with a soldering socket.

(*b*) The dielectric shall not be removed farther than is necessary. Dielectric damaged by heat shall be cut away.

(*c*) The braid, lead, or other covering over the dielectric, including the tape, shall be cut back at least half an inch from the end of the dielectric.

(*d*) With paper-insulated cables, the exposed conductor and dielectric shall be protected from moisture by suitable sealing with insulating compound.

(*e*) The ends of stranded conductors unprovided with cable sockets shall be made solid by soldering in the case of paper insulation, and in the case of rubber when exposed to damp.

Connections between Cables other than flexible cords shall be made either by soldered joints or mechanical connectors.

(*a*) Every joint shall be easily accessible and mechanically and electrically sound.

(*b*) Every mechanical connector shall be enclosed in a non-ignitable box and made impervious to moisture.

(*c*) Connections between cables and flexible cords, or between lengths of flexible cords, shall be made by mechanical connectors shrouded in non-ignitable insulating material contained within suitable receptacles.

(*d*) Joint boxes shall be constructed wholly of durable, non-ignitable, non-absorbent materials.

Main Distribution.

Control of Main Supply.—Every installation shall be adequately protected by suitable controlling apparatus easily accessible to the consumer.

(*a*) If the normal current exceeds 100 amperes, there shall be provided, in addition to a fuse or circuit breaker with overload trip, an automatic device which shall operate in the event of a leakage to earth. The leakage current must not exceed 100 amperes.

(*b*) There shall be a circuit breaker or a switch and fuse on each insulated conductor, and all such poles shall be linked.

(*c*) No fuse or unlinked single-pole switch shall be inserted in a main circuit in that conductor which is direct earthed at the supply.

(*d*) The neutral conductor shall never be interrupted while the other conductors are closed.

(*e*) Wherever single-pole switches are required, they shall be on the same pole throughout the installation.

Subdivision of Circuits.—(*a*) The maximum number of points that may be connected in parallel to a final sub-circuit shall be as follows:

Where the total rating of the points supplied from the sub-circuit does not exceed

6 amperes	. . . . .	10 points
7 ,,	. . . . .	6 ,,
10 ,,	. . . . .	4 ,,
20 ,,	. . . . .	2 ,,

(*b*) Every final sub-circuit shall be connected to a sub-distribution board.

(*c*) Every sub-distribution board shall be connected to a separate way on a main distribution board.

Earthing.

Conditions where Earthing is Necessary.—(*a*) For all pressures of supply, earthing of metal objects other than the conductors shall be effected in the following cases:—In bathrooms, lift shafts, near running machinery, and in all places where even a slight shock might lead to serious accident, all exposed metal liable to become alive, or situated so that there is risk of contact with earthed metal, shall be earthed. The metal sheathings of cables shall be earthed.

(*b*) Where the pressure of supply exceeds the limits of extra low pressure, any damp plaster, concrete, or metal shall be earthed which is liable to become alive and touched by a person making contact with earth. All metal conduits shall be earthed.

Position of Metal in Bathrooms.—All exposed metal liable to become alive shall, in addition to being earthed, be placed out of reach of a person standing in the bath. Lampholders shall have their exposed metal parts efficiently earthed, or, alternatively, all parts liable to be handled when replacing a lamp shall be constructed of insulating material.

Earthing.—(*a*) The combined resistance of the earthing lead and system should be low enough to let enough current flow to operate the fuse or earth leakage trip.

(*b*) Water pipes used as an earthing system shall have metal joints throughout.

(*c*) Pipes conveying gas or an inflammable liquid shall not be used as an earthing system.

(*d*) An earthing lead shall be of high conductivity copper protected against mechanical injury and corrosion of not less than 0·0045 sq. in. (7/·029 in.). for the main connection. For flexible cords 0·0048 sq. in. and smaller, earth lead to be the same size as live conductors.

(*e*) All connections of the earthing lead shall be easily accessible.

(*f*) All domestic appliances, whether portable or fixed, for pressures exceeding 100 volts direct or 30 volts alternating current shall be provided with a terminal or other suitable means for earthing all exposed metal.

Fittings.

Portable Fittings which have to be earthed shall be connected to earth by means of a third conductor in the flexible cord.

Ceiling Roses shall not be used for pressures exceeding 250 volts, and not more than two pairs of flexible cords shall be attached to one ceiling rose, unless it be designed for multiple pendants.

Lampholders shall comply with the appropriate British Standard Specification. Switch lampholders shall be provided with further means of control in the same room. Lampholders in open-type portable fittings must be earthed as above or insulated from fitting.

Plugs and Sockets.—(*a*) Sockets shall be so constructed that they cannot be accidentally short-circuited whether the plug be in or out, and so that a pin of the plug cannot be made to earth either pole of the socket.

(*b*) Plugs connected to apparatus for currents between 1 and 10 amperes shall be constructed with (i) a hand shield, the entry for the flexible cable being at the side of the shield remote from the hand, and, if desired, a switch being fixed adjacent to the socket, or (ii) the plug so interlocked with a switch on the socket that it is impossible to insert or withdraw the plug when the

switch is in the 'on' position, provided that every socket, or group of sockets, carrying more than 300 watts shall have a switch in an accessible position.

(*c*) Plugs having a rated capacity exceeding 10 amperes shall, in addition to a hand shield, be provided with a switch immediately adjacent to the socket and preferably interlocked.

(*d*) The dimensions and clearances shall be in accordance with B.E.S.A. Report No. 372—1930.

MOTORS.

Control of Motors.—(*a*) Every motor shall be protected by efficient means suitably placed and so connected that the motor and all apparatus in connection therewith may be isolated from the supply, except in the case of the earth point of an earthed system. For motors of less than 1 h.p. a plug and socket will be considered to be an efficient method of insulation from the supply.

(*b*) Every motor shall be provided with an efficient switch or switches for starting and stopping, so placed as to be easily operated by the person controlling the motor. Every motor rated over ½ h.p. shall also be provided with (i) means for automatically opening the circuit if the supply pressure falls sufficiently to cause the motor to stop; (ii) in the case of d.c. motors a starter or switch for limiting the current taken when starting and accelerating; (iii) in the case of a.c. motors, such starter or switch for limiting the current taken, when starting and accelerating, to the value (if any) required by the supply undertaking.

(*c*) In every place in which a machine is being driven by a motor there shall be means at hand for either switching off the motor or stopping the machine if necessary to prevent danger.

Electric Lifts.—Lifts to be operated from a circuit independent of lighting installation. Control and motor leads to be in separate conduit, and cables except trailing to be armoured or in metal conduits.

HEATING AND COOKING APPLIANCES.

Control of Heating and Cooking Appliances.—(*a*) Appliances shall be protected by a fuse on each insulated pole.

(*b*) Appliances, whether portable or fixed, shall be controlled as a whole by a switch, which in the cases of portable appliances shall be fitted on a wall.

(*c*) Where a switch or switches are fixed on the frame of a portable luminous heating appliance, at least one section of the heating element shall not be controlled by such switches, so that the luminous heating element is permanently connected to the wall-plug or similar device in order to indicate that the circuit is not broken and that the current is still flowing.

(*d*) The support and frame of every appliance shall be provided with a suitable terminal to which the earthing lead may be connected.

ALTERNATING CURRENT TRANSFORMERS, CHOKING COILS, INDUCTION COILS, STATIC CONDENSERS AND RECTIFIERS, EXCEPT RADIO APPARATUS.

(*a*) Construction shall be in accordance with B.E.S.A. specification.

(*b*) Live parts shall be guarded to prevent accidental contact.

(*c*) Suitable terminal for earthing lead shall be provided.

(*d*) Shall be placed in well-ventilated space and not exposed to risk of damage from external causes.

(*e*) All inflammable material within 24 ins. vertically above, 12 ins. vertically below, or 6 ins. in any other direction, shall be protected with non-ignitable material.

(*f*) An auto-transformer for lamps or domestic appliances shall not be used on a pressure of more than 250 volts or with earthed concentric public mains.

(*g*) In step-up transformers the primary and secondary windings shall be separate throughout; high-voltage parts shall be shielded by earthed metal, or be inaccessible to unauthorised persons. High-pressure wiring shall be a minimum, and conductors conform to B.E.S.A. specification. A multipole linked switch shall provide for the isolation of the supply to the transformer.

TESTING OF COMPLETED INSTALLATION.

NOTE.—The following tests are intended to ensure that the installation is in a satisfactory state at the time of completion.

The value of systematically inspecting and testing apparatus and circuits cannot be too strongly urged, and such periodical tests are essential if the installation is to be maintained in a sound condition and undue deterioration thereof detected.

All defects thus discovered should be made good without loss of time. The attention of consumers should be drawn to the importance of maintaining all apparatus and fittings in a clean and dry condition.

Tests to be Carried out.

Before an installation is permanently put into service the following tests shall be carried out:—

Insulation Resistances.

(*a*) The insulation resistance shall be measured by applying a direct-current pressure of not less than twice the working pressure, between earth and the whole system of conductors or any section thereof with all fuses in place and all switches on. The working pressure is deemed to be that between outer or phase conductors and neutral, where latter is earthed.

(*b*) The insulation resistance of the lighting circuits measured as in (*a*) above shall not be less in megohms than 25 divided by the number of points on those circuits, except that the insulation resistance of any final lighting sub-circuit need not exceed 1 megohm.

(*c*) The insulation resistance between the case or framework and every live part of each individual dynamo, motor, heater, arc lamp, or other appliance complete with its switch and control gear, regulating resistance, and similar accessories, shall not be less than half a megohm when measured as in (*a*) above.

(*d*) All lamps to be in place except on earthed concentric wiring systems. Heating and power circuits may be tested with appliances removed, but lamps (if any) in place.

Note.—In addition to the foregoing tests, it is advisable wherever practicable to take an insulation test between all the conductors connected to one pole or phase of a system and those on the other pole or phase.

Continuity of Metal Sheathing.

The metal conduits or metallic envelopes of cables in all cases where such methods are used for the mechanical protection of electrical conductors shall be tested for electrical continuity, and the electrical resistance of such conduits or sheathing, measured between a point near the main switch and any other point of the completed installation, shall not exceed 2 ohms.

Additions to an Existing Installation.

It must be ascertained whether the ratings of the circuits concerned is sufficient for the increased current.

Radio Apparatus Connected to Supply Mains.

(*a*) All live parts shall be guarded to prevent accidental contact.

(*b*) Cases for enclosed apparatus shall be substantial and constructed of mahogany, teak, oak (English), walnut, metal or non-ignitable material, or of other material if a non-ignitable and non-hygroscopic lining is used; shall be earthed if of metal. Air temperature inside main container not to exceed 120° F., and local temperature rise of no part of apparatus such as to impair its durability. Cables holes must not cause abrasion of cables.

(*c*) Apparatus to be protected by D.P. fuses (see fusible cutouts). A minimum size may be No. 25 S.W.G. lead-tin alloy for working current less than 2 amperes.

(*d*) When a conductor of any cable connected to radio apparatus is in direct connection to supply mains, such cable to be of type specified and installed in accordance with foregoing regulations.

(*e*) Connection between a conductor connected to mains and earthing lead of radio apparatus is permitted only through a condenser incorporated with or close to the radio apparatus.

(*f*) With D.C. supply mains, the aerial shall only be connected to the apparatus through a double wound high-frequency coupling transformer tested to 600 volts, or through a condenser in series with and adjacent to aerial terminal. On a.c. mains when a transformer is not used as above, the condenser capacity shall be not greater than 0·001 mfd. The insulation resistance test specified in (*j*) shall be applied to the H.F. coupling transformers or to the condenser, for either D.C. or A.C. supply.

(*g*) Head telephones and loud speakers shall be connected to the radio apparatus through a double-wound transformer or through a condenser on each conductor.

(*h*) Condensers referred to in (*e*), (*f*), (*g*), to conform to B.E.S.A. specification 271—1926.

(*i*) Every 'mains' transformer to be double wound with an earthed core. The winding connected to the mains to be separated by earthed metal screen or by insulation tested at 1000 volts (A.C.). All windings to have an insulation resistance of not less than 20 megohms at 500 volts (D.C.) or twice the supply pressure, whichever be the greater.

(*j*) Aerial, earth, and telephone terminals on radio apparatus, transformers, or condensers referred to in (*e*) and (*f*), where external to the apparatus, shall have an insulation resistance of not less than 2 megohms from the 'mains' input terminals measured as in (*i*).

(*k*) Batteries connected to radio apparatus and to supply mains shall have insulation resistance to earth of not less than 2 megohms tested as specified in (*i*).

The Electrical Equipment of Ships.

REGULATIONS ISSUED BY THE INSTITUTION OF ELECTRICAL ENGINEERS.
SECOND EDITION, JULY, 1926. (*Amended*, 1929.)
(*Abbreviated.*)

General.—These regulations apply to all sea-going ships except ships of war. They relate to the use of electrical energy for all purposes except the propulsion of ships, and comprise the main requirements for ensuring satisfactory results.

Nature of Supply.—Standard systems of distribution are two-wire D.C. or A.C., three-wire D.C., three-phase three- or four-wire. Standard pressures are 110, 220, 440 volts for parallel constant voltage systems; 110 volts single-phase A.C. Standard pressures have not been fixed for polyphase currents, but must not exceed 150 volts for lighting and 250 volts for power and heating, except for oil ships. On tugs, trawlers, small yachts, fishing and similar small vessels the standard pressures are 12, 50, and 110 volts. The standard frequency is 50 cycles per second.

Generating Plant.—Generators shall conform to the British Standardisation Rules, except that they shall be capable of sustaining an overload of 25 per cent. for two hours in temperate climates and cool situations on board ship. Generators shall be over-compounded 5 per cent.; they shall be placed where they do not affect the compasses, with their axis fore and aft, and the frames shall be efficiently earthed. Standard capacities for steam prime movers are 5, 10, 15, 20, 25, 30, 40, 50, 75, and 100 kW. Steam pressures for single engines 100–120 and for compound engines 150 lb. per sq. in. Excluding turbines, the momentary and permanent speed regulations respectively must be within 10 and 5 per cent. for engines up to 50 kW., and within 10 and 3 per cent. above 50 kW. For internal combustion engines, these figures are 10 and 3 per cent. up to 10 kW. and 10 and 2½ per cent. above 10 kW. Standard capacities are 2, 3, 5, 7½, 10, 15, 20, 25, and 30 kW.

Switchboards.—The requirements for conductors and distribution are in general those specified in the Regulations for the Electric Equipment of Buildings—see p. 2231, but switchboards must be constructed wholly of durable, non-ignitable, non-absorbent materials. The maximum voltage drop in wireless mains shall not exceed 1 volt plus 1 per cent. of the bus-bar pressure when carrying main and battery loads simultaneously. When the load on a conductor is intermittent, the size of cables for such ratings is given in the following table :—

Rating of Conductors Intermittently Loaded.

Nominal Area of Conductor.	Permissible Current in Amperes.					
	Rubber Cables.			Paper Cables.		
Rating :	Half-hour.	One-hour.	Continuous.	Half-hour.	One-hour.	Continuous.
0·0145	38	37	37	60	57	57
0·0225	47	46	46	79	75	75
0·03	56	54	53	94	89	87
0·04	68	65	64	113	105	104
0·06	92	85	83	151	138	135
0·075	113	101	97	180	162	157
0·1	142	124	118	225	199	191
0·12	160	138	130	252	220	210
0·15	191	164	152	303	261	246
0·2	247	204	184	376	320	296
0·25	295	244	214	453	377	343
0·3	315	283	240	523	435	385
0·4	452	357	288	663	543	464
0·5	534	422	332	804	648	540
0·6	641	499	384	960	767	624
0·75	774	604	461	1,180	930	738
0·85	900	680	512	1,325	1,045	815
1·0	1,036	803	595	1,548	1,211	932

Cables in machinery spaces or exposed to weather or the action of salt water shall be either lead covered, with or without further protection, or be run in conduit. Cables in cold-storage chambers shall pass normally through the walls or lagging with continuous lead tube flanged at each end. Such cables must be lead covered in full view throughout their length and supported by porcelain, hard wood, or other non-metallic, non-hygroscopic material. All metallic-covered

cables must have such covering earthed at both ends, except for sub-circuits, which are earthed at the supply end only.

Cables passing through decks or watertight bulkheads shall be provided with suitable glands or deck tubes—see B.E.S.A. Specification No. 94—1920. Where unarmoured cables pass through beams and non-watertight partitions, the holes must be bushed with lead or other soft non-ferrous metal. Flexible cords with silk or cotton covering must not be used in crew or cargo spaces, or where exposed to mechanical damage.

Maximum Spacing of Clips securing Lead-covered or Armoured Cables.

Overall Diameter of Cable.	Lead-covered.	Armoured.
Under 0·3 inch . .	8 inches	10 inches
0·3 to 0·5 inch . .	10 ,,	12 ,,
0·5 to 0·75 inch . .	12 ,,	14 ,,
0·75 to 1·25 inch . .	14 ,,	16 ,,
Over 1·25 inch . .	16 ,,	18 ,,

Earth Connections.—Only steel ships shall be wired on the single-conductor system. Earthing connections carrying current shall be of the same size as the corresponding insulated conductor. Earthing connections not normally carrying current shall be of copper of the same size as the working conductor (but not less than 0·003 sq. in.) up to 0·007 sq. in., and above that size not less than 0·007 sq. in. for every 50 amperes of working current or part thereof, but need not exceed 0·1 sq. in. Earth connections shall be accessible and secured by screws not less than $\frac{3}{8}$ in. in diameter.

Interference with Compasses.—Dynamos, batteries, and all apparatus producing an external magnetic field shall be fixed at such a distance from the compass that the magnetic field produced thereat has a negligible value. No single-conductor cable carrying continuous current shall be fitted within 30 ft. of standard and steering compasses. Conductors in the vicinity of the compass shall be twin, or as close together as possible. Incandescent lamps used for lighting the compass shall not consume more than 0·6 ampere, and no live part shall be less than 7 ins. from any part of the magnetic system. Effect of switching must be noted during adjustment of compass.

Navigation Lamps.—Each navigation lamp is to be separately wired and controlled by a separate switch and fuses, and to have an automatic indicator in circuit to give warning of the failure of the light. Metal filament bayonet cap vacuum lamps fixed vertically of not less than 60 watts to be employed.

Storage Batteries.—The cells shall be so constructed that the electrolyte cannot be spilled, with strong containers (not celluloid). They should be in a single tier and mounted on oil insulators. The deck must be lined with lead or other acid-resisting material, and all exposed metal protected with acid-resisting paint. The battery compartment must be well ventilated. Switches and fuses must be fixed outside the battery room. All connections in an acid battery room are to be of bare metal or lead-covered cables.

Fittings.—Fittings are to be so designed and fixed that dust and moisture cannot accumulate within them, and are not to be mounted on surfaces liable to become damp. In exposed positions they are to be weatherproof and waterproof; open type fittings must not be used near inflammable goods or where inflammable dust or gases can occur. Switches, etc., must not be installed in bunkers or where inflammable gases may be present. Weatherproof portable lamp fittings with metal frames must have all metal parts connected to earth by a third conductor in the flexible cord, and must be provided with hooks to hang them up with. Lampholders in weatherproof portable fittings must be earthed, and in open type fittings they must be insulated and protected with insulating shields. Plugs and sockets must be so constructed that they cannot be readily short-circuited or earthed or allow an arc to be maintained. Plugs must be provided with hand-shields or interlocked with switches in the sockets. Weatherproof plugs and sockets must be employed in exposed positions. See B.E.S.A. Specifications Nos. 97—1926, 98—1919, 67—1929, 372—1930, 52—1927.

Lamps.—Arc lamps must have the whole of their live parts insulated from the frame, and must be provided with wired globes or other protection; they must not be fitted near combustible goods or where gases can accumulate, and each circuit must be controlled by a switch and fuse on each insulated pole. Navigation lamps must be of the carbon or metal filament type of 32 candles.

Motors.—May be of any type enumerated in B.E.S.A. Specification No. 168—1926, or of submersible type. When larger than 1 b.h.p. they must strictly conform. Working parts must be accessible, and construction must allow of dismantling in space where fitted, unless readily removable. The carbon brush gear must be adjustable. For brushes see p. 2295 and B.E.S.A. Spec. No. 96—1922 for standard terminals. Lubrication must be automatic. Motors should be placed in well-ventilated compartments, with axis of rotation in a fore and aft direction and clear of inflammable material: otherwise they must be of flame-proof or forced-draught type with pipes taken outside compartment. Where temperature exceeds 120° F. motors to be pipe-

ventilated, induced- or forced-draught type connected to inlet air at less than 120° F. Motor starters above 3 b.h.p. must have no-volt release and replaceable resistance elements. Motors essential to the safety of the ship in the event of damage must be capable of running for a reasonable period after the compartment has been flooded.

Control Gear.—Regulators and controllers are to be of incombustible materials and guarded or enclosed in incombustible cases; contacts shall be renewable, and all live parts enclosed by metal covers, and so designed that they cannot be touched by the operator. Resistances shall not rise to an excessive temperature, and no accessible part of the apparatus may rise above 130° F. Internal connections shall not be soldered. Control gear shall not be placed where inflammable gases can accumulate, unless it is flameproof; in exposed situations it shall be totally enclosed. Resistances shall be placed in well-ventilated compartments and away from combustible material.

Heating and Cooking Apparatus.—Appliances shall be so constructed that parts which are handled cannot exceed a temperature of 130° F. The elements shall be readily replaceable; live and highly heated parts shall be guarded. Connections between the elements and switches or external leads shall be made without solder, and their temperature shall not exceed 200° F. All appliances for pressures exceeding 150 volts shall be efficiently earthed. Whether portable or fixed, appliances shall be controlled by fixed switches; portable appliances shall have at least one section of the elements controlled only on the live side of the wall plug or other connection with the fixed portion of the circuit.

Internal Communications.—Transforming apparatus and batteries shall conform with the rules for lighting and power plant. If the working pressure exceeds 25 volts, all parts of the apparatus shall be made and fitted in accordance with the rules for lighting and power circuits. Communication cables shall be made and fixed in accordance with the rules for light and power cables, and shall be kept as far as possible separate from the latter, with separate grooves when run in wood casing. Fuses need not be fitted in branch circuits for pressures not exceeding 25 volts, or in circuits supplied from primary batteries.

Lightning Conductors.—Conductors shall be fitted to all vessels having wooden masts, of copper tape or rope having not less than 0·15 sq. in. sectional area. When in dry dock, ships' lightning conductors shall be connected to an efficient earth on shore.

Oil ships.—On oil ships the pressure for all purposes shall not exceed 110 volts with A.C.; with D.C., for power 220 volts, for lighting and heating 110 volts. No part of the system shall be earthed, and double-pole linked switches shall be used on every circuit or sub-circuit. All cables shall be lead-covered, with or without armour. Distribution shall be on the two-wire, two-conductor system. No wiring shall be taken into oil-holds or coffer-dams, which must be lighted by lamps entirely outside the space and separated from it by an airtight glass bowl. Lamps in dangerous spaces shall be in gas-tight fittings, and the switches shall be outside the spaces; no portable lamps other than self-contained battery-fed lamps (mining type) may be used in dangerous spaces.

Constant-current Series System.—Special requirements include automatic circuit closer for short-circuiting generator in the event of an open circuit anywhere in the system, and forbid the use of open-type machines. All bare conductors (including heating elements) to be totally enclosed in metal cases with conductors accessible only when disconnected, and no part of the system shall be earthed.

Trials.—The insulation of the whole system, with all lamps and fuses in place, is to be tested before putting into service with direct current at twice the working pressure between the system of conductors and earth, and shall not be less in megohms than 10 divided by the number of lamps (on lighting circuits), but the insulation resistance of any final lighting sub-circuit need not exceed one megohm. The insulation resistance of every piece of apparatus, with switch or control gear, etc., shall not be less than half a megohm. The installation shall be run for six hours and all the apparatus operated, not necessarily simultaneously.

Disturbances to Wireless Installations, Direction Finders, and other Thermionic-Valve Apparatus.—Disturbances arising from commutator machinery may cover a wide area and may be reduced by employment of lead-sheathed cable for deck motor circuits, such circuits being well removed from chart-room or wireless cabin. A condenser of 2 mfd. capacity insulated to withstand four times the supply voltage may be connected with short leads across the brushes of a disturbing motor. Commutators must be kept in good condition. In motor-boats equipped with wireless, braided-metal cable should be used for sparking-plug connections, the braiding earthed at both ends and at least at one intermediate position. The low-tension earth wire should be earthed in at least two places.

Plugs and Sockets.

British Standard Specification for 5-, 15-, and 30-ampere, 2-pin wall plugs and sockets for domestic purposes (without earthing connection), No. 73—1927, gives details of materials to be used, sizes and spacing of all insulating, current-carrying, protecting, and fixing parts, together with gauges and tables of dimensions. Heavy Duty, Flame-proof type plug and socket are dealt with in Specification No. 279—1927, and Reversible Protected-type two-pin plugs and sockets with earthing connections in No. 196—1927. Charging plug and socket for vehicles propelled by Electric Secondary Batteries are dealt with in Specification No. 74—1917. Three-pin wall plugs and sockets, hand-shield and side-entry pattern (two-pin and earth type), for domestic purposes are dealt with in Specification No. 317—1928.

Cable Soldering Sockets.

Specification No. 91—1930 of the British Engineering Standards Association gives the standard dimensions of cast and wrought lug sockets, clamp sockets, and clamps, for cables from 0·003 to 1·000 sq. in. sectional area. Cast lug sockets are of brass containing 70 per cent. of copper, and tubular lug sockets are of solid-drawn copper tubing; clamp sockets are of brass or other suitable non-ferrous metal. Sweating surfaces are to be drilled and tinned, and contact surfaces machined.

Watertight glands for Electric Cables are dealt with in Specification No. 94—1920.

Cables for Installation Work.

As far as electrical installation work, and particularly conduit work, is concerned, the type of cable with which the contractor has mostly to deal is that known as a rubber insulated cable. The real properties of rubber insulated cables are more or less obscure, and the question as to what constitutes a good cable is almost impossible to answer with any degree of accuracy. To give some indication as to the quality of the rubber compound covering, a careful chemical test is required, and this combined with a stretching test will give a distinct clue as to the possible durability of the material.

Government Department Specification for Rubber-Covered Cables J, K and L Classes for Power and Light.

Designation.—Cables shall be designated by the nominal sectional area of the conductor and a letter prefix as follows: 'J Low' and 'J Low Twin,' 'J Medium' and 'J Medium Twin' for 250-volt and 660-volt respectively for V.I.R. taped and braided. 'K' and 'K Twin' for V.I.R. taped, and lead sheathed (660-volt) and 'L' and 'L Twin' for V.I.R. taped, lead-sheathed and armoured (660 volts).

Material.—The conductor shall comply in all respects with B.E.S.A. specification No. 7, and be free from all mechanical defects.

Tinning.—Each wire shall be tinned uniformly with pure tin free from lead in a bath free from contamination with copper. Before stranding, a sample shall be bent into a loop of radius not less than 12 and not more than 15 times its diameter, cleaned to remove grease and immersed for 1 minute in hydrochloric acid specific gravity 1·088 at 60° F., washed and examined under a lens for blackening. Average number of times of immersion to produce visible blackening shall not be less than 8 times for wire 0·029 to 0·048 inches diameter, 7 times for diameters above 0·048 up to 0·0808, and 5 times for diameters over 0·0808 inches. A further sample to be wrapped round a rod of diameter four times that of the wire and submitted to the above for one cycle.

Jointing.—Not more than one joint to be allowed in any wire forming 1000 yards of conductor nor a joint to be within 12 inches of another in same layer, nor in conductor after stranding. Joints shall be brazed, silver soldered or electrically welded.

Stranding.—To be uniform throughout length. 3-stranded wires to have a left-hand helical lay, 7 strand to have straight centre wire with six left-hand helically laid wires, the stranding for other numbers of wires to have alternate left-hand and right-hand lays.

Resistance of Conductors.—To conform to B.E.S.A. specification No. 7. Compensation for the laying up of twisted twin cables to be agreed between contractor and inspector.

Dielectric.—Conductor shall be covered with one layer of British plantation pure rubber applied helically or longitudinally, then a covering and jacket of vulcanised rubber consisting of pure rubber mixed with sulphur, non-hydroscopic mineral matter and wax if required. The dielectric shall contain not less than 45 per cent. by weight of pure rubber. After vulcanisation the acetone extract shall not exceed 5 per cent. by weight, of which rubber-resins are not greater than 2 per cent. and free sulphur 0·75 per cent. Maximum amount of organic matter in residue after acetone extraction not to exceed 1·0 per cent. of total dielectric. Sulphur content in rubber not to be less than 1·25 or greater than 1·75 per cent. of total dielectric. No re-manufactured rubber or rubber substitutes to be used. Cable to be rejected if blackening is shown on pure rubber or tin, or if pure rubber adheres to conductor.

Thickness.—To be in accordance with B.E.S.A. specification No. 7.

Taping and Vulcanising.—To consist of helical binding of fine cotton tape (red and black or blue for twin cables) of specified thickness, rubber proofed. The whole to be vulcanised into homogeneous body, after which rubber layers shall be inseparable but tape removable without tearing rubber.

Mechanical Tests.—After removal of the conductor, an outer covering a length of 2 inches marked on the dielectric shall be maintained stretched to 6 inches for 24 hours at an ordinary temperature not exceeding 65° F. After release, the length shall not be greater than 2½ inches, no signs having been shown of the rubber cracking or elongating irregularly. Tensile strength not to be less than 800 lb. per sq. in.

Heat Tests.—Separate samples after removal of conductor shall withstand without material deterioration a dry heat of 270° F. for 1 hr. and a moist heat of 320° F. for 3 hrs. After three days interval samples to be stretched as above from 2 ins. to 5 ins. for 6 hrs., the distance becoming not greater than 2½ ins. on release.

Electrical Tests.—(*a*) High voltage test as B.E.S.A. Specification No. 7 after vulcanising and before outer protective coverings are put on. (*b*) Insulation resistance shall not be less than that given in tables and to be measured immediately after test (*a*). Cable to be immersed in water

and measurement made with not less than 500-volt direct current after 1 minute negative electrification. The electrification must proceed regularly.

Twinning.—For circular twin, the complete cores shall be twisted with a right-hand lay, wormed with tanned jute and wrapped to approximate circular section with rubber-proofed cotton tape. For flat twin, the two cores side by side are tape-wrapped to form symmetrical cable.

Mechanical Protection.—The specification gives particulars for braiding, identification cotton, lead sheathing, bedding and serving, galvanised steel wire and steel tape armouring, tests on finished cables and packing, together with tabulated data and a temperature correction table for insulation resistance.

The foregoing details give some indication of the main features which are to be looked for in a high-class rubber insulated cable. As regards the conductors, the conductivity question is one of considerable importance, but it should be mentioned that trouble in this direction very seldom arises, because there is little difficulty in maintaining a standard purity in electrolytically refined copper. The phrase ' to show no sign of sulphide of copper after vulcanising ' refers to the appearance of the wire when it is stripped of its insulating covering. It should then present a clean, bright appearance, free from blackening.

The building up of the rubber dielectric in three layers represents the usual English practice, and the wording of the specification is designed to exclude the use of adulterants and deleterious substances, these being practically synonymous so far as the life of the rubber insulation is concerned.

The specifying of a minimum percentage of rubber in addition practically leaves no room for anything but legitimate compounding ingredients, the choice of which is left to the experience and skill of the cable manufacturer.

There is, perhaps, no industry so greatly dependent on detailed experience and skill in applying that experience as that of the manufacture of rubber, and when electrical conditions are involved this is doubly true.

Generally speaking, therefore, where considerations of quality and durability are paramount it will always be best to purchase rubber-insulated cables, either to a specification such as that instanced above, or to agreed standards such as those of the Cable Makers' Association.

The provision of suitable outer coverings for rubber-insulated wires and cables which are exposed to abnormal conditions is a point which should not be overlooked. Conversely, in a given situation, a certain method of installing may be unsuitable. For instance, condensation effects, which occur in tubes and pipes, are notoriously bad for rubber cables, and in such situations either the method of installing should be varied or special coverings, such as a bitumen sheath, should be applied to the cables. Many cable makers are prepared to advise as to the best type of covering and method of running or fixing cables under abnormal surrounding conditions.

A second quality of vulcanised rubber cable has been placed on the market to meet the demand of those users who are satisfied with a cheaper grade of cable. Such cable, known as ' non-Association ' cable, is quite satisfactory when properly manufactured, and may be safely used in cases where only a moderate length of life combined with good mechanical properties, is required, and where the conditions do not require a permanent maintenance of high elasticity of the dielectric.

Revised Specification for Cab-Tyre Sheathed Cable.*

The following specification for this class of cable is based on the British Admiralty Specification Tests for Cab-Tyre Sheathing :—

The cab-tyre sheathing shall contain not less than 35 per cent. and not more than 40 per cent. of first-grade plantation rubber or other rubber of not less good quality.

The surface of the sheathing shall not be treated or dressed without the consent of the Admiralty Overseer.

The acetone extract of the vulcanised sheathing shall not exceed 5 per cent. calculated on the weight of the complete sheathing as manufactured.

Bending Test.—A test length of the complete cable containing the conductor shall be bent round a hard rod having the same diameter as the external diameter of the cable to form a loop. The unbent portions of the test-piece shall be bound together so that the two parts touch one another closely throughout their length from their ends up to the beginning of the loop.

In cables of large size when this method is not practicable the test-piece shall be bent in a loop so that the maximum extension of the external surface at the curvature at the bend reaches 50 per cent.

The sample in either case shall be left untouched for a period of 48 hours and at the termination of this time shall show no signs of cracking.

Immersion Tests. (*a*) *Chemical.*—Test-pieces 6 ins. long shall be prepared, and after the conductor and dielectric have been removed they shall be dried at a temperature of 100° C. for one hour in an air oven ; they shall then be cooled in a desiccator for two hours, and at the end of this period shall be weighed. The ends of the test-pieces shall be plugged with rubber bungs and sealed with paraffin wax for a length of half an inch from each end. The test-pieces thus prepared shall be completely immersed in a vertical position in the following solutions for a

* Referred to in I.E.E. Rules as ' tough rubber compound ' covering, p. 2234.

period of 48 hours at normal temperature; they shall then be removed and washed in running water for a period of two hours, afterwards being wiped dry and the wax and plugs removed. The samples shall then be dried in an air oven for one hour at a temperature of 100° C. and shall be cooled for two hours in a desiccator; they shall then be reweighed. The difference in weights before and after immersion shall be calculated as grammes per 100 sq. ins. of surface exposed to the solution.

(1) *Acid Test.*—The solution shall be hydrochloric acid having a specific gravity of 1·045. The figure calculated as stated above shall not exceed 1·0.

(2) *Alkali Test.*—The solution shall be sodium hydrate having a specific gravity of 1·116. The figure calculated as stated above shall not exceed 1·5.

(*b*) *Oil Test.*—A test-piece shall be prepared 6 ins. long; one end shall be sealed so as efficiently to prevent access of oil to the interior. The test pieces shall be immersed in pure rape oil of the best quality to such a depth that not less than 2 ins. of sheathing is exposed to the action of the oil. The immersion shall be continued for six hours, the oil during that time being maintained at a temperature of 100° C. After immersion the sample shall be removed and rolled dry between blotting paper; the seal shall then be removed and the sample re-weighed.

The difference in weights shall be calculated as grammes per 100 sq. ins. of surface exposed and the figures so obtained shall not exceed 15.

The cable should be generally in accordance with *St. Helens Cable & Rubber Company's* original Patent No. 3996/1911.

Cab-Tyre Sheathed Cables are now made in C.M.A. 'Nonazo' (non-Association) quality, as well as in C.M.A. Association grade.

WIRING SYSTEMS.

Conduits and Fittings for Electrical Installations.

Conduits, with the exception of the seamless or drawn varieties, are generally made from specially selected steel strip, varying in gauge according to the diameter of the tube. The strips are rolled to a circular section of exact external and internal diameter, by a method which ensures freedom from internal roughness and an even surface to facilitate the drawing of the wires into the conduit, thus preventing injury to their insulation. Exact sizes are essential in order to obtain a tight junction between the conduit and fittings with both socket and screwed junctions.

The conduits are made in two different types, viz. Light Gauge, for socket and continuity junctions, and Heavy Gauge for screwed junctions.

Conduits are usually supplied in two standard finishes, as follows:—

(1) Stove enamelled (black); (2) Galvanised.

For all ordinary installation work in the interior of buildings, enamelled conduits will be found satisfactory in working.

The enamel employed should be flexible, acid-resisting, insulating, and durable.

The conduits must be well cleaned before being enamelled, and subsequently stored at an even temperature until the enamel is set, so that it is not liable to flake or break away when the conduit is bent. The enamel should resist any chemical action which is likely to be found in the plaster of buildings.

Where conduits are subjected to excessive chemical action, especially when buried in cinder-fill concrete, quick-drying chemical plasters, and the like, a more efficient protection is required, and this is afforded by galvanising.

British Standard Specification for Steel Conduits for Electrical Wiring.*

(No. 31—1923.) (*Abstract.*)

Steel conduits shall be either close-joint, brazed, welded or solid drawn as ordered, and shall be of mild steel and free from burrs and internal roughness. The conduit fittings shall be made of steel or malleable iron. All steel shall have a tensile strength of not less than 18 tons nor more than 24 tons per square inch of section, and an elongation of not less than 15 per cent. in a length of 8 inches. Malleable castings shall be well annealed and free from internal projections.

The conduit shall be manufactured in straight lengths of ten to fifteen feet.

For all fittings made out of *tube*, the minimum thickness of the fittings shall be equal to that of the corresponding conduit with which they are used. In no case shall 'close joint' tube be used for these fittings.

Two classes of steel conduit for electrical wiring are recognised as Standard:

Class A.—Plain.
Class B.—Screwed.

Class A consists of light gauge conduit of the thickness and dimensions given in the Table (p. 2246). Class A conduit is either close-joint, brazed, welded or solid drawn. The coupler joining the lengths of tubing is a sleeve, and neither the ends of the conduit nor the coupler are screwed.

* By permission of the British Engineering Standards Association.

In the close-joint tubes, the edges of the steel strip, though brought tightly together in the process of manufacture, are not metallically joined in any way.

Class B consists of heavy gauge conduit of the thicknesses and dimensions given in the Table. Class B conduit is either brazed, welded or solid drawn. Both ends of the conduit are screwed.

Unless otherwise specified with the order, all conduits and fittings shall be stove enamelled jet black or galvanised by the hot process both inside and out.

TABLE OF STANDARD DIMENSIONS OF STEEL CONDUITS.

Conduit outside Diameter in ins.		½	⅝	¾	1	1¼	1½	2	2½
Threads per inch		18	18	16	16	16	14	14	14
Depth of thread (ins.)		·0366	·0356	·0400	·0400	·0400	·0457	·0457	·0457
Maximum length of thread on ends (ins.)		7/16	½	9/16	11/16	¾	13/16	15/16	1 1/16
Minimum length of thread on ends (ins.)		⅜	7/16	½	⅝	11/16	¾	⅞	1
Nom. thickness—Class A (plain) (ins.)		·040	·040	·048	·048	·056	·064	·064	·072
Min. thickness—Class A (plain) (ins.)		·036	·036	·044	·044	·052	·060	·060	·068
Nom. thickness—Class B (screwed) (ins.)		·056	·064	·072	·072	·072	·080	·092	·092
Min. thickness—Class B (screwed) (ins.)		·052	·060	·068	·068	·068	·076	·088	·088
Calculated weight per 100 ft., in lb., unenamelled and not including couplers	Class A (plain)	20	26	37	50	73	100	135	191
	Class B (screwed)	27	39	53	73	93	124	192	242

TABLE OF CAPACITY OF CONDUITS.

(Type B, Screwed, B.S.S. No. 31, for the Drawing-in of Conductors.)

(From I.E.E. Rules.)

Size of Conduit (ins.)		½	⅝	¾	1	1¼	1½	2	2½
Internal Diameter (ins.)		0·388	0·498	0·606	0·856	1·106	1·34	1·816	2·316
Nominal Area of Conductor. Sq. In.	Number and Diameter (In.) of Wires.	Approximate Overall Diameter. Inch.	Maximum Number of Conductors.							
0·0015	1/·044	0·173	2	3	5	8	—	—	—	—
0·002	3/·029	0·195	—	3	4	8	—	—	—	—
0·003	3/·036	0·215	—	2	4	8	—	—	—	—
0·003	1/·064	0·197	—	2	4	8	—	—	—	—
0·0045	7/·029	0·226	—	—	3	5	7	—	—	—
0·007	7/·036	0·259	—	—	2	5	7	—	—	—
0·01	7/·044	0·287	—	—	2	4	7	—	—	—
0·0145	7/·052	0·317	—	—	—	2	5	7	—	—
0·0225	7/·064	0·359	—	—	—	2	4	7	—	—
0·03	19/·044	0·393	—	—	—	—	3	5	7	8
0·04	19/·052	0·441	—	—	—	—	2	4	7	8
0·06	19/·064	0·513	—	—	—	—	—	3	5	6
0·075	19/·072	0·596	—	—	—	—	—	—	4	4
0·1	19/·083	0·663	—	—	—	—	—	—	3	4
0·12	37/·064	0·702	—	—	—	—	—	—	2	3
0·15	37/·072	0·768	—	—	—	—	—	—	2	2

The Henley Wiring System.

(W. T. Henley's Telegraph Works Co., Ltd.)

In this system the cables carry their own metallic protection, which is of a flexible nature; it is consequently very easily and speedily erected. The wires and cables are of very small dimensions, and when painted, distempered, or stained and varnished to match existing decorations, they are to all intents and purposes indistinguishable. If desired, the cables may be buried in plaster and concealed generally. The cables may be run under floorboards, in lofts or cupboards, and dropped down between the faces of lath-and-plaster partition walls. The chasing of walls or cutting away of joists is very much less than is required with tubing.

The cables employed are of flat section of one, two, or three conductors, each conductor composed of tinned copper wires, insulated with pure and vulcanised rubber, taped with proofed tape of a distinctive colour for each core, the whole then vulcanised together, cores laid side by side, sheathed with a solid drawn tube of special alloy to protect the cores from mechanical damage. These cables may be used on either direct or alternating current.

TABLE OF SINGLE-CORE, TWIN, AND TRIPLE CABLES.

Nominal Area.	Conductor.	Minimum Insulation Resistance per Mile at 60° F.	Amperes at I.E.E. Standard.	Approximate Overall Dimensions.	Approximate Weight per 1,000 Yards.
Sq. In.	In.	Meg.		In.	Cwts.
		SINGLE CORE—CLASS ZM.			
0·0015	1/·044	2,000	6·1	0·208	3·2
0·002	3/·029	1,250	7·8	0·230	3·64
0·003	1/·064	2,000	12·9	0·232	3·76
0·003	3/·036	1,250	12·0	0·250	4·13
		TWIN—CLASS ZZMF.			
0·0015	1/·044	2,000	6·1	0·232 × 0·360	5·01
0·002	3/·029	1,250	7·8	0·254 × 0·404	5·78
0·003	1/·064	2,000	12·9	0·256 × 0·408	6·02
0·003	3/·036	1,250	12·0	0·274 × 0·444	6·66
0·0045	7/·029	1,250	17·5	0·305 × 0·486	8·93
0·007	7/·036	900	22·0	0·338 × 0·552	10·51
0·010	7/·044	900	26·0	0·366 × 0·608	12·38
0·0145	7/·052	900	31·0	0·416 × 0·688	16·82
0·0225	7/·064	900	38·5	0·458 × 0·772	20·49
0·030	19/·044	750	45·0	0·512 × 0·860	25·65
0·040	19/·052	750	53·0	0·580 × 0·976	34·85
		TRIPLE—CLASS TZMF.			
0·0015	1/·044	2,000	6·1	0·252 × 0·508	8·71
0·002	3/·029	1,250	7·8	0·274 × 0·574	10·00
0·003	1/·064	2,000	12·9	0·276 × 0·580	10·36
0·003	3/·036	1,250	12·0	0·294 × 0·634	11·43

Test Pressure.—1,000 volts alternating for 15 minutes after 24 hours' immersion in water at 60° F. (15·6° C.).

For the purpose of bonding the sheathing at switch and light points the Universal Bonding Clamp is provided. This clamp, fig. 3, has been designed so that one clamp only, instead of several, may be used at any point, thereby reducing to a minimum the necessity for stocking various sizes of clamps for various numbers and sizes of wires, etc.

When erecting it will be found more economical in many instances to run both switch and light feeds to a three-plate ceiling rose, looping from there to the next light. In the case of electroliers controlled by two switches, in many instances it will be found economical to use a four-plate ceiling rose, taking both switch and light feeds to this rose, the switches being connected by means of a triple cable. This three-core cable can be used to advantage in many other instances, which will be readily apparent when the work is planned out.

Straight through and tee joints, etc., are made by means of the boxes provided under the system.

For fixing the cables, the 'Link' clip, fig. 4, is specially convenient. These clips are made from tinned brass, and may be fixed either with nails or countersunk-headed screws.

Fig. 2 shows a form of Tee Joint Box for the twin system. These Boxes are manufactured from brass heavily tinned; the internal connectors are of the porcelain type, and are made in various forms according to requirements. The continuity of the metal sheathing of the cables is preserved by means of mechanical clamps that bond the lugs on the boxes to the cables.

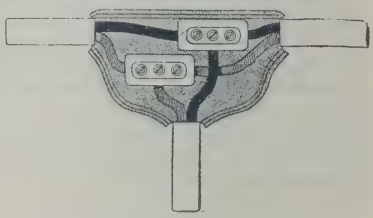


FIG. 2.

Another form of Tee Box—the 'Super' Box—is shown by fig. 5, p. 2250. In this Box the loose bonding clamps used with the box illustrated in fig. 2 are dispensed with; the cover is held in position by one unlosable self-centring nut which is fixed in the cover, and the porcelain connectors are substituted by a moulded terminal block.

The latest addition to the system is a Box of the 'Universal' type, which is illustrated in fig. 6, p. 2250. This Box works on the same principle as the Universal Bonding Clamp illustrated in fig. 7, and can be used as a straight through, tee, or four-way box as required. As with the 'Super' Box, the cover is secured by one unlosable nut. The joint is made by means of porcelain connectors.

The fixing of link clips, saddles, protective covering, or other articles, to plaster, brick, marble, slate, stone, etc., is best effected by the use of "Rawlplugs" (see p. 420).

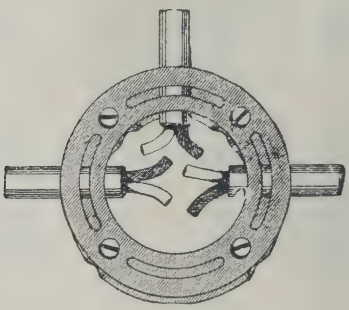


FIG. 3.

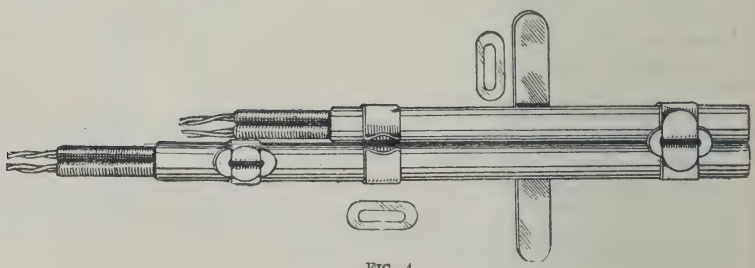


FIG. 4.

The Henley 'Nomet' (Regd.) Wiring System.

This system, whilst not new, employs Henley C.T.S. cables, and has been developed:

(*a*) To meet the wishes of customers who prefer a non-metal covered system.

(*b*) For use in abnormal situations where metal-covered wiring may be undesirable.

It is strongly recommended that, wherever possible, a metal-covered system be used, this being an economical and efficient means of electric wiring, and also a relatively safer method than any non-metal covered system.

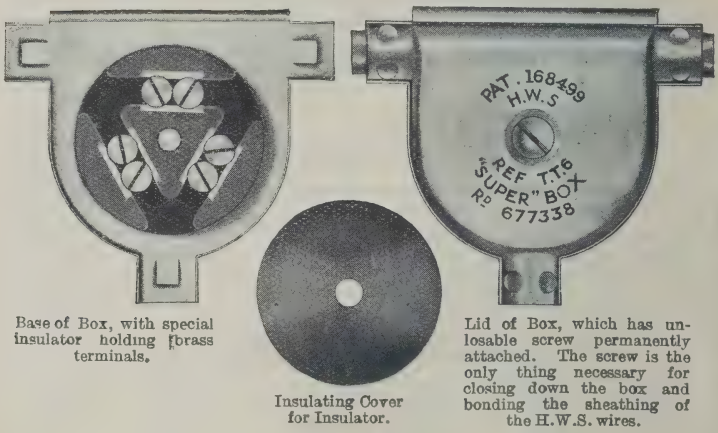
Table of Single-Core, Twin, and Triple Cables.

Nominal Area.	Conductor.	Minimum Insulation Resistance per Mile at 60° F.	Amperes at I.E.E. Standard.	Approximate Overall Dimensions.	Approximate Weight per 1,000 Yards.
Sq. in.	In.	Meg.		In.	Cwts.
		Single Core—Class S.R.X.			
0·0015	1/·044	2,000	6·1	0·212	0·86
0·002	3/·029	1,250	7·8	0·234	1·03
0·003	1/·064	2,000	12·9	0·236	1·14
0·003	3/·036	1,250	12·0	0·254	1·27
0·0045	7/·029	1,250	18·2	0·265	1·49
0·007	7/·036	900	24·0	0·290	1·91
0·010	7/·044	900	31·0	0·318	2·48
0·0145	7/·052	900	37·0	0·348	3·14
0·0225	7/·064	900	46·0	0·410	4·51
0·030	19/·044	750	53·0	0·444	5·42
0·040	19/·052	750	64·0	0·492	7·03
0·060	19/·064	750	83·0	0·564	9·89
		Two Core—Class F.S.R.X. 2.			
0·0015	1/·044	2,000	6·1	0·212 × 0·324	1·48
0·002	3/·029	1,250	7·8	0·254 × 0·388	2·05
0·003	1/·064	2,000	12·9	0·256 × 0·392	2·27
0·003	3/·036	1,250	12·0	0·274 × 0·428	2·54
0·0045	7/·029	1,250	17·5	0·285 × 0·450	2·99
0·007	7/·036	900	22·0	0·310 × 0·500	3·88
0·010	7/·044	900	26·0	0·338 × 0·556	5·05
0·0145	7/·052	900	31·0	0·368 × 0·616	6·44
0·0225	7/·064	900	38·5	0·450 × 0·740	9·54
		Three Core—Class F.S.R.X. 3.			
0·0015	1/·044	2,000	6·1	0·236 × 0·468	2·17
0·002	3/·029	1,250	7·8	0·255 × 0·525	2·67
0·003	1/·064	2,000	12·9	0·256 × 0·528	3·01
0·003	3/·036	1,250	12·0	0·274 × 0·582	3·30

Test Pressure.—1,000 volts alternating for 15 minutes after 24 hours' immersion in water at 60° F. (15·6° C.).

The 'Nomet' Wiring System cables, like the Henley Wiring System cables, are small in dimensions and carry their own mechanical protection. The cables can be easily erected, painted or otherwise treated, and the complete range of accessories (which are stocked) facilitate the wireman's work and enable a neat and satisfactory installation to be made.

THE HENLEY WIRING SYSTEM.



Base of Box, with special insulator holding brass terminals.

Insulating Cover for Insulator.

Lid of Box, which has unlosable screw permanently attached. The screw is the only thing necessary for closing down the box and bonding the sheathing of the H.W.S. wires.

FIG. 5.

FIG. 6.

The J. and P. Wiring System.

(Johnson and Phillips, Ltd.)

This system consists of single, twin (fig. 7), or three-core (fig. 8) rubber-insulated cable protected by a metallic sheathing of a special lead alloy. The cables are of flat section, and are intended for the wiring of domestic and other building interiors without further protection.

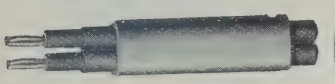


FIG. 7.

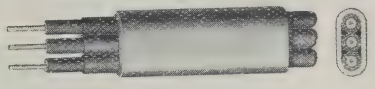


FIG. 8.

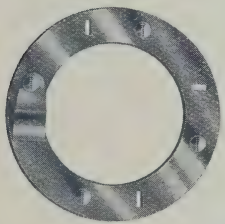


FIG. 9.

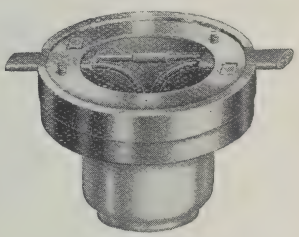


FIG. 10.

When fixed with the special fittings provided, they are neat and unobtrusive in appearance, as they lie flat against the wall or ceiling and do not sag. If used for surface fixing, they can be coloured to match the paintwork or distemper, and are then practically indistinguishable from their surroundings.

The system fully complies with the I.E.E. Wiring Rules, and with the requirements of fire insurance companies and electricity supply authorities.

The bonding fixtures are a special feature of the J. & P. wiring system. By means of bonding and bonding clips a perfect bond can be secured with the greatest ease.

The bonding ring (fig. 9) is fixed behind a wooden pattrass (fig. 10), where connections are made to switch points, etc. The ring is supplied with four ways, one or more of which may be used as required, suitable for cables up to 3/·036 twin core.

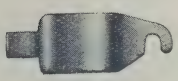


FIG. 11.

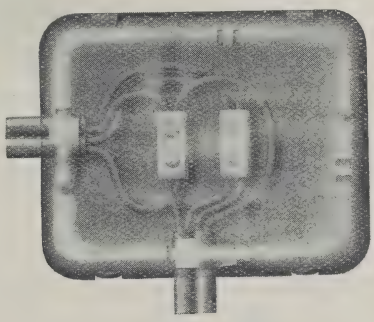
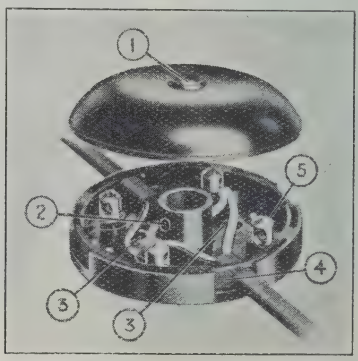


FIG. 12.

To fix, the tail of the clip, fig. 11, is simply inserted in the slot provided in the ring, the cable is placed under the clip and the notched end of the latter pushed under the screw. When this is tightened up, the metal cable sheath is securely held, the cable cannot pull out, and absolute electrical continuity is attained.

Fig. 12 shows a joint box in which twin cables can be inserted on any one or two sides if it is desired to do so.

The J. and P. 'Non-Metal Sheathed System,' designed to supplement the Metal Sheathed system described, uses C.T.S. cables which are of the same overall size as the corresponding metal-sheathed cables, and the same fixing clips and saddles, protective steel covering, etc., can be employed. C.T.S. cable is more flexible than the metal sheathed cable, and although this extra flexibility reduces the risk of kinking the cable during installation, a little more care is needed to avoid sagging on long straight runs. An ornamental wood moulding, designed to cover the cable and clips, and shaped so that it cannot be misused as a picture rail, is available for use where the appearance of a moulding would harmonise better with the decorations.



1. Blanked hole for flex.
2. Hole in pillar for flex.
3. Blanked holes for back entry cable.
4. Sliding Shutter.
5. Terminal post.

FIG. 13.

Only one size and type of Joint Box (fig. 13) is used with this system. It is hot moulded in bakelite, and has a very fine finish. Four apertures are provided for side entry cables, the apertures being reduced in size to fit any cable used for domestic installations, or closed completely by means of sliding shutters. Two blanked holes in the base can be knocked out cleanly and easily to accommodate back entry cables, and by knocking out another blanked hole in the cover the box can be used as a neat ceiling rose. The flex is passed through the central pillar, and is gripped firmly when the cover is screwed down. Loose connectors are not required with this box, four terminal pillars being moulded into the base.

The Callender 'Kalibond' Wiring System.

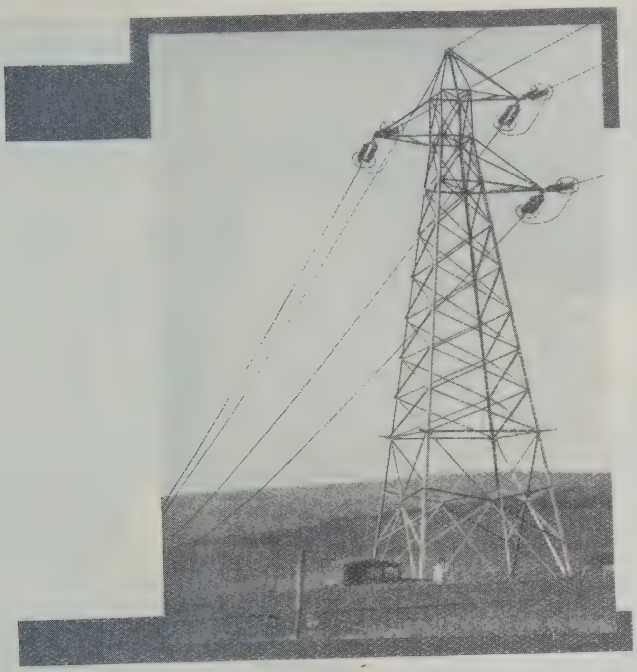
(Callender's Cable and Construction Co., Ltd.)

The system consists of single, twin, and three-core wires composed of pure copper conductors insulated with vulcanised indiarubber, sheathed with a special metal alloy, and provided with an uninsulated tinned copper bonding wire laid in intimate contact with the inner surface of the metal sheathing, together with patent metal junction boxes and all-porcelain connectors. There are no hygroscopic tapes on the wires, nor screws in the junction boxes—the system is all rubber, metal, and porcelain.

Before despatch from the works, each length of insulated core is immersed in water for at least 24 hours, then subjected for a period of 15 minutes to a pressure test of 1,000 volts alternating afterwards being tested for insulation resistance with a battery having a pressure of not less than 500 volts.

Callender 'Kalibond' metal sheated wire, although stiff enough to prevent sagging between fixing clips properly spaced, is easily bent round mouldings and cornices, and being of small dimensions has a particularly neat appearance when in position. The single core is insulated with drab rubber, the twin cores with one drab and one black, the three cores with one drab and two black.

By the use of 'braided' 'Kalibond' wire, which is made in various styles of decoration, a rearrangement of the electrical equipment of a room may be made at any time without spoiling the effect of an artistic scheme of decoration, or losing the wiring efficiency.



CALLENDER 132 KILOVOLT OVERHEAD TRANSMISSION

S.E. England Grid Scheme

The photograph shows a Single Circuit Tower at Sweet Hill, Patcham, near Brighton, being part of the Overhead Transmission Line erected by Callender's Cable and Construction Co., Ltd.

The company invite enquiries from users of electrical plant for Overhead Transmission work for small or extensive schemes in any country.

CALLENDER'S CABLE & CONSTRUCTION CO. LTD.
Hamilton House, Victoria Embankment, London E.C.4
Telephone. Telegrams, Callender, West, London.

Tele: Central 5241.

F. 2252

The Engineer.

ESTABLISHED 1856

1/- WEEKLY

The Metallurgist
IS A FREE MONTHLY SUPPLEMENT
to The Engineer,
EDITORIAL and PUBLICATION OFFICES:
28 ESSEX STREET, STRAND, LONDON, W.C.2

ANNUAL SUBSCRIPTION RATE £3 5 0

F. 2253

The Callender Junction Box.

(Pat. No. 276172.)

These boxes (made in three sizes) are of tinned brass. Push-in sections are provided at the sides of each box for the entry of cables, and pillars of tinned brass are provided for bonding the wire of the cables to the box. There are no bonding rings, screws, or loose parts to be assembled, or cutting tools to be used. The cable connections are made in the usual way with Midget all-porcelain connectors, or with standard porcelains if preferred. The bonding wires of the cables are twisted round the bonding pillars and secured by means of Midget connectors which form effective bonding clamps.

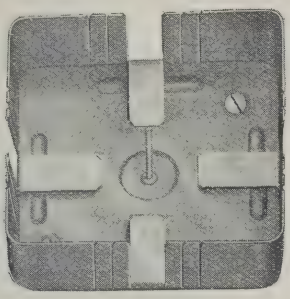


Fig. 14.

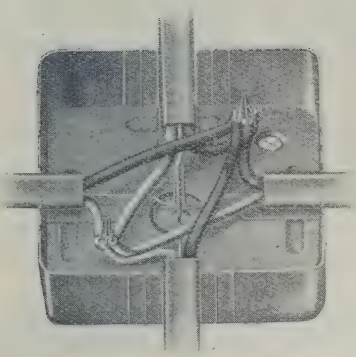


Fig. 15.

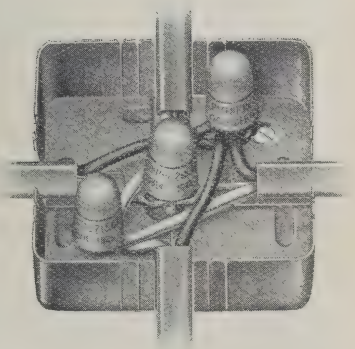


Fig. 16.

Fig. 14 shows one of the boxes with the sections pushed down to receive four twin cables, and fig. 15 the four cables brought into the box and connected up. Fig. 16 shows the twisted ends of the conductors covered with Midget connectors, which are made entirely of porcelain, and consist of a small cone having an interior thread which narrows sharply towards the apex: these connectors will joint up to three conductors each 3/·029 in size, by the simple action of screwing the connector on to the wire-ends.

Before the cables are introduced into the box a layer of a special preparation called 'Kalanoid'—an insulating and waterproof compound which remains plastic indefinitely—is placed in the box, and then, when the cables are introduced and the connectors screwed on, a further quantity of the 'Kalanoid' is worked thoroughly into the box to completely cover all the connections, and a final layer added. A cover placed over the whole is secured by means of its own natural spring without the aid of screws or clamps.

The C.T.S. Wiring System.

The cab-tyre sheath type of protected cable was originated by The St. Helen's Cable & Rubber Co. Ltd., Slough. It is generally known as C.T.S. It was designed for use in positions where ordinary protected cables often give trouble, principally as trailing or portable cables, and has since been found of value in all kinds of dangerous situations. (See also p. 2337.)

The conductors are usually insulated in accordance with the standards of the C.M.A., the whole then being sheathed with the same tough rubber material as is used in the manufacture of cab tyres. The principal characteristics of this rubber are durability, non-corrodibility, and non-absorbence. This type of cable is, therefore, now used in the numerous places where corrosion, moisture, hard wear, climatic or temperature changes are likely to be met with.

An advantage for some classes of work is that the elastic flexibility of the material prevents it from being kinked.

This class of cable is being largely used for general wiring. Being a portable solid system not needing any further protection, it is easily and cheaply run.

The use of C.T.S. without further protection of any kind is authorised by the Wiring Rules of the Institution of Electrical Engineers. In these Rules, C.T.S. is referred to as ' tough rubber compound covering.'

The Macintosh Wiring System.

(The Macintosh Cable Co., Ltd.)

The highly flexible properties, tough nature, and high permanent dielectric strength of this insulating material, known as ' Maconite,' have led to its adoption for nearly all classes of electrical conductors. The fact that twin cables of the now well-known 8 section can be mounted on the surfaces of walls, ceilings, etc., (either with or without the special wood fillet supplied) without being conspicuous, and at low cost, renders it particularly suitable for the wiring of small houses, flats, and working-class dwellings. Many housing schemes have utilised it for this purpose. Its insulation resistance is equal to high quality vulcanised rubber, but in the case of ' Maconite ' it is more constant and is unaffected by prolonged immersion in water or in a saturated solution of caustic soda. Resistance to breakdown pressure is also stated to be very high.

The following are details of tests to which Maconite has been satisfactorily subjected. Except where specifically mentioned, no tests of less duration than some thousands of hours are given :—

Extremes of temperature.—

(1) Freezing Maconite Cable in a block of ice for several days, and testing at 1,000 volts a.c. for 30 minutes. Then thawing out in running water and again testing at 1,000 volts a.c.

(2) Boiling in water and for 3 hours in a weak solution of caustic soda.

(3) Exposure for several months in temperature of 100° F. above boilers and brass furnaces.

In all these cases, not only did the cable withstand the tests, but its flexibility was found to be absolutely unimpaired at the end.

Exposure to Chemical Action.—

(1) Immersion in dilute uric acid.
(2) ,, copper pickling acid.
(3) ,, 50 per cent. solution of sulphuric acid.
(4) ,, 5 per cent. solution of nitric acid.
(5) ,, 50 per cent. solution of hydrochloric acid.
(6) ,, slaked slag water.
(7) ,, two samples of corrosive pit waters.
(8) Immersion in saturated solution of caustic soda.
(9) ,, saturated solution of common soda.
(10) ,, 10 per cent. solution of ammonia.
(11) ,, saturated solution of common salt.
(12) ,, sea-water.

In all cases the cables withstood the tests, the cables carrying direct current at 250 volts, two cables being on test in each solution, and being connected to the positive and negative poles of the system respectively. This was done in order to be certain that the material would withstand osmotic action.

The general results of the tests indicate that cables insulated with ' Maconite ' are particularly suited for installation in chemical works, steelworks, dye works, long railway tunnels, etc., etc., where corrosive fumes or liquids are always present.

The material may prove to be suitable as a sheath for cables in the place of lead alloys or bitumen. It is already being used as a sheath for a surface wiring system which requires no bonding or earthing.

Failures only occur at the points where it is jointed to other apparatus, such as junction boxes, switches, lamp fittings, and distribution boards. *The Macintosh Cable Company* have therefore produced a complete system embodying gas-tight fittings of the above nature, using patented ' Maconite ' glands which will ensure that no destructive gases or elements reach the metal fittings necessary for junction purposes.

A further use for which ' Maconite ' has proved highly efficacious is in the construction of multi-core cables for telephone and signalling purposes : its resilience and flexibility prevent destruction by the mechanical strains to which the cores of such cables are subjected. The same feature is also leading to its adoption for trailing and similar cables for colliery work, etc.

PRIMARY CELLS.

'SIEMENS' DRY CELLS.

These cells (*Siemens Bros. & Co., Ltd.*), originally known as the 'Obach,' have the exciting agent in the form of a damp paste. The following table gives the details of a few of the rectangular sizes. The outputs in ampere-hours are for continuous discharges through an external resistance of 10 ohms, until the P.D. across the terminals of the closed circuit has fallen to 0·75 volt.

Size.	Approximate Overall Dimensions (including terminal).	Approximate Weight.	E.M.F.	Initial Internal Resistance.	Output.
		Lb. Oz.	about	about	Amp. Hours.
M	$4\frac{7}{16}$ ins. × $4\frac{7}{16}$ ins. × $8\frac{1}{4}$ ins. high	10 0	1·5 volt	0·15 ohm.	260
N	$3\frac{5}{16}$ " × $3\frac{5}{16}$ " × $7\frac{1}{4}$ "	5 4	1·5 "	0·15 "	110
O	$2\frac{13}{16}$ " × $2\frac{13}{16}$ " × $6\frac{3}{4}$ "	3 0	1·5 "	0·20 "	40
P	$2\frac{1}{4}$ " × $2\frac{1}{4}$ " × $6\frac{1}{4}$ "	1 14	1·5 "	0·25 "	25

An improved form of high-tension dry battery comprises large cells capable of giving satisfactory service with multi-valve wireless receiving apparatus at a normal anode current not exceeding 10 milliamperes. Three sizes only are standardized, viz., 18 volts, 36 volts, and 72 volts. Spring clip terminals are fitted and tappings provided at suitable intermediate voltages.

SIEMENS INERT CELLS.

These cells, originally known as the 'Dura' cells, have the exciting agent in the form of an inert powder, which by the addition of ordinary clean drinking water becomes a paste, thereby giving a cell all the characteristics of a 'dry' cell, as described above.

Batteries and single cells are made up in this form, and the special utility of the construction is that the cells before being charged keep for an unlimited time in any climate, without deterioration, and may be converted at any time, as required, into a newly-made, so-called, 'dry' cell, with equivalent E.M.F. and output. This form is particularly suitable for use in those countries where it is essential to keep considerable stocks and where deterioration would be serious.

SIEMENS 'SACK' LECLANCHÉ CELLS.

These cells are wet cells using sal-ammoniac as the exciting agent, and are modifications of the porous-pot form, the depolarising material being enclosed in fabric instead of a porous-pot. This enables a lower internal resistance and higher output in ampere-hours to be obtained.

Improved results can also be obtained by using Siemens 'Siebrosal' as the exciting agent in place of sal-ammoniac crystals.

G.E.C. 'EXTRA SEC' DRY CELLS.

The cell only requires to be filled with water, without any exciting element, and the excess of water emptied out. Specially suitable for use in tropical climates as the cell can be stocked for considerable periods without deteriorating, and only becomes active when charged with water.

G.E.C. 'POROUS POT' LECLANCHÉ CELLS.

Specification.—Porous pot of high porosity, thoroughly tested and filled with a mixture of best selected peroxide of manganese and crushed carbon; positive plate of carbon throughout; without lead cap; well-amalgamated, solid drawn zinc rod of 98 per cent. purity; E.M.F., 1·5 volts.

Directions.—Place porous pot and zinc in glass jar and fill two-thirds full with solution of sal ammoniac, 2 ozs. to the pint of water.

G.E.C. 'CARSAK' LECLANCHÉ CELLS.

Specification.—Positive element consists of a porous sack which surrounds the carefully selected granules of carbon and manganese peroxide, which are very tightly packed. The 'pot' is practically unbreakable and its cylindrical form is maintained by a cap and base of hard material. The zinc is a semi-circular plate of ample dimensions, well amalgamated. 'Carsak' cells have a very low resistance, are thoroughly reliable and constant, and have a long life. They are particularly suited for telegraph, telephone, bell and signal circuits and for all positions where a low resistance Leclanché cell is required.

The 'Carsak' cell is designed to withstand severe transit conditions, and is therefore particularly suitable for export.

ACCUMULATORS.

Compiled from information supplied by the Hart Accumulator Co. and the Chloride Electrical Storage Co.

Storage cells should be placed in a well-ventilated room, preferably with louvres at the top to allow of the free escape of the gases evolved during charging; and the room should also be well lighted, both naturally and artificially, provision being always made for a hand inspection lamp to enable cells to be examined individually. The best working temperature is 50° to 80° F. Windows should face north to avoid direct sun rays. The room should not be exposed to undue draughts, as this promotes evaporation of the electrolyte, and it then becomes necessary to add an abnormal amount of water to keep the plates well covered with electrolyte.

Insulation.

Good insulation is essential. The cells should be mounted on insulators unless the boxes, if glass, are provided with feet. Above 100 volts the stands should be mounted on insulators also.

Assembly.

Cells with a capacity of less than 1,000 ampere-hours are assembled in glass boxes. The smaller sizes may be provided with feet or are mounted on separate insulators with lead discs, indiarubber washers or wood strips intervening. The larger sizes are mounted on wooden trays filled with sawdust. Above 1,000 ampere-hours capacity, cells are assembled in lead-lined wooden boxes.

The battery should be mounted on strong wooden racks or stillage protected with acid-proof paint with sufficient space between each cell to allow for insulation and cleaning.

The cells should be placed with the edges of the plates to view (if in glass boxes) and should only be arranged in double rows when they are accessible from each side.

Polarity.

The positive pole is painted red and is that to which the brown plates are connected: the negative pole is painted black and is that to which the grey plates are connected. The positive pole of each cell must be connected to the negative pole of the next cell throughout the battery, so that there will be a disconnected positive terminal at one end and a disconnected negative terminal at the other end of the battery. These are the battery terminals, which are

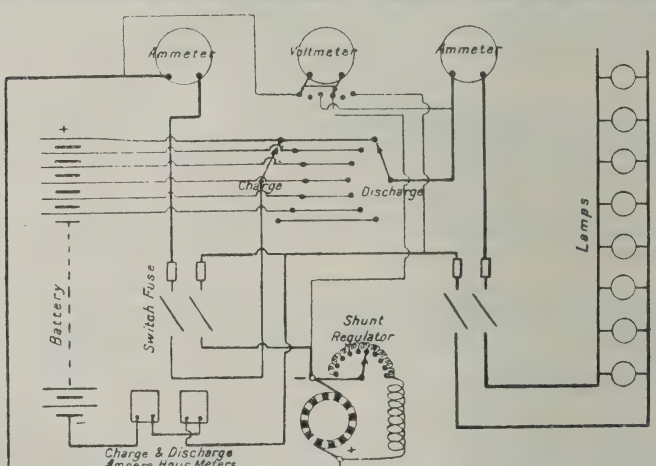


FIG. 17.

connected to the dynamo as indicated in the diagram of connections in fig. 17. Cells should never be connected in parallel so as to form single cells of larger capacity; but if necessary they may be formed into batteries, and the batteries connected in parallel by coupling their ends.

Regulating cells should be at the negative end of the battery.

An automatic switch should be provided in the charging circuit.

THE CHEMIST AND DRUGGIST

Established 1859.

Published Weekly.

The Imperial and International Journal of pharmacy and of the drug, chemical and allied trades.

CIRCULATION 10,092 COPIES WEEKLY
(Net Sales Certificate)

THE advertising medium for producers of all kinds of chemicals and pharmaceutical preparations.

Used by manufacturers of chemical and pharmaceutical plant such as sifters and mixers, autoclaves, emulsifiers, filter presses, condensers, evaporators, stills, vacuum pans and laboratory apparatus.

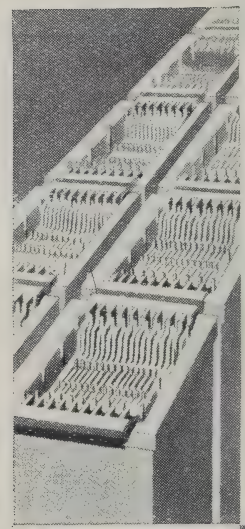
SUBSCRIPTION 20/- PER ANNUM.

Rates for advertising in plain black and white and in colour direct from The Publisher,

THE CHEMIST AND DRUGGIST,
28 Essex Street, Strand,
London, W.C.2.

Branch Offices :—
4 Cannon Street, MANCHESTER.
54 Foster's Bldgs., High Street, SHEFFIELD.
19 Waterloo Street, GLASGOW.
MELBOURNE and SYDNEY (Australia). AUCKLAND (New Zealand)

efficiency of the cell



BECAUSE you know the cell's efficiency depends on the chemical re-action inside, you take great care as to the battery-plates. But remember that the electrolyte is an equal co-partner in the action. The world's leading engineers rely on BAA—it is made from fine, natural Brimstone only, and distilled water. And it is most carefully tested.

MEDAL AWARDED - ST. LOUIS 1904

Telephone:- Monument 3874
Works:- Stratford, E., and Morriston, Glam.

BAA. REG.D
THE PUREST & BEST ACCUMULATOR ACID

Brimstone-made, by
F.W. BERK & CO., LTD
LONDON, E.C.3.

Dynamo.

The dynamo used for charging should be shunt-wound, and the electromotive force required will be 2·6 volts for each cell in series. Before connecting up the dynamo it should be tested for polarity, which may be done as follows: Place two clean strips of sheet lead in a stone jar filled with dilute acid; separate them about two inches by means of a block of wood; connect one strip to one terminal of the dynamo and the other strip to an incandescent lamp which has its second terminal connected to the second terminal of the dynamo. After the dynamo has been running for a few minutes one of the lead strips will become brown, showing that the dynamo terminal connected to this strip is the positive terminal.

Instruments.

In addition to the voltmeter and to the ammeters in the charging and discharging circuits, it is very desirable to have in circuit a pair of ampere-hour meters, one to record the charge and the other the discharge. These may be connected in series as shown in the diagram of connections, and fitted with pawls which will prevent backward running, so that there will be a continuous record of both charge and discharge.

Water.

The water used for the cells should be distilled, as this avoids any danger of impurities entering the cell.

Acid.

The electrolyte in all storage batteries—lead-acid type—is dilute sulphuric acid with a specific gravity of 1·200/1·215. The method of manufacture is immaterial provided the finished article when diluted does not contain more than 0·001 grammes per 100 c.c. of chlorine or copper, twice that amount of iron, or one-third that amount of arsenic.

On the continent, sulphuric acid prepared by the catalytic process is largely used, but as this acid is usually prepared from iron pyrites the iron contents are higher than that made from natural brimstone. In this country *Messrs. F. W. Berk & Co., Ltd.*, the manufacturers of the well-known B.A.A. Accumulator Acid, prepare it from distilled water and the best brimstone, producing thereby an acid of the highest purity and best quality.

Preparing Electrolyte.

To dilute concentrated sulphuric acid it must be mixed with about 4½ times its volume in a suitable acid-proof vessel. The acid must be added to the water, never the water to the acid on account of the heat generated. Careful stirring is necessary. Correction for temperature must be made when testing the specific gravity to the extent of one point added to the hydrometer reading for every 2·7° F. above 60° F. The mixture must be allowed to cool before using it.

Filling up.

When everything is ready for charging, the acid must be put into the cells to a height of not less than one inch above the plates. Charging must proceed as soon as possible afterwards.

Charging.

Before the battery is switched into circuit with the dynamo care should be taken that the field-magnets are properly excited and the proper E.M.F. generated. The initial charge should be continued for not less than 36–40 hours at the normal charge rate specified by the makers for the battery in use. It has been recommended that the first 12 hours' run be continuous. Some makers, *e.g.*, The Chloride Electrical Storage Company Limited, now supply their batteries in a short first charged condition, which entails only 15 hours' initial charge at a prescribed rate, or alternatively two runs of 8 hours each. During the early portion of the initial charge the specific gravity will drop 20 to 50 points, and the charge cannot be considered complete until the specific gravity has maintained a maximum stationary value for at least two hours in the case of short first charge batteries, or five hours in the case of ordinary batteries. As the plates absorb the acid slowly, the level of the electrolyte will fall at first, and to compensate for this, more acid should be added, but any subsequent evaporation should be made good by the addition of water only.

Discharge.

The permissible rate of discharge depends upon the number of hours for which the discharge is required—the longer the time the smaller the rate. The product of rate into time is, however, not constant, but is greater with low rates of discharge than with high. Thus in the Hart Accumulator, 25-plate Lighting Type, the rates of discharge for 5, 7, and 10 hours are 126, 99, and 75·6 amperes respectively, giving discharge capacities of 630, 693, and 756 ampere-hours. The

discharge rates for different times are always given by the makers, and should never be exceeded, the ampere-hours taken out being recorded on the discharge ampere-hour meter. The accompanying tables have been prepared for users of batteries supplied by The Chloride Electrical Storage Company, Limited, and may be considered applicable to all Planté batteries.

CAPACITIES OF PLANTÉ TYPE STATIONARY BATTERIES AT VARIOUS RATES OF DISCHARGE.

Capacity in Ampere-Hours (expressed as a percentage of the ampere-hours available at the 5-hour rate) When Discharging in									
1 Hour.	2 Hours.	3 Hours.	4 Hours.	5 Hours.	6 Hours.	7 Hours.	8 Hours.	9 Hours.	10 Hours.
60%	76%	86%	93·8%	100%	105·5%	110·1%	114%	117·5%	120%

Capacity in Amperes (expressed as a multiple of the amperes available at the 5-hour rate) When Discharging in									
$\frac{1}{10}$ Minute.	1 Minute.	3 Minutes.	5 Minutes.	10 Minutes.	15 Minutes.	20 Minutes.	30 Minutes.	45 Minutes.	60 Minutes.
7·5	6·0	5·91	5·82	5·61	5·35	5·10	4·47	3·58	3·0

Recharging.

To obtain longest life from a battery it should be recharged whenever the cells are about two-thirds discharged, determined by specific gravity and voltage, also by ampere-hours used (where ampere-hour meters are installed). A battery may be charged at any convenient rate between the conventional 'normal' specified by the maker, and half that value. In any case the charge should be completed at the lower rate. The ampere-hours put in should exceed the number taken out by about 10 per cent. ordinarily, but at least once a fortnight the battery should be brought up to a thorough healthy condition regardless of ampere-hour meter readings—all cells uniform and the extent of gassing, the specific gravity of the acid, the voltage, and the colour of the plates. Records should be kept on suitable forms of the individual voltage and specific gravity of each cell periodically and details of each charge and discharge.

Level of Electrolyte.

The level of the electrolyte must never be allowed to fall to less than half an inch above the top of the plates. Any loss due to evaporation must be made up by adding pure distilled water as above explained. Evaporation will be greatly retarded by the use of glass spray arresters covering the top of the plates or by an anti-spray oil film. Spray arresters must be so placed as to be easily removable for the purpose of examining the cells. Acid should not be added, except under the maker's advice.

Specific Gravity.

The specific gravity of each cell should be taken and recorded at least once a week, and if the consecutive readings so obtained are carefully compared, it is possible at once to detect a backward cell and take steps to remedy what is amiss. The hydrometer should be long enough to float in about two-thirds of the total depth of the electrolyte. The specific gravity varies to some extent with the temperature, and during the summer months will be rather lower than during the winter. The standard is taken at 60 deg. Fahr.; for every 2·7 degrees above this 1 degree of specific gravity may be added; for every 2·7 degrees below 1 degree of specific gravity may be deducted.

Weak Cells.

The indications of weak cells are bad colour of the plates, irregularity of gassing, or failure to gas, and a fall of specific gravity of electrolyte (obtained by the hydrometer), or voltage (obtained

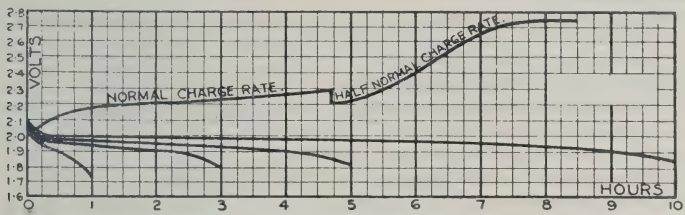


FIG. 18.—Typical Curves of Voltages during charges following 5-hour discharges, and curves for discharges at 1, 3, 5, and 10-hour rates.

by a cell-testing voltmeter) below that of other cells. Weak cells should be inspected for 'short circuits.' These may be caused by the lodgment of material between the plates or by the positive and negative plates touching each other. After removing the short circuit the weak cell should be cut out of circuit for a few discharges and put into circuit again for the following charge. Care must be taken not to allow the negative plates to dry out or heat by exposure to the atmosphere.

Voltage Characteristics.

Fig. 18 indicates the characteristic rise in voltage on charge, and fall in voltage during discharge according to the rate of discharge in amperes. The curves are applicable to any modern Planté battery.

Buckled Plates.

Buckled plates should be removed and straightened by pressing between thick boards. The positive plates are most easily straightened when fully charged, as they are then softer and not so liable to crack. The negative plates should be kept in dilute acid while the positives are being overhauled, as otherwise they may get heated by oxidisation with the air.

Battery Rooms.

SIZE OF BATTERY ROOMS REQUIRED FOR 25, 50, AND 100-VOLT 'HART' BATTERIES.

Capacity in Amp. Hours.	25 Volt.		50 Volt.		100 Volt.	
	Single Tier.	Two Tier.	Single Tier.	Two Tier.	Single Tier.	Two Tier.
	′ ″ ′ ″	′ ″ ′ ″	′ ″ ′ ″	′ ″ ′ ″	′ ″ ′ ″	′ ″ ′ ″
140	6 6×3 9	3 6×4 3	6 9×5 0	7 0×4 3	13 3× 5 0	13 3× 4 3
280	9 3×3 9	5 0×4 3	10 0×5 0	10 0×4 3	12 9× 8 9	9 9× 6 0
420	6 9×5 0	6 9×4 3	13 0×5 0	13 0×4 3	16 6× 8 9	12 6× 6 0
630	9 0×5 0	9 0×4 6	11 6×8 9	9 0×6 6	16 9×12 6	16 9× 6 6
770	10 6×5 0	10 6×4 6	13 3×8 9	10 6×6 6	19 6×12 6	13 3×11 0
1,050	13 3×5 0	13 3×4 6	17 0×8 9	13 3×6 6	25 3×12 6	17 0×11 0

In preparing this Schedule the number of cells is as follows: 25-volt circuit, 13 cells; 50-volt circuit, 27 cells; 100-volt circuit, 54 cells. The height of room in each case should not be less than 8 feet.

Idle Batteries.

When a battery is not required for several weeks it should be thoroughly charged and the plates well covered with electrolyte. The cables should then be disconnected at the ends of each row to avoid any possibility of electrical leakage. Cells will remain in this condition for several months. If a charge can be given once every six or eight weeks it is best to do so; this should be given at 75 per cent. of the normal rate and continued until the full charge is completed, and the plates gas freely. If a battery is to be used at infrequent intervals, it should be given an extended charge once every two weeks.

Brush Gear.—The brush gear is of simple and rigid construction, split horizontally, and is supported by a separate frame. The brush-holders, which are of the direct-acting box type, are bolted to a carrier arm which keeps them in strict alignment, and are easily adjustable. The insulation is micanite throughout. When the brush rocker has been once set in the working position, it remains fixed for all loads up to the maximum permissible overload.

Care of Direct-Current Generators.

Steadiness.—It is important that the machine be firmly fixed to a good foundation. A machine may be run successfully on an upper floor, if supported on solid joists either of timber or iron. The machine will spark and the brushes and commutators wear unduly if there is vibration.

Commutator and Brushes.—These will always repay careful attention. If the brushes are left to wear unequally, and the commutator is not cleaned, the machine will soon commence to spark badly, and both brushes and commutator will wear rapidly. If, however, these parts are attended to in accordance with the following instructions, the wear and tear will be very small.

Adjustment of Brushes.—The brushes must be set truly on a diameter so that the point of one is diametrically opposite the point of the other. Gauze and metal brushes as they wear must be advanced to maintain the same distance, and consequently to bear the same angle on the surface of the commutator. They must be advanced frequently, even though the wear may be very small, otherwise the brush will not bed properly on the surfaces of the commutator when advanced. Each brush must always bed equally well at the point, or toe, and at the heel. If brushes are allowed to wear too much before advancing (but this should not be), they must be taken out and filed to fit truly. In filing a brush, clamp it firmly, and file by taking long strokes with a file always in one direction. If it is the toe that requires filing, after filing to fit take off the sharp edge, leaving the toe always $\frac{1}{16}$ inch thick. One brush often wears faster than its fellow (or fellows), and must be set up accordingly. Moreover, sometimes a whole set of brushes on one side of the commutator will wear faster than those on the other side, and this must be allowed for, or the brushes will cease to be on a diameter, and sparking and flatting of the commutator may result.

Carbon Brushes are usually set radially to the commutator and therefore require no diagonal adjustment. They require periodical inspection, adjustment and cleaning, as the toes or heels usually burn away and a new surface giving the full contact area of the brush must be again obtained. The best results with carbon brushes of ample contact area and good designs fitted with flexible tails are obtained by keeping the brushes absolutely clean and free from dust and the commutator true and polished with polishing paste.

Flats.—The most common source of the formation of little 'flats' upon the surface of the commutator is the brushes being out of adjustment. Another cause is vibration of the brushes, due primarily to vibration of the machine, owing to improper foundations or a badly-jointed belt. Flats are sometimes caused by the pressure of the brushes being reduced to too low a limit.

Pressure.—It is well to have the pressure as light as possible, so long as no sparking and no vibration takes place at the brushes.

Oiling and Cleaning the Commutator.—In the case of metal brushes use a little good mineral oil or vaseline, applied with a piece of rag. Do not on any account apply oil with a feeder, or on a piece of waste. A commutator well looked after will acquire in time a dark-coloured, smooth polish. When this takes place run a piece of No. 0 0 emery cloth on the commutator occasionally before starting to work. In flour-mills and dusty places it is necessary to clean the commutator with a little benzoline every day before starting, or, if run continuously, during the run, taking care not to drench the commutator; also to apply emery cloth about once a week as just directed.

Belting.—Where practicable, the drive should be arranged so that the under side of the belt is the tight or driving side. A flexible belt should always be used. Leather-linked belting, arched on the face next the pulleys and flat on the outside, is very suitable. Such a belt should *not* be taken off after working. Single belting with double edges makes a good cheap belt for ordinary use, and may be jointed with a butt joint, secured by belt fasteners. If an ordinary single belt is used, it should be straight and of good quality. It should be jointed with a laced butt joint, until it has stretched, and then a proper spliced and cemented joint made. A dynamo belt must run evenly. A belt which lashes either up and down, or from side to side, should not be tolerated.

Oiling the Bearings.—Adjust the sight-feed lubricator to give about 3 to 10 drops per minute. The amount required varies with the quality of the oil used and the load on the machine. If the machine has been standing, give it a few extra drops before starting. It is well to use an oil feeder made entirely of copper. Keep the machine clean—free from copper dust and waste oil.

Should the commutator wear out of truth or become very uneven (scored), take out the armature and turn up the commutator in a lathe, being careful to use a fine-pointed tool and not to take heavy cuts; afterwards polish with emery cloth. If the commutator should be only scored, it may be dressed in the machine with coarse, and afterwards fine, emery cloth, finishing with No. 0. Slight 'ringing' does not matter.

ALTERNATING-CURRENT GENERATORS.

While many D.C. machines nowadays are driven by steam turbines through gearing, a similar arrangement is adopted with alternators in spite of the fact that it is not very difficult, either from an electrical or a mechanical point of view, to construct medium-size high-speed turbo-alternator sets. The geared alternator is adopted from motives of economy, as the steam turbine can then be designed for speeds of over 3,000 r.p.m.

The Mather & Platt Geared Alternator.

(Mather & Platt, Ltd.)

This machine is constructed on the following lines:—

The stator consists of a cylindrical core, built up of thin steel laminations of the highest electrical and magnetic quality, insulated from each other, and firmly clamped together in a rigid cast-iron frame.

The windings are fully insulated before insertion in the slots, the latter being usually of the open type. High-tension coils are insulated with moulded mica, and low-tension coils with standard mica sheet mechanically protected by leatheroid. In all cases the insulation forms a complete sealed tube completely enclosing the straight part of each coil, and extending for a considerable distance beyond the stator core at each end. The coils are held in position by hard wood wedges.

In normal machines, the poles are completely laminated, the core and tip being stamped in one piece. The built-up poles are usually bolted to the rim of a cast-steel hub. In special cases, however, the above construction may be modified.

For small machines with few poles, the field coils consist of cotton-covered wire or strip. Each coil is provided with terminals which relieve the last turns of wire from all stress. The coils are thoroughly dried and impregnated in vacuo with an oil- and moisture-resisting varnish. For larger machines copper strip bend on edge is used in all cases where it is practicable.

The shaft is of mild steel of ample size and stiffness. Each bearing is of the pedestal type, with a cast-iron bush of large diameter and length, lined with white metal, and lubricated automatically by two oil rings running in an oil well of large capacity. The oil channels and escape grooves are carefully designed to eliminate oil-flinging and creeping, and the overflow is so arranged that frothing or churning of oil in the oil well does not cause any drip. The pedestal and bearing bush are split horizontally.

The slip rings are of bronze, insulated entirely by mica. The brush-holders are of the direct-acting box type, and the whole brush gear is simple and strong.

ALTERNATING-CURRENT TURBO-GENERATORS.

The G.E.C. Turbo-Alternator

(with combined axial and tangential ventilation).

(The General Electric Co., Ltd.)

The following is the specification of a G.E.C. turbo-alternator of the standard type:—

The alternator is of the enclosed type, with drum-type rotor and forced ventilation. Shaft and exciter are included, but no bearings or coupling, excepting where this is specifically stated.

Stator Frame.—The stator frame is a massive casting of box section, accurately bored to accommodate and clamp tightly the laminations. Stout end-plates are used for this purpose, and are secured in such a way that no bolts pass through the stampings, thereby avoiding a considerable amount of iron loss. All longitudinal ribs are fitted with dovetailed keys to which the stampings are secured. The casing rests on two feet, which are connected to the main body by strong ribs to avoid flexure or vibration.

Laminations.—The laminations are accurate stampings of special low-loss iron, punched out of large plates with carefully ground tools to avoid burrs. They are re-annealed after punching, and insulated as a last operation. No filing or scraping takes place after the laminations are assembled, as this causes much higher iron losses due to eddies than in ordinary machines For the same reason, the insulation between plates is increased by sheets of strong paper placed at suitable intervals.

Stator Winding.—The stator winding consists of a lap winding of the diamond coil type.

The coils are completely insulated before insertion in the slots, which are of the open type. The insulation of the coils is entirely class ' B ' insulation and consists of mica and micanite.

The insu'ation of the slot portion consists of a micanite tube carefully moulded on to the coil by a special process, which eliminates the possibility of air spaces. The end turns are insulated by a suitable number of layers of mica silk tape wrapped round the coil, and the coil is finally finished off with a layer of protective tape.

Clamping of Stator Windings.—The end windings are suitably braced, which prevents any possible movement of the coil under the severe stresses which accrue should a sudden short circuit take place.

Ventilation of Stator.—Fans are arranged at both ends of the rotor, to draw air through channels in the end shields from ducts in the foundations and force it into the space enclosed by the guards of the stator winding. From there, after cooling the overhanging parts of the stator coils, the air is pressed into pockets in the stator casing, and through the vent ducts of the stator core. It enters these vent ducts on the outer periphery of the core, and is guided by suitably shaped spacing pieces first inwards to the teeth and stator coils and then outwards to the outer periphery. The air escapes from the ducts through the stator casing to the outlet opening, a funnel-shaped orifice, through which the cooling air finally leaves the machine. This orifice can be arranged either at the top or below the machine, according to local requirements. The employment of an air-cooler is recommended, but a decision as to the type can only be arrived at in each particular case with a knowledge of the local conditions.

Rotor Winding.—The winding consists of coils which are completely shaped before they are embedded in the slots. Strong and ample insulation is provided between the winding and the rotor, as well as between the conductors themselves, particular care being taken to avoid crushing action due to centrifugal force. The insulation used is practically incombustible.

Clamping of Rotor.—The rotor windings are held in the slots by metal wedges. The overhanging end-connections are enclosed by steel shrouds of ample strength to withstand with safety the centrifugal stresses at maximum speed. These are put on with a moderate shrinkage. The shrouds are located at one end on a spigot provided in the core, at the other end on strong steel end plates carried from the shaft. Balancing grooves are provided at both ends of the rotor. The windings are entirely boxed in and protected from injury.

Ventilation of Rotor.—The cooling air enters the rotor at both ends through channels underneath the stator fans and passes under the rotor end winding into axial ducts arranged below the rotor coils. It escapes from these ducts through radial vent spaces, provided at intervals in the rotor core, into the air gap of the alternator, and leaves the machines through the stator core and casing.

Slip Rings.—The current is supplied to the winding through two steel slip rings shrunk on to an iron sleeve, and insulated therefrom by a moulded-on mica bush. Any danger of distortion or of getting out of truth of the rings is thereby avoided. Particular attention is paid to the leads connecting the rings to the windings. They are sunk into a groove in the shaft and protected by a key-shaped cover.

Terminals.—The terminals are usually arranged at the bottom of the shell, the cables being brought up to them from the basement. They are thus in a naturally protected position without necessitating unsightly boxes in the engine-room. The neutral point of each phase is brought out separately.

Rotor Construction.—The rotor is of the drum type, the winding being distributed in slots all round the periphery. The rotor body and shaft are one solid forging. This design has been adopted because a solid forging gives the strongest possible mechanical construction. The slots are milled out of the solid.

Journals.—The journals are of ample dimensions to run without overheating. They are finished off after the whole of the rotor is completed to ensure a true surface. Ample oil flirts are left at both sides, working in rings supplied with the bearings by the makers of the turbine.

Exciter.—The exciter is carried from the bearings at the slip-ring end. It is specially built for high speeds, and is usually compound wound.

Tests.—Numerous tests are carried out during the process of construction to see that the materials are satisfactory at all stages. Amongst these tests are the following :—

(*a*) Breakdown test for tubes after inserting them in the core.

(*b*) Test between adjacent conductors in a slot, not less than ten times the working voltage between conductors.

(*c*) Between finished coils and between any coil and earth, twice working voltage plus 1,000 volts, but in any case not less than 2,000 volts.

(*d*) Rotor winding tested to earth with 2,500 volts when all connections are made to the slip-rings and after balancing test.

(*e*) Static balancing on flat rails.

(*f*) Dynamic balancing at full-speed.

(*g*) Synchronous run at full speed to measure iron, windage and excitation losses. During this test the open-circuit curve and the short-circuit curve are obtained.

(*h*) Overspeed test at 15 per cent. above running speed.

(*i*) Oscillograph of wave-form on open circuit.

The G.E.C. Fraser & Chalmers Turbo-Alternator Set.

(*The General Electric Co., Ltd.*)

In fig. 19 is shown a close-up view from the alternator end of a G.E.C. Fraser & Chalmers Turbo-Alternator, being one of five 22,500 kVA. alternators installed at Nechells Power Station,

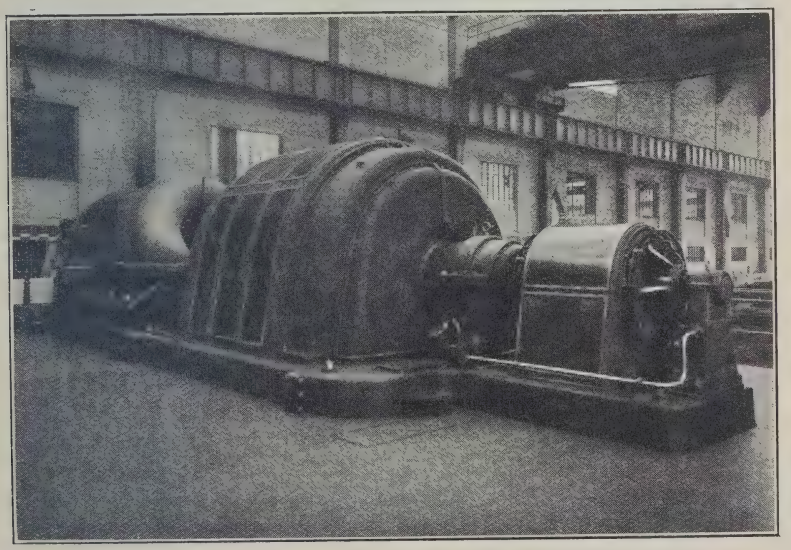


Fig. 19.—Fraser and Chalmers Turbo-Alternator Set.

Birmingham. Each set is capable of a maximum continuous output of 22,500 kVA. when running at a speed of 1,500 r.p.m. and with power factor ·66.

The G.E.C. Fraser & Chalmers Geared Turbo-Alternator Set.

Fig. 20 shows two 1,250 kVA. G.E.C. Fraser & Chalmers geared Turbo-Alternators. The turbines are of the back pressure type running at 6,000 r.p.m., and drive through gearing, two alternators, which run at 1,500 r.p.m., giving a supply at 400 volts, 3 phase, 50 cycles.

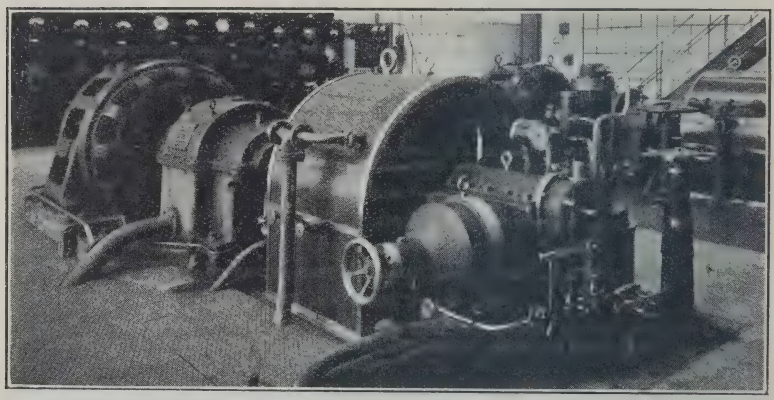


Fig. 20.—Fraser and Chalmers Geared Turbo-Alternators.

The Tirrill Automatic Voltage Regulator.

(The British Thomson-Houston Co. Ltd.)

Fluctuating loads, and the time-lag inseparable from manual control of generator voltage, may cause serious variation in a generating station voltage. This regulator (fig. 21) automatically

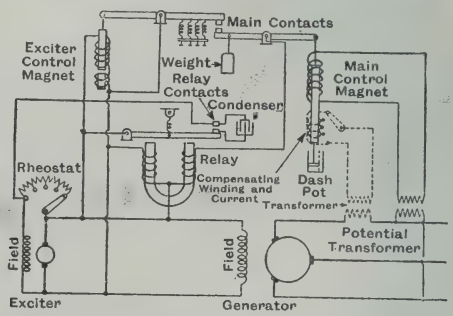


Fig. 21.—Elementary Diagram of Type T A Regulator, connected to Three-Phase Alternator, showing (in dotted lines) Compensating Coil and Current Transformer.

and instantaneously regulates the voltage of either alternating- or direct-current generators. In fig. 21, the rheostat in the field circuit of the exciter results in the alternator generating only about 60 per cent. of its normal P.D.; when it is short-circuited, the P.D. of the alternator greatly exceeds the normal value. The action of the Tirrill regulator is to vary the duration of the periods

The Engineer.

Established 1856. Registered as a Newspaper

PROVIDES
THE LATEST AND THE
MOST RELIABLE
INFORMATION
FOR
ENGINEERS
IN ALL BRANCHES OF THE PROFESSION.

The last issue in each month contains a Supplement, **The Metallurgist,** *a journal devoted to the science of metals, ferrous and non-ferrous.*

Subscriptions

			Per annum
INLAND, YEARLY			£3 5 0
CANADIAN SUBSCRIPTIONS.	Thin paper edition	...	£2 18 6
	Thick paper edition	...	£3 3 0
FOREIGN SUBSCRIPTIONS.	Thin paper edition	...	£3 3 0
	Thick paper edition	...	£3 7 6

'THE ENGINEER'
DIRECTORY AND BUYER'S GUIDE

SENT FREE ON APPLICATION.

Heenan

THE "HEENAN" SYSTEM OF REFUSE DISPOSAL AND REFUSE UTILIZATION HAS TO ITS CREDIT

The Largest Number of Successful Installations in the World.

PLANTS SUPPLIED TO MEET ALL REQUIREMENTS—LARGE OR SMALL

Specialists in

TOTAL INCINERATION PLANTS; COMBINED SEPARATION AND INCINERATION PLANTS; SCREENS AND SORTING BELTS; MECHANICAL HANDLING APPLIANCES AND STRUCTURAL STEELWORK

PLEASE SEND FOR OUR LITERATURE

HEENAN & FROUDE

LIMITED

WORCESTER • ENGLAND

F. 2267

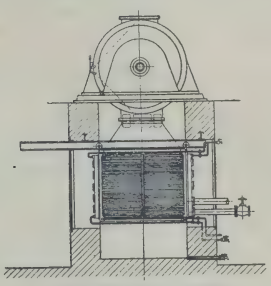
of open- and of short-circuit of this exciter field resistance. In the diagram, the two main or floating contacts are fixed respectively to two pivoted levers. The top contact lever tends to be kept depressed by springs, which are opposed by the downward pull of the exciter control magnet. The bottom contact lever has a counterweight balancing a core which is acted upon by the main control magnet energised from the alternator terminals. The pivoted armature of the relay carries one of the rheostat short-circuiting contacts which the spring tends to keep closed. The middle point of the relay winding is connected to an exciter terminal, one end to the other exciter terminal permanently, and the other end intermittently through the main contacts. When these contacts are closed there is no magnetic effect in the relay, because the two portions of the winding neutralise each other and the exciter rheostat remains short-circuited. The rising generator voltage then increases the current in the main control magnet sufficiently for the core to rise and open the main contacts. The relay magnet now becomes strongly excited, and, its armature being attracted, opens the short-circuiting contacts across the exciter field rheostat terminals, the condenser reducing sparking. The regulator is thus in a constant state of vibration and responds instantaneously to changes in alternator P.D. The function of the compensating winding on the control magnet (energised through a current transformer) is to eliminate circulating currents between alternators in parallel by reducing the excitation of an over-excited generator and *vice versâ*.

Air Coolers.

The 'Heenan' Patent Closed Circuit Air Cooler.

(Heenan & Froude, Ltd.)

This cooler, figs. 22-24, which is designed for the ventilation of turbo-alternators, comprises a tubular air cooler built generally on the lines of a surface condenser arranged in the alternator foundations and coupled up to the bottom discharge of the alternator by suitable ducting.



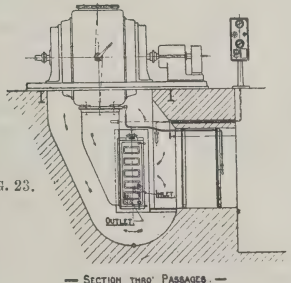
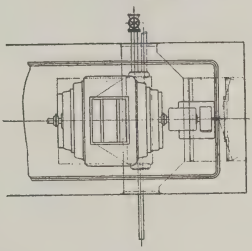


Fig. 23.



Figs. 22 & 23.—Arrangement of 'Heenan' Closed Circuit Air Cooler.

The arrows show the direction of the air.

Fig. 24.

After passing through the cooler the air returns to the alternator *via* the end shield, the fans circulating the air being carried on the rotor.

With this system the same air is continually recirculated through the windings of the alternator, so that the only dirt that can be deposited is that contained in the initial volume of air, thus the necessity for air filtration is obviated.

With an unfailing and clean supply of water the circuit can always remain a closed one, but under actual working conditions it has been found necessary to provide for cooling in the event of a failure of the water supply or other eventuality, and the arrangement, therefore, embodies a system of dampers and emergency air doors which cut out the air cooler and permit external air to enter the foundations, the hot air in this case being discharged direct to atmosphere.

Dampers are fitted to the top and bottom discharges of the alternator, which are interconnected and operated by one lever, and arranged so that if one set is closed the other set is open. Connected to the same lever is the trip gear for the emergency air doors, which are balanced and which open automatically, due to the difference in pressure, immediately the catch is released. As soon as the trouble has been diagnosed and rectified the emergency doors are closed and the alternator dampers changed over for closed circuit running.

A thermostatic alarm device is fitted in conjunction with the system, which gives visible and audible warning to the attendant should the temperature of the air leaving the cooler reach a predetermined figure. This alarm device consists of a dial type thermometer with a klaxon horn, red lamp and relay, arranged on a panel either for wall or floor fixing. The thermometer is connected to a bulb in the foundations, and as soon as the dial hand reaches a certain point the relay brings into operation the klaxon horn and lights up the red lamp, so warning the attendant that the air temperature is rising.

In regard to the construction of the coolers, this follows condenser practice in so far as the coolers consist of a nest of tubes held between tube plates, the joints being formed by screwed ferrules and packing. The water headers fitted with division plates and branches for inlet and outlet water connections are also in line with condenser practice, but the tubes in these coolers are gilled at close pitch, the gills being soldered to the tubes in one operation for each length of tube, this construction giving the highest heat transference efficiency, which has, of course, been ascertained chiefly by experimental work.

Another departure from an ordinary condenser is the addition of secondary tube plates in front of the main tube plates, so that a dead air chamber is formed at either end of the cooler into which any leakage from the main tube plates would drain instead of being carried over with the air into the windings. Rubber bushes are used in these secondary tube plates to form an air- and water-tight joint with the tubes.

Air cocks are fitted to the top of the water headers and drain pipes to the dead air chambers. The tubes are of brass (Admiralty mixture), solid drawn and tested to an internal pressure of 2 tons per sq. in.

The tube plates are also brass, of ample thickness to allow for a suitable stuffing box and screwed gland. The water headers are of cast iron of ample thickness to withstand any pressure they are likely to be subjected to, and are fitted with hinged doors, so that the tubes and tube plates can be cleaned from time to time without breaking the joints of the main header or pipes.

The gills are of thin copper, and both gills and tubes are tinned in the process of soldering the gills to the tubes, so that the danger of corrosion is well catered for.

Figs. 22-24 show the arrangement of a cooler in the foundation, from which it will be noted that the cooler itself is carried on rollers, which bear on the flanges of a steel joist, the latter being extended and supported outside the foundations, so that the complete cooler unit can be removed for overhaul or cleaning when necessary, with a minimum amount of trouble.

Inspection doors are provided on either side of the foundations to permit the removal of the cooler and also to give access to the foundations.

Air Filters.

The 'Heenan' Patent Air Filter.

(Heenan & Froude, Ltd.)

The overload capacity of turbo-alternators and generators is dependent to a considerable extent upon the supply of suitable volumes of cool air for carrying away the heat generated by electrical loss. The usual method is to mount fans on the rotor shaft, arranged to extract air through a supply system in which a combined air filter and cooler is inserted.

The 'Heenan' Patent Air Filter receives ordinary unfiltered air from the neighbourhood of the generating station, and brings it into very thorough and intimate contact with wet scrubbing surfaces, which are kept thoroughly wetted by water issuing from jets. After cleansing the scrubbing surfaces the water gravitates to a tank, from which it passes through strainers to a centrifugal circulating pump and thence to the distributing jets. The atomisation of water, forming part of certain air-filtering systems, is entirely avoided. The air leaves the filter in an entirely natural condition, containing no trace of loose moisture.

HEENAN & FROUDE

LIMITED

Engineering Works

WORCESTER • ENGLAND

Telegrams
"HEENAN, WORCESTER."

Tel. No.
391 WORCESTER (3 lines)

Specialities

DYNAMOMETERS

of the Hydraulic, Electric, and Electro-Hydraulic types, for testing the B.H.P. of any type of prime mover.

TUBULAR AIR COOLERS

for ventilating Turbo-Alternators, Generators, etc., and for general industrial purposes.

WATER COOLERS

for cooling the circulating water of Internal Combustion Engines, Air Compressors, Refrigerating Plants, Mercury Arc Rectifiers, and general industrial purposes.

OIL COOLERS

for Lubricating, Quenching and Transformer Oils, etc.

AIR FILTERS AND COOLERS

for the ventilation of Mines, Buildings, etc., also for extracting micro-organisms from air in Breweries, etc.

F. 2268

NEW TECHNICAL BOOKS

The Strength of Shafts in Vibration

By Captain J. MORRIS, B.A., A.F.R.Ae.S. (late R.A.F.). Formerly Scholar of Corpus Christi College, Cambridge. 200 pp., Illustrated. Crown 4to, cloth ... ... *Net* 30s.

An original work on a subject of outstanding importance to engineers and students. The results of rigorous applied mathematics reduced to relative simplicity by a suitable choice of notation and by the methods of attack adopted. The work constitutes an extensive monograph on resonant vibration in shafts, much of which is now published for the first time.

Railway and Seaport Freight Movement

Comprising Modern British and American Practice. By GEORGE BULKELEY, A.M.I.Mech.E.; Port Manager, Mombasa and Kenya Coast; formerly on Personal Staff of the General Manager, Great Western Railway, England; Author of "Mechanical Appliances for Handling Railway Traffic." With an Introduction by His Excellency Sir EDWARD GRIGG, K.C.M.G., K.C.V.O., D.S.O., M.C., Governor of Kenya and High Commissioner for Transport in Kenya and Uganda. Profusely Illustrated with Photographs and Diagrams. 240 pp. Royal 8vo, cloth ... ... *Net* 28s.

Design of Merchant Ships and Cost Estimating

A Treatise on Ship Design and Cost Estimating, giving up-to-date methods of arriving at correct proportions, form and power to attain minimum Capital Cost with maximum service efficiency. The work is intended for the use of Naval Architects, Marine Surveyors and Consulting Engineers, Ship Draughtsmen, Students and all who are interested in the advancement of Shipbuilding and the Mercantile Marine. By ALEXANDER KARI, M.Sc. (Dunelm), A.M.I.N.A., Associate Member of N.E.C. Institution of Engineers and Shipbuilders. Third Edition. Revised. 320 pp. 112 Illustrations. 104 Tables. Also Diagrams and Charts. Royal 8vo, cloth ... ... *Net* 36s.

Naval Architect's and Shipbuilder's Pocket-Book

Of Formulæ, Rules, and Tables, and Marine Engineer's and Surveyor's Handy Book of Reference. By CLEMENT MACKROW, M.I.N.A. Thirteenth Edition, Revised, by LLOYD WOOLLARD, M.I.N.A., Royal Corps of Naval Constructors. 744 pages. Diagrams. Fcap. 8vo, limp ... ... *Net* 21s.

Modern Bridge Construction

By F. JOHNSTONE TAYLOR. Author of "Water Power Practice," "River Engineering," etc. 232 pp., with 118 Illustrations. Demy 8vo ... ... *Net* 15s.

CROSBY LOCKWOOD & SON

Stationers' Hall Court, London, E.C. 4.

F. 2269

The filtering efficiency of this apparatus is extremely high, both heavy and fine dust being eliminated. The cooling effect in hot summer weather in temperate climates is often as high as 20° or 30° F. In tropical climates the latter figure is sometimes exceeded. The consequent reduction in air temperature is of great value in permitting the generation plant to run on heavy overloads without overheating.

Visco Air Filter.

(The Visco Engineering Co., Ltd.)

The Visco Air Filter (devised in 1916) is used in conjunction with turbo alternators, pipe ventilated motors, and general ventilating plants. It works on the principle that the air volume is divided into a large number of thin streams which are passed through a layer of specially prepared filter bodies, the surfaces placed irregularly to each other, so that the direction of the air flow is changed suddenly and a great number of times. The filter bodies are thin copper-plated steel rings $\frac{7}{16}$ in. diameter and $\frac{7}{10}$ in. long, coated with a thin film of viscinol, which is a specially prepared blend of highly viscous mineral oils. When the direction of the air flow is changed, the dust particles continue in the original direction and are thrown against and retained by the viscinol-coated surfaces.

The filter is standardised in units 20 by 20 ins. square and 3½ ins. deep, consisting of frame and cell, which latter is simply hooked on the frame. The capacity per unit is 600 cu. ft. per minute. Pressure loss is practically steady at ⅜ in. water gauge. Average consumption of oil, 1 gallon per unit per annum. Cleaning is done by rinsing in hot soda water on the average every eight weeks, depending on the original dust content and length of daily run. There are practically no other running costs.

The filter units are assembled in a convenient shape to make up any capacity, and are preferably built into the alternator foundations (thus taking up no valuable space) or as self-contained units in any position.

The *advantages* claimed are: No moving or wearing parts, all metal, fireproof, no attention, foolproof, small space.

The Ventex Air Filter.

(Ozonair, Ltd., 96, Victoria Street, London, S.W. 1.)

The latest variety of the dry or viscous type of filter is known as the 'Ventex.' It consists of a large number of specially corrugated plates spaced uniformly in a frame or holder. Owing to their form, the plates expose a large surface to the air, which in its passage between them is buffeted from side to side, thus bringing any suspended particles into contact with the viscous surfaces. The cleaning efficiency is claimed to be almost 100 per cent., whilst the resistance set up by the filter at standard air velocity is only ¼-in. water gauge. Trials made with the Ventex Filter at 50 per cent. above the normal working capacity showed that it is absolutely silent. All the parts are rigidly fixed and the coating fluid cannot flow, so that there is no danger of the filter becoming dry even at the highest velocities.

Portable Electric Blower.

(The Consolidated Pneumatic Tool Co., Ltd.)

This blower, fig. 25, is largely used for keeping the windings of motors and dynamos free from dust. Weighing, complete ,about 15 lbs., and suitable for use on either a direct or alternating

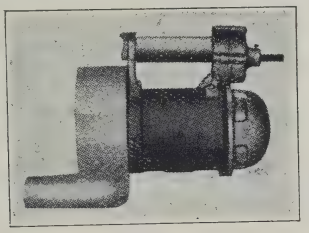


Fig. 25.

current circuit, it will maintain a steady air blast through a 1¼-in. orifice, with a pressure of about 12 ins. of water. It is very convenient to handle, and can be attached to any ordinary lamp socket.

CONVERTERS.

SYNCHRONOUS (OR ROTARY) CONVERTERS.

The G.E.C. Synchronous Converter.

(The General Electric Co., Ltd.)

The following is a specification of a Synchronous Converter of standard type :—

Frame.—The frame is constructed of cast iron or steel of high magnetic permeability. It is split horizontally and bolted together by strong steel bolts. The construction permits the top half to be readily removed for inspection or repair.

Main Pole-Pieces.—The main pole-pieces, which are bolted to the frame from the outside, are built up of laminations in order to reduce eddy-current losses in the pole faces to a minimum. The design is such as to permit the removal of a field coil without disturbing the armature or yoke. In exceptional cases cast steel poles with laminated shoes may be employed.

Armature.—The armature is built up of best quality stampings of high permeability and low iron loss.

Except in very small sizes they are mounted on a cast iron spider, and held tightly in position by heavy end-plates. The laminations are accurately punched before assembling, and are covered with a special insulating compound, which is effective in reducing eddy-currents to a minimum.

The core is held together between the end-plates by means of bolts, which do not pass through the active portion of the core. Ventilating ducts, adequate in number and ample in dimensions, are provided at frequent intervals. Similar ducts are included at the two ends between the core and end-plates.

The general construction of the armature is such as to afford a current of air to the field coils for ventilation and cooling.

The armature is composed of former wound coils, taped, varnished, and baked before assembly. The slots are lined with several layers of non-hygroscopic insulating material. Special care is taken to make the leakage surfaces as large as possible. The top of the slot is closed by an impregnated wooden wedge driven in tightly. There are no binding bands on the armature core itself, that part of the winding being retained in position by the wooden wedges.

The end windings are so shaped that the surface exposed for cooling is as large as possible. After winding, a black finishing varnish is sprayed by compressed air on to the armature until it is completely coated. A large number of equalising connections is provided to ensure that out-of-balance currents, if any, shall not pass through the brushes.

Prior to assembling in the frame the armature is balanced statically, and the final balancing is obtained dynamically.

Commutator.—The commutator is built up of hard-drawn copper segments, insulated by best mica insulation, and held tightly between two V-shaped clamping rings in such manner that the inevitable expansion of the copper and iron, due to temperature rise, may take place without in any way affecting the commutator surface. During its construction the commutator is subjected to several baking processes, with the result that there can be no possibility of moisture remaining in the mica insulation, and when completed, the whole commutator gives a true ring, which indicates that the entire mass has been formed into one rigid body. In addition, the construction is such that ' breathing ' is reduced to a minimum, and the possibility of moisture permeating the insulation is eliminated. A considerable distance to the iron parts is allowed at the front and back of the commutator in order to secure a high insulation resistance to earth, and to prevent any leakage to the iron. The radial depth of the segments is ample for the size of the commutator, and varies somewhat with the size of the machines. Before leaving the works, the mica between the segments is grooved to a suitable depth to promote good commutation. To assist in cooling, the commutator lugs which connect the segments to the armature windings are given a considerable width, and thus provide a large cooling surface.

The shunt field coils are wound to leave a clearance between the inside of the coils and the poles. Sheets of a material with high insulating properties are fixed on the inner sides of the coils and packing blocks are secured in such a manner as to maintain an air passage between the coils and the poles. It is thus ensured that the inner surfaces of the coils are efficiently cooled. Double cotton-covered wire is used, and the coils after being wound are put into a heated impregnating chamber where they are dried for several hours under vacuum, the vacuum pump drawing off all moisture. After removal from this chamber they are flooded in a duplicate chamber (while still under vacuum) with a heavy pressure at the same temperature, the insulating compound being forced right through the layers of the coils.

Series coils are wound with bare copper strip and painted with a finishing varnish. Ample leakage surfaces to the frame are provided.

The connections between the coils on adjacent poles, both for shunt and series windings, are of great mechanical strength.

Commutating Poles.—G.E.C. Rotary Converters are fitted with commutating poles of a design which permits sparkless collection of current at all loads with fixed brush position. The face of the pole is specially shaped to give an even commutating flux. For traction rotaries, and others designed for heavy overloads, the poles are made with specially high reluctance to ensure stability at all times.

The coils are wound with bare copper strip and painted with a finishing varnish. The micanite spools are moulded direct on the poles and ample leakage surfaces are provided.

Damping Winding.—A substantial damping winding is fitted to the field poles. This is instrumental in strongly opposing any tendency to hunt, whether that tendency may arise from changes in load or from any low-frequency oscillations originating in the generating plant. It consists of a pair of heavy copper rings bolted together by copper bolts between the poles and connected up by drawn copper rods passing through the pole faces and fixed at each end to the rings. In order that any pole winding may be removed, the rings are divided up between the poles and jointed in a manner which permits the joint to be readily unmade for the removal of the pole or winding, and at the same time in no way impairs the mechanical strength or the electrical conductivity of the rings. A damping winding of high conductivity and great rigidity is thus obtained which completely damps out all oscillations.

The damping winding is also necessary for improving the starting characteristics of the converter when started from the alterating current side, from transformer tappings.

Booster.—The booster, if provided, takes the form of an alternating current generator, with a stationary field. The booster armature is mounted on the main shaft side by side with the armature of the rotary converter, and is constructed on the same principle as the armature of the rotary. The current passes from the converter slip-rings through the booster armature to the armature of the rotary. The booster field-frame is split horizontally into two parts, and mounted on the bedplate by the side of the main frame. The booster field may be excited either by shunt or series windings, or both.

Six-phase and Four-phase Connections.—Rotary converters intended to work on a three-phase supply are usually wound as six-phase machines, drawing a six-phase supply from three-phase transformers in order to take advantage of the greater output and more even heating of a six-phase armature as compared with a three-phase. Similarly, converters for operating on a two-phase supply are wound either four-phase or six-phase, the latter being made possible by using a double Scott connected transformer.

D.C. Brush Gear.—The direct current brush holders are fixed to a strong cast iron arm projecting radially inwards from the brush rocker. The rocker is carried in a spigot in the yoke and can be turned by means of a screw adjusting gear. The rocker is split across the horizontal diameter so that the halves can be readily removed for inspection or repair. The copper rings connecting the brush sets of the same polarity are bolted direct on to the cast iron arms, the surface of the joint being copper plated to ensure a good contact.

Provision is made for separating the halves of the rings should it be desired to remove one-half of the brush rocker for any reason.

Each carbon brush is contained in a separate box-type holder to avoid rubbing, and a spiral spring exerts pressure on the brush. The pressure on the brush is adjustable by means of a steel lever which engages in one of a large number of slots in a quadrant.

A.C. Brush Gear.—The alternating current brush holders are mounted on semi-circular arms, which are carried on heavy insulated spindles projecting from the pedestal bearing on the slip-ring end of the rotary converter. Each brush is contained in a separate box-type holder, and a helical spring exerts radial pressure on the brush.

The pressure on the brush is maintained by means of a lever, terminating in a special pawl which wedges against corrugations on the inside of a brass tube. The pressure can be easily adjusted by slightly canting the lever, moving it up or down the brass tube and releasing it, when the pawl will automatically engage in other corrugations.

Each brush and brush holder can be easily adjusted independently of the rest of the brush gear for pressure and wear on the slip rings.

Terminals for the incoming cables are provided and fixed directly on to the arms.

Slip-rings.—The slip-rings are of phosphor bronze, and so designed as to ensure efficient cooling. The rings are given a large radial wearing depth.

A cast-iron sleeve, which is securely keyed to the shaft, forms the base for the slip-ring structure This sleeve is insulated throughout its length with compressed mica ground to shape, the slip-rings being shrunk on to this.

The connections from the armature to the slip-rings consist of solid copper bars strongly bolted into the slip-rings at one end and supported at the other by a steel ring shrunk on to the sleeve, similarly to the slip-rings. These connecting bars are covered with insulating materials, and are efficiently protected from breakdown to the other slip-rings through which they have to pass. Ample clearances are provided for cleaning.

Wherever possible, each tapping of the armature winding is brought to the rings separately in order to avoid unbalanced current. Thus the slip-rings themselves act as balancing rings.

Shaft.—The shaft is of such strength that it will carry the armature without undue flexure or vibration, and is supported by two main bearings. The tensile strength of the material is 35 tons per sq. in.

Bearings.—The bearings are either of the cylindrical or of the roller type, the former being self-lubricated by means of rings which dip into oil chambers of large dimensions. The bushes are of cast iron, lined with best quality white metal, and are split horizontally. A glass level gauge, with draw-off tap, is fitted to each bearing. The general design of the bearing is such as to preclude any oil flinging.

Roller bearings are used mainly to reduce the torque required during starting. They consist of large roller bearings of the best possible workmanship. They are supported by pedestals, and are arranged to allow the necessary axial play for expansion of the shaft. They are lubricated by grease charged through a suitable nipple.

Bedplate.—A strong bedplate of box section is provided, having heavy ribs connecting the sides at various points.

Its general construction is substantial, and ensures complete freedom from vibration.

The bedplate is grouted solid into the foundation blocks without any bolts.

Speed-Limiting Device.—All large rotary converters (and small ones, if desired) are equipped with a speed-limiting device, which will shut down the machine if by any chance the normal speed should be exceeded by a predetermined amount. This device works on the centrifugal principle. A plunger which tends to move under the action of centrifugal force is mounted on the shaft and

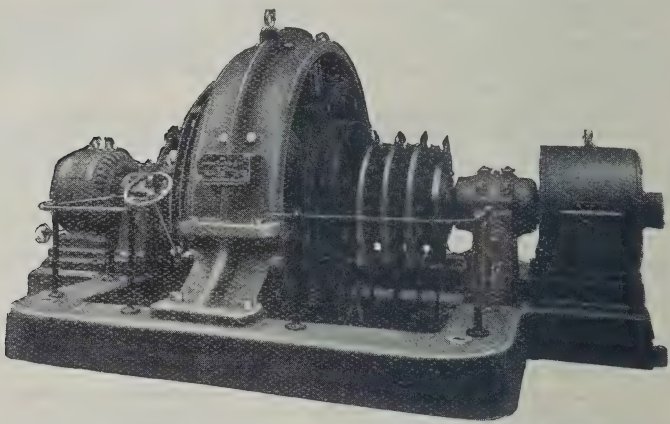


FIG. 26.—1,000 kW. G.E.C. Rotary Converter; 550–570 volts, 50 cycles, 750 r.p.m.

revolves with it. This plunger is held in position by means of an adjustable spring loaded trigger which releases at the predetermined speed. When released the plunger flies out and trips a switch device, which makes contact in a circuit, and thereby opens the direct current circuit breaker, or, if desired, cuts out resistance from the converter field circuit. The plunger automatically returns to its normal position when the machine shuts down.

Oscillating Gear.—When definitely specified, an oscillating device is supplied for moving the armature to and fro longitudinally while it is running. As the armature rotates, it is alternately pushed out of the magnetic centre and then released, thus causing a smooth oscillation. This device is not required for the satisfactory working of the machine and is not recommended.

Temperature Rise.—The rotary converter is capable of developing its full rated load continuously, with a rise in temperature of 72° F. (40° C.), in accordance with B.S.S. 172.

Overload Capacity.—The rotary converter will give an overload of 25 per cent. for 2 hours, and 50 per cent. for 15 minutes. It will also withstand overloads of 100 per cent. momentarily.

Starting Motor.—If the rotary be supplied with a starting motor, the latter is mounted on an extension of the bedplate. The motor is of large capacity, and is adequate in size to run the converter up to speed in a short time. The motor is usually of the solid rotor type. The starting

torque is sufficient to permit rapid acceleration in the first moments of starting, even though the supply voltage is below normal, as may be the case after a shut-down at the main power station.

Tests.—All material is subjected to tests at various stages of its manufacture, to ensure that no faults exist in the completed machine. Tests for static and dynamic balance are also carried out before the machine leaves the works. Standard final tests on rotaries for 650 volts direct current or under include a 2,000 volt flash test with alternating current for one minute, and six hours run on full load to ascertain the efficiency and temperature rises.

Starting, Self-Synchronising, and Controlling Synchronous Converters.

G.E.C. Methods of Starting, Self-Synchronising and Controlling Synchronous Converters.

It is usually required that synchronous converters shall be of the self-synchronising type, as the running of sub-stations is thereby facilitated. The practice can be considered standard. In the G.E.C. patented method of starting and self-synchronising, the starting current is not allowed to pass into the armature until the field is excited. Consequently, when the armature is switched on through the choking coils, the current is not strong enough to reverse the field, so that the polarity of the converter is always correct.

Fig. 27 shows the scheme of connections. The essential features are the synchronising switch and the choking coils. The rotary converter is provided with a starting motor, which is started by closing the triple-pole circuit-breaker shown at the right of the figure. This brings the set up to approximately synchronous speed. Meanwhile the switch shown in the figure is closed in its starting position, where the blades make contact on the long synchronising contacts. Under these conditions the slip-rings of the rotary are connected to the source of alternating-current supply through the choking coils, which limit the current taken by the machine. This current, however, is sufficient to pull the set into synchronism. When this has happened, the synchronising switch is pushed right home, so that the blades go into the full-on contacts. Mechanism is provided which prevents the synchronising switch being pushed at once into its fully-closed position. The device is so arranged that on slightly pulling the switch towards the 'off' position a pin releases a slotted bar, which falls under its own weight, allowing the pin to pass along the slot and permitting the synchronising switch to be pushed right home. When this happens, an

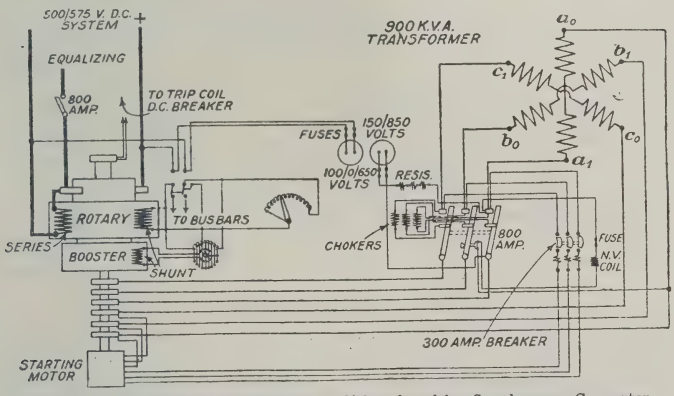


FIG. 27.—Diagram of Connections for Self-Synchronising Synchronous Converter.

interlocking contact is operated, which opens the no-volt coil of the starting-motor circuit-breaker tripping the latter, so that the converter set is left in synchronism with the starting motor disconnected.

The standard starting motor used is provided with a solid rotor. The stator windings are adjusted once for all when the set is first put into use so as to give the best value of slip to the starting motor and so as to necessitate the minimum value of synchronising current.

MOTOR CONVERTERS.

The motor converter combines the advantages of the ordinary rotary converter with those of the motor generator and to a very great extent eliminates the defects of both, while it is particularly suited to the higher frequencies now in commercial use. The machine is of the same general appearance as a motor generator, but its dimensions for a given output are considerably less. The static transformer, which is necessary in order to reduce the voltage for the ordinary rotary converter, is not required, the high-tension current being taken direct to the stator windings.

The switchgear required for the motor converter is of the simplest possible character, while starting and sychronising troubles are entirely eliminated with the use of patented devices, and the efficiency and power factor attain high values even at light loads.

The machine possesses a number of valuable features which are unattainable with any other machine; it can be designed with a voltage regulation on the direct current side of over 25 per cent., and can also without difficulty be compounded or overcompounded to suit the varied requirements met with in practice. Once synchronised a motor converter will not drop out of step, this being mainly due to the rotor being wound with a large number of phases; the exceptionally strong synchronising torque of this type of converting plant is one of its most remarkable characteristics. A motor converter, moreover, is particularly suitable for use on three-wire systems, and when working under these conditions will automatically maintain the voltage difference between the two sides within about 1 per cent. of the voltage across the outers, even with an out-of-balance current of 50 per cent. The air gap on the alternating current side is much larger than would be the case for a similar induction motor, and this advantage is of course accentuated in the larger sizes.

The motor converter consists of an ordinary induction motor with wound rotor, and a continuous current generator rigidly coupled together, the rotor windings of the induction motor and the armature of the generator being electrically interconnected. Fig. 28 shows the diagram

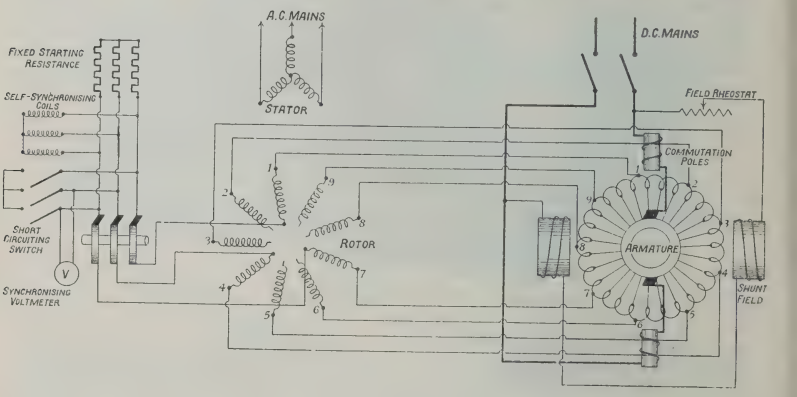


FIG. 28.—Schematic Diagram of Connections of Motor Converter.

of connections of a Motor Converter in its simplest form. The machine differs from a motor generator in that the rotor and armature are interconnected, and from a rotary converter in that the stator is direct connected to the high voltage mains and no transformer is required, although the rotating portions of the machine carry low voltages only. The high-tension stator winding may be star or delta connected, and the rotor winding is generally wound with nine phases connected to nine corresponding points of the direct current armature. Only three phases are used for starting, and only three slip rings are therefore required.

Perhaps the best method of explaining the principle of the motor converter is to consider it in conjunction with a rotary converter. A comparison between the relative merits of the two machines is thereby greatly facilitated.

In the case of a rotary converter, the supply current is first transformed to a suitable voltage by a transformer, fed through slip rings to the rotor, and the machine then runs as a synchronous

motor at a speed depending on the frequency of supply and the number of poles. At the opposite end of the rotor a commutator is provided, from which the direct current is taken, the pressure definitely depending on the pressure at the A.C. end and being adjustable only within narrow limits.

In the case of a motor converter, the supply current is fed direct to the stator winding of the A.C. part of the machine, thus setting up a rotating field.

The rotor of the machine may therefore be considered as the secondary of a transformer, the stator being the primary winding.

Assuming that both ends of the machine have the same number of poles, the rotor when electrically connected to the armature will rotate at a speed corresponding to half the frequency of the supply, that is, with 50 per cent. slip.

The whole arrangement is equivalent to a transformer, the secondary of which is allowed to rotate at half synchronous speed, and this has the effect of reducing the periodicity under which the converter part operates by half.

On account of this rotation, moreover, half of the energy taken from the mains is converted into mechanical energy in the A.C. rotor, and the D.C. armature being mechanically connected to the rotor therefore supplies half its output as an ordinary D.C. generator. The remaining half of the electrical energy drawn from the mains is transferred, as already described, from stator to rotor by transformer effect, whence, the armature being electrically connected to the rotor, it becomes available as direct current energy at the commutator brushes.

The motor converter thus acts partly as a rotary converter and partly as a motor generator.

Owing to the fact that the presure is reduced by transformer effect from stator to rotor instead of by separate transformers, no energy is supplied to the machine through brushes and slip rings which, in the case of a rotary converter of any size, carry very heavy currents on account of the low voltage. The D.C. part of the machine operating partly as an ordinary direct current generator, the output voltage may be varied between much wider limits than would be the case for a rotary converter.

The 'A.C.E.C.' Motor Converter.

The A.C.E.C. are sole licensees for Belgium of the Arnold and La Cour patents relating to motor converters, and since the British master patent has expired they have supplied a large number of these machines for use in Great Britain.

A.C.E.C. motor converters (fig. 29) are arranged for starting up on the A.C. side by means of patented devices which enable the starting and synchronising to be effected simply by closing the main switch; the machine runs up to speed and automatically comes into synchronism with the high-tension supply.

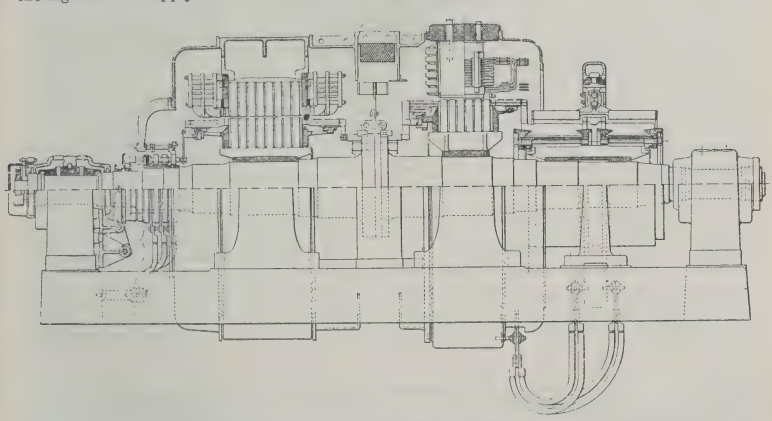


Fig. 29.—Section through Standard 1,500-kW. Double Commutator A.C.E.C. La Cour Motor Converter, showing general construction.

The rotor circuit being closed through a fixed three-phase resistance and the field rheostat being adjusted for maximum field, the high-tension switch is closed and the machine starts up as an induction motor, the current drawn from the mains being approximately one-third of normal full load current.

As the machine speeds up, its direct current voltage gradually builds up, and when this reaches a value equal to approximately two-fifths of the normal voltage, the D.C. field is reduced. Owing to the A.C.E.C. patented synchronising reactances which are connected across the rotor winding the machine then automatically falls into step, and the needle of the synchronising voltmeter comes back to zero. The rotor slip rings are then short circuited by means of a three-pole switch and the short-circuiting device for all the phases of the rotor closed. The effect of the above-mentioned reactances is to produce a resultant retarding torque when the E.M.F. of the rotor is in advance of the E.M.F. in the armature and an accelerating torque in the contrary case, so that once the motor converter has attained synchronous speed it remains in step at that speed. It will be seen that the whole operation is extremely simple, and one which can be performed without difficulty by an unskilled person.

When the motor converter is used on a three-wire system, the neutral point of the short-circuiting switch is connected to the neutral wire of the system.

Efficiency.

The values of the efficiency of a motor converter are considerably higher than those of the motor generator set, particularly at low loads. At full load the efficiency is very slightly lower than that of an ordinary rotary converter, but at three-quarter load and half load the efficiencies of the two types of machine are practically the same.

Power Factor.

Being a synchronous machine, the motor converter runs with a high-power factor at all loads, a notable advantage over the ordinary induction motor generator set. This high-power factor is maintained in spite of the large range of voltage regulation on the direct current side, whereas in the case of an ordinary rotary converter, any extension of the normal range of D.C. voltage involves a drop in the value of the power factor. The motor converter can be designed to run with practically unity power factor at all normal loads for any given value of the D.C. voltage.

As regards the general construction of the A.C.E.C. Motor Converters the *Bearings* are of the self-aligning type with automatic oil and air circulation, and are each capable of holding a large volume of oil. These bearings are so designed that the oil which is lifted by the rings passes along the shaft and falls into another chamber of the pedestal and cannot again be lifted by the rings until it has passed down to the bottom of the pedestal and up again to the surface of the oil ring chamber. The complete rotating element is very carefully balanced, and every precaution is taken to ensure smooth running without vibration.

The *Stator* frame consists of a substantial iron casting arranged with feet for bolting to the bedplate. The core is built up from suitably insulated laminations of high permeability, the slots being of the semi-closed type.

The stator end windings are supported by solid blocks of insulating material securely bolted to the frame. This work is carried out on the same general lines as are now universally adopted in the construction of turbo-alternator stators. Owing to the special methods of insulation and impregnation used, and the application of Creedy windings and Haefely wrappings, the stators can be wound for pressures up to 15,000 volts. The coils are former wound and carefully protected by massive bakelite finger blocks where they emerge from the stator slots, thus avoiding any possibility of breakdown at these points.

The laminations of the *Rotor* core are clamped between substantial end-plates. The ends of the windings are connected at the slip-ring end to short-circuiting contacts, three of them being so connected through slip rings, and at the other end to tapping points on the D.C. armature winding, through the interconnections previously described.

The *Armature* core is built up from high-permeability laminations of similar grade to those used in the rotor and stator, clamped between substantial cast-iron end-plates. The winding is of the drum type, each coil being former wound, and is provided with a number of tapping points for connection to the rotor winding. The coils are held in position in the slots by means of specially treated wedges, the end windings being kept in place by steel bands insulated with mica.

Substantial short-circuiting gear is provided, with adjustable contact blades. It is hand-operated, and so designed that the actuating portion, after pressing the sliding ring into the running position, automatically makes a small movement in the reverse direction so as to release the pressure from the ring while the machine is running. The contact blades engage with solid gun-metal blocks.

Mercury Vapour Rectifiers.

The necessity of converting alternating to direct current in many cases, and the desirability of dispensing with running machinery in sub-stations, etc., has led to the development of the rectifying properties of the mercury vapour lamp for this purpose. Essentially the rectifier consists of an arc *in vacuo* between iron or carbon anodes and a mercury cathode, which allows the current to pass in one direction only, namely, from the anodes to the cathode, as the names imply.

The exclusive use of mercury as a cathode is due to the fact that it can be vaporised, condensed and returned by gravity to the cathode during the operation of the arc, without loss of material; while the electrical conductivity of mercury vapour entails but a small voltage to maintain the arc and enables the anode to be well removed from the cathode, which is essential in order that the anode shall be kept cool.

Efficiency.

The losses within the rectifier chamber are due to the voltage drops at the anode (about 5 volts), the cathode (about 9 volts) and in the arc (about 0·1 volt per cm. of arc length), making a constant total drop of about 20 volts in the rectifier. In consequence, the efficiency of the rectifier is practically independent of the load, while the higher the voltage the higher the efficiency—hence the suitability of such rectifiers for high-voltage d.c. traction.

Transformer.

The greater the number of anodes the larger becomes the transformer for a given output, but the smaller the ripple in the d.c. voltage. Usually 6 anodes are used. The transformer for rectifier supply must be specially reinforced against short-circuits. With an absorption 'choke,' a 6-anode rectifier needs a transformer rated only for a 3-anode rectifier.

Regulation.

The rectifier has no inherent regulation, which depends largely on the connections. With 3-phase star/6-anode connections the regulation is 10–12 per cent. and 4–5 per cent. with and without an absorption choke. Induction regulators are also used to provide regulation.

Load Characteristics.

The rectifier, owing to its nature as a valve in contradistinction to that of a converting machine, is an apparatus with an inherent continuous rating. It has a large overload capacity, since it is free from mechanical inertia. The rating must be reduced where overloads of long duration are required, for although momentary overloads are easily handled, continuous overload will result in overheating with its consequent liability to the development of faults. It can respond quickly to load fluctuations owing to the comparative absence of the electrical inertia introduced by the presence of magnetic fields. A given rectifier will theoretically provide its rated current at *any* desired voltage, so that the kW. rating will depend directly upon the D.C. terminal voltage. In practice, however, the higher the rectified voltage, the greater becomes the tendency for short-circuits between anodes due to heating, and for 'arc-back,' etc., so that output current ratings are progressively reduced as the voltage is increased.

'Arc-back,' or loss of rectifying property, depends largely upon the temperature rise due to a given load. The presence of foreign gases and failure of the vacuum, reduce the voltage that the rectifier will withstand without arc-back. The fitting of an anode guard—consisting of an iron tube with open-mesh grating enclosing the anode—assists in suppressing the tendency to arc-back.

'Flashing' is momentary loss of valve property and is due to various causes such as the condensation of small globules of mercury on the cool anodes.

'Fading' occurs in glass-bulb rectifiers with long arms, and is due to charges on the glass shielding the anodes.

Cooling.

Glass rectifiers are cooled naturally, by air fan, or by immersion in oil; metal-clad rectifiers by the circulation of water. Cooling is very important, since it determines the load, overload capacity, losses, etc. Short-circuits are caused by condensed mercury running down the anodes; by overheating of the anodes; by the effect on the anodes of ultra-violet irradiation from the cathode spot; by insufficient vacuum and continuous abnormal overload causing overheating.

Construction.

(*a*) *Glass-bulb* rectifiers. The first rectifiers were built with glass bulbs, and such are still used for small powers, low voltages, currents up to 250 amperes and few anodes. Glass-bulb rectifiers have a long but unpredictable useful life. They must be discarded when the anodes have been worn sufficiently for the deposited material to hinder radiation and cause fading. The bulbs are evacuated during manufacture, when care is taken to eliminate occluded gases. Little knowledge of the state of the vacuum is possible during operation. Graphite anodes are usually employed, since they do not quickly deteriorate, and tend to absorb stray gases. Molybdenum or platinum must be embodied in the lead-in arrangements to form an efficient seal. The temperature of glass-bulbs is limited to about 110° C.; above this temperature the internal arc drop increases very rapidly. The arms of glass rectifiers are usually given a right-angle bend to protect the anodes from ultra-violet rays from the cathode and from contact with condensed mercury, both of which tend to cause short-circuits. Starting may be effected by tilting the bulb.

(*b*) *Metal-cased* rectifiers are used for large outputs and high voltages because large volumes of mercury vapour are concerned; the state of the vacuum must be continuously determinable by vacuum gauge; periodic inspection and renewal of the anodes is necessary. The cylinder is usually of steel, closed by a top anode plate which supports the main and excitation anoded and the condensing chamber. The cathode consists of a pool of mercury from which the arc rises through arc funnels and anode shields. The casing and condensing chamber are cooled by water jacket. The main anodes of special steel are insulated by porcelain bushings. Ignition (*i.e.* preliminary striking of the arc to initiate electronic emission) is accomplished by means of a plunger maintained a short distance above the cathode surface on a rod depending from the top of the condensing chamber, and capable of being lowered momentarily by the energising of a solenoid. Excitation anodes are connected to a dead load sufficient to maintain the arc when the main load falls below a critical value. The vacuum is maintained continuously or intermittently by a vacuum pump. Since it is usual to earth the negative side of a d.c. system, the rectifier must be insulated as a whole from earth, and the cooling-water connections made by long rubber pipes.

Operating Features.

Parallel operation is governed by the same principles as with D.C. generators, but no reversal of power flow is possible. The operation is independent of frequency fluctuation, so that rectifiers can safely be run in parallel from separate unconnected A.C. systems, without need of synchronisation. Automatic control is more easily applied to rectifiers than to rotary machinery. Rectifiers can be worked in parallel with converters or batteries.

Applications.

The rectifier can be widely applied, but has naturally been found pre-eminently suitable for loads undergoing considerable fluctuations, such as tramways and railways. The absence of inertia enables any load within the rating to be supplied as demanded. The traction field is at present the widest, after which come power and light, rolling mills and electro-chemistry. The rectifier is definitely out of the experimental stage, and by reason of its high efficiency is becoming a serious rival to the synchronous converter. Its limitations in size have as yet scarcely been approached. Already currents up to 20,000 amperes and voltages up to 16,000 volts have been obtained in single units.

The advantages of the rectifier may be summarised as follows :—

1. The efficiency is high at all loads.
2. Little auxiliary apparatus is needed.
3. Wear and tear is small.
4. Synchronisation is unnecessary.
5. Little floor space is needed and the sets require no solid foundations.

The disadvantages are :—

1. Ripples in the rectified wave form are objectionable in some services.
2. Larger transformers than for converting machines are required.
3. No control of the power-factor is available.
4. Reversibility is impossible.

The 'Tungar' rectifier, which is suitable for charging small storage batteries, is more closely allied to the 'valves' used in wireless telegraphy, the cathode being an incandescent filament.

MOTORS.

DIRECT-CURRENT MOTORS.

Shunt-wound Motors.

The shunt is the most common type of field winding, and is generally applied to motors designed for operating at constant speed under constant or varying loads. Nearly all commercial applications, particularly those of large capacity, require this type of motor. When

necessary, considerable speed variation can usually be secured by means of a rheostat in the field circuit, increased resistance resulting in an increased speed.

On many shunt-wound commutating-pole motors a stability winding is used on the field for the purpose of insuring stability under all conditions of operation. This is, in effect, a light series winding, which, however, does not change the speed-torque characteristics from those of an ordinary shunt-wound motor.

SERIES-WOUND MOTORS.

These are variable-speed machines particularly adapted to such uses as railway and crane service. The characteristic features of a series-wound motor are its great torque at starting and low running speeds possible.

COMPOUND-WOUND MOTORS.

For some special classes of service, in which it is necessary to start under heavy load and later operate at approximately constant speed, a series winding is added to assist the shunt field at the low-speed points. A compound-wound motor combines the characteristics of both shunt and series-wound motors. In most cases, however, the series winding has comparatively little effect except during the starting period, and may be short-circuited, if desired, at higher speeds.

Starting Motors.

Except very small machines, all direct-current shunt and compound-wound motors are started by first giving the shunt field full excitation and then connecting the armature, with a starting resistance in series, to the circuit. The resistance is then cut out in steps as the motor accelerates, the time required to attain full speed varying from about 15 seconds for small, to one minute for large motors. These limits should be used with discretion, since more time is required to start with heavy load than with light load.

After all connections are made according to instructions and all mechanical preparations are complete, see that all the field adjusting resistance (if used) is cut out and that the starting resistance is all in circuit. Close the circuit-breaker (if used), then the line switch, and lastly cut out the starting resistance step by step, allowing enough time for smooth speed acceleration. Too rapid starting results in excessive starting current and possibly flashing at the brushes. The motor should generally start when the armature circuit is first closed through the starting resistance, and failure to start when the second or third step of resistance is cut out is a signal to open the line switch or circuit-breaker at once and find the cause of the failure.

Stopping Motors.

Never move a starter or controller handle to the off position under any circumstances if by doing so the shunt-field circuit of the motor is opened. The high inductance of the motor field may generate voltage enough to puncture the insulation. If it is necessary to open a shunt-field circuit it should be done very slowly, allowing the field current to die out, through the resistance of the arc across the switch terminals. If a field circuit must be opened in this manner often, a field switch should be installed, so that a local circuit is closed through the field before the latter is disconnected from the line. Starting rheostats are usually so arranged that the field circuit is always closed through the motor armature. If the starter or controller is provided with a low-voltage release, the line switch or circuit-breaker should always be opened first for stopping; the motor speed will then decrease until the starting resistance is automatically cut into circuit.

Brushholder Adjustments.

The correct position for the brushes in either direction is clearly marked before the machine is shipped, and before starting up the position should be checked. Sparking at the brushes when the machine is operating with load can frequently be entirely removed by shifting the brushes slowly from or towards the neutral point. In any case the correct distance of the brushes from the neutral point is very slight, and the movement should be made with great care, for shifting the brushes too much will cause violent sparking.

Care of Commutator and Brushes.

Carbon brushes are used on all direct-current machines nowadays. The chief considerations in caring for such machines are to keep the commutators clean, the brushes properly adjusted, and the contact surface of the brushes good. The last condition, once fulfilled, will usually take care of itself until the brushes are worn out, provided the other conditions are complied with. At intervals depending on the service and the location, the commutator contact surface and end should be thoroughly wiped with clean cloth that is free from lint.

A few drops of oil on the cloth or a piece of paraffin rubbed lightly across the commutator surface will give all the lubrication necessary. Lubricant should be used very sparingly on a commutator, and no grease, oil, or dirt should be allowed to accumulate on the commutator or the brushes.

The ideal appearance of a commutator surface is polished dark brown or chocolate colour. Such a commutator needs no attention other than cleaning when necessary. If the surface has a raw copper colour with gradual roughening, it should be smoothed with sandpaper or, if very rough, with a piece of sandstone; emery paper or cloth should not be used for this purpose. The sandpaper should be held against the commutator by means of a block having the same curvature as the commutator. The sandstone likewise should be curved to fit the commutator. The brushes should be lifted from the commutator surface when using sandpaper or stone, and should not be placed in contact again until the commutator surface is thoroughly cleaned of all gritty particles.

Bedding the Brushes.

When making proper contact with the commutator, the face of the brush is glazed. If the glazed portion does not extend over very nearly all the brush face the brushes should be ground in by wrapping a long strip of fine sandpaper closely around the commutator with the rough side toward the brushes and then rotating the commutator and the sandpaper slowly through perhaps one revolution. Shorter pieces of sandpaper held close to the commutator surface and drawn a few times back and forth under each brush or set of brushes will answer the same purpose. The commutator surface should be wiped clean after the brushes are ground in, and lubricated with a few drops of fresh oil on a clean cloth.

Field Coils.

An open-circuited shunt field coil can be detected by touching the terminals of a voltmeter to the terminals of the individual coils while full voltage is applied to the field circuit. The voltmeter will show no deflection until the defective coil is reached and will then indicate the full voltage.

A short-circuited shunt field can be found by the same test, the voltmeter deflection being least on the defective coil. The short-circuiting of one field coil may result in burning out one or more of the others if full voltage is applied to the circuit.

In replacing a field coil make sure that the polarity of the pole differs from that of either adjacent pole. The polarity can be tested with a small bar magnet or a piece of steel wire suspended from its centre by a string. In holding the magnet near the poles successively it should reverse at each pole, *i.e.* no two successive poles should attract the same end of the magnet.

Cautions.

Never allow the load to be entirely removed from a series-wound direct-current motor while operating, or to be reduced enough to cause a dangerously high speed. A series motor should not be belted, on account of the possibility of the belt running off, thus permitting the motor to race.

Never open the shunt-field circuit of a motor in operation, otherwise the motor may race and produce serious damage.

When the shunt-field circuit of a motor or a generator is excited, never open it quickly unless a path for the inductive discharge is provided. The circuit can be opened slowly, if desired, the arc at the opening serving to reduce the field current gradually. *Do not permit any part of the body to bridge this opening*, or a serious shock will be received.

Always follow a fixed regular order in closing and opening switches, unless there are special reasons for departing from this order. A routine method will aid in avoiding mistakes. Keep all switches open when the motors are idle. Close switches carefully, keeping firm hold of the handle until completely closed.

A routine method of making inspections, replenishing lubricants, etc., will also prevent oversights and possible damage.

(*Metropolitan-Vickers Instruction Book.*)

AVERAGE EFFICIENCIES OF D.C. MOTORS.

(*Meares.*)

Output, B.H.P.	½	1	2	3	5	10	20	50	100	200
Efficiency, per cent.	73	77	82	83	84	86	87	91	92	93

SPECIFICATION FOR DIRECT-CURRENT MOTORS.

(The English Electric Co. Ltd.)

These machines are of the two end-shield type, and are supplied with standard shaft extension, but without pulley, coupling, sliding base, or foundation bolts, unless otherwise ordered.

Commutation poles are fitted as standard on all sizes from C.B. 7 upwards.

C.B. machines are usually supplied in the Protected or in the Enclosed-ventilated type, but they are also made as Pipe-ventilated, Drip-proof, or Totally-enclosed machines.

The *Protected type* motor is so designed that the internal rotating parts and live parts are protected mechanically from accidental or careless contact, a perforated metal cover being fitted to the cylindrical opening over the armature fan.

The *Enclosed-ventilated* type is similar to the Protected type, but a perforated metal cover is fitted over both cylindrical openings in the frame.

The frame is circular, of cast steel. The four main poles are bolted to the frame and are laminated except in the smallest sizes, C.B. 5 and C.B. 6, which have solid poles. Commutation poles, when fitted, are of solid wrought iron.

End shields are in the form of end plates bolted to facings on the magnet frame and form substantial supports for the bearings.

Ball or roller bearings, or in certain cases a combination of both, are used, and are of ample proportions. On the larger machines a ball-thrust bearing is fitted.

The bearings are enclosed in dust-proof housings.

This type of bearing has stood the test of lengthy experience in practical use. It requires practically no attention. As the bearings will work equally well in any position, provided the shaft is horizontal, the motor may be fixed to wall or ceiling without altering the position of bearings or end shields, as is necessary with ring-lubricated oil bearings.

The field coils of all sizes are former wound, thoroughly impregnated and taped all over. All coils are heavily insulated from the poles and the frame, to safeguard against earthing.

The armature is slotted and drum wound, and is specially designed to give thorough ventilation of the windings, core, and field coils. The core plates are keyed directly to the shaft, and are held in position by two end flanges. The windings are carefully impregnated.

The commutator is built up of segments of hard-drawn copper which are held in position by end flanges fitting into deep V grooves in the segments.

The brush rocker which carries the brush spindles is mounted on a turned seating on the end-shield. The brush holders are substantial castings, and the brushes are easily replaceable.

A terminal box is arranged on the magnet frame, and is provided with a cast-iron cover. A flange is provided on the terminal box to receive screwed conduit.

The standard direction of rotation is counter-clockwise when viewed from the commutator end; if the opposite direction of rotation is required, this must be stated on the order.

The following accessories are supplied as extras, if required :—

Pulley for rope or belt (standard or otherwise) or half-coupling or flexible coupling.
Sliding base with adjusting screws and four bolts for fixing the machine to the base.
Foundation bolts.

Speed Regulation of Direct-current Shunt-wound Motors.

There are three commercial methods of varying the speed of direct-current shunt-wound motors: (1) by altering the terminal voltage; (2) by a resistance in series with the machine; (3) by a resistance in series with the shunt circuit. The first method is not employed as a rule with the ordinary industrial motor, but is only used in connection with self-contained plants such as winding, and certain printing press control equipments. The second method is sometimes adopted where a small speed reduction is required. With it, fine regulation is difficult to obtain, while the power consumption of the motor remains practically the same throughout the whole range of speed, an amount of the power proportional to the speed reduction being dissipated in the resistance. The method is convenient where it is desired to regulate the speed down to about half for very short periods, such as for crane and haulage work. The third method of control is one which has received a good deal of attention during the last few years, and motors have been developed which are capable of speed regulation by field resistance up to 4 : 1, and in certain cases up to 6 : 1, the torque being maintained constant over the whole range.

Such large variations of speed generally require some special device to compensate for the cross magnetisation of the armature due to armature reaction, and for this purpose interpoles or auxiliary poles are employed. The poles are placed mid-way between the main field

poles and carry either the full armature current or a percentage of it. The interpole flux therefore varies directly as the load and as the armature reaction, and as the tendency of the motor to spark increases as the load, so the compensating flux increases proportionately.

The size of motors with shunt regulation is determined by the minimum speed and the maximum b.h.p., although it may be necessary for the motor to give a much lower output at the minimum speed. This is clear when it is considered that minimum speed requires maximum flux and maximum speed requires minimum flux, that the magnetic circuit must be large enough to carry the full flux, and that the dimensions of the armature are also determined by the number and size of the armature conductors. The only variable, therefore, is the field flux. The efficiency does not largely vary throughout the range of speed at a given load, owing to the fact that at the minimum speed the electrical losses are a maximum, whereas the friction and windage losses are

APPROXIMATE COMMERCIAL EFFICIENCY OF DIRECT-CURRENT MOTORS PLOTTED AS A FUNCTION OF THE BRAKE H.P.

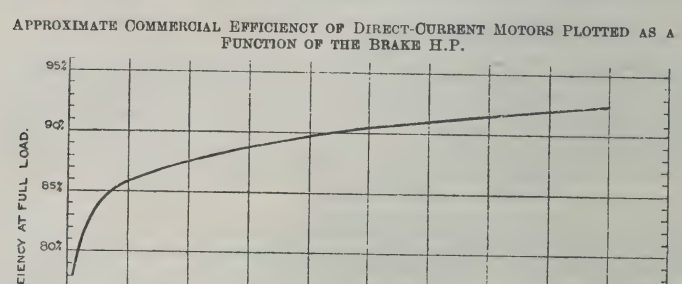


FIG. 30.

a minimum ; but at the maximum speed the electrical losses are a minimum and the friction and windage losses a maximum. The curve shown in fig. 30, giving the full-load commercial efficiency as a function of the brake horse-power, is typical of constant speed motors running at the normal speeds recommended by the British Engineering Standards Association. Motors with series resistance regulation have their sizes determined by the maximum speed and maximum output, and although the torque can remain constant throughout yet the output or power will be reduced in proportion to the speed.

Direct-Current Traction Motors.*

DIRECT-CURRENT SERIES-WOUND TRACTION MOTORS.

British Standard Specification for Traction Motors (Direct-Current, Series-wound). No. 173—1928 (*Abstract*).†

Types of enclosure are : (1) Totally enclosed, there being no air circulation between inside and outside of case ; (2) ventilated, having free circulation between inside and outside of case ; (3) self-ventilated, having air-fan ; and (4) forced ventilated (draught) type, in which air is forced through by external means.

Classes of Rating.—If excitation can be varied independently of armature current, the rating shall be associated with a stated condition of excitation and corresponding speed. (1) Nominal one-hour rating to be the h.p. output at motor shaft, at the rated voltage and constant current, which the motor can produce for one hour, the limit of temperature-rise measured by resistance being 100° C. for Class A and 120° C. for Class B insulation, and 90° C. on commutator. (2) Continuous rating is defined as the output at the motor shaft after working continuously with covers, etc., arranged as in service, the temperature-rise measured by thermometer not to exceed 85° C.

* For 'Electric Traction' see Section XIV., p. 669.
† By permission of the British Engineering Standards Association.

on commutator, and on insulated windings measured by resistance 85° C. and 105° C. for Class A and Class B insulation respectively. Class A includes enamelled wire and cotton, silk, paper, etc., when impregnated or oil immersed. Class B, asbestos and built-up and cemented mica insulation. In ventilated motors, the rating shall be the output at rated motor voltage; in totally enclosed motors the output at half the rated motor voltage.

The motor must be run in both directions with fixed brush position under specified test conditions. The statement of efficiency to be made on result of direct measurement corrected to 75° C. and not on the basis of addition of losses. The new machine must withstand, at the conclusion of the one-hour test, an alternating voltage of 1,000 plus twice the line voltage (minimum 2,500 volts R.M.S.), applied for one minute, between the windings and frame. The insulation resistance in megohms, when the high-voltage test is applied, to be not less than line volts/(1,000 + Rated B.H.P.). The specification lays down methods of carrying out performance and voltage tests, and the information as regards type, voltage, speeds, weight, and characteristics, to be supplied by the maker.

G.E.C. 'WITTON' D.C. INTERPOLE TRAMWAY MOTORS.

(The General Electric Co. Ltd.)

These tramway motors are designed for direct voltages 500 to 600 volts, and outputs from 30 to 90 b.h.p. They can be adapted for operating at 1,200 volts. The weight per b.h.p. has been reduced 28·3 lbs. While the majority of machines in this range are intended for car driving-wheels 33 in. in dia. (new), special motors are available for wheels 24 to 27 ins. in dia. The motors are designed within a minimum track gauge. Among the points of advantage incorporated are :—

1. Strongly proportioned box frame construction of high permeability cast steel.

2. Positive axial ventilation of the 'parallel flow' type (Oerlikon system), a double fan at the pinion end drawing two distinct streams of air through the motor.

3. Hoffman roller armature bearings with lubrication and grease sealing, and reliable locking.

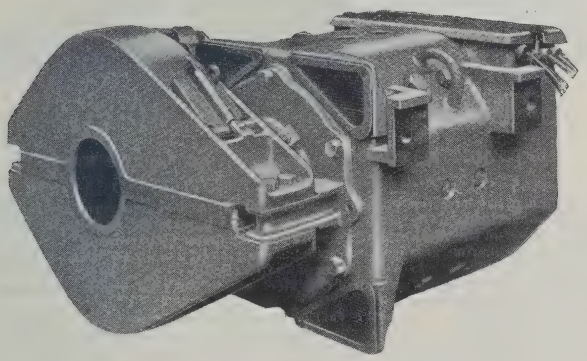


FIG. 31.

4. Armature laminations and commutator mounted on steel spider to avoid restriction of ventilation and to facilitate removal from the shaft.

5. Extensive use of mica insulation of armature coils, with mica, bakelite and porcelain brush-holder insulation.

6. Special care has been given to the design of the lubrication system and facilities for inspection of the motor.

7. Large diameter commutator tightened from the front and permitting of renewal of any segment without removal of commutator.

8. Exceptionally high efficiency.

A general view of the G.E.C. traction motor is shown in fig. 31 and the performance curves in fig. 32.

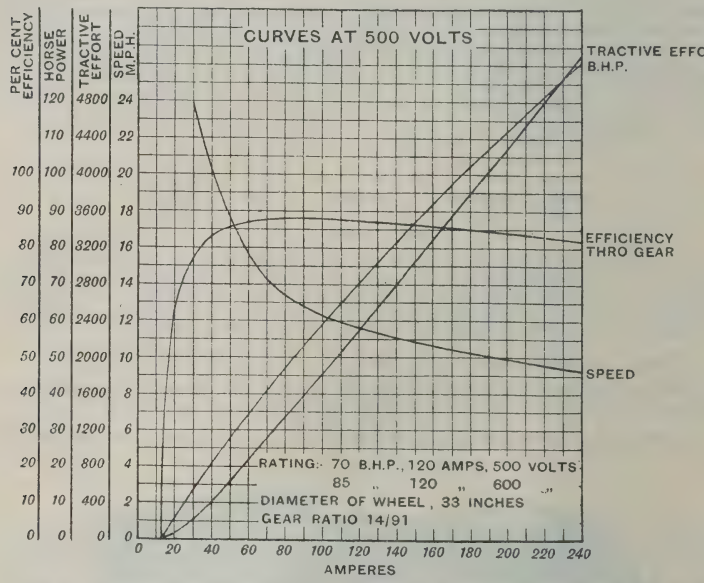


FIG. 32.—Characteristic Curves of G.E.C. 'Witton' D.C. Traction Motor, Type W.T. 32 R.

THE B.T.H. CO.'S SELF-VENTILATED, INTERPOLE TRACTION MOTOR.

(The British Thomson-Houston Co., Ltd., Rugby.)

As an example of the most modern type of light-weight direct-current series motor made by the B.T.H. Co. may be mentioned the motor rated at 52 h.p., 88 amperes at 525 volts, for a temperature rise by thermometer on the windings not exceeding 25° C. on an hourly rating. The weight of the motor complete with gearing, gear case and axle linings is 1,865 lbs. for use with wheel diameters of 24 to 27 ins. and 1,990 lbs. for wheel diameters of 30 to 33 ins.

Magnet Frame.—The magnet frame is of cast steel, and is provided with machined seatings for the pole pieces and brush holders.

Armature Bearings.—The armature is carried in roller bearings, designed to take end thrust, in addition to the radial load. The bearings are arranged for lubrication with grease; greasers are fitted with non-return valve, and screw threads for the connection of the grease gun.

Pole Pieces.—The four exciting pole pieces are built up of soft iron laminations, mounted on a steel key, into which are screwed the bolts for holding the poles to the magnet frame. The interpoles are made of drop forged steel.

Armature.—The armature is built up of soft steel laminations, assembled and pressed together hydraulically, and keyed to ensure that no rotational movement takes place with respect to the shaft. The shaft can readily be removed without disturbing the windings; the commutator can also be removed independently if necessary.

Ventilation.—The motor is self-ventilated by the parallel method. A double-sided fan is provided at the pinion end of the armature which draws air into the frame at the commutator end, expelling it at the pinion end. Air flows in two streams, one stream passing over the surface of the armature and field coils, while the other passes through the armature, by means of longitudinal ducts in the commutator shell and core punchings.

Axle Bearings.—The axle caps are of cast steel, tongued and bolted to machined surfaces on

the magnet frame. The caps have liberal-sized oil boxes designed for lubrication by means of oily waste. Auxiliary wells are provided for filling and for gauging the supply of oil.

Gear Case.—The gear case is made of malleable cast iron, bolted to the frame at three points, and provided with a door in the upper half, spring closed, for inspection and for lubrication of the gearing.

The English Electric Co.'s Direct-Current Traction Motor.

(The English Electric Co., Ltd.)

The direct-current series-wound traction motor gives in result the effect of the change-speed gear of a motor-car engine, which enables a small-powered motor to exert a big effort when required for acceleration or for climbing gradients. It is capable of working both large and small units both for short distances and main line traffic. The three features of design which have contributed to give the d.c. motor its present wide range in traction work are :—

(*a*) commutating poles ; (*b*) ventilation ; (*c*) high-tension design.

(*a*) *Commutating Poles.*—The coils are connected in series with the armature, and their effect is to counterbalance, under the brushes, the field set up by armature reaction, so that commutation under all conditions of loading is practically sparkless, even with the field shunted, thus giving a range of economical speed to traction equipments.

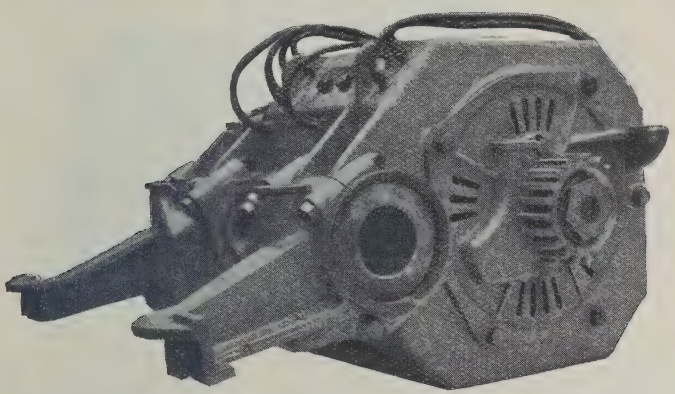


Fig. 33.—English Electric Co.'s Direct-Current Traction Motor.

(*b*) *Ventilation.*—Through the introduction of ventilation, it is now possible to transmit up to 450 h.p. on a single axle, on standard gauge, with 50-in. wheels, on the one hour rating of the motor.

All traction motors manufactured by *The English Electric Company*, ranging from 30 h.p. to 400 h.p., are fitted with commutating poles and are either self- or forced-ventilated. Self-ventilated motors range from 30 h.p. to 330 h.p. Ventilation is effected by means of a fan which forms part of the armature spider at the pinion end. The air is drawn in through apertures at the commutator end of the motor, and expelled at the pinion end. The fan is provided with two sets of blades, one of which draws the air through ventilating ducts in the armature, and the other over the surface of the armature and round the fields. Forced-ventilated motors range from 180 h.p. upwards, the ventilating air being supplied by motor-driven fans.

(*c*) *High-tension D.C. Motors.*—High-tension D.C. motors have been supplied in connection with the following schemes, amongst others :—L.M.S. Railway (England) and Cataluña Railway (Spain), both with a line voltage of 1,200 ; Midi Railway and Paris-Orleans Railway (France) ; Midland Railway (New Zealand) ; Imperial Government Railways (Japan) and The Great Indian Peninsula Railway : the last five having a line voltage of 1,500 ; Montreal Harbour (Canada), line voltage 2,400. Motors of similar design are also being supplied for the Morocco Railways for a line voltage of 3,000.

An experimental line at 3,500 volts gave entirely successful results.

The high-tension motors are specially insulated, mica and mica compound being used throughout for the insulation of armature, field coils, and brushes, so that the dielectric strength of the machines is not affected at high temperature.

On the larger sizes, such as 350 h.p. motors supplied to Chemin de Fer de Midi, the motors drive on the axles through twin gearing, the teeth of the pinions at each end of the armature being staggered to assist in giving uniformity of turning moment.

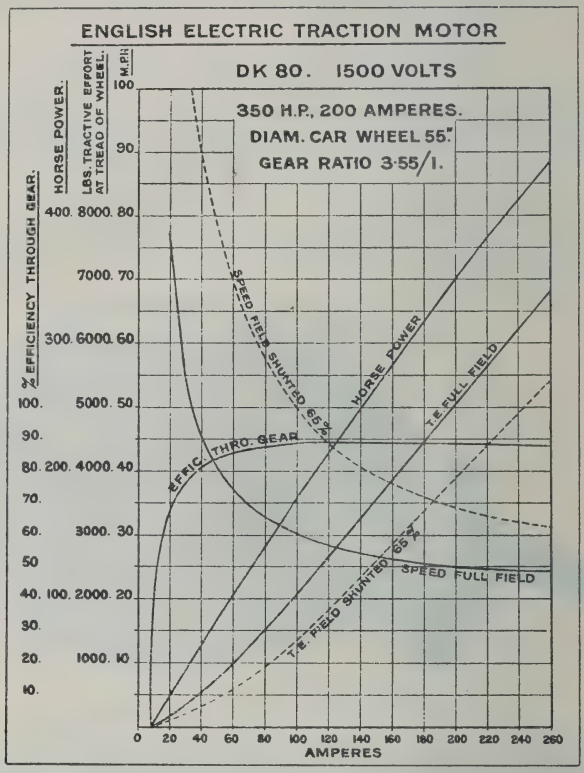


FIG. 34.

Fig. 33 shows a typical English Electric 350 h.p., 1,500 volt, traction motor, of which the characteristic curve is shown in fig. 34.

ALTERNATING-CURRENT MOTORS.

Alternating-current Motors may be divided into the following classes:—Synchronous, Induction, Induction-type Synchronous, and Commutator Motors.

Synchronous Motors.

An alternating-current generator will, if brought up to speed, excited, and connected to a suitable alternating-current supply, run as a motor at a speed in revolutions per second equal to the frequency divided by the number of pairs of poles. Such motors are called synchronous motors, and they will operate only at synchronous speed. They will not start under load, and have to be started by auxiliary means. If overloaded beyond a certain point they will fall out of step with the supply system and come to a standstill. Their sphere of usefulness is limited, and they are most commonly employed as phase regulators on transmission lines, or for driving direct-current generators.

The English Electric Co. Synchronous Motor.

(The English Electric Co., Ltd.).

The following is a specification of a synchronous motor (Class G.L.), made by the English Electric Co.

These machines are made in various standard types as follows: The Engine Base Type, the Sole Plate Type, the Bedplate and Two-Pedestal Bearing Type, the Bedplate and Three-Pedestal Bearing Type, and the Engine Shaft Type.

The *Engine Base Type* consists of a stator frame with feet, core, coils, terminals, and shrouds, one pedestal bearing, a rotor with poles and field coils, shaft, slip-rings and coupling, complete collector gear, screws for fixing stator frame and pedestal to the engine base and screws for adjusting the height of the frame.

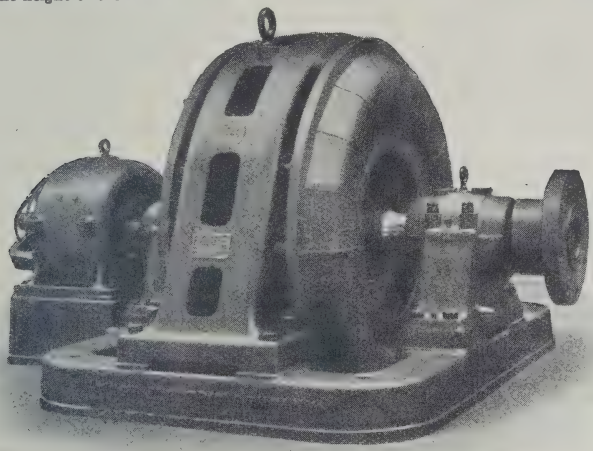


Fig. 35.—English Electric Co.'s Synchronous Motor.

The *Sole Plate Type* is practically the same as the Engine Base Type, with the addition of three sole plates suitable for building directly into the foundations.

The *Bedplate and Two-Pedestal Bearing Type* is similar to the Engine Base Type, but has a second pedestal bearing and a bedplate on which the stator frame and the two pedestal bearings are mounted. The shaft is suitable for drive with solid or flexible coupling or, in special cases, with a gear wheel, or pulley for belt or rope.

The *Bedplate and Three-Pedestal Bearing Type* is the same as the above, but has a third bearing with a correspondingly extended bedplate. The shaft is suitable for fitting with gear wheel or pulley for belt or rope.

The *Engine Shaft Type* differs from the Sole Plate Type by the omission of the shaft, coupling, and the pedestal, together with its sole plate, the rotor being suitably bored for forcing directly on to the engine shaft.

Stator.—The cast iron stator frame is of substantial design and great stiffness, the feet forming an integral part of the casting.

For the small machines the frame is cast in one piece, but on account of transport limitations the larger sizes are cast either in two or four pieces, which are held together by bolts.

Machines of small and medium size can be supplied with either high or low height of centres.

The core is built up of best quality core plate stampings of high permeability and low iron loss, effectively insulated from each other to reduce eddy currents to a minimum. In the smaller sizes the stampings are complete rings, in the larger sizes they are in segments.

The outer circumferences of the laminations are located on steel dove-tail keys, and the laminations are firmly held in position by means of heavy clamping rings. Ventilating ducts are provided at frequent intervals to allow free circulation of air.

The inner circumference of the core is slotted for the reception of the stator winding.

Semi-closed slots are used whenever possible, as they afford a mechanical support for the coils and tend to reduce the excitation loss and the eddy current loss in the pole tips.

The stator winding is usually of the hairpin type held in place in the slots by means of wedges of insulating material. The coils are amply insulated and thoroughly impregnated before being put into place.

The outhanging portions of the coils are protected from mechanical injury by shrouds secured to the frame. In all cases where the pole pitch is large or the duty gives rise to forces likely to distort the windings, these portions of the winding are supported by special brackets so as to give as strong a mechanical construction as possible.

Terminals.—Connections from the stator windings are brought out as heavily insulated flexible leads which pass through a cleat board supported on the frame. This arrangement of terminals presents the safest and simplest means of connecting the machine to the external circuit.

Rotor.—The rotor consists of a cast wheel of steel or iron to which the poles are attached.

For the largest sizes the wheel is cast in two or more pieces, the sections being held together by shrink keys and bolts. The hub is held together by shrink rings.

The hub is provided when required with register and bolt circle suitable for bolting direct to the engine makers' coupling or with a sleeve extension for bolting direct to the engine makers' flywheel. In small and medium-sized rotors, the centre, sleeve and coupling are cast in one piece.

The poles are rectangular in shape, being either steel castings or built up of steel stampings, in either case of high permeability material. They are secured to the wheel by means of bolts provided with lock washers, which pass through the rim from the inside.

For the small sizes when space does not permit of these bolts, the pole laminations are punched with dove-tails for fixing into the rotor centre. These dove-tails are held in place by two taper wedges.

The pole stampings are held together by heavy rivets passing through cast steel flanged end plates, the flanges being formed so as to serve also as supports for the field winding.

All pole pieces are built to permit the use of a damping winding when required, as, for example, in connection with synchronous motors, single-phase machines, and generators driven by gas or oil engines and generally to facilitate parallel operation with other generators. The bars which pass through the pole face are efficiently connected by welded joints to the end rings.

Field Coils.—Whenever possible the field coils consist of copper strip wound on edge and carefully insulated from the pole limb. This form of winding is specially recommended owing to the excellent cooling of the windings and its great mechanical strength. It is advisable, therefore, to select a low voltage for excitation, so that a strip-on-edge winding can be used.

Slip-rings.—These are of gun-metal and are subjected to special heat treatment which makes for good wearing quality of the rings and brushes. The rings are shrunk on to a cast-iron bush insulated by a micanite sleeve. The complete bush with rings is pressed on to the shaft. The rods connecting the rings to the rotor windings are designed so that they can be replaced if necessary without removing the rings.

Brushes and Brush Gear.—The brush holders and brushes are liberal in size, and there are usually two brushes for each ring mounted in one holder.

The brushes used are of a special composition of copper and carbon, which gives low brush resistance, low contact resistance, and low coefficient of friction.

The bearing at the collector end is provided with a seating for supporting the brush spindles which carry the brush holders.

Shaft.—The shaft is designed on ample lines. It is of forged steel finished to size by grinding on the bearing surfaces. The rotor centre, which is pressed hydraulically on to the shaft, is held in place usually by one key sunk in the shaft, but where the drive renders it advisable, tangential keys are used.

Bearings.—The bearings are of the self-aligning sleeve type liberally designed. All bearings are split horizontally to facilitate inspection and removal.

Best quality white metal only is used for the lining. The bearings are of the self-oiling type with adequate oiling rings, the housing being made with a generous reservoir of oil provided with a definite system of oil circulation for the proper lubrication of the bearing and the cooling of the oil.

The housing is furnished with overflow, oil gauge, and drain plug.

Means are provided to prevent oil creeping along the shaft outside the bearing and so reaching the windings.

Balance.—Great importance is attached to smooth running, and the rotor is balanced both during manufacture and at final erection to ensure this feature.

Ventilation.—In these machines special attention has been given to providing a positive system of ventilation. Air is driven outwards by fan blades on the rotor and is directed on to the windings and core by the special formation of the shrouds protecting the overhanging portions of the stator winding.

Exciters.—Whenever conditions permit, direct-coupled exciters are used. In most cases the direct-coupled exciter has its armature overhung on an extension of the main shaft at the collector end, no outboard exciter bearing being used.

The exciter frame is mounted either on a stool fitted to the bedplate or sole-plate, or alternatively on a cast-iron bracket bolted to the bearing housing. In exceptional cases belt driven exciters are used, in which case the exciter has to be supplied with a sliding base, belt and the necessary pulleys both on the exciter and synchronous machine.

Induction Motors.

Induction motors may be classed as polyphase or single-phase induction motors. The action of a polyphase induction motor may be likened to that of a direct-current motor in which both the fields and the armature are capable of rotation. Supposing the fields of such a motor to be fully excited and rotate with the armature at rest and short-circuited, a current of any desired magnitude may be induced in the armature, since the generation of a current depends simply on the relative motion of the fields and the armature. The torque produced by the action of the fields on the induced current will tend to turn the armature, and if the fields be kept rotating at a uniform rate, the armature will speed up until the relative motion of the fields and the armature is sufficient to generate a current that exactly balances the load ; uniform speed has then been reached. The slip, or the difference between the speed of the armature and the speed of the fields, is generally stated in per cent. of the speed that would be reached if the armature were rotating at the same speed as the fields. This speed is spoken of as the speed of synchronism. In a polyphase induction motor the rotation of the field is produced by magnetising the fixed part, or stator, by two or more alternating currents in such a way that the resultant magnetic field rotates at a uniform speed. Generally three alternating currents are used differing in phase by 120°. One advantage of such a motor over the direct-current motor is found in the absence of a commutator, for the current in the armature, or, as it is called, the rotor, can be induced by short-circuiting its coils, the current from the line being led directly into the stator without any moving contacts.

In the single-phase induction motor there is an inductive action between the field and the armature, similar to that which takes place in a polyphase induction motor : the field, however, is not a simple rotating field, but may be regarded as two fields rotating in opposite directions. The result of this is that at starting the armature is subject to an equal pull in both directions—in other words, the motor will not start by itself. But when once the armature has been started in either direction, the torque in that direction increases while that in the opposite direction decreases. As a result, the motor works as an induction machine at a speed just below synchronism similarly to the polyphase motor. This type of induction motor therefore needs some special starting device.

In a polyphase system the ratio of the power component of the current to the total current, as measured by the power factor, depends mainly upon the width of the air-gap between the fixed and moving parts of the motor. Hence it is of great importance to reduce the clearance to the smallest limit consistent with safety. This necessitates very careful construction, especially for polyphase motors on railways. Experience shows that induction motors can be constructed to run well under all conditions.

Variation of Speed with Load.—In a normal polyphase motor the speed is nearly independent of the load. This is in marked contrast to the D.C. series motor, but shows a strong resemblance to the torque-speed characteristic of a normal shunt motor.

Maximum Load.—As the load increases the speed slightly decreases, until the maximum load is reached. If the load be increased beyond this the motor is brought to a standstill. In practice the working load should not exceed 50 to 60 per cent. of the maximum load.

Speed Regulation.—If resistance be placed in the circuit of the rotor so as to increase the total resistance, say, to double its value, the slip for equal torque will be double what it was with the original resistance. Thus with any load not greater than the maximum, any required speed can be obtained by merely altering the rotor resistance. This is a wasteful method of speed regulation, and various methods are in use for obtaining economical speed control. These include :—

(1) Changing the number of poles, with or without change in the number of phases.

(2) Cascade connection with one or more motors.

(3) Addition of variable-speed set.

Starting.—There is a certain rotor resistance corresponding to standstill with maximum torque. If the rotor resistance has this value at the moment when the motor is connected to the line, it will start with its maximum torque. If the resistance is suitably taken out of the circuit as the motor speeds up, the torque during starting can be maintained constant at any value not greater than the maximum. In other words, the torque, and therefore the acceleration, can be maintained practically up to full speed.

Windings.—There are two kinds of windings. In the first the rotor winding consists of copper bars, rather longer than the rotor core, which are all connected together at their ends. In the second the rotor has a winding arranged in such a manner as to allow of an outside resistance being introduced into the rotor at starting to supplement for the time being the resistance of the winding. The former is shortly described as a ' short-circuited ' or ' squirrel-cage ' rotor, the latter as a ' slip-ring ' rotor.

Comparison of Motors.—If the full voltage is applied to the motor at starting with the rotor short-circuited there will be, owing to the low resistance of the bars, a considerable rush of current at starting. In the motor with the slip-ring rotor the current at starting may be kept quite low by means of the supplementary resistance. When giving its full-load effort at starting, such a rotor with resistance in its circuit will take its normal full-load current. A short-circuited rotor will take several times this, but the current can be kept down by using a starting transformer or star-delta switch. These appliances feed the motor at reduced pressure until it begins to get up its speed, but they render motors with short-circuited rotors suitable only for moderate starting torques.

On account of the constructional simplicity of short-circuited rotors it is preferable, whenever possible, to use motors having these. Save as regards starting current, the short-circuited rotor is superior in every respect to the wound rotor. The momentary excess of current it takes at starting does no harm to a small motor, but its effect on the regulation of the supply system may be sometimes objectionable, a point which must be carefully considered.

Effect of Starting Motors.—In choosing motors it is necessary to distinguish between those intended to work on town supply mains where the greater part of the current is used for lighting, and those suitable to a scheme in which the bulk of the current is used for power, the lighting being but incidental. In the former case it is necessary that the working of motors should not interfere with the constancy of pressure on the lighting circuits; in the latter case an occasional jump in the lighting may be of no importance.

The effect of a sudden overdraft of current at a low power factor is disturbance to a greater or less extent of the whole supply system, and fluctuations in pressure such as would cause no inconvenience in the working of a power transmission plant would not be permitted on town lighting circuits. The magnitude of the disturbance depends primarily on the amount of the overdraft relatively to the capacity of the generating plant, and to the margin of current-carrying capacity in the supply network.

Choice of Motors.—It will be evident from the foregoing that the choice of the motor depends upon circumstances, and that neither class can be pronounced the better under all conditions.

Motors with short-circuited rotors take momentarily up to five times their full-load current, if at starting they have to give their full-load effort, or from one to one-and-a-quarter times their full-load current giving 25 to 30 per cent. of their full-load effort, and using a starting device. For connection to town mains it is usual, consequent on the possibility of disturbance to neighbouring lighting circuits which might arise from the starting, to limit this class of motor to 5 h.p. or so, when it has to start at full load effort, or to 10 h.p. when operated with a starting transformer.

In a power transmission scheme where there are no neighbours to consider and the lighting is subsidiary, the short-circuited class of rotor can be employed for large sizes, provided that at starting no individual motor takes a current great enough to prejudicially affect the working of the plant. Thus, in a 500 h.p. transmission, motors of 25 h.p. may be switched right on to the mains without the use of starting transformers, while if the starting effort required is only from 25 to 30 per cent. of the full load effort, motors of 50 h.p. provided with starting transformers are quite permissible.

The conditions under which motors with slip-ring rotors should be used are determined by elimination, that is to say, motors of this class are installed in all cases where it is not advisable to use the other. They have their place on town mains for sizes above 5 or 10 h.p., and in power transmission work, when they have to start at full load effort, and their sizes are comparatively large in proportion to the generating plant. Again, in all cases where motors have to start with overload effort, a slip-ring rotor should be used. Not only do motors of this class give at starting their full load effort with only full load current, but 50 or 100 per cent. over the full load effort can be obtained with a current proportionately increased. This control of the current at starting constitutes the advantage of motors with slip-ring rotors.

The curve in fig. 36 shows the power-factor at full-load of polyphase induction motors, as designed by Messrs. Crompton & Co., plotted as a function of the horse-power per revolution. Other things being equal, the maximum power-factor that can be obtained depends upon the number of poles with a given core diameter. For the same size of machine the larger the number of poles the lower will be the power-factor. It will be noticed from the curves that as the size of the machine increases the curves become very flat, and therefore after a certain point a very large increase in size will only increase the power-factor by a very small amount. For the smaller motors, however, the power-factor increases much more rapidly with the size. The horse-power ratings shown are for a normal temperature rise, and show the maximum power factor that can be reached in these conditions.

The standard windings of these motors are arranged to work on voltages of 110, 220, 440, and 500. They can, however, be wound for any other periodicity and for any higher voltage. The machines can be supplied for polyphase or single-phase circuits with either slip-ring or short-circuited rotors. They are manufactured in 19 standard sizes and are supplied in the same types, and are of the same substantial design and construction as the direct-current motors.

The machines are designed to give a high power-factor at all loads, and to withstand heavy overloads before stalling. When required to start against a load, it is advisable to use motors with slip-ring rotors and starting resistances. Motors with short-circuited rotors are capable of exerting their full-load torque when starting, but the sudden rush of current may be as high

CURVE SHOWING THE POWER FACTOR OF POLYPHASE INDUCTION MOTORS PLOTTED AS A FUNCTION OF THE H.P. PER REVOLUTION.

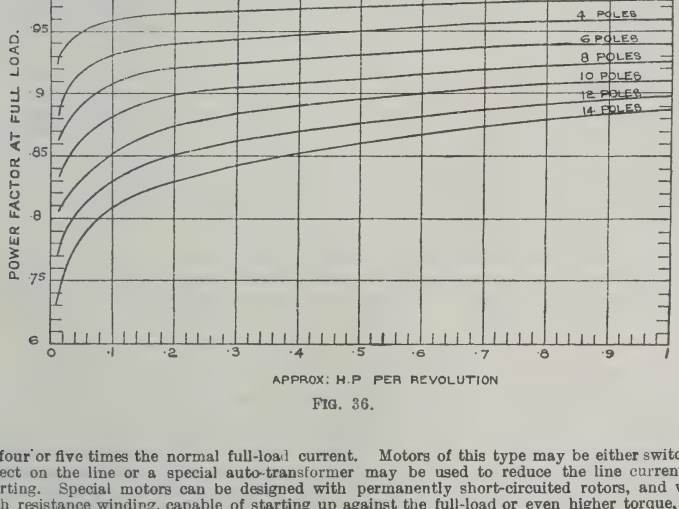


FIG. 36.

as four or five times the normal full-load current. Motors of this type may be either switched direct on the line or a special auto-transformer may be used to reduce the line current at starting. Special motors can be designed with permanently short-circuited rotors, and with high resistance winding, capable of starting up against the full-load or even higher torque, but such motors are usually required for special kinds of intermittent working, such as hydro-extractors.

SPECIFICATION FOR INDUCTION MOTORS.

(The English Electric Co. Ltd.)

The standard machine consists of stator frame with end shields, rotor, terminals, and complete internal connections. Unless otherwise stated in the quotation, the machine is supplied without pulley, coupling, sliding base, foundation bolts, or spanners.

These motors are usually supplied in the Protected or in the Enclosed-Ventilated type, but they are also made as Pipe-ventilated or Drip-proof Motors.

The *Protected type* motor is so designed that the internal rotating parts and live parts are protected mechanically from accidental or careless contact, a perforated metal cover being fitted to the cylindrical opening over the rotor fan.

The *Enclosed-ventilated type* is similar to the protected type, but a perforated metal cover is fitted over both cylindrical openings in the frame.

The stator frame is of cast iron. The stator core is built up of insulated stampings bedding directly on the stator frame. The coils are of the ' push-through ' type in the larger sizes and of the ' dropped-in ' type in the smaller sizes, the slots being nearly closed.

The rotor is built up of stampings with nearly closed slots.

In the case of squirrel-cage rotors the end rings are cast on to the rotor bars. The winding is specially designed to give a good starting torque.

In slip-ring rotors the ends of the windings are separated sufficiently to allow a free circulation of cool air.

A cast-iron terminal box is provided on the stator frame, and is fitted with a flange to take screwed conduit. Up to 650 volts, six terminals are provided, but for certain 210-volt motors the standard terminal box will take three terminals only. For pressures above 650 volts three porcelain-insulated terminals are provided.

A trifurcating box can be supplied to fit the above-mentioned terminal box. A trifurcating box can also be supplied for the rotor leads.

End shields are in the form of end plates bolted to facings on the stator frame, and form substantial supports for the bearings.

Ball or roller bearings, or in certain cases a combination of both, are used, and are of ample proportions. On the larger machines a ball-thrust bearing is fitted.

The bearings are enclosed in dust-proof housings.

This type of bearing has stood the test of lengthy experience in practical use, and is particularly advantageous in the case of induction motors, as there is no appreciable wear of bearings and consequently no change in the length of air gap after extended use. As the bearings will work equally well in any position, provided the shaft is horizontal, the motor may be fixed to wall or ceiling, without altering the position of bearings or end shields, as is necessary with ring-lubricated machines.

The slip rings are of phosphor bronze, and are arranged outside the end shield on an extension of the motor shaft. There is no earth connection to the slip rings.

The standard design permits of the addition of a short-circuiting and brush-lifting device, by which, through the movement of a hand wheel, the slip rings are short circuited and the brushes lifted out of contact with the rings, thus increasing the efficiency of the motor and reducing wear on both slip rings and brushes. With this device, a cover giving total enclosure to the slip rings is always supplied, and the brush holders are of the lever type.

If the short-circuiting and brush-lifting device is not provided, the slip rings are enclosed in a perforated metal cover and box-type brush holders are used.

The following accessories are supplied as extras, if required :—

Pulley, standard or otherwise, or half-coupling or flexible coupling.

Sliding base, with adjusting screws and four bolts for fixing the motor to the base.

Foundation bolts for fixing the sliding base to the foundations.

Spanners and spanner rack.

Cage-Rotor Induction Motor with High Starting Torque.

Although high-starting torque in cage-type induction motors can be obtained by means of high-resistance rotors, the consequent large rotor heating and reduced efficiency render such a construction unsuitable for many drives. Various designs have been evolved which impart to the robust cage-type motor a large starting torque without these drawbacks. Such motors may be started either by switching direct on to the mains or by means of a star-delta switch—a device usually preferred to the more costly and less simple auto-transformer starter. The various devices which have been developed for obtaining automatically a large inherent starting torque depend upon the effect of reactance, which is proportional to the rotor frequency or slip.

The common practice is to employ a double-cage rotor, consisting in its simplest form of two separate cage windings. The outer or starting cage has high resistance and low inductance; the inner or running cage has low resistance and—since it is sunk well into the rotor slots—high inductance. The large reactance of the inner cage at starting, due to the high frequency, keeps the current in this winding small; while the current in the high-resistance rotor cage produces a large starting torque. When running, the rotor frequency is low, and the main part of the current is carried by the low-resistance inner winding. The disadvantage of increased cost is partly offset by economies in control gear, and such a motor may cost less than a slip-ring type with its starter. The reduced power factor can be improved by suitable modifications in the design. In many drives, however, a slight sacrifice in efficiency and power factor is of small consequence compared with simplicity of control and robust construction.

Multiple-Speed Induction Motors.

The 'cascade' principle was originally applied to two motors mechanically coupled with their windings electrically connected in such a way as to make it possible to obtain two speeds. In the case of induction motors, the arrangement adopted was to connect the rotor slip-rings of motor No. 1 to the stator windings of motor No. 2, the torque of the combination being controlled by resistances placed in the circuit of the rotor of No. 2, slip-rings being used. If both machines had an equal number of poles, the 'cascade' connection gave effects equivalent to doubling the number of poles. Or if, for example, one was an eight-pole and one a four-pole motor, the speed for 'cascade' connection was that for a twelve-pole machine. Considerable use has been made of this system, notably on railways, the locomotives on the Valtellina line and for the Simplon tunnel being operated on this principle. The same effect may be secured by connecting the two rotors in series: the starting resistances may then be placed in the stator circuit and the use of slip-rings can be avoided.

In the Hunt 'Cascade' Induction Motor made by Sandycroft, Ltd., this principle has been developed so that the advantages secured hitherto only with two motors can be obtained in a single motor. In an ordinary induction motor, the stator acts on the rotor and there the inductive action stops. Starting torque and speed regulation can only be obtained by inserting resistances in the rotor winding. In the 'Cascade' motor, the rotor, by means of a supplementary magnetic field, reacts inductively upon the stator, to which the starting and speed regulating resistances are connected. The rotor in fact reflects back to the stator part of the energy received from it. This is accomplished by winding the rotor in such a way that it can, under the inductive action of the stator, simultaneously produce two magnetic fields of different numbers of poles; one of these reacts on the stator and generates in its winding induced currents. Each branch of the stator winding is divided into two parallel circuits which are tapped at suitable points. In the case of single-speed motors, or two-speed motors requiring starting or regulation with reduced rheostatic losses below two-thirds speed, these points are connected to resistances by which the torque and speed are controlled

Owing to the presence of two magnetic fields, one of which is at its maximum when the other is at its minimum, a very even turning movement is obtained at slow speeds, and a perfectly even pull with maximum torque can be obtained at as low as two revolutions per minute. This is a feature of importance with all kinds of winding engines. The slip of an induction motor depends upon the resistance of the windings in which the secondary currents circulate. In the 'Cascade' motor, these windings are in the stator and are of very low resistance, consequently the slip in these motors is very much less than that of ordinary induction motors, and the speed variation between full load and no load is only about 1·2 per cent. For constant speed work the absence of slip-rings is a matter of moment, especially in mining installations. These motors develop full-load starting torque with full-load current, and have a higher power factor than ordinary motors at low speeds where the tendency is for the power factor to become unfavourable.

The question of economical running at two or more speeds is one of great importance. The synchronous motor must run in step with its generator, and the induction motor runs at a speed very nearly that of synchronism. The speed of the former cannot be varied, the speed of the latter can only be varied by inserting resistances in the rotor at great loss of efficiency. A simple modification of the windings of the 'Cascade' motor, described above, enables two speeds to be obtained without rheostatic losses. Slip-rings are connected to a normal rotor winding in star mesh connection. When these rings are open circuited, the portion of the winding connected to the slip-rings is idle, the rotor reacts on the stator, as previously described, and the motor runs at 'Cascade' speed. When the slip-ring circuit is closed, the rotor winding becomes symmetrical, and produces only one number of poles; there is no reflex action on the stator and the motor runs as an ordinary induction motor. Thus, for example, in a twelve-pole 'Cascade' motor, running at 500 r.p.m., there are two fields, one of eight and one of four poles, the latter being the result of rotor reaction. When this is removed by current through the slip-rings, the effective number of poles is eight, and the motor runs at 750 r.p.m. without any resistance. If intermediate speeds are required, they may be obtained by the use of resistances in the usual manner.

A self-synchronising synchronous type of Hunt motor is now on the market and has many unique features. It resembles the Cascade Induction Motor, but has a modified form of stator winding. At synchronous speed the auxiliary field is stationary, and is excited by direct current supplied to the stator windings. These windings are connected to form two parallel-connected star windings, and the exciter is coupled to the neutral points. The direct current produces a field having half as many poles as the main field. The machine is started by resistances connected to the slip-rings, and when so connected the speed can be taken up to synchronous (Cascade) speed or even above it, so that all the usual difficulties of pulling-in are obviated. The motor will start and synchronise with loads of two to three times normal, and with a high starting-power factor and low current consumption. With motors designed for a leading power factor of 0·7, the starting current for full-load torque is of the order of 10 per cent. to 15 per cent. more than normal full-load current, and the starting-power factor is 88 per cent. to 93 per cent., depending upon the speed for which the machine is built. A remarkable feature possessed by this machine is its entire immunity from 'hunting' or 'phase-swinging.' Extensive tests have been made, but no trace of these troubles has ever been found.

The latest development of the Hunt machine is a self-paralleling generator. This motor closely resembles the synchronous motor, but, as slip-rings are not required, the rotor winding is of the low-voltage short-circuited type as used for the induction motor. These machines possess the same properties as the synchronous motor, and will run in parallel under conditions which would be impossible with generators of the usual type.

Other economical speed changes can be obtained in addition by Creedy's extension of the principle of the cascade motor, or by pole-changing, or by phase changing. A special application of the two-speed motor, apart from economical running at two speeds, is the avoidance of large starting peaks by two-speed starting in connection with winding and similar work, combined with a reduction in starting losses.

Induction-type Synchronous Motors.

The induction-type synchronous motor is a polyphase motor with a cylindrical rotor; it is started as an induction motor of the slip-ring type, and synchronises itself when direct current is switched on the rotor. It combines the good starting characteristics of the induction motor with the property of power-factor correction of the synchronous motor, and for this reason its

use is extending. It may with advantage be run at a leading power factor at all loads; the leading current increases somewhat as the load is reduced, and the speed is constant at all loads. A small D.C. exciter is mounted on the motor shaft. The stator is identical with that of the ordinary induction motor; the rotor may be wound three-phase or single-phase. Whilst starting, with a resistance in the rotor circuit, the machine exerts a large torque, and can be made to accelerate against more than double full-load torque. The maximum current during starting is about twice the full-load current, and persists for a very short period. The overload above which the motor falls out of step is 100 per cent. or less, depending on the power factor for which the machine is designed; on falling out of synchronism it will usually continue to run as an induction motor until the overload is reduced. In the operation of starting, the main notch is closed, and the rotor starting resistance is gradually cut out and short-circuited. The rotor circuit is then momentarily opened, and closed on the exciter; and the exciting current is adjusted to give the desired leading power-factor.

In some cases the power is supplied to the rotor, and the excitation to the stator.

The efficiency is slightly lower than that of a plain induction motor; the cost is 25 to 40 per cent. higher (including all accessories). The machine is most suitable for a continuous load of 100 h.p. or more.

AVERAGE EFFICIENCIES OF POLYPHASE INDUCTION MOTORS.

(*Meares.*)

Output, B.H.P.	1	5	10	20	50	100	150	200	300
Efficiency per cent.	80	85	86	88	89	90	91	92	93

Alternating-current Commutator Motors.

Alternating-current commutator motors are in most respects similar to ordinary direct-current motors, and may have either series or shunt characteristics. They operate under serious disadvantages, owing to the transformer e.m.f. induced in the coils short-circuited by the brushes, necessitating a low commutator voltage; but for certain purposes they are practically indispensable. For single-phase railroad traction the 'neutralised series' motor is used, the speed being controlled by a variable-ratio transformer. For smaller powers the single-phase 'repulsion' motor is very suitable, the speed being controlled by moving the brushes over a range from standstill to 30 per cent. above synchronism. Single-phase commutator motors with shunt characteristics are also satisfactory machines. The efficiency of single-phase shunt commutator motors is somewhat lower than that of direct-current motors, and the cost somewhat higher, but the weight is less.

Three-phase commutator motors are also made to work with a series or with a shunt characteristic, but have not come into extended use, except for special applications such as steel rolling-mill drives. Where variable speed is necessary, the multi-speed or the cascade induction motor can be considered as alternatives to A.C. commutator motors.

VARIABLE SPEED POLYPHASE COMMUTATOR MOTORS.

(*The British Thomson-Houston Co., Ltd.*)

This motor, known as the Schrage motor, has been developed by The British Thomson-Houston Co. for industrial applications requiring variable speed. While possessing the speed characteristic of a D.C. shunt-wound motor, it is suitable for operation on polyphase alternating current circuits.

The motors are provided with the following windings:—

(1) Primary winding located on the rotor and, by means of slip rings, connected to the supply leads.

(2) Secondary winding located on the stator.

(3) Regulating winding placed in the same slots on the rotor as the primary winding. The regulating winding is connected to a commutator and thence, by means of brushes, in series with the secondary winding on the stator.

For normal speed regulation, no external resistance is required in order to effect speed variation, and hence the motor operates with high efficiency at all speeds. Normal speed variation is obtained simply by turning the handwheel provided, which changes the position of the brushes on the commutator. The movement of the brushes can also be effected very conveniently from a distance either mechanically by means of wire rope and pulleys or by spindle and bevel gear wheels, or electrically by means of a small pilot motor coupled to the brush-shifting spindle and controlled by push buttons.

The polyphase shunt motor—unlike the D.C. variable-speed shunt-wound motor, where the maximum torque is developed at the lowest speed and decreases as the speed increases—exerts a constant torque throughout the whole speed range, *i.e.* the output is directly proportional to the speed.

The normal speed range of the standard motor is 3 : 1.

The motor operates at speeds above or below the synchronous speed and at any speed between the maximum and minimum values. When running near synchronous speed the motor operates exactly as an ordinary induction motor, the secondary winding being short-circuited through the brushes across one or more commutator segments. According to the amount that the two sets of brushes are displaced in one direction or the other, so the speed will vary either above or below synchronous speed. It is usual, therefore, to select a synchronous speed about midway between the maximum and the minimum speeds desired.

The difference between the no-load speed and the full-load speed for any brush setting is approximately 5 per cent. to 10 per cent. of the maximum speed of the motor, the exact percentage depending on the size of the motor. A creeping speed can be obtained by inserting a small resistance in each phase of the stator winding.

This motor is more stable than the ordinary induction motor when running at creeping speeds because it has normal operating speeds much below the synchronous speed.

Reversal of direction of rotation is effected by interchanging two of the line leads connected to the slip-rings.

Carbon Brushes.

*British Standard Specification for Parallel-sided Carbon Brushes for D.C. Commutator Machines. No. 96—1922.**

Definitions are given of centre line, brush dimensions, chamfer, and bevel, and these are exemplified by figs. 37 and 38, chamfer being the slope produced by removal of the four edges to prevent binding in holder box, and measured by taking one-half of the difference in side dimensions before and after chamfering (see Table II.). Flexibles to be of twisted copper strand, the

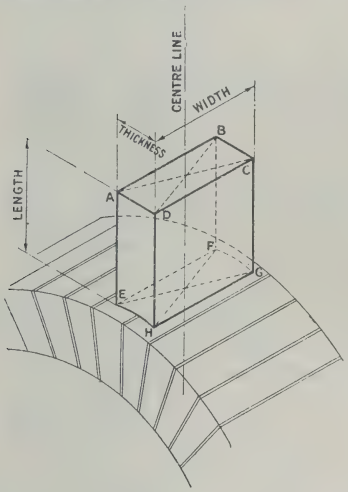


FIG. 37.
ABCD is the top.
EFGH is the contact surface.
AEFB, CGHD } are the sides.
BFGC, DHEA }
AE, BF, CG, DH are the edges.

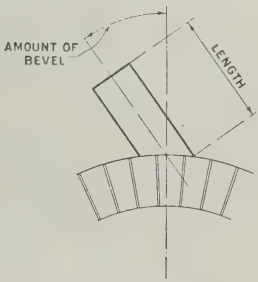


FIG. 38.

diameter of each individual wire not exceeding 0·0048 inch (0·12 mm.). One terminal only of the shoe type pressed on to the flexible shall be provided, the copper terminal shoe for all sizes being suitable for a No. 1 B.A. fixing screw (Table I.). Table III. refers to all classes of carbon and metal carbon brushes. Dimensions of compressed, unburnt graphite brushes increase with temperature. Sizes, when cold, must be stated by maker, and shall be less than values in Table III. at maker's discretion. When hot, dimensions must not exceed the values in Table III. Tolerance

* By permission of the British Engineering Standards Association.

permitted in actual dimensions stated by maker: length +0·060 inch, width +0·008 inch; thickness +0·006 inch to −0·000 inch in each case. The brush boxes shall be of the same nominal size as brushes with tolerances of width +0·008 to +0·001 inch, thickness +0·005 to +0·001 inch.

TABLE I.—FLEXIBLES AND TERMINALS.

Brush No. (See Table III.)	Number of Flexibles.	Length of Flexible.	
		Inches.	Approx. Metric Equivalents. mm.
1 to 3	1	3	76
4 to 8	1	3½	89
9	1	4	102
10 to 15	2		

TABLE II.—AMOUNT OF CHAMFER.

Brush No. (See Table III.)	Chamfer.	
	Inch.	Approx. Metric Equivalents. mm.
1 to 3	0·040	1·02
4 ,, 8	0·060	1·52
9 ,, 15	0·060	1·52

TABLE III.—NOMINAL DIMENSIONS.

Brush No.	Length.		Width.		Thickness.		
	Inches.	Approx. Metric Equivalents. mm.	Inches.	Approx. Metric Equivalents. mm.	Inches.		Approx. Metric Equivalents. mm.
1	1¼	1·250	⅝	0·625	¼	0·250	6·35
2					5/16	0·3125	7·94
3					⅜	0·375	9·53
4	1½	1·500	⅞	0·875	5/16	0·3125	7·94
5					⅜	0·375	9·53
6					½	0·500	12·70
7					⅝	0·625	15·88
8					¾	0·750	19·05
9	1¾	1·750	1¼	1·250	⅜	0·375	9·53
10					½	0·500	12·70
11					⅝	0·625	15·88
12					¾	0·750	19·05
13					⅞	0·875	22·23
14					1	1·000	25·40
15					1⅛	1·125	28·58

The dimensions of copper-plated brushes shall be measured over the plating.

Manufacturers of

CHEMICAL PLANT
LABORATORY APPARATUS
ACID-RESISTING CONTAINERS
ELECTRICAL NOVELTIES, etc.

can develop their business
at Home and Overseas
on new lines by consistent
advertising in

THE CHEMIST AND DRUGGIST

This Journal has been in existence since 1859, and is the recognised medium for the Chemical, Drug and Allied Trades.

It is the official organ for seven Pharmaceutical Societies in Ireland and in the Colonies.

10,092 COPIES SOLD WEEKLY
(Net Sales Certificate)
ANNUAL SUBSCRIPTION 20/-

Advertising rates from
The Publisher,
THE CHEMIST AND DRUGGIST,
28 Essex Street, Strand,
London, W.C.2.

Branch Offices:—
4 Cannon Street, MANCHESTER. 54 Foster's Bldgs., High Street, SHEFFIELD. 19 Waterloo Street, GLASGOW.
MELBOURNE and SYDNEY (Australia). AUCKLAND (New Zealand).

CONTROL

ENERGY uncontrolled is a destroyer — badly controlled, a menace — perfectly controlled, your friend and servant. Centralise your government with SIMPLEX Unit Panels; they give a perfection of control, an ease and mastery in working, never obtainable with widely scattered units.

WRITE FOR LIST

SIMPLEX CONDUITS LIMITED

Garrison Lane . . Birmingham

Branches at LONDON, GLASGOW, MANCHESTER, LIVERPOOL, BRISTOL, NEWCASTLE, NOTTINGHAM, LEEDS, CARDIFF, SHEFFIELD, BELFAST, and in OVERSEAS DOMINIONS

SIMPLEX

Soundest Value *Surest Satisfaction*

Electric Power Switchboards.

British Standard Specification No. 162—1928 deals with Electric Power Switchboards for indoor Installations up to and including 33,000 volts.

British Standard Switchgear.

Specifications for air-break switches and circuit breakers for pressures up to 660 volts are given in the *British Engineering Standards Association's Reports* 109—1923 and 110—1923. The specifications do not apply to switchgear for use in circuits containing high inductance. Lists of standard current ratings are given, and it is specified that the temperature rise on continuous full load shall not exceed 20° C. for switchgear rated below 100 amperes, or 30° C. for 100 amperes and upwards.

Distinction is made between: (1) A switch, being a mechanical device for non-automatically breaking a circuit carrying load-current; (2) An isolating switch for disconnecting a circuit only when carrying no load current; and (3) A circuit breaker, a mechanical device for breaking automatically a circuit under abnormal conditions such as that of short circuit.

Marble or slate slabs must be free from metallic veins, and, if of slate, must be treated all over with a damp-proof medium; the slabs are to be insulated from fixing bolts and the supporting framework. The sizes of round copper-back connections and their spacing are tabulated in the Reports. The test pressure to be applied to a switch or circuit breaker is 1,000 volts plus twice the rated pressure, with a minimum of 2,000 volts alternating.

In the case of circuit breakers, the temperature rise of coils is not to exceed 50° C. when the insulating material is impregnated cotton, silk, or paper, or enamelled wire, and 70° with materials such as asbestos, micanite, and built-up mica, and bare coils. Reverse-current devices must be capable of operating with 15 per cent. or less of the rated current; low-voltage releases must not operate unless the pressure falls to 65 per cent. or less of the rated pressure, but must operate before the pressure falls below 25 per cent. of the normal, at the rated frequency. A reverse-current circuit breaker must not be rendered inoperative if the pressure across the potential coil falls to (Class A) zero, (Class B) 50 per cent. of normal, and a reverse-current device must not operate with a forward current of any magnitude. Magnet-closing circuit breakers must operate between the limits 20 per cent. below and above normal pressure, and pressure-operated tripping coils must operate between the limits 50 per cent. below and 20 per cent. above normal pressure at 40° C.

When type tests have been made by a recognised authority, a millivolt drop test is to be accepted in lieu of a temperature test.

Specification 116—1929 gives the standard ratings, sizes, and marking, the temperature and voltage tests for oil-immersed switches and breakers for alternating current circuits. The temperature rise above the cooling air at any part must not exceed that measured at the hottest part of the oil, viz. 30° C. for ratings up to and including 2,000 amperes, 40° C. above 2,000 amperes; when the temperature of the cooling air exceeds 40°C the temperature rise must be such that the total temperature attained does not exceed 80° C. Temperature limits of coils and test voltages for switches as in Nos. 109 and 110; but, when operated from secondary of transformers the maximum testing voltage for coils to be 2,000, and for switches rated for 1,000 volts and upwards, 2,000 volts *plus* 2½ times the rated voltage. These conditions shall also be fulfilled after erection on site. Type tests specify that the circuit breaker shall interrupt its rated k.v.a. breaking capacity twice at a two-minute interval. A switch shall break twice in succession twice the rated current at rated voltage and after this carry its normal current without exceeding limits of temperature rise and then break twice the rated current a third time. A mechanical endurance test is to be carried out by operating 500 times with the main contacts carrying no current. Methods of calculating k.v.a. to be broken and notes on selection of oil circuit-breakers are given in appendices.

Specifications for totally enclosed air-break switches (No. 124—1923) and air-break circuit breakers (No. 130—1923) for circuits not exceeding 660 volts provide, in addition, for efficient earthing of containing case, arc shields and lining, and insulation of the interlocked handle. The insulation resistance must not be less than 10 megohms. No. 194—1926 specifies switchgear equipment for these circuits. No. 195—1929 specifies 33,000-volt, 3-phase switchgear.

Condition to be fulfilled by flame-proof air-break switches and circuit breakers (Nos. 126—1930 and 127—1930) require a test of flame-proofness by subjecting the apparatus to a single breaking test when filled and surrounded with the maximum explosive mixture of methane and air or the equivalent. Switches must then break a circuit carrying a current 50 per cent. in excess of normal at rated voltage on a non-inductive load; circuit breakers to operate under same conditions of current, voltage, and impedance as in service.

Flameproof enclosures for electrical apparatus and tests for flameproof enclosures are standardised in No. 229—1929. The performances required refer to conditions in the coal-mining industry.

Specification 158—1924 standardises bus-bar arrangements, connection diagrams, symbols and distinctive markings. No. 159—1925 gives sizes for bus-bars and connections constructed of bare copper and/or aluminium. Slate slabs for electrical purposes are standardised for tolerance on dimensions, shape, and flatness in No. 160—1923. Switchgear cells and cubicles

constructed of moulded stone, as regards concrete composition and preparation, strength tests on slabs and mouldings, design of cubicles, accuracy and finish are dealt with in No. 268—1926.

Specification No. 214—1929 deals with Distribution Boards up to 250 volts and 30 amperes per way.

Electric cut-outs for low pressure (Type O) are dealt with in Specification No. 88—1919.

In Specification No. 142—1927 of the British Engineering Standards Association, electrical protective relays are classified as to function and characterics. Temperature rise, limits of volt-ampere consumption and error for each type are laid down.

ELECTRO-MECHANICAL BRAKES.

Electro-mechanical brakes are used in connection with a variety of electrically driven machines, such as lifts, cranes, hoists, elevators, conveyors, rolling mills, paper reeling, slip-ways, presses, machine tools, etc., where either a quick 'stop' or adjustable gradual 'stop' is desired when the power is cut off either intentionally or unintentionally.

These brakes consist primarily of an electro magnet which operates the brake blocks through a system of levers or other medium. The magnet is so arranged that, when the current is applied, the magnet is energised and lifts the brake blocks from the drum. If for any reason the current is cut off the magnet is de-energised and the brake blocks are pressed against the drum, the requisite pressure being obtained by springs, weights, or a combination of each.

The brakes are quite automatic in action; thus, in the case of an electrically driven crane, as soon as the driver applies the current to the motor to raise the load the brake is automatically released, and when he switches off the current to stop the brake is immediately applied. Should for any reason the supply current fail, the brake automatically goes on and takes charge of the load.

It is desirable that an electro-mechanical brake should have the following features: compactness of design, robust construction, few and simple working parts, easy adjustment for brake blocks and pressure exerted for same. Coils should be liberally rated, well insulated, and protected from mechanical injury, and so placed that they are not likely to be damagee by oil exuding from the machine. The brake blocks should be readily renewable. If springs are used they should be of the compression pattern.

(*Elliston, Evans & Jackson, Ltd., London.*)

Motor Starters and Controllers.

'Starters' are devices for starting and accelerating a motor to normal speed, and comprise (*a*) Rheostatic starters (face-plate, drum, multiple switch, contactor, and liquid starters); (*b*) Compensation or auto-transformer starters for 2- or 3-phase motors; (*c*) Switch starters (star-delta connections for 3-phase starters, series-parallel connections for 2-phase machines).

'Controllers' (including shunt regulators) are used to regulate the speed of a motor, and may or may not be used for starting.

The following British Engineering Standards Association Specifications deal with starters:—

No. 117—1923. Drum starters for electric motors.

No. 118—1923. Drum controllers and resistances for use therewith, for electric motors (D.C. and A.C. slip-ring).

No. 123—1923. Face-plate controllers and resistances for use therewith, for electric motors (D.C. and A.C. slip-ring type).

No. 141—1923. Switch starters (star-delta and series-parallel) for electric motors (A.C. induction, without slip-rings, 2 to 40 h.p.).

No. 129—1923. Contactor controllers and resistances for use therewith, for electric motors (D.C. and A.C. slip-ring type) (excluding controllers for traction purposes).

No. 140—1923. Liquid starters for electric motors (D.C. and A.C. 2- and 3-phase induction with slip-rings).

No. 147—1923. Multiple switch starters for electric motors (D.C.).

No. 155—1923. Contactor starters for electric motors (D.C. and A.C. 2- and 3-phase induction with slip-rings).

No. 167—1923. Auto-transformer starters (hand-operated pattern) for electric motors (A.C. 2- and 3-phase induction without slip-rings).

No. 246—1927. Face-plate starters for electric motors (D.C.).

No. 247—1927. Face-plate rotor starters for electric motors (A.C. 2- and 3-phase induction with slip rings).

No. 280—1928. Field rheostats for electric generators, motors, synchronous converters and balancers.

AUTOMATIC CONTROL GEAR.

An automatic controller entirely eliminates the personal judgment of the operator in many cases, which is a distinct advantage where men accustomed to heavy work such as is met with in shipyards, steel mills and the like, are called upon to start up large machines. The controllers are now made suitable for practically any purpose, any horse-power, any supply, and for any particular design of motor requiring special starting apparatus.

The foolproof nature of automatic gear is not the only or the principal argument in its favour as compared with hand-operated controllers under frequent and heavy service.

An automatic controller is one in which the current supply to the motor is regulated by means of one or more switch units, each unit being operated by an electro-magnet; the operation of the magnet is controlled by an auxiliary switch, which therefore only has to deal with very small magnet-currents. This auxiliary switch can be put at any distance from the motor in the most convenient place, whereas the main automatic controller can be installed beside the motor, thus effecting the greatest economy in cable connections.

The original face-plate type of starter, which is still the standard type for moderate size motors starting up only a few times per day, has had to give place to the drum type controller for frequently operated motors as on cranes. This type of controller has been greatly improved in its mechanical and electrical construction, but its design has such inherent limitations when it has to deal with the large currents now employed that excessive wear of the contacts is produced. This necessitates constant attention to the wearing parts, and high maintenance charges. By inherent limitations is meant the fundamental fact that in order to carry larger currents the contact pressures must be increased, and such increase greatly increases the mechanical wear, so that in some cases the contacts may have to be renewed after a few weeks' operation. With an automatic controller of proper design, the wear is very much less, and does not increase with the size of the controller. Contacts operating under severe conditions will only require renewal about once or twice a year.

D.C. motors are controlled by a number of solenoid-operated switches or 'contactors' fitted with powerful magnetic blow-outs and arc shields; they are electrically interlocked so that they can only close in correct sequence, and so arranged that the motor cannot be started up except with all resistance in the armature circuit.

In some cases the time intervals between the closing of successive contactors depend upon the acceleration of the motor. This is generally determined by means of series relays fitted to each contactor except the last one. These relays are in the main motor circuit, and are adjusted so as to operate when the motor current drops to the value at which further acceleration is impossible. The operation of the relay closes the control circuit of the next contactor, which therefore closes, thereby cutting out a section of the resistance and speeding up the motor. This is repeated each time until the motor reaches full speed. It is thus evident that the motor current is kept within safe limits while the motor is accelerated to full speed in the minimum time, which means the minimum loss of power in the starting resistances.

The automatic control of any A.C. motor is now as reliable as with a D.C motor, for A.C. switches and contactors are manufactured for controlling motors of any horse-power on high or low tension supplies. A standard controller for a slip-ring motor consists of a stator switch or contactor and a number of D.P. rotor contactors complete with relays, the number depending on the h.p.

The stator switch coil is controlled by a small switch, accumulator switch or float switch, pressure regulator or other means, and on being energised closes the stator switch automatically, and the motor is connected to the mains with all the rotor resistance in circuit.

A time element controls the closing of the first D.P. rotor switch, which, on closing automatically, cuts out a section of the resistance in the three phases of the rotor circuit. As with the d.c. set, this is repeated until the motor is up to full speed with the leads to the rings short-circuited.

Pumps are arranged to stop and start according to the rise or fall of water in a tank by means of float gear and an automatic controller. The automatic control of air compressors is being universally adopted on account of the great convenience and economy effected in many cases, irrespective of the benefit of the elimination of skilled attendance.

Each plant requires individual investigation in order to ascertain the most economical method of control. On a D.C. or A.C. supply the compressor can be arranged to stop and start at maximum and minimum pressures only, or to stop and start at maximum and minimum pressures, the speed being reduced or increased between certain definite pressures, unloading at starting, or to run continuously, unloading only at maximum pressure, when the demand for air is great.

For certain duties, such as lifts, plate-edge planers or live rolls, operated by A.C. slip-ring motors, where a large initial starting torque is usual, the 'Eddystat' controller has many unique advantages. This controller is called an 'Eddystat' controller because it utilises the surplus power required when accelerating, to produce eddy current losses, just as the ordinary resistance type of controller uses up the surplus power in creating an ohmic loss in the resistances. The

controller has no moving parts, but requires a contactor for operating in the stator circuit, and when the duty is such that the motor runs continuously for a considerable period of time, it is preferable to have a double pole switch to short-circuit the 'Eddystat' controller.

The arrangement consists of three solid steel cores and insulated copper windings with tappings, which are connected across the sliprings, as shown in fig. 39. On starting the motor the rotor current is forced through the controller at line frequency, and the highest rotor volts, and produces the maximum eddy current loss. As the motor speeds up, the voltage and frequency decrease in proportion, and consequently the eddy current losses decrease automatically.

DIAGRAM OF CONNECTIONS OF 'EDDYSTAT' CONTROLLER.

Braking and Reversing.—This controller also acts as a very powerful electric brake, because if the stator circuit is reversed when the motor is at full speed, twice line frequency and twice standstill rotor voltage will be applied to the controller. For automatic reversing equipments, such as plate-edge planers or live rolls, this reduces the time of reversal, and increases the output. The maximum current when reversing in this manner is very little in excess of the current taken when starting up from standstill. The controller thus acts like a plugging resistance, but with the additional advantage that a single-speed master controller is sufficient. This eliminates the judgment of the operator, which is necessary when plugging with a three-point master controller. As the 'Eddystat' controller is always connected in the rotor circuit the motor is fully protected, a valuable feature absent in other forms of control.

Oil Immersed Type.—The 'Eddystat' controller is specially suitable for installation in fiery mines, explosives factories and chemical works, as, in addition to having no moving parts, it can be totally immersed in an oil tank, like a transformer.

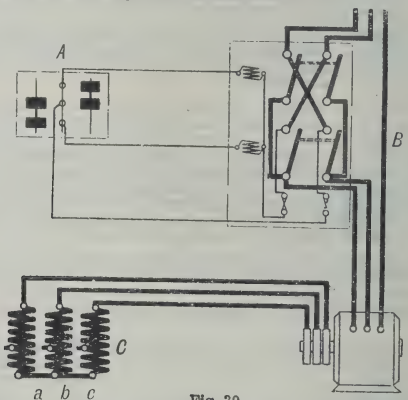


Fig. 39.

A—Master Controller. *B*—Reversing Contactors. *C*—'Eddystat' Controller.

NOTE.—An increase in the starting torque can be obtained by moving the star connection *a b c* up to the tappings immediately above.

A further increase is possible by connecting the coils in delta.

For the control of large cranes, whether the supply is direct or alternating current, the standard practice at the present time is to install contactor controllers on the crane with small master drum controller in the driver's cabin, in preference to the large drum controllers previously used.

Absolute control is afforded, and the manipulation is more efficient, with the added advantage that, due to interlocking, no damage can be caused to the gearing through carelessness. When lowering, the driver can vary the braking effect according to the load; in fact he has as efficient control as with a drum controller, but the apparatus being foolproof needs less care and attention, and frequent repairs are eliminated.

Automatic controllers are being installed in all up-to-date steel works and rolling mills, on charges, blast-furnace hoists, screw-down gear, live rolls, lifting tables, skids, and main rolls.

TRANSFORMERS.

TRANSFORMERS IN PARALLEL.

The simple distributing transformer usually has two high-voltage leads and four low-voltage leads coming out from its windings, as indicated in 1, fig. 40. It will be seen from fig. 2 that the two centre leads of the low-voltage coils are crossed inside the transformer before they are brought out through the case. This does not make any change in connecting the two sections in series on the outside of the transformer, as this is the same as connecting two batteries in series.

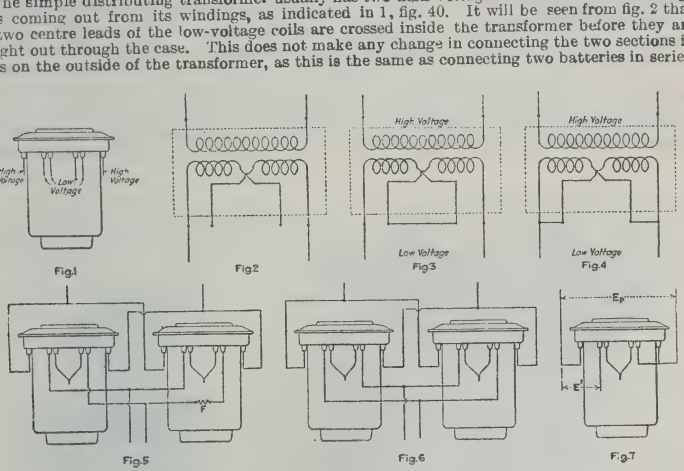


FIG. 40.

When the coils are connected in parallel, care must be exercised and adjacent leads connected together, as in fig. 4. Looking at the connections from the outside of the transformer, it would appear as though the two sections of the winding were short-circuited, but the diagram shows that the right-hand terminals of each section are connected together, also the left-hand terminals. This is just what is done in connecting batteries or other direct-current elements in parallel—like poles are connected.

Transformers to operate in parallel should be the same voltage, frequency, size, and possess the same characteristics, which means that they should be of the same make. In connecting transformers in parallel, it is only necessary to observe the same rules as used for connecting two or more pieces of direct-current apparatus in parallel; namely, connect like poles to like poles. The question now arises which are like terminals on two or more transformers. This is easily determined in most cases. After the transformers are arranged symmetrically, as in fig. 5, connect the right-hand terminals together and the left-hand terminals together and bring out a lead from each junction point, as indicated.

There is, however, a chance that one of the transformers may have its leads brought out in such a way as to cause a short-circuit when the transformers are connected as shown in the figure. To guard against this a piece of small fuse wire may be connected, as indicated at F. If the connections are correct, the fuse will not blow when the transformers are made alive without load, otherwise the fuse will blow. In case of a wrong connection it may be remedied by crossing either the high or low-voltage leads on one of the transformers, as indicated in fig. 6. A better way would be to test the transformers and find out which has wrong polarity and correct it.

The polarity may be tested as represented in fig. 7. Connect the high-voltage terminals to the line and with one terminal of the low-voltage coil to the high-voltage winding, as indicated in the figure. Measure the electromotive force at E*p* and E'. If it is greater at E' than at E*p*, the transformer has the correct polarity; if less, the polarity is wrong and either the high or the low-voltage leads must be crossed inside the transformer. The test may be made with 110 or 220-volt current.

THE ENGLISH ELECTRIC CO.'S. TRANSFORMERS.

In these transformers the windings are designed so that there is ample provision for the circulation of oil; this assures a very low temperature gradient between copper and oil, so that any danger of excessive heating is eliminated. Two patterns are manufactured, viz. the 'core' type

Self-cooled transformers can be built in sizes up to approximately 20,000 kVA. capacity, but in sizes above 4,000 KVA. a saving can usually be made by artificial cooling. Methods of artificial cooling of oil-immersed transformers include a water-cooling coil fitted inside the tank at the top; an outer sheet metal casing for cooling by air blast from motor driven fans; pumping the oil through a cooler of either the water or the air-blast type mounted outside the transformer.

Dry transformers up to about 150 kVA. single-phase and 200 kVA. three-phase are supplied protected by an expanded metal screen. Above this capacity, dry-type single-phase, air-blast transformers have been supplied to a limited extent for ratings from 300 to 1,500 kVA.

The increasing cost of erecting buildings has emphasised the advantages of installing transformers out of doors. This practice is being favoured for large power transformers in the layout of modern large generating stations and sub-stations, for which Metrovick transformers of the outdoor type have been installed for service up to 132,000 volts, and also is being adopted increasingly for power and lighting transformers in rural distribution systems.

A development that is finding increasing favour is a tapping switch mounted within the transformer and operated from outside the case. This switch enables voltage changes to be made quickly and without opening the tank, so that interruption is reduced to a minimum and even at the highest voltages perfect safety is ensured to the operator. Transformers have been supplied so arranged that tap changing can be effected without any interruption in the continuity of supply. Another convenient device for tap changing on distribution transformers is a set of sockets and adjustable plugs. This device enables transformers of a standard voltage to serve the requirements of a whole system, the voltage of individual transformers being adjusted, at the time of installation or subsequently, to compensate for voltage drop in the transmission lines.

Notes on Transformers.

To seal up holes in transformers or oil-switch cases where wires pass through, a mixture of sulphur and plaster of Paris may be used; a mixture containing equal quantities gives a material which sets very hard, and which is impervious to transformer oil. The sulphur should be melted in a ladle or pot and then an equal quantity of plaster of Paris mixed thoroughly into it. This mixture should be poured into place while hot.

Trouble is often experienced from oil being siphoned out of transformer cases on account of the braided insulation on the lead wires. When this takes place it is usually found that the insulation on the wires dips down into the oil. If the oil level is lowered below the terminal board inside the case and the insulation on the leads scraped off so that where the wire enters the surface of the oil its metallic surface only is exposed to possible contact with the oil, and the leads are properly sealed, there will be little trouble due to siphon action.

Mining type transformers are dealt with in B.E.S.A. Specification No. 355—1929.

Terminal Insulator Bushings.

Insulated terminals for transformers and switchgear require careful design for high-voltage working. The two principal types built are (*a*) oil-filled, and (*b*) condenser bushings. In each case it is the object of the construction to make the dielectric stresses or electric field densities sensibly uniform in radial distribution from the high-voltage conductor; and to avoid corona loss. This is effected in the oil-filled bushing by surrounding the conductor where it penetrates earthed metal by a long hollow porcelain cylindrical insulator, the space between the conductor (usually a hollow tube) and the porcelain cylinder being filled with transformer oil. Provision for expansion has to be made to accommodate excess oil due to heating. Condenser bushings are usually composed of bakelised paper and tinfoil wound directly on to the conductor, the tinfoil being arranged in concentric cylindrical layers, separated by bakelised paper, and forming a series of condensers of graded capacity. As far as possible the various capacities are made equal, resulting in a nominally uniform radial distribution of dielectric stress. Thus as the radius of a tinfoil condenser increases, its length is reduced. Condenser bushings are characterised by the 'stepping' of the lengths of successive tinfoil cylinders.

Static Balancers in Three-Phase Circuits.

In three-phase distribution circuits, the voltages commonly used for power purposes are often too high for lighting circuits; accordingly, where lighting is desired, it is necessary to introduce some means of obtaining a reduced voltage. A customary method of providing for this is to take advantage of the fact that by using a star connected generator or transformer having its secondary winding connected in star, it is possible to obtain between any one phase and the neutral point of either generator or transformer a reduced voltage. For example, in a circuit with 400 volts

between the phases, it is possible by taking out a connection from the star point, to obtain a voltage between this point and any one phase of $\frac{400}{\sqrt{3}} = 230$ volts only. To do this, however, the generator or transformer has to be star connected in such a manner that a cable can be brought out from the star point, and this cable must be led to the point where lighting is required. Apart from the cost of this cable, which for any distance is considerable, any out-of-balance lighting would cause unbalanced voltages due to the resistance of this cable and thus would affect the

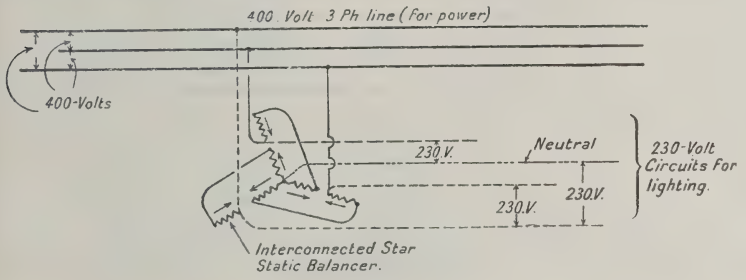


FIG. 42.

lighting detrimentally. Alternatively to the above arrangement, it is possible, by using static balancers, to save the cost of the cable referred to, and also to minimise very largely the out-of-balance effects.

The Metrovick static balancer, which is shown diagrammatically in fig. 42, consists of a three-phase auto-transformer having a lead brought out from the neutral point of the windings, and the three phases so inter-connected, that current flowing through the neutral passes equally through them.

Static balancers for ordinary industrial conditions are quite small, and of correspondingly low cost. On a 100-KVA. circuit, with a maximum out-of-balance current not exceeding 25 per cent. of the full-load current in the outers, a balancer would be similar in size to a 5-KVA. transformer, and under these conditions, the out-of-balance of the phase voltage would correspond to one-third the percentage voltage drop of such a 5-KVA. transformer. In cases where the voltage required for lighting is not exactly but only approximately the line voltage divided by $\sqrt{3}$ a combined balance and auto-transformer may be used to advantage.

The advantages gained by using static balancers are: (1) The balancer may be placed at any point on the system where lighting is required and the length of neutral wire greatly reduced. (2) The drop in the neutral wire is much reduced, so that better voltage balancing is obtained between the different phases. (3) The current is more equally distributed between the line wires. (4) A standard polyphase wattmeter can be used instead of a special four-wire wattmeter. (5) It is possible to supply a four-wire circuit from a transformer group connected in star-star or star-delta, thereby saving in the cost of the main transformers.

Induction Regulators and Booster Transformers.

(The English Electric Company, Ltd.)

The principal applications of induction regulators and booster transformers are:

1. The regulation of feeder voltage under all conditions of load.

2. The control of the reactive current in ring mains and in interconnector cables linking power areas where it is desirable that the station voltages shall be variable independently of each other.

3. The control of the alternating voltage applied to the slip-rings of synchronous converters in order to obtain voltage control on the direct current side.

Booster Transformers.

The apparatus consists of a potential and a series or current transformer. The primary winding of the potential transformer is connected across the mains and the secondary provided with a number of tappings which are brought out to a regulating switch. The series transformer has its secondary winding in series with the line and its primary excited from the variable secondary of the potential transformer, as shown in fig. 43. When the reversing arm of the regulating switch

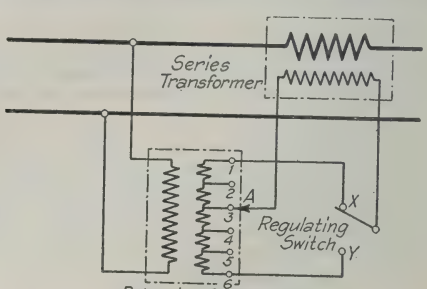


Fig. 43.

is on contact X, and the tapping contact A is on stud No. 1, the primary winding of the series transformer is short-circuited and the secondary of the potential transformer is open-circuited. The voltage of the feeder therefore is unaffected and the potential transformer takes only a magnetising current. As contact A is moved successively to studs 2, 3, etc., a voltage will be induced in the secondary winding of the series transformer and the outgoing line voltage increased. The voltage boost is thus a maximum on stud No. 6. If the operation be commenced with the reversing arm on control Y, then the tapping contact A on stud No. 6 gives normal mean voltage, and movement to successive studs Nos. 5, 4, etc., will induce in the series transformer a voltage opposing the line voltage; while on stud No. 1, the outgoing feeder voltage will be a minimum. To ensure continuity of circuit, the English Electric Co.'s booster transformer has a main circuit breaker and a pilot circuit breaker with renewable contacts which take all the arcing, the pilot breaker completing a circuit through a choke coil or buffer reactance. A 3-phase booster transformer consists of three separate identical units coupled together for simultaneous operation by hand, or motor operated from remote control, or by means of contactors. Where the supply voltage does not exceed 3,000 it is possible to omit the series transformer and to use the tapped secondary winding of the potential transformer directly in series with the line.

Induction Regulators.

Induction regulators provide a means of gradual change in the feeder voltage without the interposition of switches. They consist of two elements, one fixed and the other movable at will through 90 degs. for the 4-pole design and 180 degs. for the 2-pole. In double-range regulators the movements are through 180 and 360 degs. for 4- and 2-pole types respectively. The construction is similar to that of an induction motor, but usually with a vertical shaft. The rotor shaft projects through the tank cover and has keyed to it a worm quadrant which engages through reduction gearing with a handwheel or motor.

With single-phase regulators, the magnetic flux produced by the movable primary winding is pulsating and the voltage induced in the stationary secondary winding, connected in series with the line, is proportional to the amount of this flux linked with it. The movable primary carries a short-circuited compensating winding which is electrically and mechanically at right angles to the primary winding. As the regulator approaches the neutral position the reactance of the series winding is neutralised by this short-circuited coil.

In three-phase induction regulators the magnetic field rotates and the secondary windings are linked by a constant value of flux irrespective of the angular position. The secondary voltage is therefore constant and is added to or subtracted from the feeder voltage by varying its phase angle relatively to the feeder voltage. This is shown in fig. 44, in which OA is the line voltage, OB is the secondary voltage per phase, and AB the resultant feeder voltage. The regulated voltage AB and the unregulated voltage OA are only in phase with one another at the maximum and minimum

positions of the regulator. In ring mains or in an interconnector cable linking generating systems, phase displacement would cause circulating currents to flow. To correct this displacement it is necessary to resort to (1) the twin induction regulator, where each unit is half the total kVA. rating required, or (2) a bank of single-phase regulators.

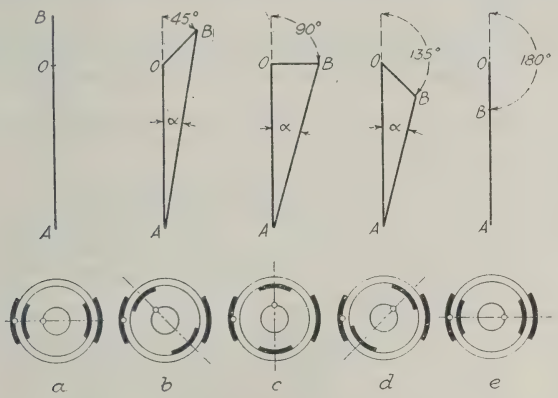


FIG. 44.

The English Electric Co.'s induction regulators are designed for air cooling and for oil self cooling for upwards of 1,000 kVA. in one unit. Very large sizes are designed for forced oil circulation. There is a growing tendency to accommodate induction regulators with the transformers and switchgear of outdoor substations. The operating gear is then enclosed in a weatherproof cover and the control panel, with its sensitive relays, mounted in an air-tight metal cabinet.

Power Factor Improvement.

With the continual growth of alternating current distribution, the necessity of improving the power factor resulting from an industrial load has become more and more apparent, as a low power factor, by reducing the working capacity of the generating plant and mains and lowering the efficiency of the system, adds to both the fixed and the running charges, besides unfavourably influencing the voltage regulation of the network. In fig. 45 curves are given for commercial induction motors at 500 and 1,000 r.p.m., showing the effect of output and speed on the power factor at various loads.

The power factor can be raised by three methods:—(1) Static condensers; (2) Synchronous condensers; (3) Phase adjusters.

(1) Static condensers afford one of the simplest and most efficient solutions of the problem, as they require no attendance and involve very little loss of energy. In the *General Electric Review* of August 1918, W. B. Taylor gives a diagram, fig. 46 (page 2309), by means of which the capacity of the static condenser required to neutralise more or less completely the wattless component of the power can be determined. The use of the diagram is best explained by an example. An a.c. feeder carries a current of 500 amperes at 500 volts, and the power factor is 0·6; required the capacity necessary to raise the power factor to 0·9. Taking on the 0·6 p.f. line the point corresponding to 250 kVA., it will be found that the wattless kVA. (ordinate) is 200 and the energy kW. (abscissa) 150. The 150-kW. ordinate cuts the 0·9 p.f. line at a height corresponding to 73 wattless kVA. Subtracting 73 from 200 gives 127 kVA. as the required condenser capacity. Similarly 88 kVA. would raise the power factor to 0·8 and 47 kVA. would raise it to 0·7.

The effect of added capacity is greatest when the power factor is lowest; owing to the relatively large capacity required to bring the power factor up to unity, it is not commercially advisable to attempt to raise it much above 0·9. The condensers should be fixed near the load, in order to improve the power factor in the mains as well as at the generator. The condensers should be disconnected at times of light load, if the leading current they can then take be not required in other parts of the system. High voltage and high frequency are favourable to the use of static condensers; they are made for pressures up to 750 volts by the *British Insulated Cables, Ltd.*, and for higher pressures they are connected in series. In the case of low pressures such as 400 volts, it may be found economical to install transformers to raise the

voltage at the terminals of the condensers, thus reducing the capacity required. Another method is to insert a suitable amount of inductance in series with the condensers, whereby the pressure across the latter is increased. To protect the condensers, the control switches should be fitted with overload trips, and to prevent accidents due to the stored-up charge, provision should be made for discharging the condensers through resistances when they are switched off. The efficiency of condensers exceeds 99·5 per cent.

(2) Synchronous condensers are simply synchronous a.c. motors, over-excited so as to take a leading current, in which condition they act like static condensers. The motor may be run without load, but in this case the method is costly, and it is preferable to use a loaded motor. A rotary converter can be employed for the same purpose. The necessary capacity to produce

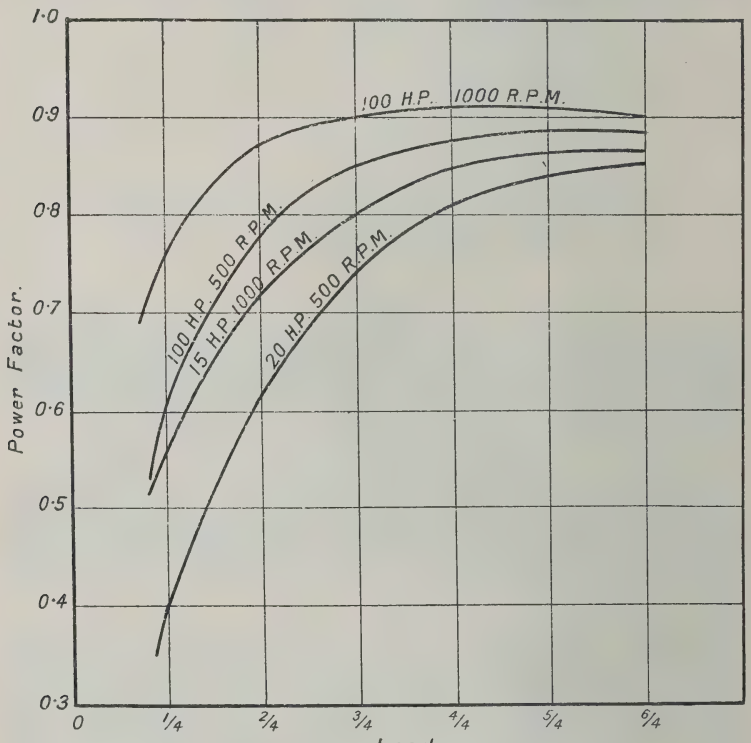


FIG. 45.

a given improvement in the power factor of the system is determined in the same way as that of static condensers, using fig. 46; the greater the mechanical load on the motor, the less will be its condenser effect, the rule being that the square of the kVA. capacity of the machine is equal to the sum of the squares of the kW. load and of the wattless kVA., so that the motor is used to the best advantage when the wattless output = the power output = 0·71 of the kVA. rating of the motor. The synchronous motor is usually started up as an induction motor with the aid of an auxiliary squirrel-cage winding embedded in the poles of the field system, or by means of an induction motor coupled to the same shaft, the field being unexcited until full speed is attained; direct-current excitation must be provided, either by a direct-coupled exciter or by an external source.

(3) The phase adjuster is a special machine, resembling a commutator induction motor, which is coupled to the slip-rings of an induction motor whose power factor needs correction, and is mechanically driven, either from the shaft of the main motor or by a small separate motor, the power taken being only that required to overcome friction and windage, and being fully compensated by reduced losses in the main motor. The function of the phase adjuster is to supply the magnetising current for the motor, thus raising the power factor to unity, or even passing that

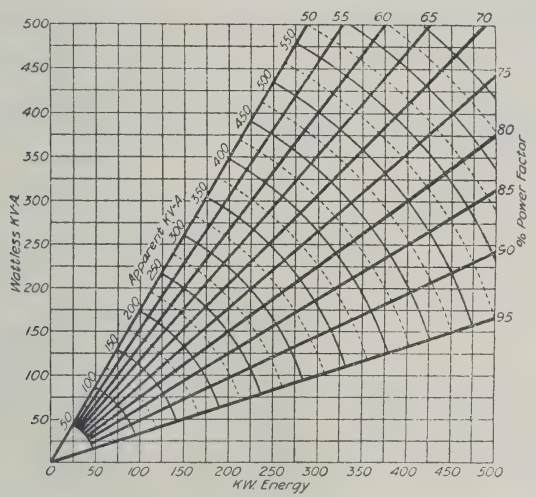


FIG. 46.

point and supplying current to the system to compensate for low power factor in other machines. An oscillatory type of phase adjuster is also made. A great advantage of this method is that the power factor of the individual motor is improved, so that a smaller motor can be used, with reduced losses. Advantages common to all methods are that the losses in the generators and cables are reduced, and the load factor of the plant is raised so that fewer generators need be run, and considerable economy of fuel as well as of capital outlay can be effected.

SIZES OF ENAMELLED COPPER WIRE FOR ELECTRICAL INSTRUMENTS.

A 'tentative' British Standard Specification No. 156—1922, has been issued, giving the dimensions and resistances of enamelled copper wire for instruments and apparatus. The sizes of the bare wires correspond with those given in Specification No. 128—1929. Tests are prescribed as follows:—

Mechanical.—The enamel must not be easily removed from the wire with the thumbnail. Below 0·0084 in. diameter, a 10-in. sample shall be extensible to 11 ins. without cracking the enamel; above that diameter, a sample shall be wound round a mandrel not more than four times the diameter of the bare wire without cracking the enamel.

Chemical.—The enamel shall not be dissolved when immersed for one hour in absolute alcohol or dilute sulphuric acid (sp. gr. 1·25); and the samples shall pass the mechanical tests after immersion as above.

Baking.—After baking at 266° F. for one hour the samples shall pass the mechanical tests.

Electrical.—Test samples twisted together shall withstand an arc test at a pressure varying from 150 volts (R.M.S.) up to 0·0032 in. to 1200 volts above 0·029 in. diameter. A sample passed through a mercury bath at a p.d. of 6 volts d.c. shall not make more than forty contacts in 50 yards.

The thickness of the coating prescribed ranges from $\frac{1}{4}$ mil on the smallest wires to $\frac{1}{2}$ mil at 6 mils diameter, 1 mil at 28 mils diameter, and 1·1 at 48 mils, but wide limits of variation are permitted. Tables of minimum and maximum dimensions are given.

Electric Measuring Instruments.

Ammeters and Voltmeters.—The types available are (1) Moving-coil, permanent-magnet; (2) Moving-iron; (3) Hot-wire; (4) Dynamometer; (5) Induction type; (6) Electrostatic voltmeters.

The following British Engineering Standards Association Specifications deal with electric measuring instruments:—

No. 81—1927. Instrument transformers.

No. 89—1929. Indicating ammeters, voltmeters, watt-meters, frequency and power factor meters.

No. 90—1929. Graphic (Recording or Chart-Recording) ammeters, voltmeters, and watt-meters.

Limits of Error in Ammeters, Voltmeters, and Wattmeters.

B.E.S.A. Specification No. 89—1929 classifies ammeters, voltmeters, and wattmeters into grades of sub-standard, first-grade and second-grade accuracy. The permissible limits of error are tabulated below:—

Table I.

Sub-standard Ammeters, Voltmeters, and Wattmeters.

Instrument.	Limit of Error over the Effective Range expressed as a percentage of the Maximum Scale Value.		
	Permanent-Magnet Moving-Coil Type.	Dynamometer Type.	Other Types.
Voltmeter, single-range, self-contained or with external resistance	0·2	0·3 *	0·5 *
Voltmeter, multi-range, self-contained or with external resistance	0·3	0·4	0·6 *
Voltmeter, for use with voltage transformer	—	0·3	0·5
Ammeter, single-range, self-contained	0·5	0·5	0·5
Ammeter, single-range, for use with separate shunt or transformer	0·3	0·5	0·5

	Single Element for D.C. or Single-Phase or Three-Phase Balanced Load.	Double Element for Polyphase Unbalanced Load.
Special dynamometer wattmeter, as described in Clause 30	0·25	—
Dynamometer wattmeter with single self-contained current-range, and single voltage-range, self-contained or with external resistance	0·5 †	1·0 †
Dynamometer wattmeter, but with single current-range and multi-voltage-range	0·6 †	1·2 †
Dynamometer wattmeter, but with double current-range by series-parallel connection and single voltage-range	0·5 †	1·0 †
Dynamometer wattmeter, but with double current-range by series-parallel connection and multi-voltage range	0·6 †	1·2 †

* For voltmeters with rated voltage exceeding 75 volts.

† Provided that the rated voltage be not less than 75 volts, and the rated current not more than 10 amps., nor less than 1 amp.

TABLE II.

FIRST- AND SECOND-GRADE AMMETERS, VOLTMETERS, AND WATTMETERS.

Instrument.	Limit of Error.			
	From the Maximum of the Effective Range to Half the Maximum Scale Value expressed as a Percentage of the INDICATION.		From Half the Maximum Scale Value to the Lower Limit of the Effective Range expressed as a Percentage of the MAXIMUM SCALE VALUE.	
	First Grade.	Second Grade.	First Grade.	Second Grade.
Voltmeter, single-range, self-contained or with external resistance	1·0	2·0	0·5	1·0
Voltmeter, multi-range, self-contained or with external resistance, permanent-magnet moving-coil type	1·2	2·4	0·6	1·2
Voltmeter, multi-range, of other types	1·5 *	3·0 *	0·75 *	1·5 *
Voltmeter, for use with voltage transformer	1·0	2·0	0·5	1·0
Ammeter, single-range, permanent-magnet moving-coil type, for use with shunt	1·0	2·0	0·5	1·0
Dynamometer ammeter, self-contained, for A.C. 5 or 10 amperes				
Ammeter, single-range, self-contained, or for use with transformer	2·0	4·0	1·0	2·0
Wattmeter, with single self-contained current range, and single voltage-range, self-contained, or with external resistance	2·5	5·0	1·25	2·5
Wattmeter, but with single current-range and multi-voltage range	3·0	6·0	1·5	3·0
Wattmeter, but with double current-range by series-parallel connection, and single-voltage range	2·5	5·0	1·25	2·5
Wattmeter, but with double-current range by series-parallel connection, and multi-voltage range	3·0	6·0	1·5	3·0

To determine the variation in the indication of a wattmeter for use with a 3-phase unbalanced load, the error is found with a balanced load and then with one current coil interrupted, but the same instrument-reading. The difference, expressed as a percentage of the maximum scale value, shall not exceed 0·5 per cent. in a sub-standard instrument, 2 per cent. in a first-grade, and 4 per cent. in a second-grade instrument.

* Provided that the rated voltage of the lowest range is not less than 100 volts, and the ratio of the highest to the lowest range not more than 5 to 1.

Variations due to Temperature.—The standard temperature is 20° C. The percentage variation in the indication of an instrument due to a change of 1° C. in the temperature of the surrounding air shall not exceed :—

Instrument.	Sub-standard.	First Grade.	Second Grade.
Voltmeter . .	0·05	0·1	0·2
Ammeter . .	0·10	0·2	0·4
Wattmeter . .	0·075	0·15	0·3

Variations due to Power Factor.—The variation in the indication of a wattmeter, for a change in the power factor of the load from unity to 0·5 shall not exceed 0·5, 1·0, and 3·0 per cent. of the reading in sub-standard, first-grade and second-grade wattmeters.

Limits of Error in Frequency and Power-Factor Meters.—The errors in a frequency meter shall not exceed 1 per cent. of the mean of the frequencies shown on the scale, nor a change of voltage of 10 per cent. produce more than 1 per cent. change.

The limit of error in a power-factor meter tested from 0·5 to 1·0 power factor at any current not less than one-fifth of the maximum, and at the rated voltage, shall be 2° of angle ; and a voltage change of 10 per cent. from rated value shall not cause a change in the indication of more than ½° of angle.

Weston Standard Ammeters and Voltmeters.

Direct Current.

In these instruments, although the scales are remarkably uniform, no particular 'law' of deflections is assumed, as must be done when printed or engraved scales are used. A skeleton of each scale is laid out by actual comparison with an accurate standard, and the lines afterwards are inked in. Finally, the work is checked by comparison with the same standard. No multiplying constant is necessary (except with high-range voltmeters), as simple inspection of the position of the pointer on the scale gives the value in amperes or volts. Time in making readings is therefore saved, and errors of calculation are avoided. The scales of all direct-current instruments have practically uniform divisions from zero to the top mark. The range is thus greater than is common, while the uniform size of the scale divisions permits of their visual subdivision with uniform accuracy. The pointer comes to rest at once, although absolutely no friction can be detected, thus reducing the time required for readings, and preventing unnecessary wear of the moving parts.

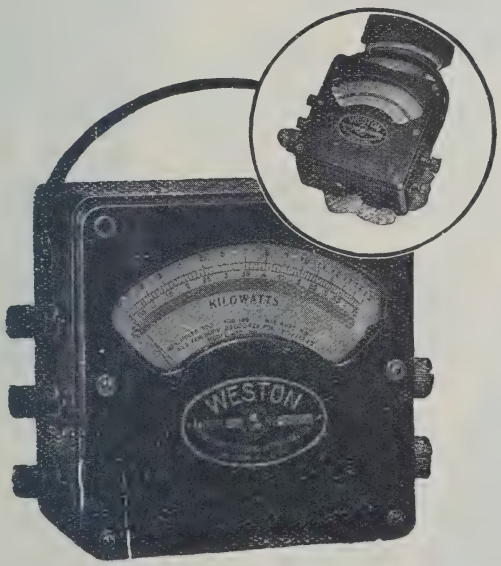
There is no 'magnetic lag,' giving rise to different deflections with the same current or potential, as in instruments having iron in the moving parts. The 'temperature correction' is negligible, being for all voltmeters having a range above 15 volts, not greater than ¼ per cent. for a range of 35° F. above, or 35° below, 70°, and for the ammeter, less than 1 per cent. for the same range of temperature. The instruments of the 'Direct Current' type may be kept in circuit any length of time, as the heating effect, except in the higher ranges, is so small as to be inappreciable. In the voltmeters provided with a 'calibrating' coil, changes in scale-value resulting from accidental injury are easily checked ; this is a very valuable feature, as all instruments are subject to occasional accidents and to rough usage, which may prove injurious. The user of these instruments can at any time roughly check their scale-values by applying a single cell of some constant form to the terminals of the calibrating coil.

Alternating and Direct Current.

These instruments can be used in measuring the current or voltage of both alternating and direct-current circuits with equal accuracy, being of the electrodynamometer type. Owing to the principle of operation, they cannot be made with a uniform scale throughout, but the upper two-thirds of the scale is practically uniform. To prevent errors from parallax, mirror scale-plates and knife-edge pointers are provided. The entire movable and field coil system is mounted in a double closed iron shield, which effectually protects it from external magnetic fields and electrostatic influences. Damping is accomplished by means of an air damper, which is very effective.

WESTON

ELECTRICAL INSTRUMENT CO. LTD.



SMALL TYPE PORTABLE A.C. INSTRUMENTS.

These new instruments are supplied as Wattmeters, Voltmeters, Ammeters and Mil-ammeters.

They are guaranteed accurate to ½% on D.C. Circuits or A.C. Circuits of any commercial frequency.

The scale length is 3·8″ and a set of four instruments, including our new Model 461 Current Transformer may be conveniently carried in a small suit case.

Write for List A. 7.

15 GREAT SAFFRON HILL, LONDON, E.C. 1.

Telephone No: 2029 Holborn. Telegrams: 'Pivoted, Smith, London.'

The Engineer.

ESTABLISHED 1856

1/- WEEKLY

The Metallurgist

IS A FREE MONTHLY SUPPLEMENT

to The Engineer.

EDITORIAL and PUBLICATION OFFICES:

28 ESSEX STREET STRAND, LONDON, W.C.2

ANNUAL SUBSCRIPTION RATE £3 5 0

F. 2313

In voltmeters it is necessary for great accuracy to have the current taken by the instrument as small as practicable; the Weston Model 341 voltmeter is designed with this object in view, and the current required is much less than is usual in this type of instrument. For ordinary measurements no corrections are necessary for temperature or for frequencies up to 133 cycles per second, nor is it necessary to take reverse readings on direct-current circuits. When great accuracy is required, however, such corrections can be made. The instruments may be left in circuit continuously. In general, they are guaranteed to have an accuracy of ½ of one per cent. of full-scale value on the working portion of the scale.

Similar characteristics are found in the portable wattmeters made by the company.

All ammeters are really milli-voltmeters, as they merely measure the drop of potential across a constant resistance. This constant resistance is termed a shunt. The potential drop is directly proportional to the current passing through the shunt, and if the shunt is correctly designed, the instrument can therefore be graduated in amperes. The resistance of these shunts is very low: for a Weston switchboard ammeter of 100 amperes the resistance of the shunt is approximately 0·0005 ohm. The shunt must not be strained when fixing to the bus bars, for straining might affect the resistance and so alter the reading of the instrument. For this reason shunts are usually securely fastened to a base of hard wood. Low-range ammeters up to about 200 amperes can have the shunt contained within the instrument case. The resistance of the instruments does not exceed about 2 ohms, and about 25 to 30 milli-amperes is required for full-scale deflection.

Voltmeters which are placed directly across the circuit must be high-resistance instruments. These only require about 10 or 15 milli-amperes for full-scale deflection. For special testing work Weston voltmeters can be obtained which have a resistance of 2,500 ohms per volt, and only require 0·0004 ampere for full-scale deflection. It is important to have a high-resistance voltmeter for testing the E.M.F. of dry batteries, etc.

Owing to the temperature coefficient of copper being high, all copper in the instrument should be reduced as much as possible to eliminate temperature errors. The moving system should be as light as possible to reduce the wear upon the pivots, which should work in good jewels. The air gap must be small to ensure the permanence of the magnet, which must be specially aged and carefully made. In Weston instruments the air gap is approximately 0·04 in. Any number of different range shunts may be used with moving coil ammeters, so that accurate readings may be taken over a wide range.

More current is required to operate moving iron instruments than is necessary for moving coil instruments. For this reason moving iron voltmeters are not suitable for testing dry cells. The resistance per volt of moving iron instruments is about 12 to 20 ohms, against 60 to 100 or higher for moving coil instruments. The pointers of moving iron instruments always move in one direction, irrespective of the direction of flow of the current, and cannot therefore be made with centre zero. They will not have such uniform scales as moving coil instruments, the scale being contracted at one end, and in some instruments at both ends.

In reading instruments, the eye, the pointer, and the mark on the scale should be in a line at right angles to the plane in which the pointer moves, in order to avoid errors due to parallax. In edgewise instruments this line of vision must pass through the centre of the arc swept out by the pointer; one eye must be closed. In multi-range instruments each range should be a simple multiple of the other ranges; and the figures upon the scale of the instrument should preferably indicate the number of scale divisions. An instrument having a range of 150 or multiples thereof should be calibrated to have 150 divisions, figured 0–150. There is less liability of error in reading direct in scale divisions, and afterwards multiplying by a correct constant, than in mentally calculating the value of a fraction of a scale division, especially if there is a rapidly fluctuating load.

The accuracy of an instrument is best expressed as a percentage of the maximum load the instrument is intended to measure. For instance, an instrument having a range of 500 volts with an accuracy of $\frac{1}{10}$ per cent. should be correct at every point upon the scale to within half a volt. If the instrument has 100 scale divisions, this corresponds to $\frac{1}{10}$ of a scale division, which could be accurately read upon a portable instrument with a knife-edge pointer working over a mirror to eliminate errors due to parallax. If, however, the accuracy is taken as a percentage of the actual reading, when reading this instrument at 100 volts the error would not have to exceed 0·1 volt, which would correspond to $\frac{1}{50}$ of a scale division, and would not be readable by the naked eye. If the instrument is supplied with an additional range to read 0–100, an accuracy of $\frac{1}{10}$ per cent. upon this range would represent 0·1 volt, and would correspond to $\frac{1}{10}$ of a scale division.

The Synchroscope

Apart from special methods only applicable to polyphase machines, two devices are in general use for synchronising alternators, (*a*) the synchronising lamp or voltmeter, and (*b*) the rotating field synchroscope. The former is more simple, but suffers from the defect of being insensitive as regards the exact moment of switching in, which should be when the machines are in exact coincidence of phase. Machines may be as much as 16° out of phase before the synchronising voltmeter shows 1 per cent. of variation from its maximum reading, so that a little error of judgment may result in a considerable shock on switching in. A further objection is that although the rapidity of fluctuation of the lamp or voltmeter reading indicates the difference between the frequencies of the machines, there is nothing to show which of the two machines is the faster, and considerable time may thus be wasted. In order to make the lamp method of real service, therefore, it is necessary to combine it with some device for indicating exact coincidence of phase and also whether the incoming machine is fast or slow; and this is what has been done in the Weston Synchroscope.

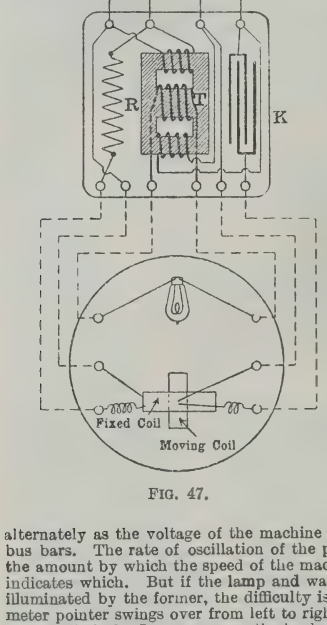


FIG. 47.

If an ordinary dynamometer wattmeter has both its fixed and moving coils wound with fine wire, and two alternating voltages are applied to the terminals of these coils, it will show very accurately when these two voltages are in exact *quadrature*, as the wattmeter will indicate zero in this case, and deflect to right or left if the phase difference is greater or less than 90. By connecting a condenser in series with one of the coils, and introducing a small amount of inductance into the other, it is easy to arrange that the wattmeter indicates zero when the two voltages are in phase instead of in quadrature, and this can be checked by connecting the two circuits to the same supply. Under these circumstances, if one circuit is connected to the bus bars and the other to the incoming machine it is only necessary to see that the synchronising lamp is bright and the wattmeter pointer at zero to be able to throw in without any shock.

But this only indicates when the machines are in exact synchronism, and not whether the incoming machine is too slow or too fast. If the speed of the incoming machine is low or high, the lamp will flicker in the ordinary manner and the wattmeter will also deflect to right and left alternately as the voltage of the machine alternately lags and leads with respect to that of the bus bars. The rate of oscillation of the pointer, like the rate of fluctuation of the light, shows the amount by which the speed of the machine is high or low, although neither of them by itself indicates which. But if the lamp and wattmeter are combined, so that the dial of the latter is illuminated by the former, the difficulty is at once solved. For if the machine is fast the wattmeter pointer swings over from left to right while the lamp is bright, while when it swings back the lamp is dark. In consequence the back swing is not seen, and instead of the pointer appearing to oscillate it appears to rise with the dawn of light at the left-hand side of the scale, to swing across in full light and to disappear on the right of the dial with the extinction of the light, almost exactly like the rising and setting of the sun, and as if it were steadily revolving in one direction. When the machine is slow, the lamp is bright as the pointer swings from right to left, and the direction of rotation is therefore apparently reversed. As far as its practical application is concerned, the instrument resembles in every way the rotating field synchroscopes (except that the fluctuation of the light is visible), and the arrangement is extremely simple, and free from any loose contacts.

The actual indicating instrument is almost identical in construction with the switch-board wattmeter, the only differences being that the dial carries a bold central mark with two arrows indicating 'Fast' and 'Slow,' and that the synchronising lamp is inserted behind it. This lamp can be easily withdrawn and replaced. An auxiliary box contains an ordinary synchronising transformer, T, for the lamp, and a non-inductive resistance, R, and a condenser, K, for connecting in series with the two wattmeter coils respectively. A small inductive coil for adjusting the exact quadrature is included in the instrument case. Fig. 47 is a diagram of the connections of the instrument and box, which will be readily understood from the principle of the instrument.

METERS

In all commercial motor meters there is necessarily a certain amount of friction, which as time goes on tends to increase more or less according to the design, and to cause the meter to slow down. This defect manifests itself particularly at low loads, and is the more serious as the majority of meters are worked at points very low down on their load curves. For long periods the working point is frequently at or below one-tenth full load capacity, and in many instances, particularly where metal filament lamps are used, the working load is liable to be between one-twentieth and one-fiftieth the full load capacity of the meter for many consecutive hours daily.

Pointer type clocks should have light machine cut gears, and should run with an exceedingly small amount of friction, which, being constant, is easily compensated for. This fact constitutes an important advantage in favour of clocks of this type.

Roller cyclometer clocks have more friction than the pointer type, and moreover possess the defect that as the friction is a varying quantity it cannot be compensated for. A meter may have to move round one counter only, or again it may have to move up to five counters at one time.

Jumping figure cyclometer clocks are a modification of the roller type, having the friction load more evenly distributed through the use of a falling weight device. This weight is wound up slowly by the meter. It is released at the proper time, and in falling carries round the additional counters which it is necessary to move.

This partial equalisation of friction makes cyclometer clocks which have springing figures better than the roller type, although not so good as the pointer type clocks, where the friction is a constant quantity. An important feature of springing figures in comparison with roller type counters is that with the former no halving of figures takes place.

Manufacturers are generally willing to guarantee the accuracy of their meters, so that a.c. meters can be obtained which are capable of easy adjustment within plus or minus 2 per cent. from 50 per cent. overload down to one-fiftieth of full load.

The voltage drop in the main coils should be as low as possible. The drop varies with the size of the meter, being greatest in the smaller sizes and diminishing as the ampere capacity increases, due to the fact that in meters of large capacity fewer turns are required to create a similar torque. The loss in the shunt circuit should also be small, but should not be reduced at the expense of the torque, as in meters where the torque is reduced in consequence of the current in the shunt circuit being very low, the retarding effect of friction becomes great, affecting the accuracy of the meter under low load conditions. Moreover, with low torque more compensation will be required to balance the friction, and under such conditions trouble from 'creeping' through vibration, due to street traffic or other external causes, will be much more liable to occur.

A record should be kept of the life of every meter on the system of any supply undertaking. As meters are received from the makers they should be checked against a standard which should be certified from time to time by some recognised authority such as the National Physical Laboratory. This check should be made at various loads, say 125 per cent., $\frac{3}{4}$, $\frac{1}{2}$, $\frac{1}{4}$, $\frac{1}{10}$, $\frac{1}{20}$ and $\frac{1}{50}$ full load, at unity power factor, and at half load 50 per cent. power factor, and the results obtained should be plotted on a card; a note of any subsequent attention required by the meter should be added to its record, and at the end of recognised intervals—say every two or three years—each meter should be brought in for overhaul and tested carefully before breaking any seals. The result obtained should be added to the record, thus affording a means of continuously studying the behaviour of every meter.

The following tests, arranged in order of their relative importance, are easily made:—

(1) Accuracy at varying currents from 2 per cent. to 150 per cent. load with voltage frequency and power factor constant. (2) Accuracy at varying voltage with constant current and frequency. (3) Accuracy at varying power factor with voltage, frequency, and current constant. (4) Accuracy at varying frequency with constant load and voltage.

British Standard Specification for Electricity Meters.

(*No.* 37—1930.) (*Abstract.*)

Two grades of accuracy—'sub-standard' and 'commercial'—are classified. The following provisions apply to both unless otherwise stated.

The construction of the meter shall be mechanically sound and suitable for its purpose and shall be such as to give reasonable assurance of permanence in all mechanical, electrical, and magnetic adjustments, and shall not be seriously affected by external magnetic fields.

The parts of the meter within the case shall be carried on a substantial stiff framework forming, if desired, part of the case. The framework shall not be liable to distortion by changes of temperature, presence of moisture, or other cause. They shall be completely enclosed in a substantial dust-tight metal case, the front and sides of which may be glazed. The case shall be of sufficient strength to afford adequate protection to the working parts against mechanical injury with reasonable usage, and capable of being sealed in such a manner that access to the working parts and the electric or magnetic adjustments of the meter can only be obtained by breaking the

seals. Care shall be taken to ensure that the mechanism as well as the interior of the case is free from dust or other extraneous matter immediately prior to sealing. Provision to be made for earthing any accessible metal part when case is closed.

One or more substantially glazed dust-tight windows shall be provided in the front of the case in such a manner as to give a clear view of the register, and also of those working parts the observation of which is necessary for rapid testing of the meter. It shall be impossible to remove the glass without breaking the seals of the case or breaking the glass itself. The case, etc., to be protected by an efficient coating of paint or suitable material.

The terminals of the meter itself shall be insulated with non-hygroscopic material and, unless of the switchboard type, shall be enclosed by a separate cover suitable for sealing, or contained in an external compartment of the case.

All meters of sizes above 100 amperes shall be provided with soldering sockets, except where provision is made for coupling with strip or bar connections. The left-hand terminal seen from the front shall be the positive in D.C. meters, and be connected to the incoming main in A.C. meters (or to transformer secondary of same polarity).

The pressure circuit shall be connected to the supply side of the main circuit, and a disconnecting device shall be provided in the terminal compartment, to enable the pressure circuit to be entirely disconnected from the main circuit, in order that it may be supplied with current from a separate source for purposes of testing. The disconnecting device is not necessary where separate terminals are provided for the pressure circuit.

The meter, unless of the switchboard type having external connections, shall be provided with separate facilities enabling the purchaser to seal independently the terminal compartment and the meter case. Facilities may also be provided for independent sealing by the manufacturer.

Every meter, the accuracy of which is affected by small changes in level, shall be provided with means of ascertaining, without breaking the seals, that it is correctly levelled.

A diagram of connections to be attached to the meter.

The circuits of the meter shall be capable of carrying full load continuously without injury to any part of the meter, without any permanent change in its accuracy, and without the temperature of any accessible part of the case exceeding 10° C. above the temperature of the surrounding air, except when a shunt is employed in a separate compartment forming an integral part of the meter, when the temperature of any accessible part of the case of such compartment shall not exceed 30° C. above the temperature of the surrounding air.

Every external resistance shall be completely enclosed in a strong metal case.

Every external resistance shall be so constructed that when the marked pressure is applied to the meter and resistance combined and is maintained continuously, no injury shall be done to any part of the resistance, and the temperature of any accessible part of the case shall not exceed that of the surrounding atmosphere by more than 30° C. The full-load speed shall not exceed 100 r.p.m. for a D.C. and 50 r.p.m. for an A.C. meter.

The insulation resistance between all the electric circuits of the meter coupled together, and the containing case, or other metal not intended to be insulated when the meter is in use, shall be not less than 5 megohms.

The insulation resistance between the main circuit and the pressure circuit (if any) shall be not less than 2 megohms.

The insulating material between all the electric circuits and the containing case, or other metal not intended to be insulated when the meter is in use, shall withstand, for one minute, an alternating pressure equal to twice the pressure of the circuit for which the meter is intended, plus 1,000 volts for voltages above 660, and 2,000 volts (alternating) on meters for circuits up to 660 volts, and on all meters used with transformers.

The insulation resistance shall be measured with a pressure of not less than 200 volts nor more than 500 volts (direct), applied for a sufficient time for the reading of the insulation indicator to become practically steady.

The rate of loss of energy in the pressure circuit at marked pressure shall not exceed the values given in the following table :—

Meter (Commercial Grade).	Marked Pressures up to 250 volts.	Marked Pressures above 250 volts.
Direct current . . .	5 watts	2 watts per 100 volts
Alternating current . .	2 watts below 50 cycles 1·8 watt, 50 cycles or above	1 watt per 100 volts

The rate of loss of energy in the meter current circuit for all meters having a marked current not exceeding 50 amperes shall not exceed the following values; with the further restriction that the pressure drop shall not exceed 1 volt.

Marked Current.	D.C. Meters.	A.C. Meters.
	Watts.	Watts.
Up to and including 10 amperes . .	5	2
Above 10 amperes and up to 50 amperes .	5	3

If the marked current exceeds 100 amperes, the voltage-drop in the shunt and meter combined when carrying the marked current shall not exceed 0·25 volt.

When the meter has more than one element the above requirements shall apply to each element independently. This clause shall not apply to electrolytic and commutator ampere-hour D.C. meters.

The forward direction of rotation of the rotor, if any, shall be clearly indicated and shall be from left to right, viewed from the front.

The following shall be the standard sizes of meters:—2·5, 5, 10, 20, 40, 50, 100, 150, 200, 300, 400, 500, 600, 800, 1,000, 1,500, 2,000, 3,000, 4,000, and 5,000 amperes.

When a current equal to 0·5 per cent. of the marked current of the meter (or when this is less than one-twentieth of an ampere, then one-twentieth of an ampere) traverses its main circuit, the rotor shall start and continue running steadily, the pressure circuit, if any, being energised at the marked pressure to give a condition corresponding to unity power factor in the main circuit.

The error at the standard or marked temperature, voltage, and frequency shall not exceed the following values. These accuracies shall be obtained with shunt, transformers, or auxiliary apparatus connected.

A.C. METERS, COMMERCIAL GRADE.

Conditions of Test.		Limits of Error.	
Current expressed as a Percentage of the Marked Current.	Power Factor.	Meters without External Shunts or Transformers.	Meters with External Shunts or Transformers.
	Per Cent.	Plus or Minus. Per Cent.	Plus or Minus. Per Cent.
125 to 20 . . .	1·0 to 0·5	2·0	2·5
20 to 10 . . .	1·0	2·0	2·5
At 10	0·5	2·5	2·5
At 5	1·0	2·5	2·5

D.C. METERS, COMMERCIAL GRADE.

Size of Meter.	Conditions of Test.	Limits of Error.
		Per Cent.
Below 10 amperes .	125% to 10% of marked current	± 2·5
10 amperes and over .	125% to 10% of marked current	± 2·0
	At 5% of marked current	± 2·5

For sub-standard meters of all types the error shall not exceed ± 1·5 per cent. under any of the following conditions :—

Type.	Percentage of Marked Current.	Power Factor.
		Per Cent.
A.C.	100 to 20	1·0 to 0·5
A.C.	20 to 10	1·0
D.C.	100 to 10	—

In addition, the total change of error over the range of the above conditions shall not exceed 1·5 per cent.

If the meter has a pressure circuit, a variation of 5 per cent. in the case of D.C. meters and 10 per cent. in the case of A.C. meters, above or below the marked pressure at any load from full to one-tenth shall not cause a change in the rate of registration in respect of such variation of more than 1 per cent.

Except D.C. mercury type for marked currents of 10 amperes and less, the temperature coefficient shall not exceed 0·2 per cent. per degree C. ; for A.C. induction meters 0·1 per cent. per degree C. at unity power-factor and 0·15 per cent. at 0·5 lagging at the marked current and 50 frequency ; for mercury meters for 100 amperes and above, 0·1 per cent.

The change in rate of registration from the time the main marked current is switched on to the time at which the rate of registration becomes constant shall not exceed the following values ; the voltage having been continuously applied for not less than six hours and no current in excess of one quarter of the marked current passed through the main circuit during this period.

	Commercial Grade.	Sub-standard Grade.
	Per Cent.	Per Cent.
Induction watt-hr. meters :		
At unity power factor .	0·75	0·5
At 0·5 power factor (lag)	1·5	0·75
All other types . . .	2·0	1·0

A variation of ± 5 per cent. from the marked frequency shall not cause a greater variation than 0·5 and 1·5 per cent. at unity and 0·5 (lag) power factors respectively in commercial grade, and 0·25 and 0·5 per cent. in sub-standard meters.

The meter shall not be injured nor accuracy impaired by overloads for 30 minutes of :—100 per cent. for whole-current meters rated up to 50 amperes ; 50 per cent. for whole-current meters from 50 to 1,000 amperes ; also all meters with current transformers and shunts ; 25 per cent. for whole-current meters rated above 1,000 amperes ; with compound-filled transformers, 25 per cent. The meter shall also withstand the passage of 30 times the marked current for not longer than half a second, the rotor, if any, being free to rotate, and the pressure circuit energised at marked pressure. No test current need exceed 15,000 amperes.

The meter shall maintain all these specified accuracies for 1 year after calibration, and other requirements for 3 years.

The accuracy of a 3-wire D.C. meter shall not change more than 1 per cent. when no current flows in one circuit : a polyphase meter shall comply with specification on balanced and unbalanced loads.

The meter shall not be injured, and its accuracy shall not be permanently impaired, by the application of a pressure 25 per cent. above the marked pressure for a period of half an hour.

Every meter having a pressure circuit shall not register with its main circuit open when the pressure circuit alone is acting, either when marked pressure is applied or when this pressure is increased to 10 per cent. above the marked pressure.

Insulation Measuring Instruments.

(Evershed & Vignoles, Ltd.)

These instruments are very largely used for the measurement of the resistance of insulators and conductors; they embrace ranges from 1 microhm up to 10,000 megohms. All are direct reading in the strictest sense, and the ruling feature of each is to provide a means of measurement which shall be so simple that it can be used by any unskilled person, so rapid to use that a test shall occupy but a few seconds, and so portable that the instrument can be taken anywhere to do the work required.

The 'Megger' Testing Set consists of a moving-coil type ohmmeter combined in one box with a high-pressure generator driven by a folding crank handle. The general principles of the construction and the electrical connections of the set are shown in the following diagram.

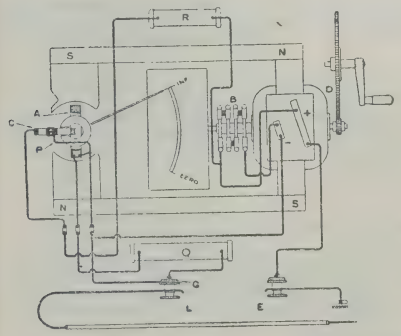


Fig. 48.—Megger Testing Set. Diagram of Connections.

A = current coil; P = pressure coil; C = compensating coil; N, S = magnets; Q, R = resistances; B = brush gear; D = driving gear; + − = generator terminals; G = guard plate; L = external 'line' terminal; E = external 'earth' terminal.

In external appearance the 'Megger' testing set is similar to the 'Bridge-Megger' testing instrument shown by fig. 49, but without the terminals at the end of the case. To carry out an insulation test on a system, connect the 'line' terminal to the circuit and the 'earth' terminal to a good earth. If testing between two wires, connect one wire to each terminal.

When the handle of the generator is turned at about 100 r.p.m. the rated voltage will be developed, and the value of the insulation resistance under test will be indicated directly by the pointer over the scale. The instruments are fitted with either variable or constant-pressure-generators; the constant-pressure type should always be used when testing the insulation of circuits having considerable electrostatic capacity, such as large house-wiring installations, supply and feeder cables, etc. The constant-pressure device consists of a centrifugal friction-clutch, which secures that when the handle is turned above the slipping speed, the speed of the generator, and consequently the terminal E.M.F., remains constant. All 'Megger' and 'Bridge-Megger' testing sets are provided with a system of guarded insulation to ensure that no internal or surface leakage can affect the readings. Any error in tests of high insulation due to leakage across the surface of the material under test can be eliminated by the use of a guard wire connected to the 'guard' terminal, which is provided on all high-range instruments.

The Bridge-Megger Testing Set, fig. 49, is a combined ohmmeter and generator of the usual 'Megger' type for insulation testing, but also arranged so that the ohmmeter can be converted into a Bridge Galvanometer by means of the change-over switch fitted. The current for the Bridge tests is provided by the generator (constant-pressure type), and a direct-reading

resistance box of special and convenient design is supplied as a standard of comparison. The whole combination forms the most rapid known means of measuring that range of resistances for which the Bridge method is adapted, and the value of the resistance is given plainly in a row of figures, which prevents errors. The series of ranges covers Bridge measurements from 0·01 ohm

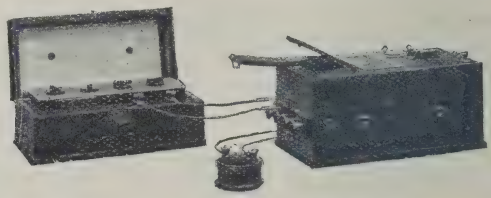


FIG. 49.—Bridge-Megger Testing Set in Use.

upwards, and insulation measurements up to 2,000 megohms. In addition to the measurement of insulation and conductor resistances, the set may be used for the localisation of faults on telephone and telegraph cables, or on electric light cables if their resistance falls within the limits of Bridge methods.

THE 'DUCTER' POTENTIAL OHMMETER TESTING SET provides for the measurement of resistances from 1 microhm up to 100 ohms by the direct deflection method. The current for this set is supplied by a separate battery, and a pair of duplex handspikes are used for making contact on the resistance under test. With these instruments the resistances of switch and fuse contacts, armature windings, rheostat coils, rail-bonds, sheathing and earth circuits in collieries, joints and the like, are easily and rapidly determined.

MEG INSULATION TESTER.

The most recent development in Evershed & Vignoles' resistance-measuring instruments, is the 'Meg' Insulation Tester (fig. 50), which was introduced in 1922. While the Megger and Bridge-Megger continue to be the standard instruments where the measuring instrument must be capable of dealing with a very long range of resistance values, it was felt that there was room

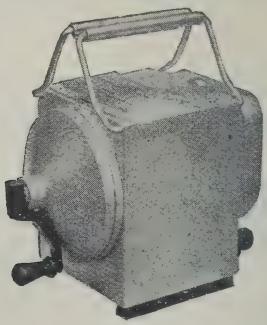


FIG. 50.

or a smaller, lighter, and less costly instrument of the Megger type. The 'Meg,' with its ranges up to 20, 50 or 100 megohms, is quite suitable for most classes of insulation testing, and is used very extensively for switchboard wiring, testing generator and motor coils, etc. Two models of the 'Meg' insulation tester are made for variable pressure and constant pressure, the latter having a speed-control device which ensures constant voltage, an important feature where the capacity of the insulation to be tested exceeds 0·5 microfarad, as, for instance, house wiring, insulation, etc.

ELECTRIC LAMPS.*

Metal Filament Lamps.

The gas-filled lamp, like the vacuum lamp, which it is rapidly replacing, has a filament of tungsten, the melting point of which is over 3,000° C.; this fact enables the filament to be operated at a higher temperature than any other metal, with the result that a high proportion of its total radiation is in the visible spectrum. While the melting point of tungsten is very high it oxidises readily at quite a low temperature (about 300° C.). When new it can be readily bent into any desired shape. It is this property which facilitates the manufacture of the filament, both in the ordinary up and down winding and also in the spiralised form.

In a tungsten lamp an increase in voltage produces an increase in resistance (in a carbon filament the resistance under such variation remains practically constant). Fig. 51 shows the relationship between watts and voltage of a modern Tungsten electric lamp, while the lumen characteristic is shown in fig. 52.

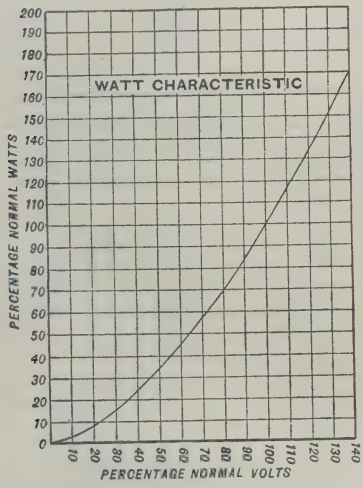


FIG. 51.

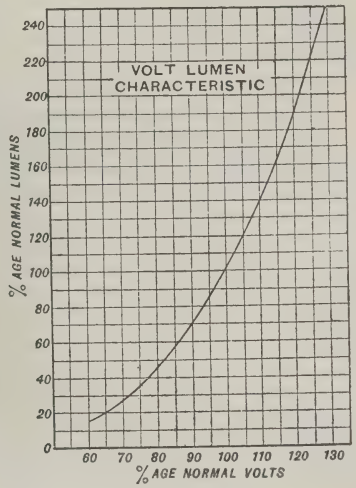


FIG. 52.

An increase in voltage produces an increase in temperature of the filament, so that a very great volume of light is obtained. An increase of 10 per cent. volts brings about an increase of 41 per cent. in light output, while a reduction of 10 per cent. reduces the light output by 32 per cent. For this reason under-running is to be greatly deprecated in view of the great loss of luminous energy. On alternating current of low frequency a flickering is obtained dependent upon the heat capacity of the filament itself, but the remedy in such cases is to eliminate the use of small lamps by installing a smaller number of larger units. This effect of flickering is less conspicuous in the tungsten lamp than in the carbon lamp because of the temperature coefficient. As far as possible it is recommended that where difficulties are experienced in this direction that lamps of not less than 100 watts be employed.

In the case of the gas-filled lamp the bulb temperature is considerable, while the lamp which is not gas-filled remains comparatively cool.

All lamps manufactured in this country by the members of the Electric Lamp Manufacturers Association are made in accordance with specifications which ensure that a uniform product is obtained of a standard quality.

* See also Sect. XXXVIII., Part III, Lighting.

Standard Tungsten Vacuum Electric Lamps.*

The following table † (from the B.E.S.A. Specification No. 161—1930 (*Abstract*)) gives figures relating to standard tungsten vacuum lamps :—

Rating.		Standard Caps.	Maximum Overall Dimensions.		Torsion Test for Bayonet Cap.
Volts.	Watts.		Length.	Bulb Diameter.	
1		3	4	5	6
100, 105, 110, 115, 120, 125, 130	10	B.C.	mm. 122	mm. 61	
100, 105, 110, 115, 120, 125, 130; and 200, 210, 220, 230, 240, 250	20	B.C.	140	63	29 kg. cm. (25 pound-inches.)
	30	,,	140	63	
	40	,,	140	63	
	60	,,	150	70	

Standard Tungsten Gas-Filled Electric Lamps.*

The following table † (B.E.S.A. Specification No. 161—1930 (*Abstract*)) gives figures relating to standard tungsten gas-filled lamps :—

Rating.		Standard Caps.	Overall Dimensions.				Neck Diam.	Light Centre Length with Standard Cap.	Tolerance on Light Centre Length.	Torsion Test.
			Length.			Diameter of Bulb. Tolerance ± 1 mm.				
Volts.	Watts		Overall.	Limits.			Max.			
				Max.	Min.					
1	2	3	4	5	6	7	8	10	11	12
100, 105, 110, 115, 120, 125, 130	30	B.C.	mm. 110	mm. 115	mm. 106·5	mm. 60	mm. 44	mm. 80	mm. ±3	
100, 105, 110, 115, 120, 125, 130; and 200, 210, 220, 230, 240, 250	40	,,	110	115	106·5	60	44	80	±3	29 kg. cm. (25 pound-inches.)
	60	,,	117	122	114·0	65	45	85	±3	
	75	,,	125	130	121·5	70	51	90	±3	
	100	,,	140	145	136·5	80	53	100	±3	
	150	B.C. or E.S.		170		(Max.) 92	45	120	±4	
	200	E.S.		215		103	48	150	±5	
	300	G.E.S.		250		113	55	180	±6	
	500	,,		280		133	60	205	±6	52 kg. cm. (45 pound-inches.)
	1000	,,		340		173	70	250	±8	
	1500	,,		340		173	70	250	±8	

* Report No. 33—1930 of the British Engineering Standards Association deals with carbon filament glow lamps.

Report No. 377—1930 deals with Miners' Handlamp Bulbs.

The following tables * (B.E.S.A. Specification No. 161—1930 (*Abstract*)) give figures relating to Traction, Train, and Bus Lamps:—

STANDARD VACUUM TRACTION LAMPS (SERIES). STANDARD VACUUM TRAIN LIGHTING LAMPS.†

Rating.		Standard Caps.	Maximum Overall Dimensions.		Rating.		Standard Caps.	Maximum Overall Dimensions.	
Volts.	Watts.		Length.	Diameter of Bulb.	Volts.	Watts.		Length.	Diameter of Bulb.
1	2	3	4	5	1	2	3	4	5
			mm.	mm.				mm.	mm.
100					16	15	B.C.	76	52
110	30	B.C. or E.S.	110	61		20	B.C.	76	52
120					24	15	B.C.	76	52
130	40	B.C. or E.S.	110	61		20	B.C.	76	52

STANDARD GAS-FILLED BUS LAMPS.

Rating.		Filament Shape.	Standard Cap.	Type of Bulb.	Overall Dimensions.			
					Length.		Diameter of Bulb.	
Volts.	Watts.				Max.	Min.	Max.	Min.
1	2	3	4	5	6	7	8	9
					mm.	mm.	mm.	mm.
14	14	V	B.C.	Clear Pipless.	72	63	52	48

Note—The 'Torsion Test' for Bayonet or Edison Caps for all three tables is 29 kg. cm. (25 pound inches.)

MAZDA LAMPS.

(*The British Thomson-Houston Co., Ltd.*)

Mazda lamps, the filaments of which are made from pure-drawn tungsten wire, give a brilliant white light. They can be operated satisfactorily in any position, but in the larger sizes it is better to use the lamps set vertically. The lamps operate equally well on either continuous or alternating current circuits.

Mazda lamps, including both the vacuum and gas-filled types, have made it possible to design lighting units of suitable sizes for all lighting conditions, and of such high efficiency as to compete effectually with all other types of artificial illuminants. Users of electric light have a choice of sizes ranging from 5 watts to 1500 watts. Every size is adapted, as regards filament position and shape of bulb, for use in conjunction with specially designed glass or metal reflectors.

The gas-filled lamp is claimed to give, for the same current consumption, considerably more light than the previous best incandescent lamp. The recent development of 30 and 40 watt gas-filled lamps in standard supply voltages has brought the advantages of the gas-filled principle within reach of the ordinary householder.

The strength and durability of the drawn wire filament ensure satisfactory results from Mazda lamps under all conditions of service. They have successfully withstood the severe conditions of use aboard battleships during target practice.

* By permission of the British Engineering Standards Association.

† The Committee responsible for the preparation of this Schedule recommends the use of the 24-volt Lamp for Train Lighting.

The flexibility of the Mazda drawn tungsten wire has made possible the production of special forms of filaments which could not be produced with a brittle squirted tungsten wire used in previous lamp bulbs. Helical coil filaments, as used in Mazda headlight bulbs, are better in regard to focus than the types of filaments hitherto employed.

The intrinsic brilliancy of the lamp is stated to be eight times that of ordinary metal filament lamps.

The gas-filled lamp is now made in small sizes for headlight use. The master patents governing the manufacture of Mazda vacuum and gas-filled lamps are the property of the makers of Mazda lamps.

Siemens Lamps.

(Siemens & English Electric Lamp Co., Ltd.)

These lamps have drawn tungsten wire filaments, and the Standard Vacuum type operate at an efficiency of, approximately, 1·2 watts per c.p. The 'gas-filled' lamp represents the latest development in metal filament lamp manufacture, and the larger sizes attain an efficiency of, approximately, 0·5 watts per c.p. Both types of lamps are made in an extensive range of voltages and candle-powers.

Siemens Standard Type Lamps.—These are made in voltages from 20 to 260, and in wattages from 5 to 60, and cover the lamps in general demand for ordinary electric lighting schemes of average intensity.

Siemens Traction Type Lamps.—This type of lamp has been introduced to meet the requirements experienced in situations where lamps are subjected to severe shock and vibration, and is now largely used on the leading tramway systems and tube railways. A totally new arrangement of the filament is employed, which ensures maximum strength and adequate support. The complete length of filament is spiralised and carried in a series of loops on a special spider in the usual Siemens manner. The distance between the top and bottom supports is considerably reduced, which ensures maximum strength, while neighbouring lengths of filament are more widely separated, thus obviating short-circuiting of adjacent lengths of filament, which sometimes occurs in the ordinary type of lamp subjected to severe vibration. The filament has no sharp bends, and will not 'sag,' so that the lamp may be used in any position. The vertical candle-power of the lamp is increased, and the light emitted is uniform in all directions other than towards the cap end. It is claimed for this design that these additional advantages have been obtained without loss of efficiency in operation.

Siemens Gas-filled Lamps.—The gas-filled lamp was originally introduced in high candle-power types only, but the range has been increased, with the result that these lamps are now obtainable for standard lighting voltages from, approximately, 30 to 1,500 watts. Owing to their high efficiency, these lamps became known to the public as 'Half Watt' type lamps. It is, however, an essential condition to their production that they should be gasfilled in order to obtain a satisfactory life at the efficiency mentioned. The leading manufacturers are therefore stamping the word 'gasfilled' on the bulbs of these lamps, so that the consumer may be protected should he be offered lamps which, though resembling genuine high-efficiency lamps, are not gas-filled.

The range of Siemens gas-filled lamps now includes automobile head-light lamps, from 4 to 14 volts, from 8 to 48 watts; Siemens gas-filled lamps for railway lighting, 25 volts, can be obtained from 15 to 100 watts. The 50- to 60-volt gas-filled lamps are supplied from 15 to 500 watts; 100- to 130-volt from 30 to 1,500 watts; and the high voltage gas-filled lamps, 200- to 260-volts, from 40 to 1,500 watts. These high candle-power lamps are being employed in large numbers for general lighting in stores, workshops, factories, offices, shops, etc., and for street lighting; they produce illumination without flicker, and require no attention after installation, save occasional cleaning.

Siemens Automobile and Battery Type Lamps.—In addition to the range of gas-filled automobile head-light lamps mentioned above, Siemens Battery Lamps are made for automobile lighting, sign lighting, omnibus and yacht lighting; while there is also a variety of types suitable for use in miners' hand and rescue lamps. The voltages of these vary from 2 to 14, and the wattages from, approximately, 1 to 36 watts.

Osram (G.E.C.) Lamps.

(The General Electric Co., Ltd.)

The Osram (G.E.C.) lamps have filaments drawn from tungsten; they have also pipless bulbs, which greatly improves their appearance and renders them less liable to be broken, both in transit and in use.

In addition to the low c.p. sizes of Osram lamps, a new tungsten lamp has been introduced in which a gas-filled bulb is used in place of vacuum; the efficiency of these lamps approximates to ½ watt per candle, as compared with vacuum lamps which consume about 1 watt per candle. The Osram gas-filled lamp is known as the 'Atmos' type lamp, and is made from 30 to 3,000 candle-power. The high candle-power lamps are largely used in the place of arc lamps.

The Osram standard or 1-watt lamp is made in sizes from 4 to 400 c.p., and, in addition standard types have been supplemented by special types for different purposes, such as with concentrated filaments for projection and focusing work; with 'Axial' type filaments for downward illumination, such as over desks, etc.; and for use from low voltage batteries, such as miners' and general portable hand-lamps and automobiles (including motor-buses). The Osram (G.E.C.) lamp 'Atmos' type is also made for automobile headlights.

A modification of the gas-filled lamp is the 'White Osram,' which is a standard clear Osram lamp, the bulb of which is coated with a film of unglazed porcelain. This coating has the effect of efficiently diffusing the light, and eliminating all glare from the intensely bright filament. White Osram lamps should be used in all positions where the lamp itself is visible.

For industrial lighting, bowl sprayed White Osrams—that is, lamps having only the lower part of the bulb coated—when used with top reflectors, such as G.E.C. enamelled steel reflectors, are specially recommended.

A further development of the White Osram is the 'Colour Sprayed Osram' lamp, which is produced in six standard colours—red, orange, yellow, green, blue, and flame tint. Colour sprayed lamps are the most convenient means of obtaining colour light for both indoor and outdoor decorations, such as dances, concerts, restaurants, seaside promenades, etc.

In addition to the Osram lamps for ordinary lighting, they are made for a wide variety of special purposes, such as the 'Daylight' lamp for colour matching, projection lamps, ship type lamps, candle type lamps, striplite lamps, automobile type lamps, miners' lamps, etc.

The lamps are an entirely British product, every component part, including the brass caps, being produced within the G.E.C. organisation.

The Osglim Discharge Lamp.

(Osram G.E.C. Lamp Works.)

Whereas discharge lamps usually require at least 2,000 volts, by filling the lamp with neon gas the discharge will start on about 200 volts. The current flows through the neon from one electrode to the other, the gas glowing round the cathode with a soft rosy light. By using a resistance inside the cap, the consumption of the lamp is reduced to 5 watts. The faint light from this lamp (known as the 'Osglim') makes it ideal as a night light, or wherever subdued illumination and low consumption are important. By making the cathode in the form of letters or figures, effective indicators or advertising signs are obtained. The lamp can also be made suitable for use in the photographic dark room.

Efficiency of Filament Lamps.

Type of Lamp.	Average Efficiency in Lumens per Watt.	Average Efficiency in Watts per Spherical Candle.
Carbon	3·3	4·1
Tantalum	6·0	2·1
Tungsten (Vacuum)	8·0	1·5
Tungsten (Gas-filled)	12·5	1·0

Intrinsic Brilliancy of Lamp Filaments.

The candle power per square inch of the filament of carbon 3·1 w.p.c., tantalum 2 w.p.c., and tungsten 1·25 w.p.c. lamps is 480, 625, 750, and 1,000 respectively, while the gas-filled lamp is 5,000.

Quality of Light.

The light of tungsten lamps more closely approaches daylight than that of other illuminants. It naturally follows that tungsten lamps cause colours to retain more nearly their true values than any other artificial light source.

Attachment of Flexible Cords to Sockets.

The majority of short circuits occurring in pendant sockets and the attachment-plug caps of portable electrical appliances, according to the Underwriters' Laboratories (America), are due to loose strands of the flexible cord, which become separated from the terminals when the binding screws or terminals are loosened. This loosening is frequently the result of the omission of a knot or its equivalent in the cord inside the cap of the socket or attachment plug, the strains due to pulling separable connections apart or dragging portables around by their cords being transmitted directly to the small binding-screw connections.

The so-called 'Underwriters' knot,' fig. 53, provides an easily-tied strained relief forming a rather flat knot, which will fit snugly in the recess provided for the purpose inside the cap of pendant sockets and attachment plugs.

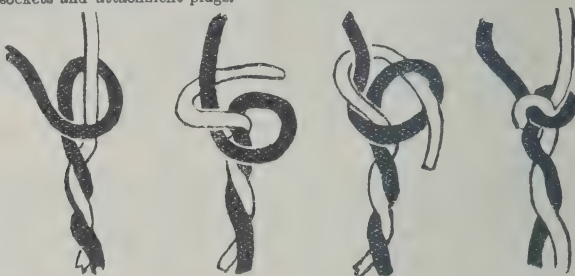


FIG. 53.

Arc Lamps.

In a D.C. arc lamp the top or positive carbon burns about 1½ inches per hour, and twice as fast as the bottom or negative carbon of the same diameter. A 1,000-candle-power lamp requires carbons about $\frac{9}{16}$ inch in diameter and consumes about 1,000 watts.

SIZES OF CARBONS.

Current of Lamp, in amperes .	4	5	6	7	8	9	10	12	14	15	20
Diam of Plain Carbons, in mm. .	7	7	8	8	9	9	10	11	12	12	13
Diam. of Cored Carbons, in mm. .	10	12	14	14	16	16	17	19	21	21	23

THE ANGOLD MAGAZINE FLAME ARC LAMP.

(The General Electric Co. Ltd.)

The Angold Magazine Flame Arc Lamp is arranged to burn carbons of comparatively short length (maximum 15 ins.) without copper coating or metal core ; twelve pairs of these carbons can be filled in the magazine at one time, and thereby, according to the type of lamp, 120 to 150 hours be obtained.

The carbons are so arranged that as soon as one pair is consumed the next pair comes into operation. The magazines can be filled up if only part of the trim has been consumed, so that at any time the full burning hours obtainable with a full magazine can be assured. The positive carbons cannot be put into the negative magazine, and when the last carbons are consumed, the feeding gear is automatically put out of action. Partly burnt carbons need not be removed. The interval of time during which the light is extinguished does not exceed two seconds.

The direct current lamp is manufactured for 8 or 10 amperes and can be supplied with self-contained equivalent resistance and automatic cut-out, or it can be supplied without these accessories. In the latter case, however, the lamps can only be used singly, on suitable voltage, or two in series on 100-120 volts. For longer series (or even for two lamps on higher voltage) equivalent resistances and automatic cut-outs have to be provided. If it is inconvenient to have these accessories self-contained they can be supplied in the suspension type, or if it is necessary to lower the lamp from contact gear a box type is recommended.

Both 8-ampere and 10-ampere lamps are designed for 9 mm. positive and 8 mm. negative carbons, 15 ins. long. Each trim of carbons consists of 12 pairs.

An alternating-current lamp of this make will burn in parallel singly on any voltage between 100 and 250 volts—25 cycles to 100 cycles. Each lamp is fitted with a self-contained transformer choker so that it can be connected directly to the mains.

THE STUDIO ARC LAMP.

(The General Electric Co., Ltd).

The G.E.C. Studio Arc Lamp is of the semi-enclosed type, and the high voltage long arc produced yields the highest possible actinic values.

It is equally suitable for portraiture and blue printing, as, owing to the purity of the light and to the absence of shadows, exceptionally good results are obtained.

It is supplied for use on A.C. or D.C. circuits.

To obtain the high voltage necessary the lamp is run singly on 220-250 volts. For average work a 16-ampere with these voltages is generally used.

ELECTRIC CLOCKS AND TIME SERVICE.

The Pul-syn-etic System.

The 'Pul-syn-etic' system, designed by *Gent & Co., Ltd., Leicester*, is arranged to operate any number of impulse clocks, workmen's time recorders, turret clocks, etc., installed in any large building or institution and connected electrically in series. The hands of such dials are all caused to advance simultaneously at half-minute intervals by means of step-by-step impulse propulsion movements (fig. 54) (or where inaudible working is required, the Silent Triangle Movement fig. 55) which, taking the place of the ordinary spring, or weight, and train of wheels, are operated by electric currents sent out periodically by a MASTER PENDULUM

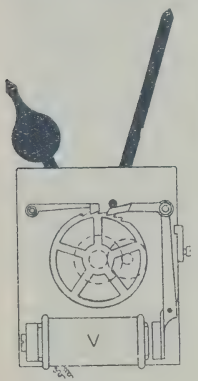


Fig. 54.

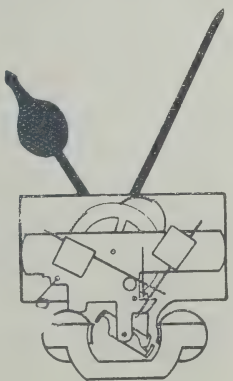


Fig. 55.

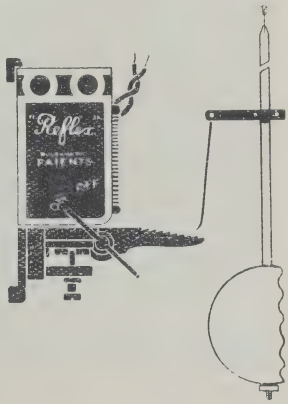


Fig. 56.

or TRANSMITTER. The latter (fig. 57) consists of a pendulum, which, by means of a pawl, causes a ratchet count wheel to revolve tooth by tooth—one tooth at every vibration. One of the teeth on the wheel is cut deeper than the others, so that a prolongation of the pawl is enabled to engage with and move aside a spring detent, which it misses during the passage of the other teeth. The detent, when moved aside, allows a weighted gravity lever to descend, and this through the medium of a friction roller carried thereon and an inclined plane fixed on the side of the pendulum, gives an impulse to the latter, imparting energy sufficient to enable the pendulum to swing during another revolution of the wheel—that is, for a period of half a minute. This gravity impulse is, of course, constant and invariable, so that other factors, such as temperature variation, etc., being suitably provided for, the time-keeping qualities of the pendulum are exceptionally good. The descent of the gravity lever is caused to close an electrical circuit, which fulfils a dual function: (*a*) energising an electro-magnet, which causes the restoration of the gravity lever to its potential position, and (*b*) sending an electrical impulse through the external circuit to operate the impulse movements. The duration of this contact is about 0·05 second; but, depending upon the total self-induction of the circuit, its duration is self-regulating, in that the contact is not broken until the self-induction has been overcome and all the dials operated.

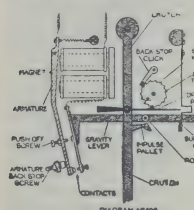


Fig. 57.

The system includes in series with the transmitter and the clocks a weak battery warning-bell which rings automatically every half minute should the battery run down and so endanger the safe working of the system. This warning is given before the actual breakdown point is reached, affording a safe margin of time in which to recharge or renew the battery. The 'Pul-syn-etic' system is noted for its low current consumption, and it is claimed that it takes the least current of any known Impulse system.

The system lends itself conveniently to the automatic operation of sound signals for the starting and ceasing of work, and also to the convenient operation of output and idle machine recorders. This class of recorder has proved itself useful in connection with textile machines and machines which should not be idle during working hours.

Workmen's Time Recorders are now controlled from the half-minute impulses of the system by means of the 'Pul-syn-etic Reflex' Pendulum Control (fig. 56). With this control the recorders remain independent units, and no mechanical alterations are made, but with the 'Reflex' fitting, as shown in fig. 56, attached to their pendulums, all recorders are compelled to keep in step with the transmitter.

Even if the electrical connections be removed the recorders continue to work under their own power as before, but uncontrolled. This 'Reflex' control is also applicable to the pendulums of existing large public clocks, under the name of Duflex Control, and its introduction is a marked advance in the science of the Unification of Time.

For large turret clocks, the accuracy of whose performance is subject to interference from variation in wind pressure or from accumulations of snow and ice on the hands, a special WAITING TRAIN movement is used, in which the hands are driven continuously by a special type of electric movement sufficiently powerful to overcome any such resistance ordinarily to be expected. The hands are caused to travel at a slightly faster rate than usual, to gain in fact about two seconds per half minute. Every half minute, however, the driving train disconnects itself from the hands, which then wait until the driving power is reconnected by means of the usual half-minute electric impulse current sent out by the transmitter driving the other impulse clocks in the building. The waiting period is practically unnoticeable—so that in very large turret clocks immunity from the usual weather interruptions is achieved, together with a greater degree of accuracy in timekeeping than in clocks operated in the usual manner. Efficient electric striking, or striking and chiming apparatus, can be supplied for bells of any size.

This WAITING TRAIN movement has been chosen for driving the hands of the large electric clock, in the towers of the offices of the Royal Liver Friendly Society at Liverpool. This has four 25-ft. dials, and the weight of a pair of hands is 5 cwt. The minute hands are 14 ft. long, 2 ft. 6 ins. wide at the greatest width, and 9 ins. thick where attached to the spindle. The combined weight of each pair of hands, spindles, and internal counter-weights, is a trifle over half a ton. The area of the hands exposed to snow and wind is very great, nevertheless the type of mechanism employed has shown itself able to withstand the disturbing effect of several severe storms since its erection in 1911. The size of the Liver Clock has now been exceeded by the new 'Singer Clock' in the tower of the works of the Singer Mfg. Co., Ltd., Clydebank, which clock has four 26-ft. dials, all driven by a single 'Waiting-Train' movement.

For ship use special clocks are provided, the Impulse Transmitter being fitted with an automatic arrangement, which enables the whole of the ship's clocks to be automatically advanced or automatically retarded, while sailing east or west. These 'Pul-syn-etic' marine clocks are installed in a large number of liners and ocean-going steamers.

LIGHTNING CONDUCTORS.

CODE OF RULES FOR THE ERECTION OF LIGHTNING CONDUCTORS.

(Lightning Rod Conference, 1882, *and Lightning Research Committee, Royal Institute of British Architects*, 1905.)

Two main lightning rods, one on each side, should be provided, extending from the top of each tower, spire, or high chimney stack by the most direct course to earth. Horizontal conductors should connect all the vertical rods (*a*) along the ridge, or any other suitable position on the roof; (*b*) at or near the ground line. The uppermost conductor should be fitted with aigrettes or points at intervals of 20 or 30 feet. Short vertical rods should be erected along minor pinnacles and connected with the uppermost horizontal conductor. All roof metals, such as finials, ridging rain-water and ventilating pipes, metal cowls, lead flashing, gutters, etc., should be connected to the horizontal conductors. All large masses of metal in the building should be connected to earth, either directly or by means of the lower horizontal conductor. Where roofs are partially or wholly metal lined they should be connected to earth by means of vertical rods at several points. Gas pipes should be kept as far away as possible from the positions occupied by lightning conductors, and as an additional protection the service mains to the gas meter should be metallically connected with house services leading from the meter.

Material for Rod.—Copper, weighing not less than 6 oz. per foot run, either in the form of tape or rope of stout wires, no individual wire being less than No. 12 S.W.G. Iron may be used but should not weigh less than 36 oz. per foot run. Subsidiary conductors for connecting metal ridging, etc., to earth may with advantage be of iron and of a smaller gauge, such as No. 4 S.W.G. galvanised iron.

Points.—The point of the terminal should not be sharper than a cone of which the height is equal to the radius of its base; but a foot lower down a copper ring should be screwed and soldered on to the upper terminal, in which ring should be fixed three or four sharp copper points, each about 6 ins. long. The number of conductors or points to be specified depends upon the size of the building, the material of which it is constructed, and the comparative heights of the several parts; no general rules can be given, but even ordinary chimney stacks, when exposed, should be protected by short terminals connected to the nearest rod, as accidents often occur owing to the good conducting power of the heated air and soot in a chimney.

Fixing.—Rods should preferably be taken down the side of the building which is most exposed to rain. They should be held firmly by holdfasts of the same metal as the conductors (not by insulators), but not so tightly as to prevent free expansion and contraction due to changes of temperature. They should not be bent abruptly round sharp corners; the vertical rods should be carried at a certain distance from the wall to prevent dirt accumulating, and to avoid the necessity of running them round projecting masonry or brickwork. In some cases it may be necessary to carry a rod through a string-course or other projecting stonework to avoid curvature, in which case the hole should be large enough to give ample clearance to the conductor, but it is better to run outside, keeping the rod away from the structure by means of holdfasts as mentioned above.

Joints.—Joints in series of conductors should be as few as possible. As a rule they should only be necessary where the vertical and horizontal conductors are connected, and the main conductors themselves should always be continuous, without artificial joints. Connections between vertical and horizontal conductors should always be in places readily accessible for inspection. Every joint, besides being well cleaned, screwed, scarfed, or riveted, should be thoroughly soldered, and should be protected from the action of the air, especially in cities.

Painting.—Iron rods, whether galvanised or not, should be painted; copper rods may be painted or not, according to architectural requirements.

Protection.—Copper rods to the height of 10 ft. above the ground should be protected from injury and theft by being enclosed in an iron pipe reaching some distance into the ground.

Earth Connections.—It is essential that the lower extremity of the conductor be buried in permanently damp soil; hence proximity to rain-water pipes and to drains is desirable. It is a very good plan to make the conductor bifurcate close below the surface of the ground, and adopt two of the following methods for securing the escape of the lightning to earth: a strip of copper tape may be led from the bottom of the rod to the nearest gas or water *main*—not merely to a lead pipe—and be soldered to it; or a tape may be soldered to a sheet of copper 3 ft. sq. and $\frac{1}{16}$ in. thick, buried in permanently wet earth, and surrounded by coke, charcoal, or pulverised arc-lamp carbons (coke, however, is likely to set up corrosive action on the earth-plate), or many yards of the tape may be laid in a trench filled with coke, well rammed. Where iron is used for the rod, a galvanised iron earth-plate should be employed. A tubular earth connection driven into moist ground makes an effective earth, especially if it is connected to a rain-water pipe. (See also 'Earth Connections,' *below*.)

Masses of Metal.—As far as practicable the conductor should be connected to all extensive masses of metal, such as ornamental ironwork, hot-water pipes, etc., both external and internal; but it should be kept away from all soft metal pipes, and from internal gas-pipes of every kind. Church bells and turret clocks should be connected with the conductor. In the case of a long line of metal ridging, each end of the ridging should be directly connected to earth by a rod. Where the ridge is non-metallic, a horizontal conductor should be run at a short distance above the ridge and be connected to earth.

Factory Chimneys.—These should have a copper band round the top, and stout sharp copper points, each about 1 ft. long, at intervals of one or two feet all round the circumference. The rods should be connected with all bands and metallic masses in or near the chimney. As an alternative, the rods above the band may be curved into an arch provided with three or four points. Two lightning rods should be carried down from the band to earth.

Collieries.—Undoubted evidence exists of the explosion of firedamp in coal mines through sparks from atmospheric electricity being led in to the mine by the wire ropes of the shaft and the iron rails in the galleries; hence the head-gear of all shafts should be protected by proper lightning conductors.

Inspection.—Before giving his final certificate the architect should have the conductor carefully examined and tested by a qualified person, as injury to it often occurs up to the latest period of the works from accidental causes, and often from the carelessness of the workmen.

Earth Connections.

In making earth connections for electrical circuits, lightning rods, etc., it is desirable to provide several paths to earth, as far apart as possible. Connection should be made with the stratum of permanently moist earth nearest the surface of the ground, rather than to deeper layers or even to the water in a deep well. The best earth connection is a water pipe. For most purposes galvanised iron pipes, about 1 in. in diameter, give satisfactory results; in a series of experiments such pipes, driven into stony clay soil, gave the following average results per pipe:—

1 ft. deep	149 ohms	4 ft. deep	50 ohms.
2 ft. „	88 „	6 ft. „	36 „
3 ft. „	64 „	8 ft. „	29 „

Generally, if moist earth is reached, pipes need not be driven deeper than 10 ft. The resistance of two earth-pipes in series, driven 10 ft. deep, becomes practically constant at any distance apart from 10 ft. upwards. Earth-pipes connected in parallel should be at least one foot apart. They need not be larger than 2 ins. in diameter.

If earth-plates are used it is better to bury two small plates 20 ft. apart, connected by a stout wire, than one large plate. They should be on edge, with the soil well rammed on both sides. In shallow soil over rock, long narrow strips of metal buried in trenches should be used. Patented devices offer no special advantages. Two plates in parallel, each 2 ft. by 4 ft., 6 ft. deep, 5 ft.

apart in stony clay soil had a joint resistance of 16 ohms. A strip of galvanised iron 1·5 ins. wide and 20 ft. long, in a trench 12 ins. deep, had a resistance of 17·7 ohms. The resistance of a water pipe is usually less than 1 ohm, averaging $\frac{1}{4}$ ohm. The addition of salt to the soil round the earth connection greatly reduces the resistance; so does the packing of coke or charcoal round the plate, but both these devices hasten the corrosion of the plate. As a high potential gradient may exist on the surface near the earth connection, unless it is of the buried type, the earth-pipe and connecting lead should be guarded to prevent accidental contact with them. The joint between the lead and the earth-pipe or plate should be embedded in concrete for protection, and the earth-wire should be covered throughout its length. All earth connections should be inspected and tested at least once a year.

Phœnix Office Rules for Lightning Conductors.

The Phœnix Fire Office, in their rules for lightning conductors, point out that the only way to obtain absolute security against lightning is to enclose the building in a metal framework constituting the 'bird-cage' system of conductors. This can be accomplished generally by utilising the rain-water pipes and gutters as part of the system. The ordinary conductors must be fixed in addition, and the whole system effectively earthed. The material of the conductor should be of copper tape, measuring at least 1 inch by $\frac{1}{8}$ inch, or, if copper cable be adopted, it should not consist of less than seven strands, each $\frac{1}{4}$ inch diameter. Soft iron may be used in place of copper, and, except for chemical reasons and for permanence, the sectional area need not be greater than in the case of copper. Conductors should be kept a certain distance away from walls so as to prevent the accumulation of dirt, but they must not be insulated. If a copper plate is used for the earth connection it should not be less than 3 square feet in area and $\frac{1}{4}$ inch in thickness, and should be adopted only where it will always be in water or moist earth; otherwise connection should be made to the water mains, or the 'tubular' system should be employed. The number of connections depends on the ground area of the building, but there are few structures where less than two are necessary. The elevation rod of factory chimneys should be a circular band placed round the top of the shaft, with points fixed at an angle of 45°, and extending outwards, or an arch of metal, provided with points, may be fixed across the mouth of the chimney.

Domestic Uses of Electricity.

It is seldom that lighting is the only domestic service obtained when an electricity supply is available. Provided the cost of electricity is not exorbitant, certain valuable appliances are to be met with in most households, *e.g.* flat iron, vacuum cleaner, desk fan, toaster, coffee percolator, milk boiler, kettle, warming pan, hair dryer, bowl fire, etc., etc. Such services can be had at a very small running cost, since the amount of energy consumed is generally small. Where electric energy can be obtained at a low rate, *e.g.* at an overall charge of $\frac{3}{4}d.$ to $1\frac{1}{2}d.$ per unit, main domestic services such as heating, cooking, and hot-water supply are available at costs comparable with other methods, such as coal or gas. Even where the cost of electricity may be prohibitive for general use, it may be low enough for occasional use. In addition the advantages of cleanliness, convenience, comfort, labour-saving, etc., are valuable assets. Reliable appliances for all the above-mentioned services are now available, and when the capital costs of wiring and of apparatus are suitably reduced or embraced in hire-purchase systems, etc., a great expansion of domestic electrification can be looked for.

Data concerning heating, cooking, and hot-water supply in private households will now be given. In Table I figures are reproduced from p. 305, vol. 64, of the *Journal I.E.E.* These figures were given by Prof. A. H. Barker in the discussion of a paper on 'An All-Electric House,' by Prof. S. Parker Smith, and are intended to show how many heat units are needed per person per annum for the stated services:

Table I.—Net B.Th.U. used per Person per Annum.

Description.	Estimate for well-to-do Person.	Estimate for Artisan.	From Barker's Experiments for One Person Living Alone.	From Parker-Smith's All-Electric House.
Heating . .	3,000,000	1,000,000	4,500,000	3,400,000
Hot-water supply and laundry . .	8,000,000	2,000,000	2,100,000	4,150,000
Cooking . . .	400,000	180,000	800,000	474,000
Total .	11,400,000	3,180,000	7,400,000	8,024,000

In all domestic electrification, the Regulations of the I.E.E. concerning Electrical Equipment of Buildings should be observed—particularly as regards earthing in bathrooms and kitchens.

AN ALL-ELECTRIC HOUSE.

In Tables II.–IV. the consumption and costs in the 'All-Electric House' just referred to are given.

The differences between the totals in Tables II. and III. are due to meter errors.

TABLE II.—TOTAL CONSUMPTION IN UNITS RECORDED ON TWO-RATE MAIN METER.

Year of 52 Weeks.	1st.	2nd.	3rd.	4th.	5th.	Average p.a. In Units.	Average p.a. Percentage.
Day load	10,299	9,713	9,079	10,727	11,720	10,307	64
Night load	6,043	5,827	5,241	5,848	5,897	5,771	36
Total for year	16,342	15,540	14,320	16,575	17,617	16,078	100
House closed	19	10	13	8	7 days		

TABLE III.—ANALYSIS OF CONSUMPTION IN UNITS AS OBTAINED FROM SEPARATE METERS.

Year of 52 Weeks.	1st.	2nd.	3rd.	4th.	5th.	Average p.a. In Units.	Average p.a. Percentage.
Heating	5,954	6,636	6,585	7,781	8,265	7,044	42·0
Hot water { day	1,242	1,245	1,103	993	1,315	6,951	41·5
Hot water { night	6,043	5,827	5,241	5,848	5,897		
Cooking	2,777	1,858	1,706	1,963	2,037	2,068	12·5
Lighting	568	687	717	785	863	724	4·0
Total for year	16,584	16,253	15,352	17,370	18,377	16,787	100

TABLE IV.—TOTAL ANNUAL COST OF ENERGY (BASED ON FIGURES IN TABLE I.).

Year of 52 Weeks.	1st.	2nd.	3rd.	4th.	5th.	Average p.a. In Cost.	Average p.a. Percentage.
Day load at ½d. per unit	£ 21 s. 9 d. 2	£ 20 s. 4 d. 8	£ 18 s. 18 d. 3	£ 22 s. 7 d. 0	£ 24 s. 8 d. 4	£ 21 s. 9 d. 6	50
Night load at ¾d. per unit	9 8 10	9 2 1	8 3 9	9 2 9	9 4 3	9 0 4	21
Fixed charge	12 10 0	12 10 0	12 10 0	12 10 0	12 10 0	12 10 0	29
Total for Year	43 8 0	41 16 9	39 12 0	43 19 9	46 2 7	42 19 10	100

Comments on above Results.

1. For the third year, an extra of £9 0*s.* 3*d.* falls to be added for the general strike of 1926.

2. The increase in the fifth year is to be accounted for by the exceptionally severe winter of 1928–29. No discomfort was experienced, but fires had to be switched on for a longer period before the room was occupied. The longer time taken for cooking operations and heating water during this cold spell was pronounced.

3. The fact that the extremes fall within 10 per cent. of the mean justifies the conclusion that a definite measure of the annual consumption and cost has been obtained.

4. Experience soon showed that undue care in consumption was not necessary. Extravagance was, of course, avoided, but except during the general strike, no particular economy in heating water, &c., was practised, while occasional heating of bedrooms has greatly increased.

5. The number of regular occupants in the house varied from four to six, whence it can be concluded that for such small numbers it is meaningless to express consumption or cost in terms of 'per person per day.'

6. The average overall cost per unit over the five years worked out at 0·64*d.*, or under two-thirds of a penny.

7. The consumptions for heating and for hot water are almost identical; cooking is only one-eighth of the total, while lighting is negligible.

8. Over one-third of the total consumption was taken between 11 P.M. and 7 A.M.—a valuable asset to stations with a small night load.

An All-Electric Flat.

A three-apartment flat with bathroom and kitchen was converted from gas and coal to all-electric working. The consumptions and costs for two years are given in Table V.

TABLE V.—Total Annual Consumption and Cost of Energy.

Year of 52 Weeks.	1st.	2nd.	Average	
			In Kind.	In Per Cent.
Consumption in units . . .	6,862	6,987	6,925	—
	£ *s.* *d.*	£ *s.* *d.*	£ *s.* *d.*	
Cost at ½*d.* per unit . . .	14 5 11	14 11 2	14 8 6	76
Fixed charge	4 11 0	4 11 0	4 11 0	24
Total for Year	18 16 11	19 2 2	18 9 6	100

Average consumption per week . . . 133 units
Average cost per week . . . . . 7*s.* 4*d.*
Average cost per unit . . . . . 0·66*d.*

The Artisan's All-Electric Home.

Figures were published by Dr. S. Parker Smith in *The Electrician* (vol. 96, p. 346). Allowing for losses in the hot water and cooking appliances, the estimate for an artisan given in Table I. is increased to 3,750,000 B.Th.U. or 1100 kw. hours per person per annum, making 5,500 kw. hours per annum for a household of 5 persons. At an overall cost of 1*d.* per unit, this works out at slightly under £23 per annum—a rather high figure for some families. In many areas, however, the cost per unit averages ½*d.* to ¾*d.*, bringing the annual outlay down to a figure between £12 and £17, which is well within the reach of many artisans' families, and compares favourably with alternative methods.

For large kitchens in restaurants, works canteens, etc., various sizes and combinations of apparatus are made. The following data are taken from actual installations.

Class of Kitchen.	Total kilowatts.	Annual Consumption units.	No. of Meals per week.
Restaurant	247·6	146,500	9,518
Staff feeding	120·6	97·140	2,944
,, ,, ...	40·5	21·141	991
,, ,, ...	32·8	20,690	640

British Standard Specification No. 106—1920 gives particulars of electric cooking ranges or 5 and 10 persons respectively.

See also 'The Domestic Applications of Electricity,' by A. J. Milne, Grad. I.E.E. and R. H. Rawll, A.M.I.E.E. (*Journal of the Institution of Electrical Engineers*, No. 379, July 1928.)

Large Electric Baking Ovens.

Asbestos in block form is the most convenient insulating substance for baking ovens. Slag wool can be used for horizontal surfaces, but for vertical situations block or slab asbestos should be employed. Slag wool mat can be exposed to temperatures as high as 1350° F. without losing its fibrous structure.

The following table gives *mean* temperature, *i.e.* the average between the hot and cold faces of a slab of slag wool mat when equilibrium had been attained.

Mean temperature.	Thermal conductivity in B.Th.U. per sq. ft. per hour per 1 in. thickness for 1° F. difference.
° F.	
300	0·48
400	0·56
500	0·64
600	0·72
700	0·80
800	0·88
900	0·96

Resistor.

The following can be taken as desirable electrical, mechanical and thermal properties of a resistor suitable for baking ovens.

Specific resistance . .	105 microhms per cm.³	Specific heat . . .	0·106.
Temperature coefficient .	0·00012.	Specific gravity . .	8·35.
Tensile strength . .	59 tons per sq. in.	Coefficient of expansion .	0·000015.
Melting point . .	1375° C.		

The above is a pure nickel-chromium alloy containing 20 per cent of chromium with but the merest trace of iron as an impurity.

Heating elements should dissipate their full rated energy at normal temperature without reaching red heat.

Long life and efficiency are the essential conditions demanded of a heating element, and attention must be given, when inserting them in the oven, to good heat transmission, sound end-connections, easy accessibility and low current density per unit of surface.

Elements embedded in cement are protected against oxidation and have an advantage over exposed elements, particularly when temperatures of 650° C. to 700° C. are exceeded. This method has a marked influence on the life of the element.

Cost of Operation.

Taking an average from over 4,000 electric bakery installations varying from 300 to 14,000 loaves daily output, the average power consumption is 90 watts per pound of bread baked. A typical bakery with 2 baking ovens having a total connected load of 41 k.w. has a daily output of 4,000 lbs. of bread and a monthly consumption of 3,500 kw.h.

One bakery in which tests were made had a monthly production of 3,000 cakes and 24,000 loaves of bread. Both gas and electric equipment were used for comparison. An average saving of 5 to 6 ozs. per cake, or a total of 156½ lb., was made by the electric oven. The bread savings totalled 500 lbs. It was further found that the gas-baked cakes shrank 5 ozs. each more than the electrically baked cakes in a period of 5 hours after baking. In another instance a bakery claims a saving of ½ to ¾ oz. per pound of bread baked.

To assure maximum economy in operation the smallest possible oven should be installed that will turn out the required production in the longest number of hours during continuous operation. This will keep the maximum demand to a minimum. It is shown that a smaller oven operating over a longer period of hours can be operated much more economically than a larger oven operated over a shorter period. Assume 1½ to 3 hours to preheat the oven, according to size. After preheating it takes from 50 to 60 per cent. of the maximum load of the oven to bake the heaviest products. Assuming the oven to be efficiently insulated, considerable heat is retained long after the current has been switched off.

The following is an example of the estimated cost of operation :

Connected load	10·8 kw.
Baking surface	30 sq. ft.
Capacity (1 lb. loaves)	80
Hourly capacity (1 lb. loaves)	120
Capacity when operating for 7 hours (1 lb. loaves)	840
Power consumption during preheating (2 hours)	21·6 kw.h.
Operating to maximum capacity for 7 hours (10·8 × 7 × 50 per cent.)	37·8 kw.h.
Total power consumption	59·4 kw.h.
Cost of current at 0·75*d.* per kw.h.	3*s.* 8½*d.*
Monthly cost (26 days)	£4 16*s.* 5*d.*
Monthly output of bread	21,840 lbs.
Electric current cost per 100 lb. of bread	5·3*d.*

It can be accepted that a good oven has an hourly capacity of 1½ times its deck capacity for bread baking, and twice the deck capacity hourly for rolls and buns, the hourly pastry depending upon the variety.

Records have established that baking-oven installations consume, on an average, 150 kw.h. per month for each kw. of connected load.

The maximum operating demand averages 50 to 60 per cent. of the connected load on the equipment. In some cases installations are consuming 200 kw.h. per month for each kw. of connected load.

An analysis of existing representative installations is given below.

	kw.h. per month per kw. of connected load.
82 baking oven installations	101
13 baking oven installations	146
15 baking oven installations	184
A wholesale bakery	185
	616
Average kw.h. per month per kw. of connected load	154
Load factor	40 per cent.
Power factor	Unity

(*Ellerd-Styles, Journal of the Institution of Electrical Engineers, No.* 378, *June* 1928.

ELECTRICAL PRECIPITATION OF SMOKE AND DUST.

In recent years the Lodge system of precipitating dust particles or fumes in gases, such as the flue gases from metallurgical works, cement mills or boiler furnaces, has been greatly developed by the Research Corporation, U.S.A., under Cottrell's patents. The apparatus consists of iron pipes through which the gases are led, preferably downwards, with a steel or nichrome wire suspended in the axis of each tube ; the wires are charged to a pressure of 30,000 to 100,000 volts and the pipes are earthed. The electric charge is derived from a.c. transformers with mechanical rectifiers driven in synchronism with the alternating pressure so that the polarity of the wires is always negative. The process can be carried on without cooling the gases and requires little power, from 5 to 50 kilowatts being usually sufficient. Resistance should be inserted in the low-pressure circuit to limit the short-circuit current and to provide means of regulation, and reactance in the high-pressure circuit to prevent surging of the electric charge.

From 80 to 98 per cent. of the dust can be deposited; in some cases the introduction of steam is necessary. In the case of blast furnace and cement mill gases, large quantities of potash can be recovered, particularly if salt is added to the charges of the blast furnace. Particulars of some successful installations are given below:—

Nature of Process.	Number of Pipes.	Length and Diameter of Pipes.	Pressure volts.	Rate of Flow of Gases.	Remarks.
Copper converter	608	10 ft. × 5 in.	30,000	10 ft. per sec.	97·25 per cent. recovered.
Silver furnace .	Chamber 8 ft. × 6 ft. × 16 ft.		50,000	—	Electrodes 6 in. lead strip.
Cement mills .	800	15·5 ft. × 12 in.	70,000	3·5 ft. per sec.	Temp. 900° F. 20 tons per day.
Copper and lead smelting works	384	15 ft. × 12½ in.	60,000	10,000 cub. ft. per day	Plant increased to 1,312 pipes.
Detarring coal gas	—	—	40,000 to 80,000	30,000 cub. ft. per hour	Unrectified alternating current.

Careful tests should be made before deciding on the dimensions, etc., of the apparatus to be installed. Corrosive substances can be dealt with satisfactorily. At a certain cement factory the potash recovered is worth more than the cement. The kenotron is an efficient rectifier up to 250,000 volts. A similar process is used for separating oil from water.

ELECTRICITY IN AGRICULTURE.

The principal use of electricity in agriculture is to drive machines, in addition to lighting the farm buildings; it is also experimentally used for stimulating the growth of plants. Motors are used either to drive stationary machines indoors from counter-shafting, or, being mounted on wheeled trucks or stretchers, to drive portable machines out of doors, through belting. A farm load has an excellent diversity factor, and as it does not come on during the evening peak load it is very acceptable to a public supply system. Experience shows that for remunerative supply, at least 25 per cent. should be arable or dairy land. At Hereford the average revenue from farms is about 20*l.* each per annum. With an electric milker one person can milk six cows at once, using 20 watt-hours per cow. To churn 165 lb. of butter requires 1 kw.-hour. Cutting 1 ton of ensilage consumes 1·38 kw.-hours. Threshing corn requires 70 watt-hours per bushel of oats, 100 watt-hours per bushel of barley, 160 watt-hours per bushel of wheat. Grinding a bushel of oats takes 420 watt-hours. Elevating 1 ton of hay, 200 watt-hours. The most suitable type of machine is the three-phase induction motor, on account of its simplicity and safety from fire; fire risk is a most important consideration in connection with farm work.

Electric ploughing is a special application of electricity under exceptional conditions; it is usually effected with hauling gear mounted on trucks, which travel along the sides of the field, hauling the plough (with three to five shares) to and fro across the land. The tractive effort required for ploughing in light soil is 740–820 lb. per sq. ft. of section ploughed; in heavy soil 1,100–1,250 lb. For ploughing a furrow or furrows w ft. wide and d ft. deep, at v ft. per second, the horse-power required = $kwdv/550$, k being a constant as above, depending upon the soil. The speed v should not exceed 5 or 6 ft. per second. The motor should be capable of hauling double the normal load at times. Furrows up to 650 yds. long can be ploughed in this way. On a large farm the capital charges amount to 12*s.* per acre, wages to 2*s.*, energy at 1*d.* per unit to 2*s.* 6*d.* per acre for deep ploughing (12–14 ins. deep), total 16*s.* 6*d.* per acre. For ordinary ploughing the cost is 11*s.* 4*d.* per acre. The hauling gear can be used also for harrowing and rolling the land, and for driving other machinery. The motor for ploughing should be of 25–30 h.p. (*Delamarre.*)

A large-sized plough can work 27 acres per day against one acre with horses.

For stimulating plant growth, the silent discharge from fine bare wires suspended above the crops at a pressure of 50,000 volts or more, d.c., has been much used for experimental purposes. The Agricultural Discharge Co. suspends the wires 10/12 yds. apart, at a height of 16/18 ft., using one pole to 3 or 4 acres.

The Board of Agriculture reported in 1918 that experiments extending over six years showed an average increase of 22 per cent. in the yield of spring cereals; some tests gave over 30 per cent. increase. The cost of a 100-acre equipment is estimated at 4*l.* per acre.

In a paper by R. Borlase-Matthews (*Journal I.E.E.*, 1926, vol. 64, p. 801) recent advances are described and the results of much experimental work given. Details of the results obtained with electric ploughing, rotary tillers, and the reduction of plough friction by passing a current between the coulter and ploughshare are recorded. Crop treatment has resulted in the production of a better milling wheat and superior malting barley. The electrical method of preserving green fodder (silage) has been improved by the addition of tubular heaters. In this case, the load is kept constant during the whole time of treatment, energy consumption averaging 1½ kw. hours per cwt. of material. Central manuring plants are in use in which the contents of the manure tank are agitated by electrically-driven revolving arms or by circulation through centrifugal pumps. The liquid is pumped out to stands in the fields and distributed by a hose pipe. The mains used for manuring can be employed for irrigating. The consumption for combined watering and manuring is stated to be 90 units per acre. By illumination of live-stock sheds and poultry houses, half an hour per day per man is saved in the feeding of stock, and an increase of 15 to 43 per cent in egg production during winter months is obtained. Lighting and heating in connexion with beehives has resulted in an additional 17 lb. of honey per hive per annum.

Agricultural type motors have been developed on the Continent and machines suitable for the very severe conditions which obtain are available. Such a machine is totally enclosed and embodies moisture-proof vacuum impregnated windings and internal ventilation. Ball bearings should be used, simple over-load circuit breaker provided, while the starter should be enclosed in the motor frame. Since the motor should be capable of working in any position the cable connection should be made through a plug and socket. A 5-h.p. motor is the largest size required on most farms except for ploughing, 2 h.p. being a generally useful size. This entails the use of suitable farm machinery. Recent practice tends towards the use of permanently installed motors on individual lines. Belt drives should be avoided by the use of self-contained compound reduction gearing, a friction drive replacing gear-wheels for motors of less than 5 h.p. Conductors are generally run outside the buildings under roof-eaves of farm buildings. For interior wiring the use of tough rubber-protected cable is advisable.

Electrical Heating of Soil for Horticultural Purposes.

This consists in heating the soil electrically by means of buried lead-covered cables enclosing high-resistance wire circuits. The success attained has made the system a commercial one. In Norway alone there are about 200 nurseries which make use of this method. In Great Britain (1928) there are two installations, one at the Cheshunt Horticultural Research Station and the other on the farm of Mr. Borlase Matthews at East Grinstead.

It is not necessary to maintain a constant supply of energy, as at 75° F., the standard temperature, the soil retains its heat for a long period.

Much less labour is required when the soil is heated electrically, and the plants are as healthy and strong as when the best manure hotbed is employed. In Sweden the method has been tested on a large scale. The standard electric beds are 92 ft. long and the cables are enclosed in earthenware ducts to prevent them being damaged by the gardener's spade. Strawberries have been raised to maturity in two months by this means. Potatoes planted in April were ready for digging at the end of June, two months before the ordinary crop. The commercial results obtained have proved satisfactory.

TELEPHONY.

Telephonic Transmissions.

The actual frequencies in normal speech range from about 200 to 2,500 periods per second, the mean frequency is assumed to be 800 for all transmisson calculations. Electromagnetic waves travel along telephone cables with a velocity depending on the electrical characteristics of the cable—i.e. resistance, inductance, capacity, and leakance; hence there is a time-lag between the application of an e.m.f. at the end of a cable and the time when it affects a point some way along the cable. At a certain distance a complete period will have been lost; the length to this point is called the 'wave-length' of the cable (α).

The magnitude of the e.m.f. also suffers attenuation, owing to the loss of energy due to resistance and leakage, according to a logarithmic law, involving the 'attenuation constant' of the cable (β). These constants are found by the following formulæ:—

$$\alpha = \sqrt{\tfrac{1}{2}\left\{\sqrt{(R^2+\omega^2L^2)(G^2+\omega^2C^2)} - (RG-\omega^2LC)\right\}}$$

$$\beta = \sqrt{\tfrac{1}{2}\left\{\sqrt{(R^2+\omega^2L^2)(G^2+\omega^2C^2)} + (RG-\omega^2LC)\right\}}$$

per mile in each case, where,

R = resistance in ohms per mile;
L = inductance in henries per mile;
G = leakance in mhos per mile;
C = capacity in farads per mile;
ω = angular velocity in radians per second = $2\pi f$, where f is the frequency in periods per second.

These constants are of the greatest importance, as they measure the speaking qualities of the cable.

The relative attenuation of the different frequencies present in the voice—i.e. the 'distortion'—is measured by the rate of change of the attenuation constant β with frequency, and is designated by δ. Measurements are made at frequencies giving ω values of 3,000, 5,000, and 7,000 radians per second, and the distortion is given by the formula:

$$\delta = 25\,(\beta_{7000} - \beta_{3000})/\beta_{5000} \text{ per cent.}$$

Transmission can be improved by decreasing R and C and by increasing L; the two former methods are limited by the resulting increase in the weight and size of the cable. β would be independent of frequency, and the cable therefore distortionless, if R/L were equal to G/C; for an ordinary cable R/L is of the order of 5,000, and for gutta-percha G/C is about 120, so that inductance has to be added, to diminish both attenuation and distortion. This is done either by winding iron wire uniformly in a close helix along the length of the conductor, or by inserting inductance coils at intervals; in the latter case there should be at least nine coils per wave-length at 800 cycles per second.

Loading has been extensively adopted for land-lines, but is more necessary for cables, underground or submarine; for the former, coils are used, but for the latter, both methods are employed.

To provide a standard of comparison for different types of cable, the Post Office has adopted a 'standard cable' having the following characteristics (unloaded) per mile: Total resistance, R, 88 ohms; inductance, L, 1 millihenry; capacity, C, 0·054 microfarad; leakance, G, 1 micromho; G/C, 18·5; wave-length constant, α, 0·1119; wave-length, 56 miles; attenuation constant, β, 0·1062; distortion, 9·5 per cent.

The British Post Office standard for long-distance communication is that the audibility should not be worse than over 35 miles of standard cable. Allowing 10 miles for the local lines, the balance for the transmission line proper is 25 miles. Examples of the lengths of certain types of line giving this equivalent are given in the table, all being 150 lb. per mile.

	Loading.	R ohms	L μH	C μF	G, μ mhos	G/C	α	Wave-length, miles.	β	Standard Cable Equivt.	Length = 25 m.s.c.	Distortion.
Underground cable	nil	11·73	1·0	0·065	1	15·4	0·537	117	0·0356	0·335	74·5	5·5
Aerial wire	nil	11·73	3·76	0·0084	1	120	0·0292	215	0·0087	0·082	304	2·6
Submarine cable	nil	11·73	1·25	0·15	18	120	0·0851	74	0·0525	0·494	50·5	6·0
Submarine cable	coils, 80 m.H. per naut.	16·08	70	0·15	18	120	0·513	12·2	0·0179	0·169	148	11·6
Submarine cable	coils, 80 m.H. per naut.	16·08	70	0·15	3	20	0·513	12·2	0·0127	0·120	208	5·0
Submarine cable	Continuous, 3 layers 0·008 inch iron	13·21	21	0·18	3·6	20	0·308	20·4	0·0199	0·188	133	5·0

The equivalent length of standard cable is now being superseded, as a means of expressing the inefficiency of transmission lines, repeaters, etc., by the 'transmission unit' or 'T.O.' When the input power (P_i) to a system and the output power (P_o) are such as to make the expression $10 \log_{10} \frac{P_i}{P_o}$ unity, the system is said to possess 1 T.O. of inefficiency. It is equivalent to 1·088 miles of standard cable at 800 cycles per second.

The thermionic valve (p. 2343) is an almost ideal relay for telephonic transmission. For perfect action the impedance of the line must be the same in both directions and for all frequencies; for this purpose each side is balanced by an artificial line, enabling even cables to be relayed on to aerial lines, but in this connection continuously loaded cables are much more suitable than coil-loaded cables. A double-relay repeater gives an improvement equal to 15–25 miles of standard cable; with the aid of successive repeater stations, good speech is now possible between New York and San Francisco, a distance of about 3,000 miles.

(*A. Rosen, Journal I.E.E.*)

Loaded and Repeated Telephone Cable Circuits.

(*Standard Telephones & Cables, Ltd.*)

The introduction of the telephone repeater to the sphere of practical telephony has made it possible to give high-grade telephone transmission over very long distances, and also to reduce

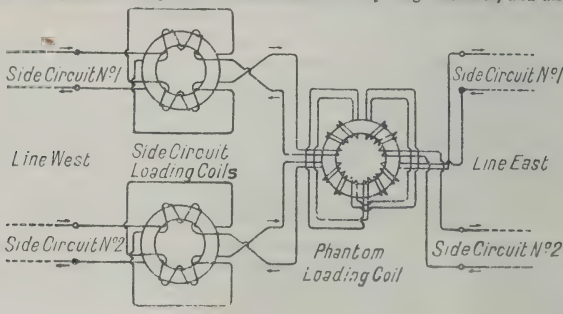


FIG. 58.

the cost and annual charges of many of the medium-length circuits found in the ordinary telephone plant. Telephone repeaters may be applied to improve the transmission of practically all circuits over which transmission is not normally satisfactory; but, when used in conjunction with carefully balanced and loaded telephone cable circuits, a system can be obtained which is as near the ideal as possible, both with regard to transmission qualities and to economical considerations.

Details of modern loading equipment and modern repeaters are briefly described under the following sub-headings.

LOADING.

The loading coils manufactured by *Standard Telephones & Cables, Ltd.* (formerly Western Electric Co., Ltd.), are designed to give the highest efficiency under all conditions of service. They have toroidal cores, which are constructed of a special magnetic material, rendering them immune from magnetisation troubles. The arrangement of the two side circuit coils and the phantom circuit coil, comprising a 'unit of loading,' is shown in fig. 58.

Loading coils can be designed to meet almost any inductance requirement. It has, however, been found convenient to establish certain standard types of coils, which cover the range of inductance required in most cases. These are, broadly speaking, classified (according to the amount of inductance) as 'extra light,' 'light,' 'medium,' and 'heavy' loading coils. The inductance of these coils covers a total range of from 25 to 250 millihenrys. When repeaters are used upon loaded lines it is necessary that the inductance of the coils be maintained within fine limits.

The coils, in suitable numbers, are contained in cast-iron cases, fig. 59, which are approximately of square cross-section, and which contain vertical spindles on which the loading coils are mounted, and are completely surrounded by insulating compound. The cases can contain up to and including 80 units of loading. In accordance with the latest practice, the coils are joined to the main cable through a single lead-covered stub cable.

FIG. 59.

Telephone Repeaters.

The three-electrode vacuum tube is the special feature of the modern telephone repeater as it is the special feature of modern radio-telephony and telegraphy. This vacuum tube is the most successful device for amplifying telephone currents that has yet been produced.

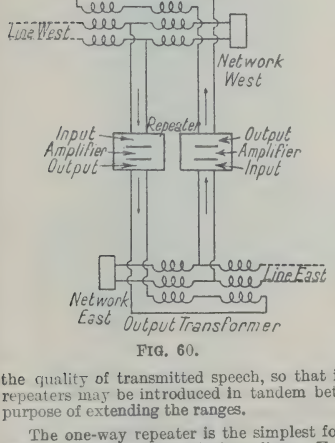


FIG. 60.

With the conducting and magnetic materials available to-day, the transmitted current on the best loaded line ultimately becomes attenuated beyond the value needed for commercial operation of the best telephone receiver. When this condition is reached the limit of improvement by loading alone has been obtained. This limit of distance is reached far short of the distances over which commercial telephony is desired, and other means for extending the range must be sought.

These are found in the telephone repeater, which is an amplifier or a device placed in the line at appropriate intervals for the purpose of adding new electrical energy to the circuit. In general, the current after passing the repeater is not better in wave form than the attenuated current which comes to it, and to this extent the repeater cannot serve one function of loading, which is to reduce the distortion. The modern telephone repeater functions so well, however, that the amplified current is to all intents and purposes an exact magnified image of the control current which it receives. In other words, it does not itself impair the quality of transmitted speech, so that if the line distortion is not too great a number of repeaters may be introduced in tandem between the transmitting and receiving stations for the purpose of extending the ranges.

The one-way repeater is the simplest form of repeater circuit. It is a device which can be connected between two telephone lines. Voice waves enter the input terminals from the transmitting ends of the circuit, and act upon circuit of the repeater in such a way as to set up an electromotive force in the output circuit which causes an output current to flow through the output impedance of the repeater to the receiving end.

Since it is essential that telephone lines should transmit equally well in both directions, it was found necessary to combine two of these one-way repeaters by special circuit arrangements, and the resulting apparatus is known as the two-way repeater (fig. 60). It will be seen that in addition to the two one-way repeaters there is additional apparatus supplied to prevent the flow of currents from the output circuit of each amplifier into the input circuit of the other. This consists of a three-winding transformer and a network, which are placed at each point where the circuits branch. Each network is made to balance its respective line, and consequently the output circuit of one amplifier and the input of the other have been effectively separated, while at the same time the inputs and outputs of the amplifiers have been left connected to the telephone circuits.

The two-way or 22-type repeater is usually used on circuits up to a length of 350 miles. Above this length the four-wire repeater is used in conjunction with a 4-wire circuit, which consists of two one-way transmission paths connected together at the ends by suitable transformers. The one-way paths may be either physical or phantom circuits. If these are physical circuits, the four-wire circuit really uses four conductors; while if the one-way paths are phantom circuits, the four-wire circuit has eight conductors. In principle the four-wire repeater consists of two one-way repeaters, each of which contains two vacuum tubes connected in tandem.

Telegraph and Telephone Lines.

Report No. 139—1921, of the British Engineering Standards Association, provides a specification for red fir poles. They should be felled when the sap is down, and contain the natural untrimmed butt of the tree. After seasoning, the poles should be impregnated with a minimum of 12 lbs. of creosote oil per cu. ft., the specific gravity of the oil being between 1·015 and 1·07 at 38° C. The properties of the oil are detailed in the Report, together with the methods of determining them, and the limiting dimensions of light, medium, and heavy poles, from 16 to 85 ft. long, are tabulated.

Report No. 134—1927 deals with iron and steel tubular poles.

Report No. 204—1924 gives terms and definitions used for telegraphs and telephones.

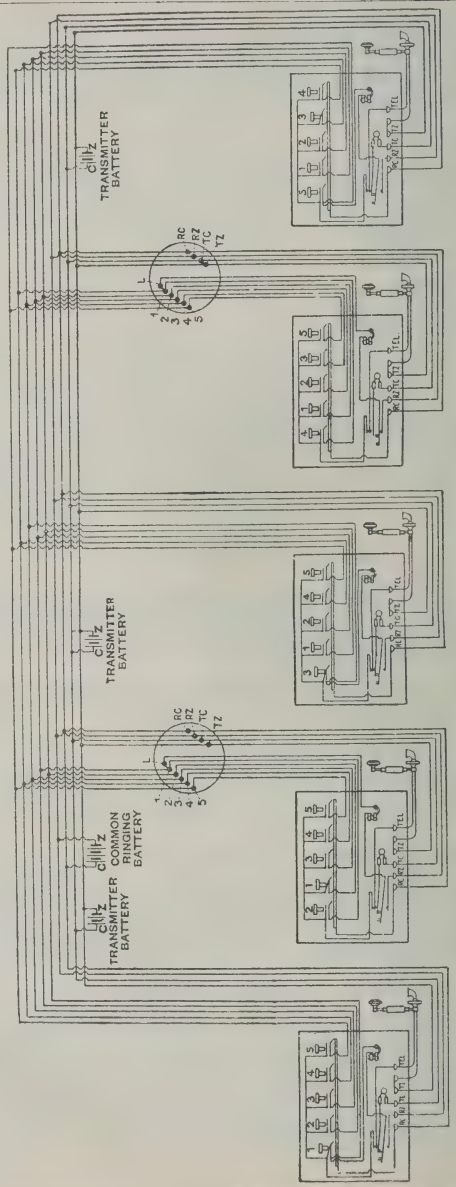


FIG. 61.—Wiring Diagram for G.E.C. Intercommunication Telephone System.

SPEAKING AND RINGING BATTERIES SHOULD NOT BE LESS THAN THREE CELLS EACH.)

Intercommunication Telephones.

(General Electric Co., Ltd.)

The circuit diagram of the G.E.C. Intercommunication Telephones is given in fig. 61, and a wall instrument (metal-cased, electro-magnetic type without induction coils) in fig. 62.

FIG. 62.—G.E.C. Wall Pattern Instrument.

This type of instrument has been specially designed by the G.E.C. with a view to strength and smooth and reliable working.

All working parts are easily removed should it be necessary at any time to make adjustments without disconnecting any of the internal wiring.

The push button line selector, line and instrument bar springs, together with the spring operated by the cradle switch, are all mounted on a steel frame by machine-cut screws, the frame being securely fixed on a stout hard wood base by means of metal bridge pieces at each end. Faults caused by warping woodwork or loose wood screws are thus obviated. The instrument possesses several unique points. It can be used either for a table or wall instrument, the cradle being of a design allowing the instrument to be fixed in either a horizontal or vertical position. This facilitates the keeping of stocks and ordering for export.

The instrument is made from one standard pattern, and can be readily extended from the minimum number of five lines to a maximum of twenty lines, by simply adding extra push buttons and springs, which are supplied at a reasonable price and in banks of five per set. These extra lines can be added without removing the instrument from its fixed position.

It is equipped with a hand combination, buzzer wound to 25 ohms, impedance coil and cradle switch, the cover being of black enamelled pressed steel.

To reduce inductive cross-talk, which is so noticeable with some types of intercommunication telephones, the G.E.C. have introduced the 'electro-magnetic' circuit in which no induction coil is used. Diagram fig. 69 shows a complete system with internal and external wiring. It will be noticed that more than one speaking battery of two cells is used, in order to keep the voltage constant on the common speaking battery leads. One battery only will suffice, providing all the instruments are within 50 or 60 feet of the battery. One ringing battery of three cells is usually sufficient to operate the calling signal. All lines are fitted with a black button with the exception of the home line, which is usually fitted with a red button.

With the electro-magnetic circuit the receiver, transmitter, and impedance coil are in series and connected across the speaking battery terminals marked T. C. and T. Z.

The receiver is constructed with a soft iron core for one pole, and the case as the other, and is only magnetised when the current is allowed to flow through the coil winding. The usual practice, of course, is to have permanent magnets, but if this type of receiver were used with the electro-magnetic circuit the permanent magnets would in time become depolarised unless care were taken to ensure the receiver and battery being so connected as to allow the current to flow in the right direction. To answer an incoming call the hand-combination must be removed from the cradle rest and the red button pressed down; it is automatically held in this position by a sliding bar which is drilled with the necessary number of holes to allow each of the plungers attached to the buttons to pass through, the bar being kept under tension by a spiral spring which forces the bar back and automatically holds the button down and at the same time brings the instrument spring in contact with the line spring and completes the line circuit. The bank of springs which are operated when the hand-combination is removed from the cradle rest automatically complete the speaking circuit.

To call out, the button corresponding to the tablet on which the name of the person required is written should be depressed; this movement temporarily connects the line spring on to the common ringing battery bar, allowing the current to pass along the line and operate the buzzer of the called instrument.

When the pressure is removed from the button the ringing circuit is disconnected and the line and instrument springs are brought into contact, this circuit being completed in the same manner as an incoming call. The button last pressed remains down, but is automatically replaced when another is depressed.

RADIO COMMUNICATION.

The services required include communication between ships and between ships and shore, long-distance telegraphy and telephony, broadcasting, direction-finding and radio beacons, and aeroplane communications.

Radio-Telegraphy.

Transmission systems may be classified as continuous-wave and spark systems. Continuous-wave systems comprise stations employing arc transmitters, high-frequency generators, or thermionic valves. In them a continuous wave of constant amplitude is produced. This wave requires less power and smaller aerials than the older spark systems.

Spark Transmitters.

In spark telegraphy a succession of groups of damped waves is radiated. Though still the most important from the point of view of the number of messages handled, these systems will eventually be superseded because of the disturbance caused to other radio-communications by the number of frequencies simultaneously radiated by spark transmitters. A typical spark transmitter is shown

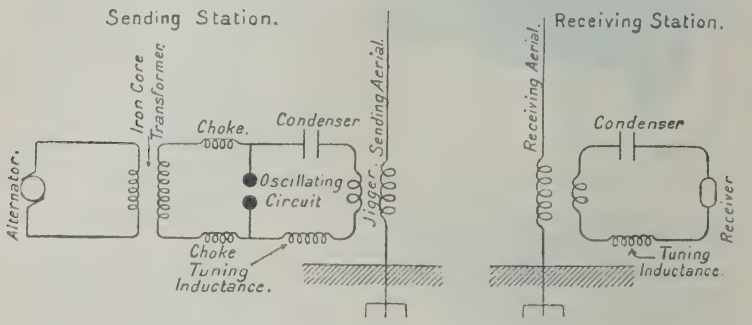


FIG. 63.

in fig. 67. The condenser is charged by the alternator through a step-up transformer. At every peak value of the voltage the spark gap breaks down, and the frequency with which successive sparks follow one another determines the pitch of the note heard in the receiver. This frequency depends on the speed and number of poles of the alternator, and may be from 100 to 1,000 per second. At every discharge of the gap a high-frequency wave-train is radiated by the aerial which is magnetically coupled to the oscillating circuit through an air-core transformer or 'jigger.' The frequency of this oscillating circuit is determined by the values of the condenser capacity and the tuning inductance. The aerial is tuned to this frequency. The corresponding wave-length is given by

$$\lambda \text{ (metres)} = 1885 \sqrt{\text{L (microhenries)} \times \text{C (microfarads)}}.$$

The heating of the spark gap causes difficulty at those higher note frequencies which are preferable for reception. By the use of a rotary spark in which projections carried by the alternator shaft come successfully opposite to a fixed insulated sphere, the discharge takes place only for a short time, and the automatic cooling prevents return of energy from the aerial. In the quenched spark system the discharge takes place between very short narrow gaps formed between thick parallel insulated metallic plates provided with radiating fins. The conducting spark is thus 'quenched' directly the voltage falls below the peak value. Spark systems are a source of great interference to neighbouring wave-lengths.

Reception of spark signals is by means of a telephone receiver connected in a receiving circuit tuned to the wave-length of the transmitter station.

Continuous-wave Transmitters.

Arc Transmitter.

The ' negative-resistance ' characteristic of the electric arc is employed to produce a continuous oscillation, the arc itself being connected in the aerial circuit which is tuned to the desired frequency by the loading inductance L in conjunction with the aerial capacity. In fig. 64 R is the stabilis-

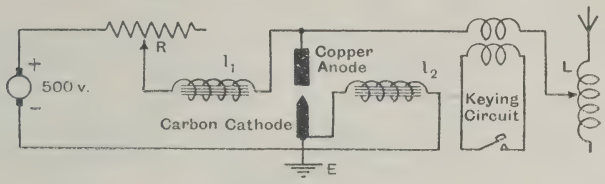


FIG. 64.

ing resistance and l_1 and l_2 are inductances which prevent the high-frequency currents passing through the alternator and also provide a powerful magnetic field across the arc to assist its extinction at each oscillation. The hollow copper anode is cooled by internal water circulation and the carbon cathode is slowly rotated. The arc is enclosed in an atmosphere of hydrogen or coal-gas. For signalling, the arc is not switched off, but ' marking ' and ' spacing ' waves are transmitted by the short-circuiting of a coil coupled to the aerial inductance. The system has the disadvantage that the apparatus is heavy and bulky, and a large number of intense harmonics are associated with the radiated wave.

' *High-frequency Alternators* ' for the direct generation of an oscillating voltage of comparatively low frequency for telegraph stations promise to have relatively little importance in the future, owing to the expense, complication, and restricted frequency range compared with valve transmitter stations.

Thermionic Valves.

In the three-electrode valve a filament heated by the passage of a current forms the cathode, the plate or anode being maintained at a potential positive relatively to the filament. A third electrode, the grid, is placed between the plate and filament. These electrodes are enclosed in a bulb exhausted to a pressure of about 0·00001 mm. of mercury or less. Electrons are emitted by the heated filament, and, owing to the high vacuum in the bulb, they are the only current carriers. The variation of the conductivity of the space between anode and cathode, therefore, is independent of the nature of any residual gas, and thus control of the anode current is effected by variation of the electron stream. This control is the function of the grid, while the amplification factor of a valve is determined by the relative dimensions and spacing of the electrodes.

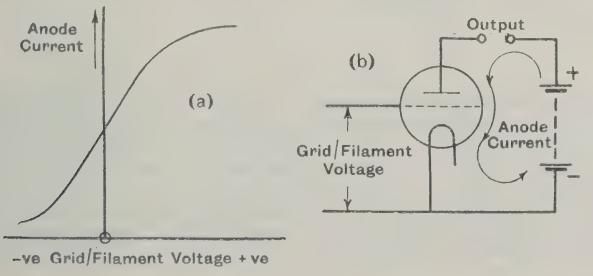


FIG. 65.

A typical valve characteristic curve and a diagram of valve circuits are shown in fig. 65 (*a*) and (*b*). An oscillating potential applied to the grid causes corresponding fluctuations in the anode current, which produce magnified fluctuations in the voltage across an output circuit connected between the anode and high-tension battery. The valve thus acts as an *amplifier*.

The valve may be used also as an *oscillator* by coupling the anode circuit inductively to that of the grid, as shown in fig. 66.

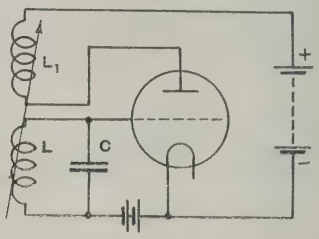


FIG. 66.

An increase in voltage across the tuned circuit LC applied to the grid causes an increase in the anode current in the coil L_1. This increase of current induces an e.m.f. in L, which can be arranged further to increase the grid voltage. At a value of the mutual inductance between L_1 and L greater than a critical value, determined by the valve characteristics and the resistance and inductance of the circuits, a condition of unstable equilibrium results, and the variation of grid voltage extends the whole length of the characteristic corresponding to the working plate voltage. The valve thus becomes a generator of oscillations, the necessary power being obtained from the high-voltage battery in the anode circuit.

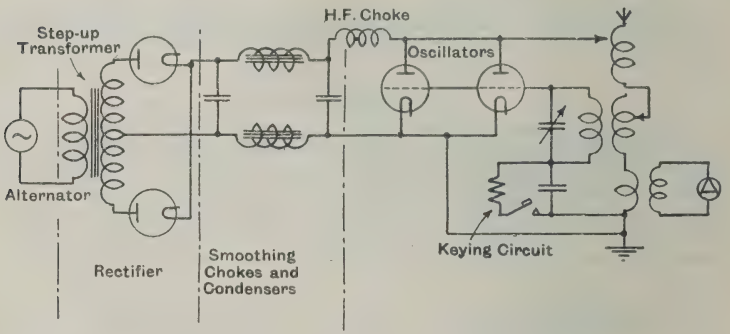


FIG. 67.

A typical *continuous-wave valve transmitter* is illustrated in fig. 67. Keying is effected by disconnecting the grids of the oscillating valves, the resulting negative potential acquired by the grids stopping the oscillation. The size of the valves employed is limited only by the rate of emission of heat from the anode. Silica valves are constructed up to about 12 kw. In larger sizes the containers are partly of metal, which form the anodes, and are water cooled. The input power to the valves may be obtained from special direct-current generators, or from the high-voltage side of step-up transformers through two-electrode rectifying valves. Frequency stabilisation may be effected by tuning-fork control, in which an amplified harmonic of the fork is applied to the grids of a bank of large power valves in parallel forming the output stage of the transmitter. For a description of a large modern valve station 'The Rugby Radio Station of the British Post Office,' by E. Shaughnessy (*Journ. I.E.E.*, vol. 64, pp. 683–713), should be consulted.

Radio Telephony.

Wireless telephony has become commercially possible by the introduction of the three-electrode thermionic valve. Valve oscillators provide a radiation of constant amplitude and frequency to

form a carrier wave upon which the low-frequency speech wave is superimposed. Fig 68 shows how the microphone M modifies the transmitted high-frequency wave. The speech wave of

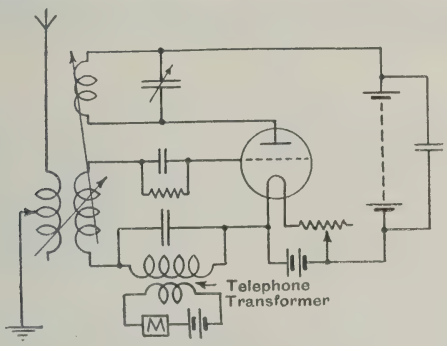


FIG. 68.

frequency f_n, superimposed on the carrier of frequency f, adds the two radio frequencies of $(f + f_n)$ and $(f - f_n)$ to the original wave. As f is increased, the smaller will become the percentage difference between $(f + f_n)$ and $(f - f_n)$. If f_n represents the highest note to be transmitted, the band of frequencies between $(f + f_n)$ and $(f - f_n)$, which are known as the upper and lower side-bands, must be allocated to one particular station to avoid interference with other stations. With high-power low-frequency (long-wave) stations this allocation is becoming difficult. From Rugby, the largest valve station in the world, in addition to world-wide telegraphic transmission on a wave-length of 18,740 metres, a successful trans-oceanic commercial telephone system is operated. Economy in the frequency band occupied and a certain degree of secrecy is obtained by filtering out one of the side-bands and radiating only the other. The necessary carrier frequency is inserted at the receiving station.

Broadcasting.—For the faithful reproduction at the receiving end of broadcast music, the band of frequencies must extend to about 10,000 cycles on either side of the carrier wave, whereas 2,000 cycles on either side suffices for commercial telephony. To avoid interference between the large number of broadcast stations working in Europe, international agreements now govern the allocation of the carrier frequencies of these stations and endeavour to ensure that these frequencies are maintained within very narrow limits. A modern broadcast receiver is shown in fig. 69, in

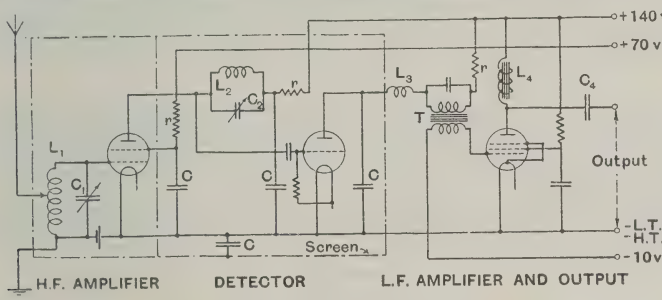


FIG. 69.

which L_1C_1 and L_2C_2 are tuned circuits, L_3 is an H.F. choke, T is an intervalve transformer, L_4C_4 are output filter circuits, and rC are anti-coupling filter circuits. The 4-electrode screened-grid valve amplifies the incoming signals, which are rectified in the second valve. The audio-frequency output from the second valve is amplified by the 5-electrode valve which provides for the operation of the loud-speaker. Screening and resistance-capacity filters are necessary, since

the H.F. valve has an amplification of about 100, and any trace of back coupling would lead to instability. These valves represent the latest advance. With 4- and 5-electrode valves employed in this fashion, no more than three valves are necessary for the reception of distant stations, distance being limited only by atmospherics and interference from other stations. Compared with the most suitable 3-electrode valve, the 5-electrode or pentode valve operates a loud-speaker with only one-tenth of the input A.C. grid voltage and one-half the anode voltage.

Directional Wireless.

The development of short-wave transmitters has recently culminated in their commercial application in the beam systems for trans-oceanic communication. In one system the waves are collected and concentrated in the required direction by an arrangement of reflectors. In another system the radiations from a number of transmitters interfere and nullify one another, except in the one wanted direction in which they reinforce one another. An aerial system designed for directional transmission is effective for receiving in the same direction. At the Bridgwater station in Somerset, aerials for receiving from Canada and from South Africa respectively are each supported from five masts, and the reflectors, consisting of vertical screens parallel to the line of oscillators, are tuned to the oscillator wave-length. Energy radiated in the direction of the reflectors is absorbed and retransmitted in the proper direction. Short-wave stations for long-distance communication are far less costly than long-wave stations, while directional transmission is less affected by atmospheric disturbances because of the short wave-lengths employed. Other important applications of the wireless beam are to surveying and direction-finding, which is of great assistance to navigation when approaching the coast during foggy weather. By the installation of directive receivers in land stations which are linked up by ordinary telephone, it is now possible to communicate its bearings to aircraft inside a radius of some 300 miles within about a minute of the pilot's inquiry.

Standard Specifications for Radio Work.

The British Engineering Standards Association have issued the following specifications:

No. 234—1925. Ebonite for Panels for Radio Reception Purposes.

No. 271—1926. Fixed Condensers for Radio Reception Purposes.

No. 289—1927. Plug-in Coils for Radio Reception Purposes.

The I.E.E. Rules covering radio apparatus connected to supply mains are dealt with on p. 2239.

INSULATING MATERIALS.

British Engineering Standards Specifications No. 231—1925, for Pressboard, and No. 216—1926, for Vulcanised Fibre for Electrical Purposes, refer to the composition, dimensions, tensile, shearing, tearing, and electrical strengths, freedom from conducting particles, and other physical properties. The methods of determination are specified. Specification No. 234—1925, for Ebonite for Panels for Radio Reception Purposes, in addition to the above information, gives standard values and tests relating to sulphur, acetone, and ash contents, the breaking strength, surface leakage, dielectric loss, and electric strength. Specification No. 157—1923, refers to Moulded Flat Top Insulating Bushes, and No. 316—1929 to Synthetic-resin Varnish-Paper Boards and Tubes.

DEXONITE.

This material (*Dexine Ltd., Stratford, London, E.*) possesses the properties of vulcanite and ebonite, but unlike the two latter it retains its normal degree of hardness under high temperature. It is easy to tool, screw, and tap, is not brittle, and is capable of being moulded into any shape or size. It has a high electrical resistance, and is non-corrosive and acid resisting. *Specific Gravity* at 60° F., 1·2765. *Tensile Strength:* Stress per sq. in. at maximum load 6,765 lbs., equivalent to 3·02 tons.

Articles made from Dexonite include insulating parts for electrical apparatus, medical requisites, binocular parts, etc. It is also supplied in sheet, rod, or tube form to any convenient sizes.

MACONITE.

See p. 2254.

WILLESDEN PAPER.

(*The Willesden Paper and Canvas Works, Ltd.*)

This material is largely used for electrical insulation, chiefly on the field magnets and armatures of continuous current dynamos, and is found by manufacturers to be of great value, as it is less

hygroscopic than other materials formerly employed. It can also be used with advantage instead of vulcanised fibre in many places such as small switches, washers on fuse and switch boards, and for telephone work, etc.

Resinous Insulating Materials.

(The Ioco Rubber and Waterproofing Company, Ltd.)

' Linapex Varnished Fabrics.'

Linapex Varnished Cloths are manufactured from carefully selected cambrics which are coated with a hardened film of a special oil varnish. They form one of the most extensively used insulating materials for the taping of field and transformer coils, etc., and for repair work generally. They are manufactured in thicknesses of 4–12 mils and supplied in the full width of 36 ins. or in tape form of any desired width, cut either on the straight or on the bias. A dielectric strength of 1,000 volts per mil is guaranteed.

Linapex Varnished Silk is a similar type of material, except that silk is employed as a base fabric instead of cotton. Such Varnished Silks have a higher dielectric strength than Varnished Cambrics, namely, about 1,400 volts per mil. These are also supplied in tape form, and are largely used for instrument work and where space is limited. All Linapex Varnished Cloths and Silks are oil and moisture resisting.

' Formapex Varnishes.'

These are solutions of the well-known brand of Phenol-Formaldehyde Synthetic Resin (Bakelite) employed mainly for impregnating paper, cloth, asbestos, etc., for the manufacture of Formapex Miocarta Tubes and Sheets, and also for impregnating coils, etc., affording a protective water-resisting coat of high dielectric properties. To apply, the coil is first dipped in the varnish, allowed to drain and air-dry, and is finally stoved in a moderate temperature.

' Formapex Boards and Tubes (Bakelite Type).'

These are manufactured from paper or cloth which has been treated with Formapex Phenol-Formaldehyde Synthetic Resin. In the case of sheets, the material is cut into plies, which are built up and placed in a heated hydraulic press and subjected to high pressure and temperature. For tubes, the treated paper is rolled on to a steel mandril between the heated rollers of a tube winding machine, and afterwards baked and varnished with a water- and oil-resisting varnish. Formapex Paper Boards and Tubes are characterised by their high dielectric strength and low water absorption value. Unlike Ebonite they do not soften with heat. They are readily machined and are used for insulators in all high-tension work. Formapex Condenser Bushings are used for transformers and switchgear. Formapex Cloth Boards are, on account of their remarkable mechanical strength and resistance to water, steam, oil, acids, etc., largely employed for mechanical purposes, such as silent gears, railway track insulators, etc.

Insulating Oils for Electrical Purposes.

British Standard Specification for Insulating Oils for Electrical Purposes.*

(No. 148—1927.) *(Abstract.)*

Classification and Temperature Limits.

Two classes of oil, Class A and Class B, are referred to in this Specification.

2. Class A oils, sometimes referred to as non-sludging oils, are suitable for use in transformers in which the working temperature of the oil, measured by a thermometer, exceeds 80° C. (176° F.).

They are also suitable for use in oil switches in which the temperature exceeds 70° C. (158° F.).

They may be used with advantage in all oil switches of above 500 amperes capacity having their main contacts under oil and operated for long periods near their rated current or in hot situations.

Class B oils are suitable for use in transformers in which the working temperature of the oil does not usually exceed 75° C. (167° F.).

They are also suitable for general use in oil switches not subjected to conditions requiring the use of Class A oils.

3. The oil of each class shall be a pure hydrocarbon (mineral) oil, clean and sufficiently free from moisture or other matter likely to injure its insulating properties, and shall satisfy the requirements of the specification.

* By permission of the British Engineering Standards Association.

4. The characteristics shall be within the limits shown in the following Schedule :—

Class of Oil.	Sludge (maximum).	Viscosity at 15·5° C. (60° F.) (maximum).	Electric Strength (minimum).	Flash Point (minimum).	Evaporation (maximum).	Cold Test (maximum).
	Per cent.	Redwood Secs.	Volts.		Per cent.	
A.0 A.10 A.30	0·1	200	30,000	145° C. (293° F.)	1·6	0° C. (32° F.) — 10° C. (14° F.) — 30° C. (— 22° F.)
B.0 B.10 B.30	0·8	200	30,000	145° C. (293° F.)	1·6	0° C. (32° F.) — 10° C. (14° F.) — 30° C. (— 22° F.)

6. The oil shall not contain any inorganic acid. The acid value per gramme of oil due to organic acids shall not exceed 0·2 mg. KOH.

7. The saponification value per gramme of oil shall not exceed 4 mg. KOH.

8. The oil shall not cause discoloration of copper.

B.E.S.A. Specification No. 358—1929 gives rules for measurement of voltage with sphere gaps.

Test for Moisture in Transformer Oil.

While a thorough test for the suitability of oil for transformers and oil switches can be made only with high-tension testing apparatus, very good indications of the presence of moisture have been obtained by Mr. J. K. Mackie, superintendent of the Connecticut Power Co., as follows :— A sample of the oil to be tested is drawn from the bottom of the transformer, oil switch, or storage tank. (Samples from the upper parts of the tank are not considered suitable for the test, as water is heavier than oil and usually collects at the bottom.) In the sample thus taken is placed powdered anhydrous copper sulphate. If moisture is present the copper sulphate will be dissolved, producing a blue colour that will diffuse through the oil if moisture is in suspension. Since enough copper sulphate is added to ensure a saturated solution, the intensity of the blue tint will be a measure of the amount of moisture present. As small percentages of moisture have a very deleterious effect on the dielectric strength of oil, however, the lightest shade of blue is sufficient indication that the oil should be dried by filtering. It may be pointed out that other substances giving a deeper tint than copper sulphate may be used, the only requirement being that they dissolve quickly in water but not in oil.

'No-Mag' Non-Magnetic Cast Iron.

This is a patented non-magnetic cast iron, which combines the non-magnetic property with a high electric resistance, thereby giving great freedom from hysteresis and eddy currents.

The efficiency of 'No-Mag' non-magnetic cast iron is shown as follows :—

Metal.	Magnetic Permeability.	Specific Resistance.
	Maximum.	Micro-ohms per cm. cube.
Ordinary cast iron . . .	330·00	95·0
'No-Mag'	1·03	140·0
Brass	1·00	7·5

The 'No-Mag' can be applied for the purposes of oil switch covers, bus bar clamps, brackets, cable boxes, transformer covers, separating pieces, sealing bells, etc. ; by using 'No-Mag' for end plates and shields for generators and motors the losses due to stray flux are reduced to a minimum, thus allowing economies in design.

Anchorage for Overhead Line Structures.

In the *Electrical World* Mr. A. W. Malone describes a cheap and effective method of making anchorages for overhead telegraphic or electric light line structures. A small hole is bored or drilled into the ground at a suitable angle, and an explosive is ignited at the bottom of it; the explosion makes a cavity, which eventually contains a ball of concrete, as shown in

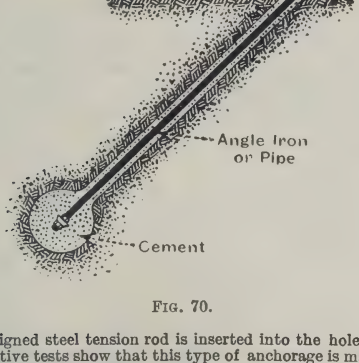


FIG. 70.

fig. 70. A specially designed steel tension rod is inserted into the hole, which is then filled up with concrete. Exhaustive tests show that this type of anchorage is many times as resistant to uprooting as the ordinary type, whilst it is far cheaper in labour, time, and material. A Malone anchor having a concrete 'bulb' 26 ins. in diameter, 9 ft. below the ground surface, offered a resistance of 56,000 lbs. after yielding 2 ins., whereas a grillage type of footing had a resistance of only 15,000 lbs.

This type of foundation has been adopted by the Central Electricity Board for some of the grid towers. The tower leg-stubs are fitted into position by means of a template and the cavities are filled in with concrete. The operation is simple, economical, and accurate.

X-Rays in Industry.

(C. Norman Kemp, B.Sc., F.R.S. (Edin.), A.I.C.)

The term 'opacity' has reference to a characteristic of certain substances in their reaction to ordinary light (λ about 8,000-3,900 Ångström units).

No substance known is totally opaque to X-rays, owing to their extremely small wave-length, although, of course, a sufficient thickness of a dense material may, in a given case, effect complete absorption, this being proportional to atomic weight and thickness.

From Planck's equation there may be derived the simple and useful relation that the product of the *maximum* tube voltage and the *minimum* resulting X-ray wave-length is a constant, the wave-lengths being expressed in Ångström units (Å.U.),
i.e.

$$\lambda_{min.} \cdot V_{max.} = 12{,}400,$$

whence

$\lambda_{min.}$ at 12,400 volts = 1.0 Å.U.; $\lambda_{min.}$ at 124,000 volts = 0·1 Å.U.

Within the range of wave-lengths from 1 to 0·1 Å.U. a vast amount of modern radiological work is done, commonly employed tube voltages being 45, 60, and 100 kv.

For work on metals, however, the last figure is often largely exceeded, values up to 500 kv. being used, and 1000 kv. having been proposed, if not actually attempted, with the object of reducing exposure times, and increasing the thicknesses which may be examined.

Industrial applications are based on the employment of X-radiations having a penetrating power (a function of λ) and an intensity related to the atomic weights (or numbers) of the constituent elements of the material under examination, so that, as indicated, very high tube voltages are necessary for the radiography of iron and steel products.

At present, definite information as to the homogeneity of such materials may readily be secured with commercially available equipment with an exposure of 30 minutes, this permitting of the detection of flaws in steel down to a sixteenth of an inch in size and up to three inches in thickness. Some of the practical uses of X-rays may be conveniently grouped as follows:—

1. Examination of natural objects or substances for information as to internal structure.

2. Examination of substances or objects from stage to stage in manufacturing processes with a view to more effective control, tending towards the elimination of waste, with consequent cheapening of production.

3. Examination of manufactured products or assembled articles for homogeneity or freedom from foreign matter in the one case, and correctness of assembly in the other.

4. General research in the foregoing and other directions whereby new applications and new methods of attacking existing problems are sought.

One of the most recent applications of radiological technique is the study of the internal structure, or, as it has been called, the ' mineral skeleton ' of coals, anthracites, cokes, etc., which may be visually studied, and also photographically recorded if required.

Particularly valuable in this connection is the use of stereoscopic radiography, whereby the interior of the specimens may be viewed in natural relief, and the ash-forming constituents examined as regards nature, quantity, and distribution.

The outstanding feature of X-ray methods lies in the provision of an easy, rapid, and non-destructive test of opaque materials, a field capable of almost unlimited extension as a result of continued research.

Electrical performance of transformers for X-ray purposes are dealt with in B.E.S.A. Specification No. 326—1928.

APPLICATIONS OF ELECTRICITY.

References to application will be found in the following sections :—

Subject.	*Section.*
Canal traction	XXIX
Electric traction	XIV
Fuels	XXXIV
Furnaces	XL
Gas and oil engines	XXXI
Lifts	XVIII
Lifting magnets	XVIII
Lighting	XXXVIII
Magnetic clutches	XIX
,, chucks	XXI
,, separators	XVIII, XXXIX
Mining	XXXIX
Pyrometers	XL
Railways	XIII
Ship propulsion	XXXVI
Signalling, railway	XIII
,, mining	XXXIX
Steam plant	XXX
Stoneware	X
Turbines, steam	XXX
,, water	XXV
Welding	XL
Winding engines	XVIII

ADDENDA.

Cables for High Voltage. (See p. 2209.)

'H' and 'SL' Cables.—For medium and high voltages the use of the so-called ' H ' (Hochstädter) and ' SL ' (Self-Leaded) type cables has become general in the last few years. In the H-type each core has a thin metal-foil sheathing, while the SL construction provides a separate thin lead sheath for each core. This arrangement ensures a radial electric field (economising insulating materials) and obviates any discharge when the cores are inadvertently separated. The sheaths acquire earth potential by contact, so that the fillers carry no electric stress : they can therefore be made of cheaper material.

SO-Type Cables.—Cables of the ' SO ' type have been designed to improve the mechanical and thermal properties of high-voltage cables, the internal form being roughly triangular corresponding with the three conductors. The insulation of the conductors remains unchanged, but the filler area is much reduced, resulting in a shortening of the periphery and a reduction in the lead sheath and armouring.

Oil-Filled Cables.—Three-conductor cables, with paper insulation impregnated with highly viscous oil-resin compound, are satisfactory up to about 22 kV., but have given considerable trouble when used for higher voltages. This is due to the combined action of two processes : (1) fluid dielectrics experience mechanical forces tending to move them away from regions of high electric field density towards the sheathing ; (2) the viscosity of impregnating compounds decreases with rise in temperature.

These difficulties have been overcome in the oil-filled cable. Single-conductor cables are generally used, although SL-type 3-conductor designs are possible. The conductors are wound with graded paper, the impregnating medium being a low-viscosity mineral oil. The system is arranged so that the expansion of the oil due to temperature rise is accommodated by means of external reservoirs which return the oil to the cable under a small hydrostatic pressure when the temperature falls.

Oil-filled cables at 132 kV. are in operation between Kidbrooke and Deptford, Wimbledon and Battersea, in London, for connection to the national grid network.

Transformers. (See p. 2301.)

The out-door transformers on the 132 kV. system of the National 'Grid' System have standard capacities of 7,500, 10,000, 15,000, 20,000, 30,000, 45,000, 60,000 and 75,000 kVA. Those of 30,000 kVA. rating or less are of the 3-phase type, larger capacities consist of single-phase units. Natural cooling is used on the smaller sizes: the larger transformers have forced air-blast cooling when working above half full-load. The 132 kV. windings are star-connected with neutral point directly and permanently earthed: lower voltage windings are delta-connected. The voltage regulating equipment can be operated under load and provides for ratio variation of $\pm$ 10 per cent. in steps of not more than 1·43 per cent. Automatic interlocking ensures that all transformers in a sub-station must have the same transformation ratio before being paralleled.

All insulating materials are required to withstand oil temperatures of 95° C. continuously and 110° C. for short periods. The insulation is graded to withstand between turns for 3 seconds full-line voltage on 1 per cent. of the end turns, at each end of the phase windings, $\frac{1}{3}$ line voltage on 5 per cent. of the end turns, and $\frac{1}{4}$ line voltage on the remainder of the main winding. Tapped coils and 3 per cent. of the winding on each side can withstand $\frac{1}{3}$ line voltage between turns.

The terminal bushings must withstand when dry a test pressure of 300 kV. for 1 minute when mounted as in service, and no corona must appear below 1·4 times normal voltage. All bushings are mechanically interchangeable. Conservator tanks with calcium chloride breathers are fitted to exclude moisture from the oil. In addition to an alarm device which operates when a winding temperature of 100° C. (or other value) is reached, the circuit breaker is automatically tripped at a temperature of 10° C. above that at which the alarm is given. All ratings are in accordance with B.S.S. No. 171—1927, and oil is defined in B.S.S. No. 148—1927.

Electric Clocks and Time Service. (See p. 2327.)

In connection with frequency indication and control in central stations, the Warren self-starting hysteresis synchronous motor has been combined with a standard clock. The latter has one set of hands driven by the clock and another set by gearing from the motor. Any error is indicated by a differential mechanism. The standard clock keeps time to 1 second a day, and a difference of 0·005 per cent. in the mean frequency can be detected. A standard time service can, therefore, be given from central stations, the frequency of which is standardised, to synchronously-driven repeater clocks connected to the supply mains.

Other types of synchronously-driven clocks have also been brought into service.

Porcelain Insulators for Overhead Lines. (See p. 2225.)

For the national 132 kV. transmission scheme, insulators of the cap and pin type with ball and socket fittings have been selected. Each suspension string consists of 9 units and for tension insulators 10 units. The factor of safety is not less than $2\frac{1}{2}$ when the insulator is supporting its maximum working load (4,000 lbs. on suspension and 8,000 lbs. on tension insulators) with full normal voltage applied to it. The minimum flash-over voltage of a unit is 85 kV. dry and 60 kV. wet. The minimum puncture voltage is 145 kV. per unit. The complete strings with all fittings have minimum dry flash-over voltages of 395 kV. and 450 kV. for suspension and tension types respectively, and 385 kV. and 440 kV. wet flash-over voltages. Since the normal voltage to earth is 76 kV., this gives a nominal factor of safety of 5. Each chain of insulators has a guard ring at the conductor end and arcing horns at the earthed end with the object of equalising the voltage distribution over the units and preventing damage to clamps, conductors, and insulator when a flash-over occurs. Two insulator strings in parallel are used at tension points.

Electric Measuring Instruments. (See p. 2310.)

The most notable advance in recent years has been the introduction of nickel-iron alloy having a high permeability at low flux densities and very small hysteresis loss. Used in the cores of instrument transformers, current transformers have been constructed with a secondary burden of 12·5 VA., a ratio, constant to 1 part in 10,000 between 0·1 and 1·2 times the rated current, and a phase displacement of 0·75 minute. The very exacting limitations imposed by the Central Electricity Board for bar-type current transformers which include a maximum error of 1 per cent. between 5 and 125 per cent. of rated current with a change of ratio of only 0·3 per cent. have been met. Applied to soft-iron or moving-iron instruments it has permitted first-grade accuracy on a.c. or d.c. with angular scales of about 120° carried down to one-tenth of the maximum reading and almost uniformly divided down to about one-fifth. The instruments have very large overload capacities and are accurate on all frequencies from 25 to 500 cycles per second.

A recent introduction is the 'A.C. Test,' in which the movement of the iron takes place in the gap of a projecting laminated ring through which the current-carrying cable passes. No terminals are required, and by winding the cable the appropriate number of times through the ring, any desired range below that for which the instrument is scaled may be obtained. Ranges from 100 to 300 A. for a single conductor are available. A coil attachment enables the same instrument to be used in conjunction with current transformers, or as a voltmeter taking 75 mA. at 100 volts. The 90° scale reads to about one-tenth of the maximum, and first-grade accuracy is given between 25 and 50 cycles.

Moving-coil permanent magnet instruments are now adapted to the measurement of alternating current and voltage by the incorporation of thermo-junctions and copper-oxide rectifiers. The employment of vacuum thermo-couples has resulted in instruments being available for all ranges from 25 mA. to 10 A. and above, and for radio frequencies up to 108 cycles per second. For commercial sensitive a.c. instruments up to 10,000 cycles, the copper-oxide type has the advantages of a nearly linear law, a long range and low power consumption. On most ordinary wave-forms a 2 per cent. accuracy on frequencies from 20 to 6,000 cycles is claimed. In these instruments the rectifier, consisting of plates of copper with oxidised surfaces in contact with a soft metal such as lead made up into units, is contained in the instrument case. Recording instruments of this type are also available.

Photo-Electric Cells and their Uses.

Photo-electric cells include the following principal types: (*a*) *Photo-Conductive*, in which a change of ohmic resistance occurs on exposure to light. Selenium cells belong to this class. They are employed where the time lag is not a disadvantage. (*b*) *Photo-Voltaic*, in which certain metals, immersed in an electrolyte, develop an e.m.f. when illuminated. (*c*) *Photo-Electronic*, based on the emission of electrons from the surface of alkali electro-positive metals, such as sodium, potassium, rubidium, and caesium when exposed to light. The following notes refer to this type, which became of practical importance when the thermionic valve amplifier was developed, by permitting the small currents available from p.e. cells to be amplified to any desired magnitude. In its simplest form, the p.e. cell comprises a glass or quartz bulb, with an inner deposit of some photo-sensitive metal and a separate anode of wire. The wave-length or colour sensitivity of the cell depends upon the cathode metal and its base, *e.g.* potassium deposited on a base of the same metal has a maximum sensitivity in the violet and ultra-violet region, whereas potassium on a copper base has a maximum sensitivity in the orange-red range. There are many available coatings, including caesium, rubidium, barium, and various organic dyes.

Both vacuum and gas-filled cells are manufactured. The former has the higher output up to saturation, after which any increase in the cathode-anode voltage has no effect on the output. Gas-filled cells having a trace of argon are more sensitive on account of the increased emission due to the ionised gas. Their output is limited by glow discharge, consequently a protective resistance of some 10,000 ohms must be used in series with them.

SECTION XXXVI

NAVAL ARCHITECTURE AND MARINE ENGINEERING

(Contributed and Revised by Professor Percy A. Hillhouse, D.Sc., M.I.N.A., and James Thomson, M.I.N.A.)

SECTION XXXVI

NAVAL ARCHITECTURE AND MARINE ENGINEERING. DIMENSIONS AND GEOMETRY OF SHIPS—DISPLACEMENT, STABILITY, AND TRIM—WEIGHTS OF HULL AND MACHINERY—DEADWEIGHT—RESISTANCE OF SHIPS—ADMIRALTY COEFFICIENTS — ECONOMICAL SPEEDS — COAL CONSUMPTION—RUDDERS—ESTIMATING HORSE-POWER—DESIGNING MARINE ENGINES AND BOILERS—TURBINES—REDUCTION GEAR—TURBINE MACHINERY—CONDENSING PLANTS—DIESEL ENGINES—PROPELLERS.

(Contributed and Revised by Professor Percy A. Hillhouse, D.Sc., M.I.N.A., and James Thomson, M.I.N.A.)

DIMENSIONS OF SHIPS.

Moulded Dimensions.

LENGTH L = 'Length between Perpendiculars' = Length on load water-line measured from the fore side of the stem ('F.P.,' or 'Fore Perpendicular') to the aft side of the stern post ('A.P.,' or 'After Perpendicular'), or fore side of rudder stock if there is no stern post.

BREADTH B = 'Moulded Breadth' = Breadth measured over the widest part of the frame at the middle of the length L.

DEPTH D = 'Moulded Depth' = Depth measured from top of keel to top of deck beam at side at the middle of the length L. To avoid error the name of the deck to which the depth is measured should be stated.

Registered Dimensions.

LENGTH = Length from fore part of stem head to aft side of stern post, or fore side of rudder stock if there is no stern post.

BREADTH = Breadth measured over the outside of the shell plating at the widest part.

DEPTH = 'Depth of Hold' = Depth measured from top of ceiling to top of deck beam at the centre line of the vessel at the middle of the length. This depth is measured to the 'tonnage deck,' which is the 'second deck from below.'

Overall Dimensions.

LENGTH = 'Length Overall' = Length measured from the foremost point of the stem or figure-head (but excluding bowsprit if any) to the aft side of the taffrail, or cruiser stern.

BREADTH = 'Breadth Extreme' measured over the outside of the shell plating or belting or paddle sponsons, if any.

DEPTH = Depth from top of keel to top of deck amidships.

Drafts of a Ship.

The DRAFT d of a ship is the distance of the lowest point of the keel (or of the line of the keel produced) below the water line.

The MEAN DRAFT is half the sum of the drafts at the forward and after perpendiculars or at the stem and the stern post.

The MOULDED DRAFT δ is the distance of the top of the keel (or of the line of the top of keel produced) below the water line.

The LIGHT DRAFTS of a vessel are the drafts forward and aft when the vessel is floating complete, with water in boilers, but without crew, fuel, cargo, stores, water, or other loads.

The LOAD DRAFT of a vessel is the mean draft of the vessel when floating complete and ready for sea and with all fuel, cargo, passengers, crew, baggage, stores, spares, water, and other loads. The maximum load draft is determined by the freeboard tables of the Board of Trade; for passenger vessels the actual draft is limited to that determined by the spacing of the water-tight bulkheads, in accordance with the recommendations of the Convention for the Safety of Life at Sea. This draft is called 'The Bulkhead Draft.'

The FREEBOARD of a vessel is the height of the top of the wood deck at side at the middle of the length above the water line. The minimum freeboard allowable for any vessel entering or leaving a British port is fixed by law. It is assigned for each vessel by the Board of Trade or Classification Society, and must be marked upon each side of the vessel amidships by the 'Freeboard Disc' or 'Plimsoll Mark.'

The 'TRIM' or 'LEAD' of a vessel is the difference between the drafts forward and aft. It is said to be 'by the head' or 'by the stern' according as the draft forward or the draft aft is the greater. When the drafts forward and aft are equal, the vessel is said to be 'on even keel.' 'Change of Trim' is the arithmetical sum of the changes of draft forward and aft produced by the shifting of any weight.

Displacement.

The DISPLACEMENT of a vessel is the amount of water displaced by the underwater portions of her hull. It may be stated either by volume or by weight. The weight of the water displaced by a floating vessel is equal to the total weight of the vessel and her contents. If the vessel be floating in salt water, her displacement in tons, Δ, is equal to $\frac{1}{35}$ of her volume of displacement, ∇, in cubic feet. If in fresh water, $\Delta = \frac{\nabla}{35{\cdot}8}$.

The TONS PER INCH of a vessel at any draft is the number of tons weight which must be placed on board in order to increase the mean draft one inch.

$$\text{Tons per inch, in salt water} = \frac{\text{area of water plane}}{420} = \frac{A}{420},$$

$$\text{Tons per inch, in fresh water} = \frac{A}{430},$$

where A is the area of the water plane in square feet.

Coefficients of Displacement.

The BLOCK COEFFICIENT, or *Coefficient of Fineness*, C_b, is the ratio of the immersed volume of displacement to the volume of the circumscribing parallelepipedon.

$$\text{Block coefficient} = C_b = \frac{\nabla}{L\,B\,\delta} = \frac{35\,\Delta}{L\,B\,\delta} \text{ in salt water} = \frac{35{\cdot}8\,\Delta}{L\,B\,\delta} \text{ in fresh water},$$

where ∇ = Immersed volume of displacement in cubic feet,
Δ = Displacement in tons,
L = Length on load water-line, in feet,
B = Moulded breadth in feet,
δ = Moulded draft in feet.

Builders of cargo vessels usually estimate block coefficient with reference to length B.P., breadth over shell plating, and draft including flat plate keel.

The PRISMATIC COEFFICIENT, C_p, is the ratio of the immersed volume of displacement to the volume of the circumscribing cylinder having the same form of cross-section as the vessel below the water-line amidships.

$$\text{Prismatic coefficient} = C_p = \frac{\nabla}{La} = \frac{35\,\Delta \text{ or } 35{\cdot}8\,\Delta}{La},$$

where a is the immersed area of the midship section to outside of frame in square feet.

The MIDSHIP SECTION COEFFICIENT, C_m, is the ratio of the area of the midship section up to the water-line to the area of the circumscribing rectangle.

$$\left.\begin{matrix}\text{Midship section coefficient} \\ \text{or midship area coefficient}\end{matrix}\right\} = C_m = \frac{a}{B\,\delta}.$$

From the above values for the three coefficients it is easily seen that

$$C_o \times C_m = C_p.$$

Mr. W. H. Riddlesworth, M.Sc., gives the following approximate formulas connecting draft, displacement, and block coefficient:—

Let α = water plane area coefficient $= \frac{A}{LB}$.

β = block coefficient.
δ = moulded draft.
Δ = displacement in tons.

Then, approximately,

(1) The water plane area coefficient is equal to one-third plus two-thirds of the block coefficient,

i.e. $$\alpha = \tfrac{1}{3} + \tfrac{2}{3}\beta$$

(2) The change (c) in block coefficient per foot change in draft is equal to the complement of the block coefficient divided by three times the draft,

i.e. $$c = \frac{1-\beta}{3\delta}.$$

(3) The complement of the block coefficient varies inversely as the cube root of the draft,

i.e. $$1-\beta = \text{constant} \div \sqrt[3]{\delta},$$

the constant being determined for each ship from any one known draft δ_o and the corresponding block coefficient β_o; constant $= \sqrt[3]{\delta_o}(1-\beta_o)$.

The above three relations have been found to be both useful and reliable in practical work.

Curves of Displacement, etc.

As the draft of a vessel changes, the displacement, area of water plane, coefficients, etc., also vary, and it is customary to record these changes by means of curves whose vertical abscissæ represent draft, while the horizontal ordinates represent areas, displacements, coefficients, etc.

The ordinate of the curve of displacement at any draft is proportional to the area of the curve of water plane areas up to that draft.

Centre of Buoyancy.

The Centre of Buoyancy, B, at any draft, is the centre of gravity of the volume of displacement at that draft. The vertical position of the centre of buoyancy is the same as that of the centre of gravity of the curve of water plane areas, and its longitudinal position is the same as that of the centre of gravity of the prismatic curve. (See next page.)

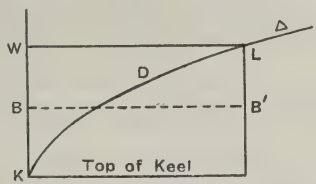


Fig. 1.

To find the height, K B, of the centre of buoyancy above the top of keel, draw the line BB′, across the displacement curve so that the area of the rectangle WB′ is equal to the area W L D K above the displacement curve.

Morrish's Rule.

Very approximately $KB = \frac{5}{6}\delta - \frac{V}{3A}$;

where K B = height of centre of buoyancy above top of keel, in feet;
δ = moulded draft, in feet;
V = displacement, in cubic feet;
A = area of the water plane, in square feet.

$\frac{V}{A}$ is called 'Rankine's Mean Draft.'

KB is approximately 0·55 of the draft.

Wetted Surface.

The WETTED SURFACE or IMMERSED SURFACE of a vessel is the total area of the surface of the shell plating, keel, bilge keels and rudder in contact with the water in which the vessel is afloat.

Approximately,

$$\left.\begin{aligned} S &= L(1{\cdot}7\,\delta + C_b B) && \text{(Mumford)} \\ S &= 3{\cdot}4\,\nabla^{\frac{2}{3}} + 0{\cdot}5\,L\nabla^{\frac{1}{3}} && \text{(Froude)} \\ S &= 15{\cdot}4\sqrt{L\nabla} && \text{(Taylor)} \end{aligned}\right\} + \text{surfaces of keel, bilge keels, and rudder.}$$

where L = length between perpendiculars, in feet;
δ = moulded draft, in feet;
C_b = block coefficient;
B = moulded breadth, in feet;
Δ = displacement, in tons; ∇ = displacement in cubic feet;
S = wetted surface, in square feet.

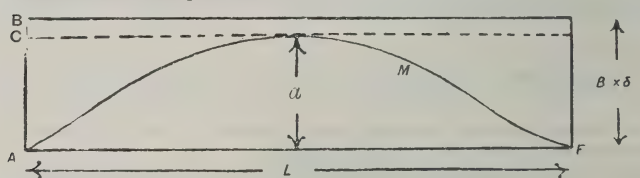


FIG. 2.

The PRISMATIC CURVE or CURVE OF AREAS for any draft is a curve whose horizontal base represents to scale the length of the vessel on load water-line, and whose ordinate at any point is proportional to the immersed area of the corresponding transverse section of the vessel. This curve shows the longitudinal distribution of the displacement, which has an important bearing upon the resistance.

Let A M F (fig. 2) be a prismatic curve, inscribed within a rectangle C F of height representing the area of the midship section, and surrounded by another rectangle B F of height representing the area of the rectangle circumscribing the midship section. Then

Area of prismatic curve is proportional to displacement.

$$\frac{\text{Area of prismatic curve}}{\text{Area of rectangle C F}} = \frac{\nabla}{L \times a} = \text{Prismatic coefficient.}$$

$$\frac{\text{Area of prismatic curve}}{\text{Area of rectangle B F}} = \frac{\nabla}{L \times B \times \delta} = \text{Block coefficient.}$$

$$\frac{\text{Height of rectangle C F}}{\text{Height of rectangle B F}} = \frac{a}{B \times \delta} = \text{Midship section coefficient.}$$

Centre of Flotation.

The CENTRE OF FLOTATION is the centre of gravity of the water plane. The two waterlines defining any small change of trim or heel intersect at the centre of flotation.

Transverse Metacentre.

If a vessel be slightly inclined to one side the centre of buoyancy, B, will move slightly from the middle line towards that side, say to B′, and the vertical line through B′ representing the upward pressure of the water will cut the inclined middle line at the point M called the 'Transverse metacentre.'

$$BM = \frac{I}{\nabla}.$$

where,

BM is the height of the transverse metacentre above the centre of buoyancy,
I is the transverse moment of inertia of the water-plane about its middle line,
and ∇ is the volume of displacement.

BM is approximately equal to ·075 to ·085 × breadth^2 ÷ draft, at load draft.

The METACENTRIC HEIGHT, GM or m, in any condition of loading is the vertical distance between the transverse metacentre, M, and the centre of gravity, G, of the whole vessel and her contents. If, in the upright condition, G lies above M the vessel is unstable and will not remain upright; if G lies below M, the vessel is stable and will return to the upright position if slightly inclined and set free. GM is a measure of the vessel's 'Initial Stability.'

The value of GM can be obtained either by separate calculation of the heights of G and M or by an inclining experiment.

Inclining Experiment.

In carrying out an 'Inclining Experiment' on a vessel, to determine the height, KG, of the centre of gravity, the following procedure should be observed :—

Have the vessel free from wind, tide, current, transverse mooring ropes, gangways, and particularly from loose water or other weights liable to move when the vessel is inclined. Have no men on board except those engaged on the experiment. Read the drafts carefully at each end from a small boat. Move any known weight w (enough to produce an inclination of about 2°) transversely across the deck through a total distance, a, and observe the corresponding deflection d, of a simple pendulum of length l, suspended anywhere on board. Several shifts of the weights are usually made to each side of the ship and the mean value of d taken. From the draft readings and the curves of displacement and transverse metacentres we can obtain the value of Δ, and of KM.

Then GM as inclined $= \dfrac{w\,a\,l}{\Delta\,d}$, w and Δ being in tons, GM and a in feet, and l and d in inches,

and KG as inclined = KM − GM.

These results apply only to the vessel in the inclining condition. To obtain the metacentric height for any other condition it is necessary to make suitable corrections for the different amounts and positions of the various items of loading and for the new values of Δ and KM.

It is usual to estimate the value of GM for the following conditions :—

Light Condition.—Ship and machinery complete, boilers and condensers full, but no fuel, cargo, passengers, crew, baggage, stores, spares, fresh or salt water, or reserve feed-water on board.

Docking Condition.

Load Condition.—Fully loaded condition; ship and machinery complete, boilers and condensers full, and with all fuel, passengers, crew, baggage, stores, spares, fresh and salt water, reserve feed-water, and holds filled with homogeneous cargo of such density as to bring the vessel to her load draft.

Spent Condition.—Same as full load condition, but with all fuel, water, and stores consumed.

Ballasted Condition.—Same as spent condition, but with water-ballast tanks filled.

Any other special conditions likely to arise in practice.

Rolling Period of a Vessel.

In Smooth Water.

K = Radius of gyration of the vessel and all weights on board, about a longitudinal axis passing through her centre of gravity.

m = Metacentric height;

2 T = Time of double roll, say from port to starboard and back again, in seconds;

g = Force of gravity = 32·2.

then

$$\mathrm{T} = \pi\sqrt{\frac{\mathrm{K}^2}{g\,m}} = \cdot 554\frac{\mathrm{K}}{\sqrt{m}},$$ neglecting water and air resistances.

The period 2T of a double roll varies from about 3 to 30 seconds; in large Atlantic liners it is about 20 seconds.

Stability.

The Initial Stability of a vessel is the measure of the tendency of the vessel to return to the upright if she is forcibly inclined to a small angle, θ, of not more than about 10°.

Initial Stability or Righting Moment at the inclination $\theta = \mathrm{RM}_\theta = \Delta\,m\,\theta$ foot-tons.

where,

Δ = displacement in tons;

m = metacentric height in feet;

θ = angle of heel in circular measure

= inclination in degrees ÷ 57·3.

CURVE OF STABILITY.—When a vessel is inclined to any angle θ (fig. 3), the centre of buoyancy moves to one side (B to B'). The upward force of buoyancy, Δ, now acts through B', while the weight acts downwards through G, the centre of gravity. These two forces form the righting couple, whose arm is GZ (the righting arm) and whose moment is W.GZ

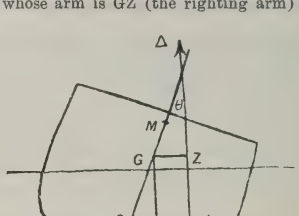


FIG. 3.

(the righting moment). When θ is small, B'Z passes through M, as already stated, and $GZ = m\theta$; at larger angles GZ must be separately calculated, and it is customary to obtain its value at each 10° of inclination and to connect the series of righting arms by a curve which is called the curve of stability (fig. 4).

OR is called the 'Range of Stability,' H the 'Maximum Righting Arm,' occurring at some

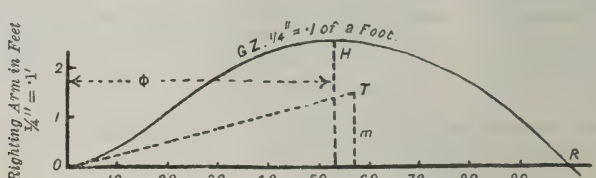


FIG. 4.

angle ϕ, while the initial slope of the curve is measured by the metacentric height, m, set up at 57·3° ($\theta = 1$), as shown in the figure, where OT is tangent to the curve at the origin O. (The scales marked are those to which the curves are usually drawn, but of course do not apply to the reduced figure.) The area of the curve of stability up to any angle is proportional to the work which must be done (by wind or other external forces) in order to incline the vessel to that angle. The work so done is called 'Dynamical Stability.'

Longitudinal Metacentre.

If a vessel be slightly inclined about a transverse horizontal axis, the centre of buoyancy will move longitudinally, and the point in which the vertical line through the new centre of buoyancy cuts that through the centre of buoyancy corresponding to even keel is called the 'longitudinal metacentre.'

The height of the longitudinal metacentre above the centre of buoyancy = LBM = $\frac{I}{V}$, where I is the moment of inertia of the waterplane about a transverse axis through its centre of gravity, and V is the volume of displacement.

The LONGITUDINAL METACENTRIC HEIGHT, LGM, is the height of the longitudinal metacentre above the centre of gravity = LBM − BG = M.

The longitudinal GM is used in calculating changes of trim. If any weight, w, be moved longitudinally through a distance, a, the 'moment to change trim' is wa.

The 'moment to change trim one inch' or 'inch trim moment' $= {}''M = \frac{\Delta M}{12L} \doteq \frac{I}{420L}$

Approximately, $''M = \frac{30\,t^2}{B}$

where, $''M$ = inch trim moment, in foot tons.
Δ = displacement, in tons.
L = length between draft marks, in feet.
t = tons per inch.
B = moulded breadth, in feet.
I = longitudinal moment of inertia of water plane.

The *change of trim* in inches due to any moment $w\,a$ is $\frac{w\,a}{''M}$. Of this total change of trim half is usually added to the draft at the end towards which the weight w was moved, and half deducted from the draft at the other end.

If instead of merely moving a weight already on board, we desire to find the alteration of drafts due to placing a new weight on board, first imagine the weight to be placed at the centre of flotation F. This will increase the drafts both forward and aft equally by an amount $\frac{w}{t}$. Then imagine the weight moved longitudinally to its assigned position and calculate the change of trim. Then combine these changes of drafts with the additional immersions $\frac{w}{t}$ and so find the total changes of draft forward and aft.

Curves connecting the mean draft with the position of the centre of flotation, the tons per inch, and the inch trim moment should be prepared to facilitate such calculations.

For large additions of weight the above method is not accurate, and it is necessary to prepare a complete list of all the items making up the displacement, together with their moments forward or aft of midships. Then

$\frac{\text{Total moment}}{\text{Total weight}} = \text{G}\,\boxtimes$ or $\boxtimes\,\text{G}$ = distance of centre of gravity aft or forward of midships.

From a curve connecting draft with the distance of the centre of buoyancy for even keel from midships, we can find the value of B $\boxtimes$ or $\boxtimes$ B, and hence the value of GB or BG, that is the longitudinal distance of the centre of gravity aft of or forward of the centre of buoyancy. The moment to change trim is then $\Delta \times$GB, and the change of trim $\frac{\Delta . \text{GB}}{''M}$.

RALSTON'S STABILITY AND TRIM INDICATOR.

This useful piece of apparatus determines the metacentric height and trim of a vessel by mechanical means instead of by calculation. It consists of a capacity profile of the vessel

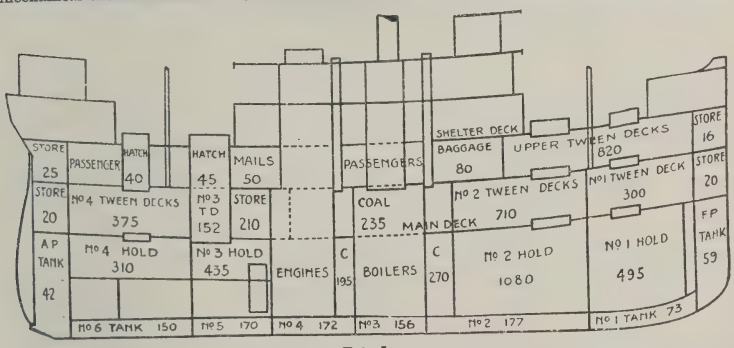


FIG. 5.

(fig. 5), the vertical scale of which, for convenience, is made greater than the longitudinal scale. This profile is engraved on a steel plate or table upon which are mounted two balance weights

* $\doteq$ is the symbol for 'approximately equal.'

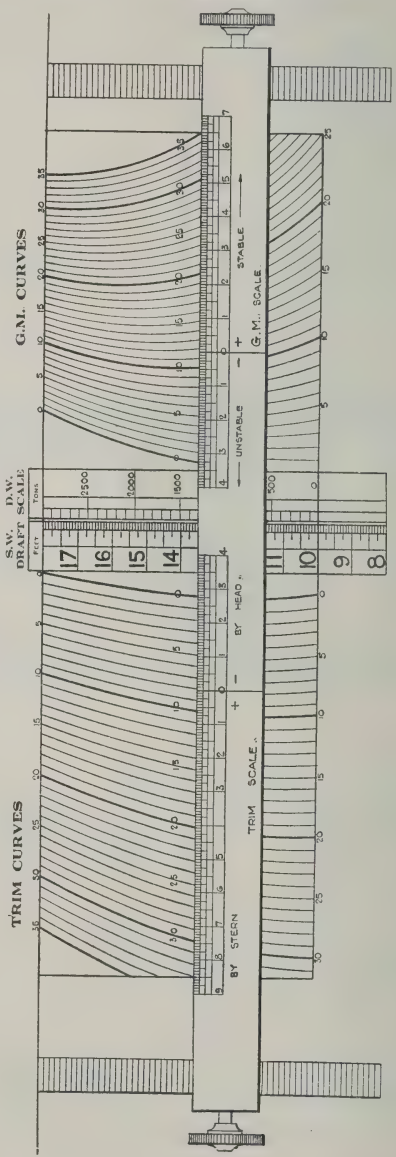

RALSTON'S STABILITY AND TRIM INDICATOR.
TRIM CURVES
G.M. CURVES
S.W. DRAFT SCALE
FEET
D.W. SCALE
TONS
2500
2000
1500
17
16
15
14
11
10
9
8
BY HEAD.
BY STERN
TRIM SCALE.
G.M. SCALE
STABLE
UNSTABLE

FIG. 6.

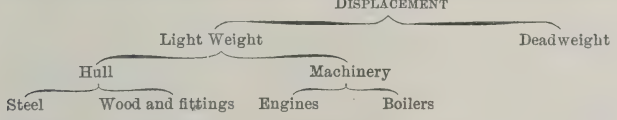
or sliders, one for stability and the other for trim. These are moved by rack and pinion over vertical and horizontal scales of 'index numbers,' engraved on the edges of the plate. The combined weight of table and sliders is made proportional to the light weight of the ship and in the zero position of the sliders the common centre of gravity corresponds in position to that of the ship in the light condition. The weights of cargo, fuel, stores, water, etc., are represented by a series of small brass weights which are placed in position on the capacity profile so as to represent any condition of loading for which it is desired to know the metacentric height and trim.

A separate diagram (fig. 6) gives mean draft and deadweight together with a series of G.M. and trim curves distinguished by index numbers corresponding to those over which the sliders move. The G.M. and trim scale movable over this diagram can be set to any draft and deadweight by means of racks and pinions. In use the table is supported firstly upon a longitudinal axis passing through the light C.G. and the vertical slider moved until balance is obtained and the 'index number' noted. The G.M. scale is then set to the correct deadweight and draft and the metacentric height read at the curve having the same index number. The table is then transferred to a vertical axis passing through the light C.G. and a similar process repeated to obtain the trim.

The instrument is simple, ingenious, and accurate; it takes into account all the features of the vessel's form and weight-distribution, obviates a considerable amount of calculation, and is being extensively adopted in the mercantile marine.

Weights.

The total weight or displacement of a merchant vessel may be subdivided as follows:



The *Weight of Steel* includes:—

Steel and iron plates and bars of all kinds.
Structural forgings and castings, whether of steel or of bronze.
Pillars.
Steel masts.

The *Weight of Wood and Fittings* includes:—

Carpenter work.	Deck machinery and piping.	Cement.
Joiner work.	Electric lighting installation.	Insulation.
Plumber work.	Refrigerating machinery.	Rigging and sails.
Wood masts and spars.	Heating installation.	Canvas covers and awnings.
Upholstery.	Ventilation.	Sheet iron work.
Boats.	Galley outfit.	Fixed ballast.
Smithwork.	Paint.	Deck Coverings. Etc.

The *Weight of Engines* includes:—

Main propelling engines.	Sanitary and fresh-water pumps.	Piping in engine-room and tunnel.
Shafting and propellers.	Floors, ladders, and gratings in engine spaces.	Water in condensers and piping.
Condensers and circulating pumps.	Feed water.	Tools, etc.
Air-pumps and bilge-pumps.	Oil in forced lubrication system and drain tank.	Blowers, silencers.
Electric generators.		Air bottles.

The *Weight of Boilers* includes:—

Main boilers and donkey boiler.	Piping in boiler rooms.	Floor-plates, ladders, and gratings in boiler-rooms.
Uptakes and funnels.	Feed-pumps.	Tools.
Fans and fan engines.	Fuel pumps.	Etc.
	Hot water in boilers.	

Deadweight includes:—

Spare gear.	Reserve feed water.	Stores and provisions.
Crew and effects.	Water ballast.	Bedding and napery.
Passengers and baggage.	Bilge water.	Cutlery and crockery.
Coal in bunkers and on firebars.	Cooling water.	Glass and silverware.
Fresh water for ship's use.	Mails and cargo.	Weights added by the owner.
Salt water for ship's use.	Oil fuel in tanks and pipes.	Etc.

Hull weight may be estimated either in detail or as a whole. Detail calculation requires that complete plans of the vessel are available, together with reference lists giving weight-rates or actual weights of all varieties of fittings. An accurate record of the finished total weights of the various items of the outfit of previous completed vessels is of great value in this connection.

For preliminary work, where a rapid estimate of the weights is required, the method of coefficients is largely relied on. In this method the whole weight of the hull, or of certain sectional portions thereof, is compared with some arbitrary parameter or parameters of simple form, depending upon the principal dimensions of the vessel or of her sectional parts. For each vessel completed an accurate record of these weights is filed, and the values of the 'Coefficients of Weight' $\left(= \frac{\text{weight}}{\text{parameter}} \right)$ worked out. Then for any proposed new vessel the weight of the hull, or of any of its parts, is found approximately by multiplying the proper parameter by a suitable coefficient chosen from among those of similar finished vessels.

The most commonly used parameter is the *Cubic Number*, which is the product of the Length, Breadth, and Depth, divided by 100.

$$\text{Cubic number} = N = L\,B\,D \div 100.$$

where L = Length between perpendiculars, in feet,

B = Breadth moulded, in feet,

D = Depth moulded, in feet, taken to the highest deck to which the shell plating extends continuously or almost continuously throughout the whole length of the vessel;

then,

$$\text{Weight of Hull, } H = N \times \text{Coefficient of Weight of Hull}$$
$$= N \times C_h$$

Similarly,

$$\text{Weight of Steel, } S = N \times C_s$$

$$\text{Weight of Wood and Fittings, } W = N \times Cw.$$

The weights of deck machinery, refrigerating machinery, insulation, etc., may either be included in the wood and fittings coefficient or separately estimated.

The following table gives average values of these coefficients :—

Type of Vessel.	C_s	Cw.	Ch.
Torpedo Boat Destroyer	·27	·10	·37
Cruiser	·34	·12	·46
Battleship	·33	·12	·45
River Paddle Steamer.	·20	·10	·30
Channel Steamer	·25	·13	·38
Coasting Steamer	·32	·20	·52
Cargo Tramp	·35	·12	·47
Intermediate Liner	·42	·17	·59
Fast Mail Liner	·40	·16	·56
Oil Tanker	·38	·06	·44

Other parameters sometimes used for estimating the weight of wood and fittings are

$$(L \times B) \text{ and } L\,(B + D)$$

Weight of Machinery.

In making preliminary estimates, the weight of machinery can be taken as proportional to the horse-power, the number of horse-power per ton of total machinery weight being recorded for all vessels completed and tried. A more accurate method is to record separately the weight of engines and the weight of boilers per horse-power, and to employ these rates when estimating the weights of proposed machinery of similar type. The weight of reciprocating engines may be taken as proportional to the product of the square of the diameter and the square root of the stroke of the high-pressure cylinder. For cylindrical boilers a suitable weight parameter is the product of boiler volume and square root of working pressure. For the greatest accuracy the weights of both engines and boilers must be made up in detail from the drawings and the detailed weight records of previous jobs.

Deadweight.

The total deadweight may either be specified by the owner or must be made up in detail from the particulars of the service to be performed.

Spare Gear.—Make up the weight in detail from the specified list of spares or allow a total approximate amount by reference to the actual weights carried by similar steamers. The amount may vary from about two tons in river paddle steamers and fast Channel steamers to about 120 tons in an Atlantic liner.

Crew and Effects.—Allow about one ton per ten men.

Passengers.—Allow one ton for every fifteen persons in ordinary cases and for every twenty persons in excursion crowds including women and children.

Baggage.—For the hand baggage of excursion passengers allow about one ton for every 250 persons. For long voyages allow about three tons for every ten persons.

Fuel in Bunkers = horse-power at sea, including that of auxiliaries × pounds of fuel consumed per horse-power per hour ÷ 2240 × distance between fuelling stations farthest apart ÷ service speed, + coal for galleys etc. + allowance for use in port + a margin of 10 to 20 per cent. for emergencies.

For rates of coal consumption see tables on pages 2375, 2392.

Coal on Bars.—Allow about 35 lbs. per square foot of grate area.

Fresh Water.—Allow about one ton per day for every 25 first or second-class passengers and about one ton per day for every 75 third-class or steerage passengers and crew.

Salt Water.—For sanitary purposes a tank containing from ½ ton to 5 tons according to the extent of the sanitary accommodation.

Reserve Feed Water.—Allow about 4 tons per 1,000 H.P. per 24 hours for short voyages and about half as much in vessels where distilling apparatus is carried.

Provisions.—Allow about one ton per day for every 75 first- and second-class passengers and about one ton per day for every 250 third-class or steerage passengers and crew.

Water Ballast.—As may be required for purposes of immersion, stability or trim during the voyage. Say one-half of weight of consumables.

Bilge Water.—Is more or less inevitable, but should not amount to much if proper care in pumping is continually taken. Allow about one-fifth of the vessel's 'tons per inch.'

Mails, cargo, and stores.—As required by owners.

Bedding and napery
Cutlery and crockery } Together about one ton for every 15 persons on board.
Glass and silverware

Resistance of Ships.

The total resistance to the ahead motion of a ship is made up of four parts :—

1. 'Skin-friction resistance,' or 'skin resistance.'
2. 'Wave-making resistance,' or 'wave resistance.'
3. 'Eddy-making resistance,' or 'eddy resistance.'
4. Air resistance.

2 and 3 are usually considered together under the name 'residuary resistance,' or 'wave resistance' merely, since eddy resistance should be very small. An allowance of 10 per cent. is usually added to the resistance of the 'naked' hull to account for the resistances of bossings, bilge-keels, rudder and other appendages, and air resistance.

HORSE-POWER.

The horse-power required to overcome any of these resistances is proportional to the resistance multiplied by the speed, and is called the 'effective horse-power.' It is from about ·45 to ·57 of the corresponding indicated or shaft horse-power developed within the vessel, the remainder of the power developed being absorbed by engine and shaft friction and losses in the propeller.

The ratio of effective horse-power to indicated or shaft horse-power is called the 'propulsive efficiency,' or 'propulsive coefficient.'

$$e = \frac{\text{E.H.P.}}{\text{I.H.P.}} \text{ or } \frac{\text{E.H.P.}}{\text{S.H.P.}}$$
$$= \text{from } \cdot 45 \text{ to } \cdot 57.$$

Skin Effective Horse-Power.

Skin E.H.P. is directly calculable :—

$$\text{Skin E.H.P.} = \text{E}_s = \frac{\text{S}}{100} \times \text{F}_{10} \times \left(\frac{\text{K}}{10}\right)^{2 \cdot 83}.$$

where,

S = wetted surface in square feet.
F_{10} = Skin E.H.P. per 100 square feet at 10 knots.
K = speed in knots.

The following table gives the values of F_{10} and of $\left(\frac{\text{K}}{10}\right)^{2 \cdot 83}$ for various lengths and speeds :—

L.	F_{10}.	L.	F_{10}.	K.	$\left(\frac{\text{K}}{10}\right)^{2 \cdot 83}$	K.	$\left(\frac{\text{K}}{10}\right)^{2 \cdot 83}$
50	1·999	450	1·832	10	1·00	18	5·28
100	1·918	500	1·826	11	1·31	19	6·15
150	1·888	600	1·816	12	1·68	20	7·11
200	1·873	700	1·807	13	2·10	25	13·37
250	1·863	800	1·798	14	2·59	30	22·40
300	1·854	900	1·790	15	3·15	35	34·65
350	1·846	1000	1·782	16	3·78	40	50·56
400	1·839			17	4·49		

From these figures curves may be drawn upon squared paper from which intermediate readings can be obtained.

Wave Effective Horse-Power.

The wave effective horse-power is not directly calculable, although several empirical formulæ and curves of coefficients have been published.

Taylor gives :—

$$\text{Wave E.H.P.} = \text{E}_w = \text{B}_\bullet \frac{\Delta^{\frac{2}{3}}}{\text{L}} \times \cdot 0030707 \, \text{K}^5.$$

Where $\text{B}_\bullet$ = from ·35 to ·55, according to the degree of fullness of the vessel and her ratio of length to breadth.

The most accurate method of estimating E_w is by using the results of model experiments or of actually tried vessels, and Froude's 'Law of Comparison.' The Law of Comparison may be stated thus :—If two vessels of exactly similar form, but of different size, are run at speeds proportional to the square roots of their lengths, then their wave effective horse-powers will be proportional to the products of their displacements by their speeds. If, then, we are designing a vessel n times as long as a model or similar smaller ship whose speeds and corresponding wave E.H.Ps. we know, the following ratios which various elements of the design bear to corresponding elements of the prototype may be used :—

Linear dimensions	n	Speeds	$\sqrt{n}$
Surfaces	n^2	Wave resistances	n^3
Displacements	n^3	Wave E.H.Ps.	$n^3 \sqrt{n} = n^{3 \cdot 5}$

Speeds related as above are called 'Corresponding Speeds,' and the ratio $\frac{\text{K}}{\sqrt{\text{L}}}$ of speed to square-root of length is called the 'Speed-Length Ratio.'

Model experiments give us the total E.H.P. at any speed. From this the wave E.H.P. can be obtained by separately calculating the skin E.H.P., as above described, and deducting it from the total. The wave E.H.P. for the full-size ship will be $n^{3\frac{1}{2}}$ times that of the model, and the skin E.H.P. for the full-size ship can be separately calculated. The sum of these two results gives the

total E.H.P. for the full-size ship. It is then finally necessary to assume a suitable value for e, the propulsive coefficient, and calculate the I.H.P. or S.H.P. by dividing the total E.H.P. by e. If we are designing from the trial results of a similar completed vessel, the procedure is slightly different :—

I.H.P. or S.H.P. of Type Ship $\times\ e$ = E.H.P. of Type Ship

$\frac{S}{100} \times F_{10} \times \left(\frac{K}{10}\right)^{2\cdot83}$ = Skin E.H.P. of Type Ship } at K knots.

Difference = Wave E.H.P. of Type Ship

Multiply by $n^{3\frac{1}{2}}$

and so obtain Wave E.H.P. for Design

$\frac{S'}{100} \times F'_{10} \times \left(\frac{K'}{10}\right)^{2\cdot83}$ = Skin E.H.P. for Design } at $\sqrt{n}$ K knots.

Sum = Total E.H.P. for Design

Divide by e

To obtain I.H.P. or S.H.P. for Design

where,

S = wetted surface of type ship, in square feet,
$S' = n^2 S$ = " design "
K = speed of type ship, in knots,
$K' = \sqrt{n}\,K$ = corresponding speed of design, in knots,
F_{10} = skin E.H.P. per 100 square feet at 10 knots for type ship,
F'_{10} = " " " for design.

D. W. Taylor's book, 'The Speed and Power of Ships,' which contains a series of curves based on model experiments and covering a large range of proportions of dimensions and speeds, will be found to give very accurate estimates of effective horse-power.

For fast, fine vessels, the paper 'On some Results of Model Experiments,' by R. E. Froude, published in the 'Transactions of the Institution of Naval Architects for 1904,' may be advantageously consulted. Many other experimental results have now been published both in Great Britain and America.

Admiralty Coefficients.

The 'Admiralty Coefficient of Performance' $= \frac{\Delta^{\frac{2}{3}} K^3}{P} = C$

where,

Δ = displacement, in tons,
K = speed, in knots,
P = I.H.P. or S.H.P.

Conversely $P = \frac{\Delta^{\frac{2}{3}} K^3}{C}$, so that if C is known or can be correctly assumed, P can readily be calculated. In similar ships at 'corresponding speeds' $\Delta^{\frac{2}{3}} K^3$ varies as $(n^3)^{\frac{2}{3}} (\sqrt{n})^3$, that is, as $n^{3\frac{1}{2}}$. Hence, if P were also to vary as $n^{3\frac{1}{2}}$, the values of C would remain the same. And also, if for any given vessel P varied as K^3, C would be the same for all speeds. For these reasons C is sometimes called the 'Admiralty Constant,' but as the above relations are theoretical assumptions only, and C is found to vary from speed to speed and from ship to ship, it is better to use the term 'Admiralty Coefficient.'

Although wave-making power follows the law of comparison, and therefore varies as $n^{3\frac{1}{2}}$, skin-friction power does not obey the same law and only increases at the lesser rate

$$n^2 \frac{F'}{F} (\sqrt{n})^{2\cdot88} = \frac{F'}{F} n^{3\cdot415}$$

And hence larger ships have relatively less power and larger values of C than smaller vessels. In the case of large vessels, also, a lesser proportion of power is lost in overcoming internal friction and eddy-making, and a further increase of C is the result.

In spite of its known defects, the 'Admiralty Coefficient' is largely used for estimating horse-power, the assumed value being based upon that for a similar, or nearly similar, completed vessel at 'corresponding' speed, and suitable allowances made for variations in size and proportions,

If the prismatic coefficient of the type ship differs from that of the proposed vessel, we may still find suitable values for C by modifying the 'corresponding speed' of the type ship, thus :—

$$K = K' \sqrt{\frac{L}{L'} \cdot \frac{1 - C_p}{1 - C'_p}}$$

where K, L, and C_p are the corresponding speed, length between perpendiculars, and prismatic coefficient respectively for the type ship, and K', L', and C'_p the desired speed, length, and prismatic coefficient for the proposed vessel.

Economical Speeds.

For each vessel there is a certain 'suitable' or 'economical' speed, determinable in relation to her length and fineness, above which she would be wasteful of power or 'overdriven' and below which she would be unnecessarily fine.

The following table (A) expresses these relations, and Table B gives possible values of the Admiralty coefficient on trial for vessels agreeing with Table A :—

TABLE A.

C_p. Prismatic Co-efficient.	$\frac{V}{\sqrt{L}}$ Speed Length Ratio.	C_p. Prismatic Co-efficient.	$\frac{V}{\sqrt{L}}$ Speed Length Ratio.
·61	·925	·73	·758
·63	·897	·75	·730
·65	·869	·77	·703
·67	·842	·79	·675
·69	·814	·81	·647
·71	·786		

TABLE B.

L. Length on Waterline.	C. Admiralty Coeffi-cient.	L. Length on Waterline.	C. Admiralty Coeffi-cient.
100	137	500	307
150	188	550	310
200	227	600	312
250	255	650	315
300	275	700	317
350	289	750	319
400	297	800	321
450	303	850	323

Table B assumes an efficiency, *e*, of about 55 per cent.; for large turbine work *e* appears to be about 50 per cent., so that in estimating shaft horse-power in such cases C should be taken at about 10 per cent. less than is given by the above table. For single-reduction geared turbines *e* = about 53 per cent.; for double-reduction geared turbines *e* = about 55 per cent.

The formula $C_b = 1{\cdot}06 - \frac{K}{2\sqrt{L}}$ has also been proposed as expressing a suitable connection between block coefficient and speed-length ratio.

Admiralty Coefficient applied to Fuel Consumption.

A variation of the Admiralty Coefficient much used by superintendent engineers, when comparing performances at sea, consists in substituting the daily fuel consumption in tons for horse-power in the formula $C = \frac{\Delta^{\frac{2}{3}} K^3}{P}$. We thus get;—

$$\text{Fuel Coefficient of Performance} = \frac{\Delta^{\frac{2}{3}} K^3}{T}$$

where Δ = mean service displacement in tons,
K = mean service speed in knots,
T = daily fuel consumption in tons.

Rudders.

The area of a vessel's rudder is determined by dividing the product of her length and load draft by about 30 to 40 in the case of warships, 60 to 70 for fast liners, 90 to 100 for slow cargo

tramps. The most efficient angle is about 35° from the middle line, at which angle the total pressure upon the rudder in tons is about $\frac{AK^2}{700}$ for single screw ships, $\frac{AK^2}{1000}$ for twin screws, where A is the rudder area in square feet and K the ship's speed in knots. The centre of pressure at 35° is found by assuming that the pressure on each elementary horizontal strip is applied at a point four-tenths of the breadth of the strip from its leading edge, summing the moments for each strip, and dividing by the total area.

Let h be the distance of the centre of pressure from the axis in inches, and P the total pressure upon the rudder in tons. Then T, the twisting moment in inch-tons, $= Ph$, and

$$D = \text{diameter of stock, in inches} = \sqrt[3]{\left(\frac{16T}{\pi F}\right)}$$

where F = maximum allowable shearing stress in material in tons per square inch.

$$\text{If } F = 5{\cdot}1,\ D = \sqrt[3]{T}$$

In practice the diameters of the rudder stocks of all classified vessels are determined by the rules of the Classification Societies.

Estimate Sheet.

When making preliminary estimates of dimensions, weights, power, etc., the following form will be found convenient:—

Number of Inquiry. Type of Vessel. Owners. Date.

Conditions.

Length × Breadth.

Type. Length × Breadth × Depth. Cubic Number.

S Steel Weight. Steel Coefficient.

W Wood and Outfit Weight. W. & O. Coefficient.

M Machinery Weight. H.P. per Ton.

D Deadweight.

$$\frac{\text{Displacement}^{\frac{2}{3}} \times \text{Speed}^3}{\text{Admiralty Coefficient}} = \text{Horse Power.} \qquad \frac{\text{Length} \times \text{Breadth} \times \frac{\text{Draft full}}{\text{Keel}}}{35} = \text{Block Number} = \frac{\text{Block Coefficient}}{\text{Displacement}}$$

For example, in figures :—

Inquiry No. 42. Cargo and Passenger Steamer. Messrs. Brown and Black. 3.3.30.

To carry 6000 tons and steam 13½ knots on 24 ft. draft. Class, 100 A1 at Lloyd's. Sixty first-class. 100 second-class.

	19200		
T.S.S.	400 × 48 × 30		5760
S.	2200	·382	
W.	850	·147	
M.	750	4·54	
D.	6000		

$$\frac{9800^{\frac{2}{3}} \times 13\tfrac{1}{2}^3}{313} = 3400 \text{ I.H.P.}$$

$$\frac{400 \times 48 \times \overset{24'\ 0''}{\underset{2''\ \text{keel}}{23'\ 10''}}}{35} = 13080 \times {\cdot}75 = 9800$$

These figures are arrived at by a process of trial and error-correction until a satisfactory balance is reached between weights and displacement, block coefficient and speed length ratio, horse-power and weight of machinery.

PARTICULARS OF 13-KNOT ATLANTIC CARGO STEAMERS TO ILLUSTRATE THE RELATIVE FUEL ECONOMY OBTAINED BY INCREASE OF DIMENSION.

Dimensions.				Block Coefficient.	Load Displacement.	Horse-power on Service.	Light Displacement.	Dead-weight.	Coal used on Voyage.	Coal in Bunkers.	Stores and Water.	Cargo.	Tons of Cargo per ton of Coal used.
Length.	Breadth.	Depth to Shelter Deck.	Draft.										
ft.	ft.	ft. ins.	ft. ins.		tons.		tons.	tons.	tons.	tons.	tons.	tons.	
300	40	31 0	22 0	·742	5,600	3,250	2,335	3,265	500	550	300	2,415	4·8
350	45	34 6	24 2	·762	8,250	3,950	3,260	4,990	610	675	350	3,965	6·5
400	50	38 0	26 4	·780	11,750	4,800	4,440	7.310	740	825	400	6,085	8·1
450	55	41 6	28 6	·796	16,000	5,750	5,900	10,100	890	975	450	8,675	9·8
500	60	45 0	30 8	·810	21,300	6,750	7,670	13,630	1,040	1,150	500	11,980	11·4
550	65	48 6	32 10	·822	27,500	7,850	9,820	17,680	1,210	1,350	550	15,780	13·1
600	70	52 0	35 0	·832	35,000	9,000	12,370	22,630	1,390	1,550	600	20,480	14·7

British Government Standard Cargo Vessels (Fig. 7).

Type.	Moulded Dimensions. L.	B.	D.	Gross Tonnage.	Steel Wght. — Coefft.	Wood and Outfit Wght. — Coefft.	Wght. of Engines.	Light Wght.	Dead Wght.	Load Dispt. — Block Coefficient.	Load Draft.	H.P. at Sea. — Admiralty Coefficient.	Sea Speed.	Engine.				Boilers. No.	Boilers. Diam.	Boilers. Lgth.	Parent Builders.
A	400	52	31	5,030	2,225 0·346	405 0·063	570	3,200	8,175	11,375 0·762	25-1	2,500 268	11	27	44	73	48	3	15-6	11-6	Henderson, Glasgow.
B	400	52	31	5,030	2,325 0·361	405 0·063	570	3,300	8,075	11,375 0·762	25-1	2,500 268	11	27	44	73	48	3	15-6	11-6	Henderson, Glasgow.
C	331	46-6	25-6	3,000	1,390 0·354	300 0·077	460	2,150	5,050	7,200 0·760	21-8	2,200 258	11·5	25	41	68	45	3	14-0	11-6	Tyne I. S. Co., Willington Quay.
D	285	41-9	21-3	2,300	1,110 0·440	240 0·095	420	1,770	2,980	4,750 0·732	19-0	1,900 226	11·5	25	41	68	46	2	16-6	11-9	Austin, Sunderland.
E	376	51-6	29	4,400	1,920 0·342	400 0·071	570	2,890	7,020	9,910 0·749	23-9	2,500 280	11·5	27	44	73	48	3	15-6	11-6	Duncan, Port Glasgow.
F	411-6	55-6	29	6,440	2,565 0·388	520 0·079	620	3,705	10,795	14,495 0·775	28-8	3,650 281	12	27	45	75	54	3	16-0	12-3	Thompson, Sunderland.
*	400	53	35-5	5,680	1,920 0·256	430 0·057	620	2,935	9,000	11,935 0·745	26-4	3,650 246	12	27	45	75	54	3	16-3	12-6	Northumberland Co., Howdon.
G	450	58	40	8,000	3,200 0·306	650 0·062	1,350	5,200	10,800	16,000 0·740	29-0	5,500 253	13	26½	44	73	48	3	16-3	20-6	Workman, Clark & Co., Belfast.
H	303	43	23	2,800	1,150 0·384	300 0·100	360	1,810	3,960	5,670 0·780	19-6	1,400 261	10·5	22	36	59	39	2	15-6	10-6	Austin, Sunderland.
†	385	52	29	4,600	2,070 0·357	200 0·035	550	2,820	7,520	10,340 0·750	24-0	2,500 287	11·5	27	44	73	48	3	15-6	11-6	Russell, Port Glasgow.
N	411-6	55-6	38-2	6,500	2,600 0·299	520 0·060	570	3,690	10,500	14,190 0·770	28-0	2,300 204	10·5	Geared Turbine				3	Babcock		Swan, Hunter, Wallsend.
A′	400	52	31	5,250	2,600 0·404	350 0·054	570	3,520	7,885	11,405 0·760	25-1	2,500 269	11	27	44	73	48	3	15-6	11-6	Laing, Sunderland.
Z	400	52	31	5,800	2,360 0·367	317 0·049	570	3,400	8,000	11,400 0·762	25-1	2,500 269	11	27	44	73	48	3	15-6	11-6	Swan, Hunter, Wallsend.

* Shelter deck. † 'Tween deck. A′ Converted oil carrier. Z Oil carrier. All single-screw vessels.

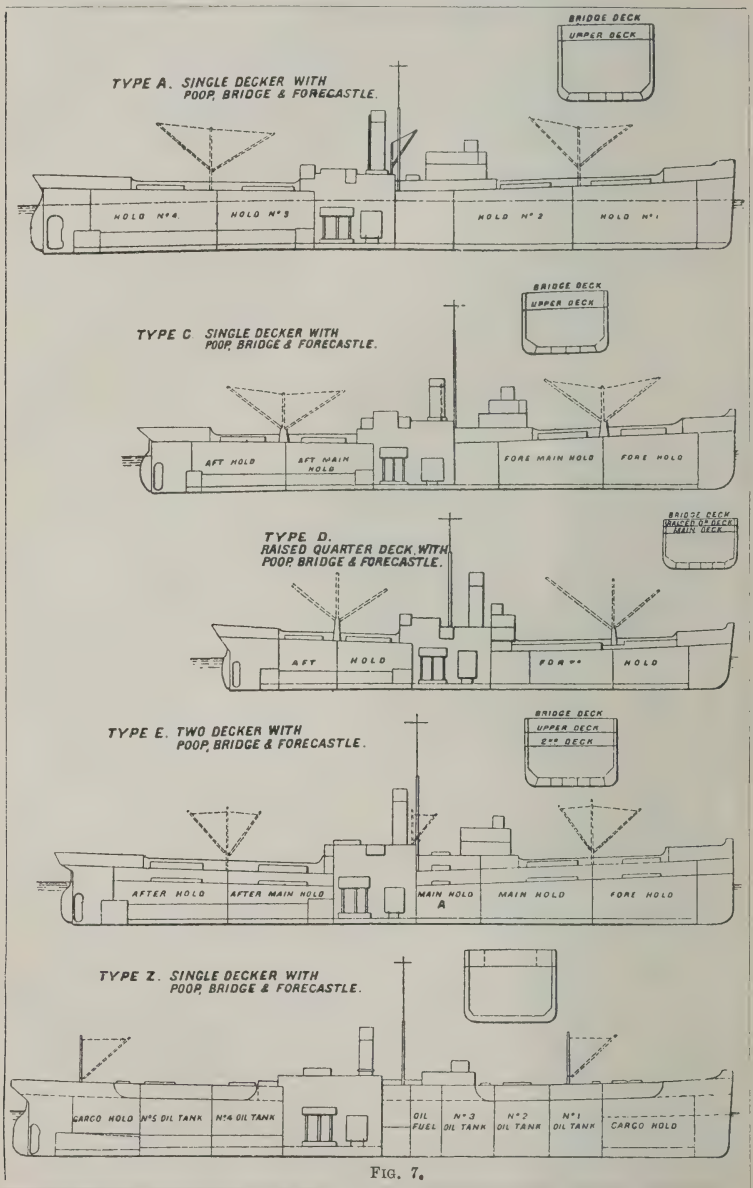

TYPE A. SINGLE DECKER WITH POOP, BRIDGE & FORECASTLE.
BRIDGE DECK
UPPER DECK
HOLD N°4.
HOLD N°3
HOLD N°2
HOLD N°1
TYPE C. SINGLE DECKER WITH POOP, BRIDGE & FORECASTLE.
BRIDGE DECK
UPPER DECK
AFT HOLD
AFT MAIN HOLD
FORE MAIN HOLD
FORE HOLD
TYPE D. RAISED QUARTER DECK WITH POOP, BRIDGE & FORECASTLE.
BRIDGE DECK
RAISED QR DECK
MAIN DECK
AFT
HOLD
FOR
HOLD
TYPE E. TWO DECKER WITH POOP, BRIDGE & FORECASTLE.
BRIDGE DECK
UPPER DECK
2ND DECK
AFTER HOLD
AFTER MAIN HOLD
MAIN HOLD A
MAIN HOLD
FORE HOLD
TYPE Z. SINGLE DECKER WITH POOP, BRIDGE & FORECASTLE.
CARGO HOLD
N°5 OIL TANK
N°4 OIL TANK
OIL FUEL
N°3 OIL TANK
N°2 OIL TANK
N°1 OIL TANK
CARGO HOLD

FIG. 7.

Nationality.	British.			Ex-German.		German.		French.	Italian.	
Owners.	Cunard Steamship Company.		White Star Line.	Cunard S.S. Co.	White Star Line.	Hamburg South American S.S. Co.	N.D. Lloyd.	Cie. Générale Transatlantique.	Cosulich.	Navigazione Generale Italiana.
Names.	Aquitania.	Mauretania.	Olympic.	Berengaria.†	Majestic.‡	Cap Arcona.	Bremen.	Ile de France.	Saturnia.	Augustus.
Built at	Clydebank	Wallsend	Belfast	Hamburg	Hamburg	Hamburg	Bremen	St. Nazaire	Trieste	Sestri Ponenti
Date	1914	1907	1911	1912	1922	1927	1930	1927	1927	1927
Length	865′	760′	850′	880′	912′	660′	888′	758′	599′	664′ 6″
Breadth	97′	88′	92′ 6″	98′ 6″	100′ 6″	84′ 3″	101′ 9″	92′	79′ 6″	82′ 6″
Depth	64′ 6″	60′ 6″	64′ 9″	63′	64′ 2″	55′ 5″	62′ 4″	61′	46′ 6″	51′ 6″
Gross tonnage	45,647	30,696	46,459	52,226	56,551	27,560	51,656	43,548	23,990	32,500
Draft	36′ 2″	36′ 1·5″	34′ 7″	39′	38′ 10″	28′ 5″	32′	32′	29′	30′
Displacement	53,130	41,590	52,310	63,100	64,509	—	51,009	41,000	—	32,000
Passengers I.	757	589	947	686	875	636	800	677	279	300
,, II.	635	399	498	714	725	274	500	617	566	578
,, III.	1,998	1,150	1,100	2,690	2,216	402	900	346	1,352	1,300
Engines	Turbine	Turbine	Combination	Turbine	Turbine	Turbine	Turbines	Turbine	Diesel	Diesel
Cranks	—	—	2 × 4	—	—	—	—	—	2 × 8	4 × 6
Cylinders No.	—	—	2 2 2 2	—	—	—	—	—	16	24
Cylinders Dia.	—	—	54 84 97 97	—	—	—	—	—	33	30·2
Stroke	—	—	75	—	—	—	—	—	59	47·2
Revolutions	180	180	78, 160	185	180	125	180	220	125	120
Boilers	21 D	23 D : 2 S	24 D : 5 S	46 Yarrow	48 Yarrow	8 D. W.T.	11 D : 9 S : W.T.	24 W.T.	—	—
Pressure	195	195	215	235	240	300	325	228	—	—
Furnaces	168	192	159	—	—	—	217	144	—	—
Heating surface	138,596	157,520	144,142	203,000	219,504	52,500	225,243	125,591	—	—
Grate area	3,542	4,048	3,466	3,767	4,013	—	—	—	—	—
Draught	Howden	Howden	Natural	Howden	Forced	—	Howden	Howden	—	—
Fuel	Oil	Oil	Oil	Oil	Oil	Oil	Oil	Oil	Oil	Oil
Horse-power	60,000	70,000	46,000	76,250	66,000	24,000	92,500	52,000	23,000	28,000
Speed	23	27·625*	21 §	22·5	25 §	20	26·25‖	23·5	20	20·75

* 633 miles in 24 hours, July 1928. † Ex-Imperator. ‡ Ex-Bismarck. § Estimated. ‖ Contract Speed: Europa 27·91 in March 1930.

BASES OF ENGINE AND BOILER DESIGN.

Cylinder Ratios.

Considerations of weight and space usually limit the dimensions of cylinders for the machinery of high speed cross-channel vessels. For ordinary merchant and mail steamers these considerations are of minor importance; coal consumption however is of the utmost importance, and consequently a higher ratio of expansion of steam is utilised, involving larger cylinder dimensions, in order to obtain maximum economy of coal consumption. In fast passenger vessels with high revolutions per minute, four-crank triple-expansion engines are invariably adopted, there being one H.P., one M.P., and two L.P. cylinders. This arrangement admits of a satisfactorily balanced engine being obtained, reducing vibration to a minimum, and the weight of the moving parts of the L.P. cylinders is considerably reduced compared with the ordinary three-crank triple engine or four-crank quadruple. The following table of cylinder area ratios is in accordance with modern practice for various types of marine engines:—

TABLE SHOWING CYLINDER AREA RATIOS.

Type of Engine.	Boiler Pressure. Lbs. per Sq. In.	Ratio of Cylinder Areas.			
		L.P. / H.P.	2nd M.P. / H.P.	1st M.P. / H.P.	H.P.
Quadruple expansion . .	200 to 220	8 9	4·0 4·3	2·0 2·08	1 1
			M.P. / H.P.		
Triple expansion, ordinary merchant practice . . .	160 to 180	6·5 to 7·5	2·5 to 2·7		1
Cross-channel steamers . .	180 to 200	6·25	2·4		1
Compound engines, screw .	100 to 140	4 to 5	—		1
,, ,, paddle .	90 to 130	3·5	—		1

The reciprocating engine is now, however, being displaced by direct or geared turbines in all classes of vessels, from fast passenger and mail steamers and cross-channel boats to cargo vessels, even of the tramp class, owing to the reduction obtainable thereby in fuel consumption.

Mean Pressures and Expansion of Steam.

The total power developed in an engine may be varied by altering the point at which steam is cut off in the H.P. cylinder, and generally provision is made to enable the cut-off in each cylinder to be independently adjusted if necessary to equalise the powers developed in each cylinder. For the maximum designed power the cut-off in the H.P. cylinder takes place at from ·65 to ·75 of the stroke, and the total number of expansions of the steam is

$$= \frac{\text{Volume of L.P. cylinder}}{\text{Volume of H.P. cylinder at point of cut-off}};$$

or

$$\frac{\text{Ratio of } \dfrac{\text{L.P. cylinder area}}{\text{H.P. cylinder area}}}{\text{Cut-off in H.P. cylinder (expressed in fraction of stroke)}}.$$

The total expansion of the steam is usually considered with clearance neglected and neglecting also the fact that release takes place before the end of the stroke. In estimating the mean pressure obtainable in any multiple-stage expansion engine it is referred to the L.P. cylinder, the mean pressure thus obtained being that necessary if the whole of the power were developed in the L.P. cylinder. The actual referred mean pressure is, therefore, in the case of a triple expansion engine equal to

$$\text{Mean pressure in L.P. cylinder} + \frac{\text{Mean pressure in M.P. cylinder}}{\text{Ratio of L.P. to M.P. cylinder area}} + \frac{\text{Mean pressure in H.P. cylinder}}{\text{Ratio of L.P. to H.P. cylinder area}}.$$

The theoretical referred mean pressure may be calculated, assuming that the steam is moderately moist, the curve of expansion being a hyperbola. It depends upon the designed number of expansions and the absolute admission pressure of the steam.

Let Pa = absolute initial pressure,
Pm = absolute theoretical referred mean pressure,
r = total number of expansions,
$\frac{1}{r}$ = cut-off, or reciprocal of r,

then

$$Pm = Pa \frac{1 + \text{hyp. log. } r}{r}.$$

Pressure of Steam in Cylinders of Compound and Triple-expansion Engines.

The values of $\frac{1 + \text{Hyp. Log. } r}{r}$ for Various Expansions are given on page 1705.

The initial pressure, Pa, should be taken 5 to 10 lbs. below the absolute boiler pressure to allow for the drop between boilers and engines, and the back pressure, which must be deducted from Pm as found above, should be taken at not less than 4 lbs. per square inch absolute.

Diagram Factor.

Owing to the losses which occur while the steam is expanding in the various cylinders the actual referred mean pressure obtained from indicator diagrams is less than the theoretical referred mean pressure, and consequently the latter must be multiplied by a diagram factor which takes into account the losses due to clearance, condensation, radiation, friction of ports, compression, drop in pressure between the cylinders, and other losses. The value of this factor is obtained by experience, and, when possible, should be derived from engines similar to the one projected, and working under similar conditions. The values for various types of engines are approximately :—

Quadruple-expansion engines	·63 to ·68
Triple-expansion engines, merchant service	·6 to ·65
„ high-speed cross-channel engines	·56 to ·63
Compound engines	·63 to ·70

When cylinders are jacketed the diagram factors may be increased by about 5 per cent.

The power is usually divided equally among the cylinders of an ordinary compound, triple, or quadruple-expansion engine, and if N is the number of cylinders and P the referred mean pressure (*i.e.* theoretical mean pressure × diagram factor) the mean pressure in the various cylinders of a triple-expansion engine, for example, will be :—

$$\text{L.P. cylinder mean pressure} = \frac{P}{N}$$

$$\text{M.P. „ „ „} = \frac{P}{N} \times \frac{\text{L.P. cylinder area}}{\text{M.P. cylinder area}}$$

$$\text{H.P. „ „ „} = \frac{P}{N} \times \frac{\text{L.P. cylinder area}}{\text{H.P. cylinder area}}$$

In the case of four-crank triple-expansion engines the power developed in each L.P. cylinder is usually five-eighths that of the H.P. or M.P. cylinder.

Piston Speed.

The mean piston speed is usually reckoned in feet per minute and is equal to stroke in feet × 2 × revolutions per minute.

The power which may be obtained from a given weight of engine varies almost directly with the piston speed ; the wear and tear of the engine, however, is increased with the higher speeds on account of the increased inertia stresses involved

Piston speed may be increased either by increasing the stroke and keeping the revolutions constant or by increasing the revolutions, keeping the stroke constant.

By adopting the former method the cost, dimensions, and weight of the engines are increased, whilst by adopting the latter method there is practically no increase in any of the items mentioned. It is evident, therefore, that in cases where weight and space are limited high piston speed must be obtained by high revolutions; these have a limit, however, on account of the stresses due to the inertia of the reciprocating parts.

The piston speed and revolutions of engines fitted in various types of vessels are as shown in the following table :—

TABLE SHOWING PISTON SPEED AND REVOLUTIONS OF RECIPROCATING MACHINERY FOR VARIOUS TYPES OF STEAMSHIPS.

Description.	Number of Cranks.	Piston Speed in Feet per Minute.	Revolutions per Minute.
Large mail steamers	4	800 to 900	75 to 90
	3	600 ,, 750	70 ,, 80
Cargo vessels	4	650 ,, 800	70 ,, 100
	3	550 ,, 650	60 ,, 100
Channel steamers	4	800 ,,1000	170 ,, 200
	3	750 ,, 900	130 ,, 180
Paddle steamers	—	550 ,, 650	45 ,, 55

(Also see remarks about propeller revolutions for geared turbines.)

Calculation of Cylinder Diameters.

In determining the diameter of cylinders required for an engine of a given I.H.P., the type of vessel for which the engine has to be designed to a certain extent determines the piston speed and cylinder ratios, due consideration being given to the boiler pressure, the number of expansions being increased as higher boiler pressures are adopted.

Having decided the piston speed and the mean pressure referred to L.P. cylinder as already described, the area of the L.P. cylinder may be calculated, and from the cylinder ratios decided upon the areas of the remaining cylinders are determined.

Let piston speed = S feet per minute;

referred mean pressure = P lbs. per square inch,

then,

$$\text{Indicated H.P.} = \frac{\text{Area of L.P. cylinder} \times P \times S}{33{,}000}$$

or

$$\text{Area of L.P. cylinder} = \frac{\text{I.H.P.} \times 33{,}000}{P \times S};$$

$$\text{Area of M.P. cylinder} = \text{Area of L.P. cylinder} \times \text{Ratio of } \frac{\text{M.P. cylinder area}}{\text{L.P. cylinder area}};$$

$$\text{Area of H.P. cylinder} = \text{Area of M.P. cylinder} \times \text{Ratio of } \frac{\text{H.P. cylinder area}}{\text{M.P. cylinder area}}.$$

Combined Recipro-Turbine Machinery.

In vessels of say under 18 knots the direct coupled turbine installations are not suitable, since the comparatively low rotational speed necessary for an efficient propeller necessitates for an efficient turbine a considerable increase in dimensions and hence increase in weight and cost. In such cases a combination of reciprocating engines and turbines was employed, there being two reciprocating engines to deal with the high-pressure steam and one or two turbines to expand the exhaust steam down to the lowest possible pressure. In the steamer 'Otaki' this system is credited with a saving of about 17 per cent. in the coal consumption; it has also been fitted to the White Star Liners 'Laurentic' and 'Olymp.c,' the Orient Liner 'Orama,' and other vessels.

In the 'Reina Victoria Eugenia' a four-shaft 'combination' arrangement was adopted, the two inner shafts being driven by reciprocating engines and the two outer shafts by low-pressure turbines. At low powers the reciprocating engines produced about three-quarters of the total power and the turbines one-quarter; at full output the proportions were about two-thirds and one-third respectively. The steam consumption was only about 11 lbs. per h.p. per hour or about 20 per cent. less than that required for quadruple expansion reciprocating machinery.

The revolutions of the reciprocating engines were about 113 per min., those of the turbines about 180. The sister vessel 'Infanta Isabel de Borbon' had the usual three-shaft combination the revolutions of the turbines at full power being 276 per min. and of the reciprocating engines 116 per min.

In a three-shaft combination installation with 92 revolutions per min. for the reciprocating engines and 250 for the turbines, the reciprocating portion of the machinery produced 56 per cent. and the turbine portion 44 per cent. of the total full power, but the propulsive effect of the turbine propeller is rather disappointing. This machinery has been displaced by the geared turbine.

TURBINES.*

The development of the steam turbine as a motor for ship propulsion has progressed rapidly in the last decade. In the earlier stages of its adoption, when the turbine was directly coupled to the propeller shafting, the high revolutions necessary to obtain suitable blade speeds for low steam consumption without abnormally increasing the size and weight of the turbines entailed small propellers which were inefficient except for fast vessels, and even in these the turbines were run at an uneconomically low speed in order to improve the propellers.

The introduction of single-reduction gearing between the turbines and propellers enabled advantage to be taken of high blade speeds and fairly efficient propellers for vessels of over about 16 knots speed, and for all speeds of vessels double-reduction gearing permits the turbine and propeller designers to select the revolutions most suitable for their individual purposes so as to obtain the maximum efficiency of turbine and propeller respectively. The revolutions of the most efficient propeller for a vessel depend on horse-power, speed and form of vessel, and should be worked out in conjunction with experimental results such as those published by Taylor and Froude, but the following method gives very satisfactory results for turbines, gearing and propellers :—

If

R = revolutions of propeller per minute;
SHP = shaft horse-power of each shaft;
V = speed of vessel in knots on trial;

then, $$R = C\sqrt{\frac{V^6}{SHP}}.$$

VALUES OF C.

—	Direct Coupled.	Single Reduction Gearing.	Double Reduction Gearing.
Merchant vessels—ocean	12 to 13	7 to 8	5 to 6
,, ,, channel	13 ,, 14	9 ,, 10	5 ,, 6
Battleships	10 ,, 12	7 ,, 8	—
Light cruisers	11 ,, 13	9 ,, 10	—
Destroyers	13 ,, 15	7 ,, 8	—

RELATIVE I.H.P., S.H.P. AND COAL CONSUMPTION FOR ALL PURPOSES ON AN OCEAN LINER.

—	Reciprocating		Turbine		
	Triple.	Quadruple.	Direct.	Single Gear.	Double Gear.
Effective horse-power	5,000	5,000	5,000	5,000	5,000
Propulsive coefficient $\frac{EHP}{SHP}$	·6	·6	·50	·54	·6
S.H.P.	8,333	8,333	10,000	9,260	8,333
I.H.P.	8,950	8,950	—	—	—
Lbs. steam per S.H.P.	—	—	14·25	13·75	13·5
,, ,, ,, I.H.P.	17·1	16·2	—	—	—
Lbs. coal per S.H.P.	—	—	1·425	1·375	1·35
,, ,, ,, I.H.P.	1·71	1·62	—	—	—
Tons coal per hour	6·8	6·5	6·4	5·7	5·0

* See also 'Steam Turbines,' p. 1767.

PARTICULARS OF THE 36 HOURS COMPARATIVE TRIAL OF THE 'CAIRNROSS' & 'CAIRNGOWAN.

	'Cairnross' Geared Turbines.	'Cairngowan' Triple-expansion Engines.
Draught on trial, mean	23 ft. 8½ ins.	23 ft. 4½ ins.
Displacement on trial . . . . . tons	9,950	9,891
Revolutions per min. (mean for 36 hours)	61·76	61·68
Coal per day . . . . . tons	27·8	32·7
I.H.P.	—	1,790
S.H.P.	1,570	—
Ratio of S.H.P. 'Cairnross' to I.H.P of 'Cairngowan'	·877	—
Lbs. of coal per I.H.P. per hour, all purposes	(equiv.) 1·45	1·704
" " S.H.P. " "	1·65	(equiv.) 1·94
Estimated water consumption per hour, all purposes	22,000 lbs.	27,200 lbs.
" " " per I.H.P. hour	(equiv.) 12·3 lbs.	15·18 lbs.
" " " per S.H.P. hour	14 lbs.	(equiv.) 17·3 lbs.
Hot-well temperature	79° Fah.	104° Fah.
Feed temperature	203° Fah.	221° Fah.
Estimated lbs. of water per lb. of coal from feed temp.	8·5	8·97
Percentage of ash	12·5	9·36
Pressure, steam pipe in engine room	158 lbs.	175 lbs.
Initial pressure h.p. turbine	138 lbs.	—
Vacuum	28·75 ins.	26·8 ins.
Circulating water temperature, inlet	50° Fah.	50° Fah.
" " " outlet	70° Fah.	95° Fah.

COMPARATIVE RESULTS OF PARSONS SINGLE REDUCTION GEARED TURBINES AND RECIPROCATING ENGINES IN SIMILAR SHIPS.

Name of Vessel.	Type of Machinery.	Dimensions and Tonnage (Approx.).		Speed (Knots).	Coal per Day.	Saving in Coal for Turbines.
		Gross.	Displacement.			
					Tons	
(1) Vespasian .	(A) Turbine single screw .	275×38·75	2,150	4,350	9½	14·17	17%
	(B) Recipro. " " .	"	"	"	9½	17·07	
(2) Cairnross .	(C) Turbine single screw .	370×50¾	4,000	10,000	10·15	27·8	15%
Cairngowan	(D) Recipro. " " .	"	"	"	10·15	32·7	
(3) Mahanada .	(E) Turbine single screw .	470×58	7,200	13,500	12·138	54·8	19%
Malakuta .	(F) Recipro. " " .	"	"	"	11·533	67·4	
(4) Cumberland	(G) Turbine twin screw .	474×60	9,470	15,600	13·11	77·5	15%
Devon. .	(H) Recipro. " " .	473×60	9,661	"	12·8	91·5	

Note.—In 3 and 4 the speeds of the turbine vessels are higher than the corresponding vessels with reciprocating machinery. If the speeds had been equal the saving would have been greater than as recorded.

Arrangement of Turbines.

Turbine machinery may be arranged to drive one, two, three, or four lines of shafting.

In mercantile vessels with direct-coupled turbines three lines were adopted, the centre shaft being driven by a high-pressure turbine, the steam from which was exhausted into two low-pressure turbines driving the two wing shafts. Astern turbines were incorporated in the aft end of the l.p. ahead turbine casings, and during normal ahead running revolved idly in vacuum. A separate high-pressure steam connection was fitted to each astern turbine, and also to each low-pressure ahead turbine. When manœuvring in and out of harbour the wing turbines only were employed, each turbine being controlled independently of the other.

In some installations of turbine machinery in large mail steamers four shafts were employed and the steam expanded in series through an h.p. and i.p., and two l.p. turbines. This arrangement, while introducing more complication in the intermediate receiver connections, enabled a higher efficiency to be secured than in the usual three- or four-shaft arrangement.

In torpedo boats and torpedo destroyers the practice was to adopt a three-shaft arrangement similar to that described above. Most of the vessels fitted in this manner were provided with h.p. and i.p. cruising turbines in order to maintain economy at low speeds, and these turbines were cut out of action as the power increased.

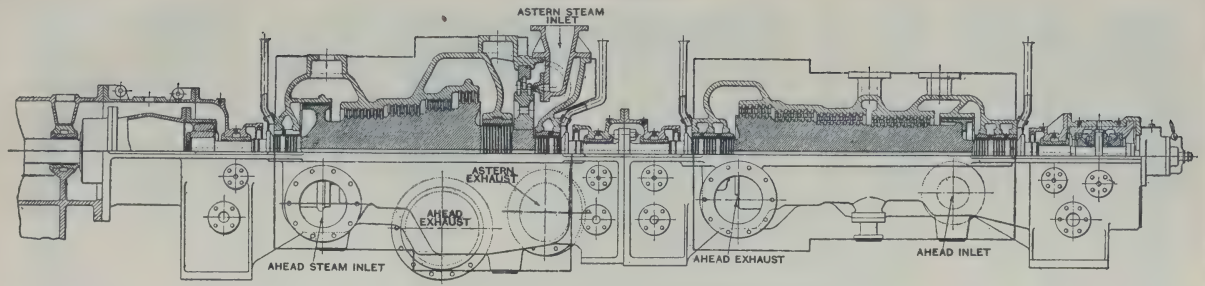


FIG. 8.—Parsons' Turbines, Section through H.P. and M.P. Turbines with H.P. astern incorporated.

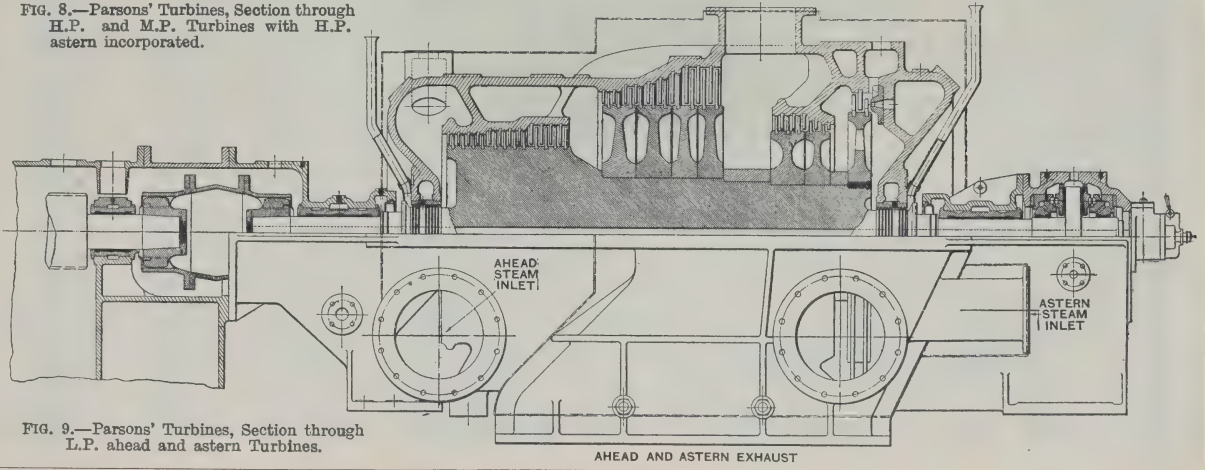


FIG. 9.—Parsons' Turbines, Section through L.P. ahead and astern Turbines.

More recent experience has shown that there are several objections to this arrangement, and in particular to the weight of these cruising turbines, and as a consequence the 'independent' type of turbine has been adopted. Where direct turbines are still fitted the installation usually consists of twin screws driven by turbines of the Parsons' or Brown-Curtis types, having the astern turbine incorporated in the same casing as the ahead turbine.

In light cruisers the machinery usually consists of two independent sets of single-geared turbines fitted in compartments separated by longitudinal or transverse bulkheads, and so arranged that in the event of damage to the one set the other will be available for use. Geared cruising turbines have also been fitted for low power.

In battle cruisers and battleships the turbine installation usually consists of two complete sets arranged in two or three compartments divided by longitudinal bulkheads. In the latter arrangement the h.p. turbines are placed in the wing compartments and the two l.p. turbines in a common central compartment, while in the former the bulkhead is placed between the l.p. turbines.

On each outer shaft there is a h.p. ahead turbine forward and an independent h.p. astern turbine aft. On each inner shaft there is a l.p. ahead turbine, having incorporated with it in the same casing a l.p. astern turbine. All four shafts are thus in use for manœuvring either ahead or astern, the steam exhausting from the h.p. ahead or astern to the l.p. ahead or astern turbine, as the case may be and from thence to the condenser.

With geared turbines each shaft is driven by an independent unit consisting of one h.p. ahead turbine and one l.p. ahead turbine with an astern turbine incorporated in the same casing with the latter.

With double-reduction gearing the high pressure unit is sometimes divided into an h.p. and i.p. in separate casings so as to reduce the length of rotor between bearings and enable higher revolutions to be run without 'whipping.' Where manœuvring requires greater astern power than can be transmitted by one pinion an h.p. astern turbine is fitted at the end of the h.p. or i.p. ahead turbine separately or in the same casing with a diaphragm to prevent astern steam entering the ahead turbine and vice versa. This arrangement is also preferred with superheated steam to minimise the change of temperature in the astern turbine when reversing.

For marine propulsion the two principal types which have been adopted are the Parsons and the Brown-Curtis. In the merchant service turbines of both types with single-reduction gear have been extensively fitted in high-speed steamers for cross-channel mail and passenger service, and the more recent development of double-reduction geared turbines is now being extensively adopted in vessels where the speed is lower, and on that account the propeller revolutions of single-reduction gear are not suitable.

The direct-coupled turbine is rarely fitted now owing to the many advantages and greater economy of the geared installations.

For naval service, so far as this country is concerned, either Parsons' turbines or Brown-Curtis turbines with single-reduction gearing are now being fitted in all battleships, cruisers, or torpedo-boat destroyers. With superheated steam it is found that an economy of about 1 per cent. in steam consumption is obtained for each 10° Fahr. of superheat.

With superheated steam and the latest aids to economy, turbine machinery is a successful rival to Diesel engines at large powers. The liner *Empress of Australia*, ex *Tirpitz*, originally fitted with turbo-hydraulic propelling machinery and water-tube boilers, was recently re-engined by the Fairfield S. & E. Co., Ltd., with single reduction geared turbines of the Parsons' type and cylindrical boilers with superheaters. The speed of the vessel was increased from 16½ to 19 knots and at the same time the fuel consumption decreased from 205 to 150 tons per day. The consumption of oil fuel for propelling purposes is less than 0·7 lbs. per s.h.p. per hour. As boiler fuel oil is usually much cheaper per ton than Diesel fuel oil, and the weight, first cost, and maintenance of a turbine installation less than Diesel's, turbine machinery is more economical and suitable for installations of this class for powers above 15,000 s.h.p.

The latest design of machinery for ocean liners consists of Parsons' turbines with single reduction gear, water-tube boilers of the Yarrow type generating steam at 500 lbs. pressure, and 750° F. temperature, stage feed heaters using steam bled from the turbines, and the majority of the auxiliary machinery electrically driven by current from Diesel-driven generators.

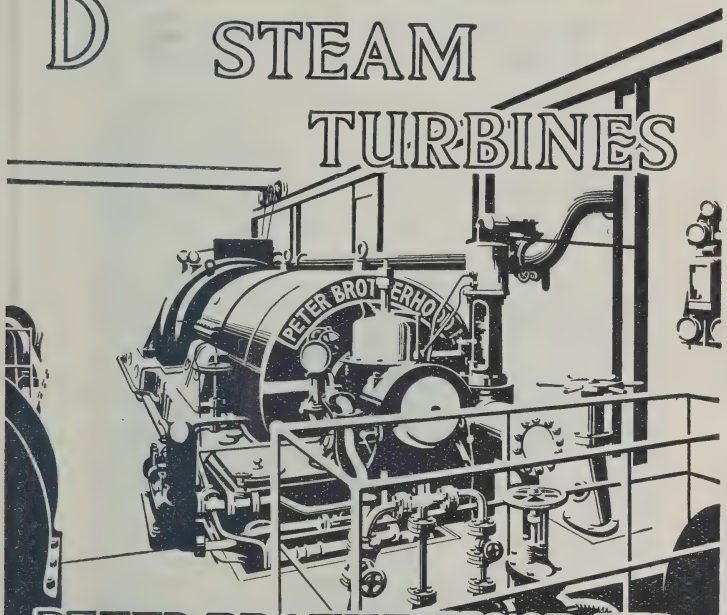
With this arrangement of machinery the consumption of oil fuel for all purposes does not exceed 0·65 lbs. per S.H.P. per hour. The earlier difficulties with water-tube boilers were largely due to impurities in the feed water, but these have been overcome by Mr. J. Johnson, of the Canadian Pacific Steamships, Ltd., by fitting one or two cylindrical boilers to generate steam for auxiliary and heating purposes, the condensate being used as make-up feed for the water-tube boilers. Condenser leakage has been practically eliminated by the use of Monel metal or cupro-nickel condenser tubes.

Parsons' Marine Turbines.

Longitudinal sections through typical Parsons' reaction turbines are shown in figs. 8 and 9, and these turbines would be used in the arrangement shown in fig. 13, page 2380, the H.P. and M.P. tandem turbine taking the place of the H.P. turbine shown in the figure.

A set of Parsons' turbines of 3,000 s.h.p. with double-reduction gearing is shown in figs. 10, 11, and 12, p. 2379.

BROTHERHOOD STEAM TURBINES



PETER BROTHERHOOD LTD.

PETERBOROUGH

Brotherhood Turbines, of the impulse type, are made for all powers up to 15,000 S.H.P. and for all conditions of service

Branch Offices :

34 Victoria Street
LONDON, S.W. 1

324 Harrogate Road
LEEDS

17 St. Ann's Square
MANCHESTER

48 West Regent Street
GLASGOW

PARSONS TURBINES

VERY HIGH ECONOMIES ARE BEING OBTAINED WITH HIGH-PRESSURE TURBINE INSTALLATIONS.

F. 2379

PARSONS' TURBINES.

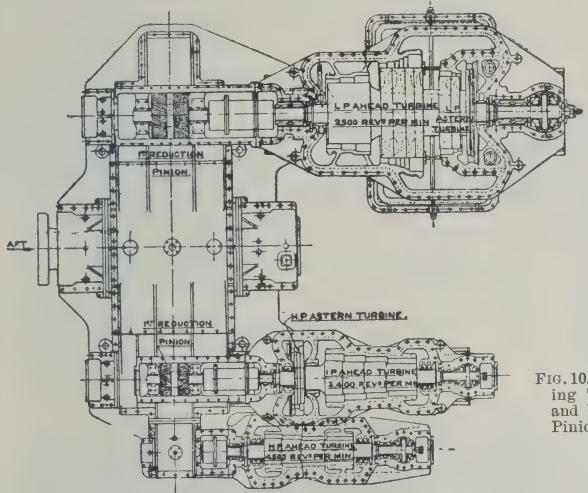


FIG. 10.—Plan View showing Turbine and I.P. and L.P. 1st Reduction Pinion Covers removed.

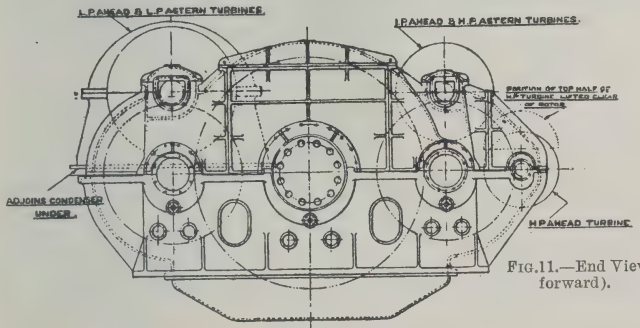


FIG. 11.—End View (looking forward).

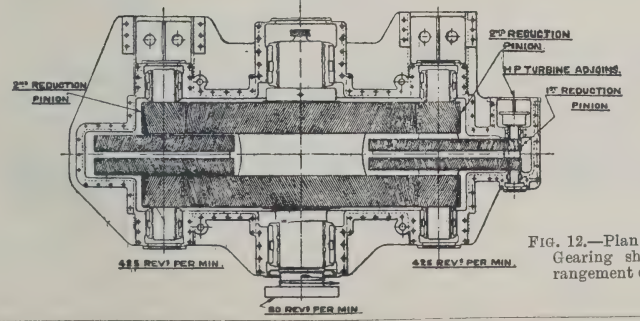


FIG. 12.—Plan View of Gearing showing arrangement of Wheels.

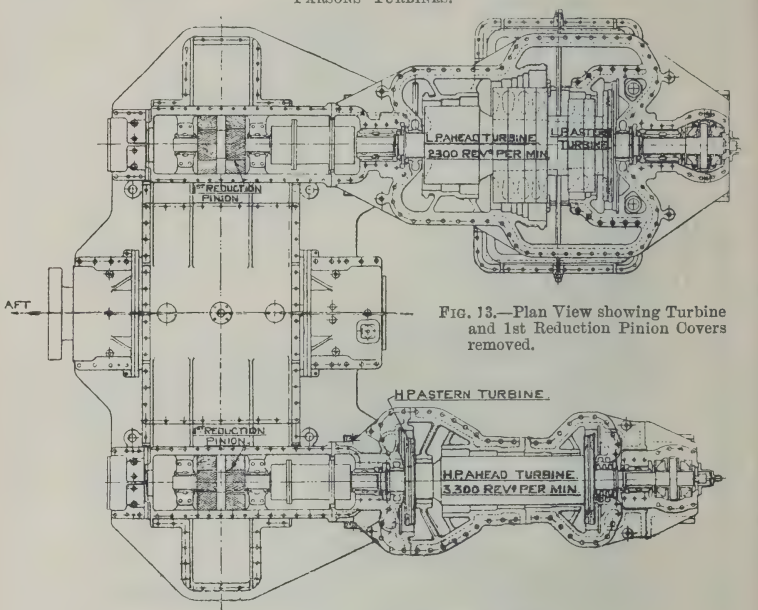


FIG. 13.—Plan View showing Turbine and 1st Reduction Pinion Covers removed.

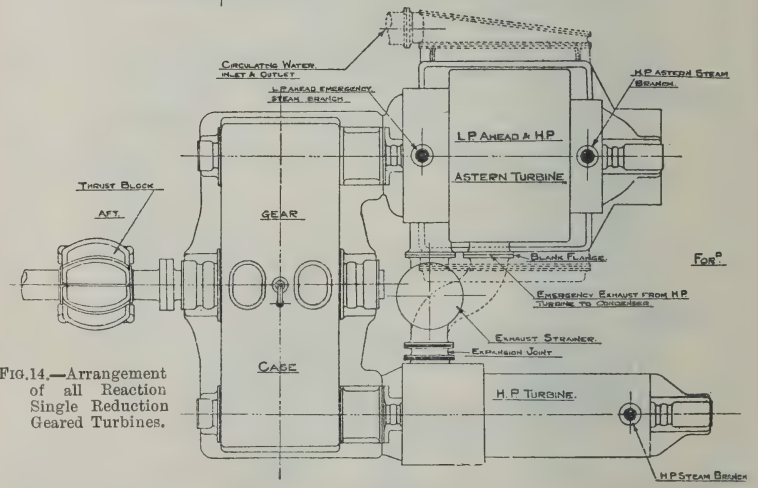


FIG. 14.—Arrangement of all Reaction Single Reduction Geared Turbines.

There is one h.p. and one i.p. ahead turbine each developing approximately one quarter of the total power and driving separate pinions geared to one first reduction gear-wheel. The l.p. turbine develops one half power and is geared to the first reduction gear-wheel on the opposite side of the main wheel. An h.p. astern turbine is incorporated in the same casing as the i.p. ahead turbine but separated from it by a diaphragm, and the l.p. astern turbine is also incorporated with the l.p. ahead.

Owing to the high revolutions at which turbines can run with double-reduction gearing, the desired blade speed can be obtained with a small diameter of rotor and consequently the end thrust due to steam pressure is low enough to be taken entirely on a thrust block of the pivoted type, thus dispensing with dummies and the leakage caused by them. In addition, the blade heights to pass the steam are greater than in large rotors of equal power, but as the clearance is not increased the percentage loss due to leakage past the tips is much less in the geared turbine. As a result of these and other advantages the steam consumption is much reduced from that required for direct-coupled turbines, and if superheat is used, say 200° F., the consumption for turbines only, exhausting to 28 inches vacuum, will be from 8 to 8½ lbs. per hour.

Figs. 11 and 12 show the arrangement of the double-reduction gearing for this set of turbines.

Fig. 13 shows an alternative arrangement with two turbines instead of three. There is one h.p. and one l.p. ahead turbine with a corresponding astern turbine in each casing.

This arrangement is simpler, but the consumption is slightly higher than in the three-turbine arrangement.

Fig. 14 shows a set of single-reduction geared turbines with one h.p. and one l.p. ahead turbine and an h.p. astern turbine incorporated with the latter.

This arrangement is suitable for low powers or where the propellers can be run at higher evolutions as in channel or shallow-draught steamers.

Brown-Curtis Turbines.

Figs. 15, 16, and 17 show a set of Brown-Curtis turbines for a double-reduction geared installation for one shaft of a liner.

The h.p. turbine is sometimes designed with three wheels each with two rows of buckets or moving blades, and at others with one two-bucket wheel and several single-bucket wheels. The m.p. and l.p. ahead turbines have single bucket wheels, and in the astern turbines there are usually two three-bucket wheels, or if a greater proportion of astern power is required, one three-bucket and two two-bucket wheels.

The turbines are arranged with the h.p. and m.p. in line and coupled together, driving one pinion of the gearing and the l.p. turbine driving a separate pinion. This arrangement enables the designer to choose the revolutions most suitable for each turbine.

The astern turbines are arranged with the first or h.p. stage in the low pressure end of the m.p. ahead turbine casing, but separated from it by a diaphragm, and the l.p. astern in the corresponding l.p. ahead casing exhausting through the common pipe to the condenser. The astern drive is thus distributed over both pinions, and especially with superheated steam the change in the casing temperature of the first stage when astern steam is turned on is not so great.

The casings and diaphragms are of cast iron, except where superheated steam is used, when the casing of the h.p. is of cast steel and the diaphragms of special bronze.

The rotors and wheels are of mild steel and specially designed to avoid 'whipping' and to withstand the stresses caused by the high speed of rotation. The blades or buckets are of special bronze capable of maintaining its strength at high temperatures, and the nozzles are formed of steel plates embedded in the diaphragms during casting. The rotors pass through glands at each end of the casing, which are packed with carbon rings to prevent the egress of steam at the high pressure or inlet of air at the low pressure stages.

The end thrust of the rotor is small and is taken up by a thrust block, usually of the Michell type.

Reduction Gear for Propulsion.*

The development of the turbine has led the way to the adoption of indirect transmission between the turbine and propeller, in order to utilise to the fullest extent the economy to be derived from running the turbines at high relative speeds and the propellers at reduced speed, as compared with direct-driven turbines. The importance of saving a few ounces of coal per h.p. per hour, and five per cent. or more in propeller efficiency in modern large power ships, either naval or mercantile, is obvious. In the former the radius of action with available fuel supply is increased, and in the latter a substantial reduction is made in the annual fuel bill.

The following systems have proved successful, and may be now considered beyond the experimental stage :—

(*a*) Parsons' mechanical gearing.
(*b*) Fottinger's hydraulic transformers.
(*c*) Electric transmission.

(*Continued on p.* 2386.)

* See also pp. 930, 1818.

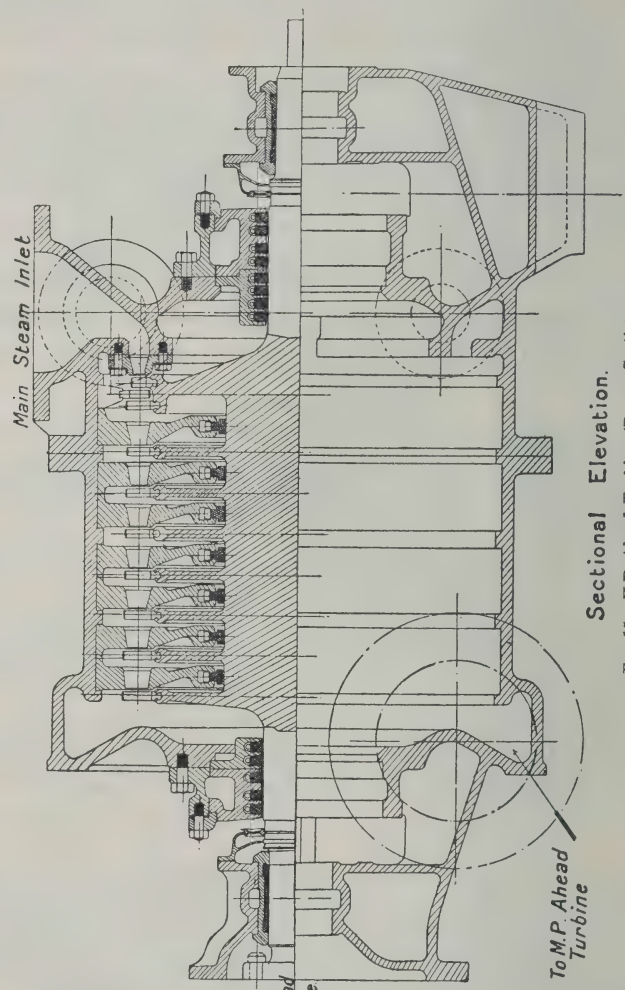


FIG. 15.—H.P. Ahead Turbine (Brown-Curtis).

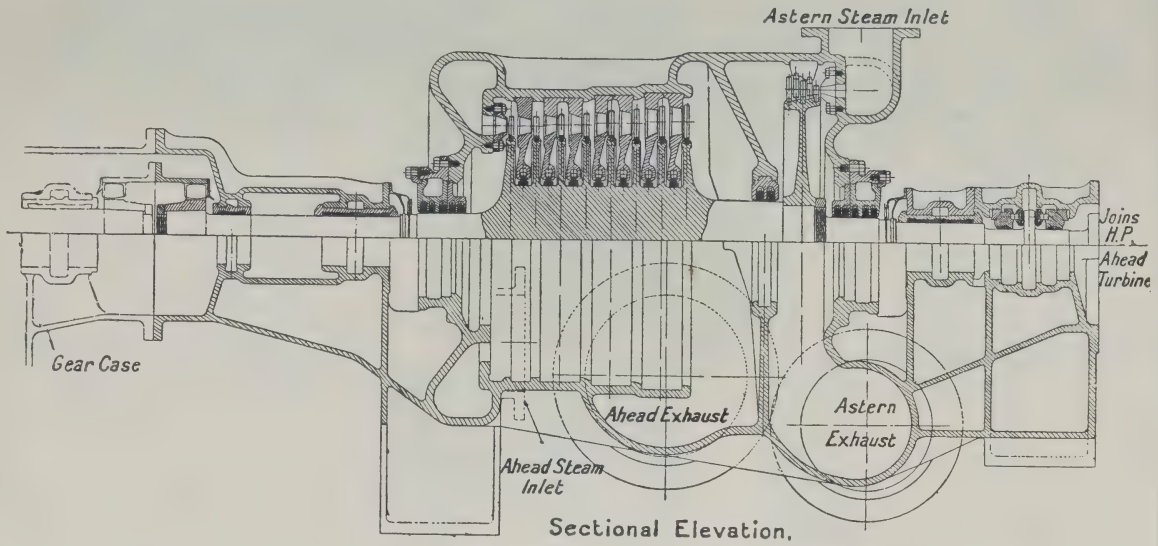


FIG. 16.—M.P. Ahead and H.P. Astern Turbine (Brown-Curtis)

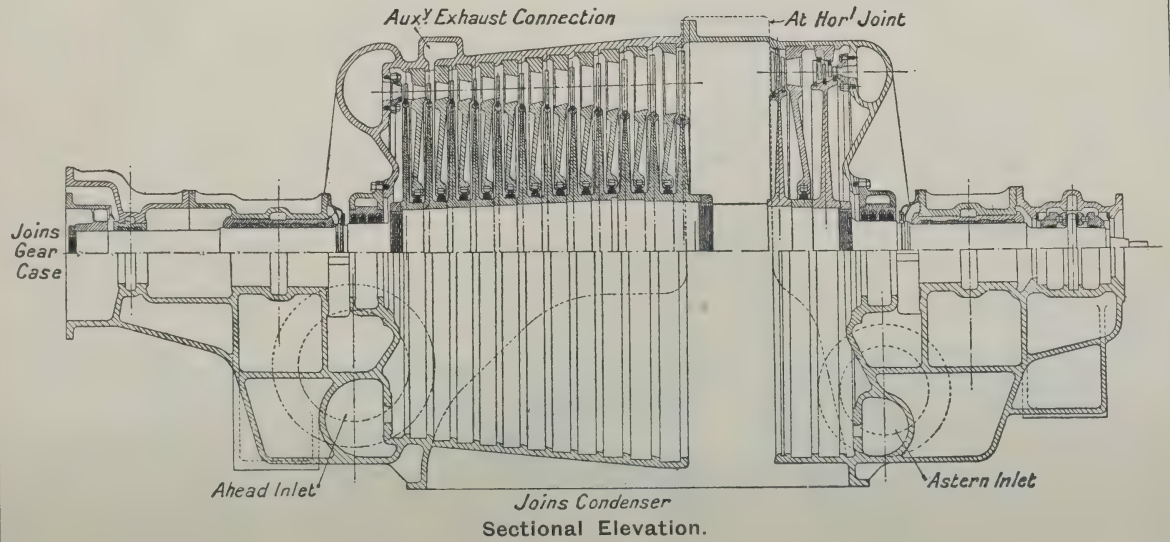


FIG. 17.—L.P. Ahead and Astern Turbine (Brown-Curtis).

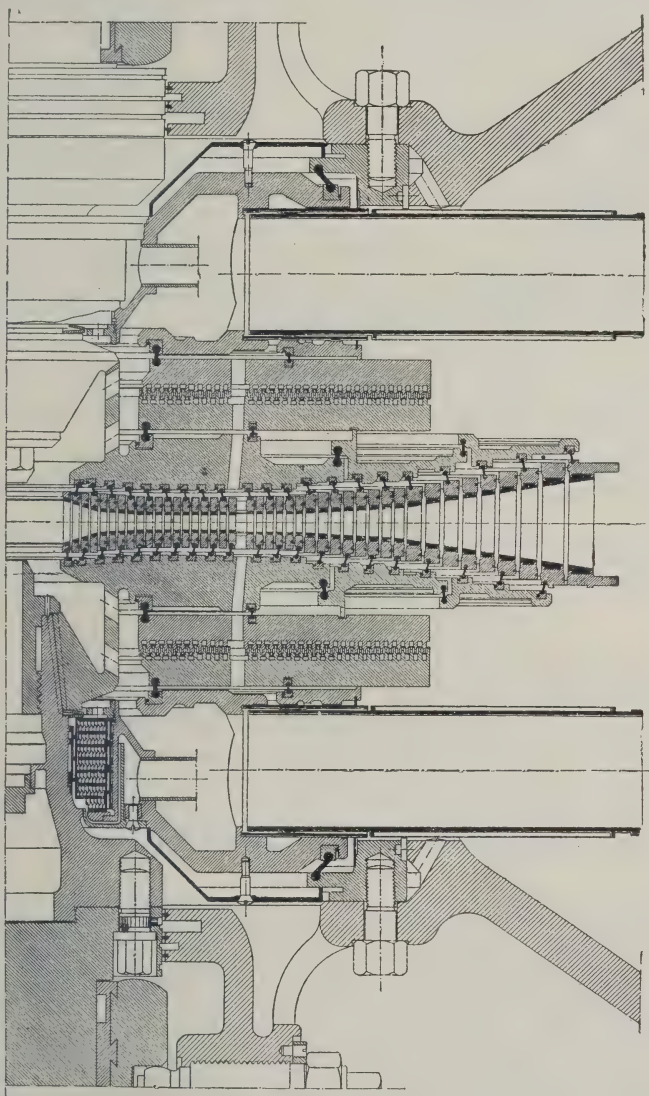


FIG. 18.—Longitudinal Section of a 1,000 K.W. Ljungström Turbine (Lower part).

The details of these systems are described in the following pages.

Each system will have certain fields of application where its particular advantages render its use desirable.

Some of the relative advantages and disadvantages of the systems are as follows:—

(*a*) MECHANICAL GEARING.

Advantages.

(1) Reduction in weight of installation from 10 to 15 per cent. as compared with direct-coupled turbines.

(2) Reduction in fuel consumption of 10 per cent. on same basis.

(3) High mechanical efficiency of gearing, 97 to 98 per cent. for double and single reduction gears respectively.

(4) Large speed reduction ratio permissible, so that greatest propeller efficiency can be obtained without impairing turbine efficiency.

(5) Simplicity of mechanism.

Disadvantages.

(1) Separate turbines for astern running are necessary which are only capable of developing about two-thirds of ahead power.

(2) Noise in engine room. This is very slight with the recent improvements in tooth cutting, and is not noticed outside of machinery space.

(3) Wear in teeth. With properly designed gears and properly cut teeth this is very slight.

(*b*) HYDRAULIC TRANSFORMERS.

Advantages.

(1) Astern turbines dispensed with and about 85 per cent. of ahead power available for astern running with reversible transformer.

(2) Absence of noise.

(3) Relatively no wear.

(4) Practically no limitation in power to be transmitted.

(5) Increased manœuvring power due to greater astern power and practically instantaneous reversal.

Disadvantages.

(1) Lower efficiency as compared with mechanical gearing, about 89 per cent.

(2) Limitation in speed reduction ratio, the maximum being about 6 : 1, and in consequence a reduction of efficiency of both turbines and propellers as compared with geared turbines, but not to the same extent as with direct-coupled turbines.

(*c*) ELECTRIC TRANSMISSION.

Advantages.

(1) Astern turbines dispensed with and full power available for astern working.

(2) Absence of noise.

(3) No wear and minimum cost of upkeep.

(4) No restriction in revolutions, so that highest efficiency of turbines and propellers can be realised.

(5) Great economy in multiple screw ships at low power, as one generator can be cut out entirely.

Disadvantages.

(1) Greater first cost than geared turbines.

(2) Higher consumption at full power than geared turbines—say 6 per cent.—due to transmission losses in generator and motor being greater than in gearing.

PARSONS' GEARING.*

This gearing consists of a large gear wheel with double helical teeth about $\frac{7}{8}$ in. pitch, coupled to the propeller shaft, and driven by either one or two pinions, coupled to the turbines. The connection between the pinion spindle and turbine spindle is by means of a claw coupling, which under running conditions permits of any slight lateral movement due to the expansion of the turbine rotor under steam being taken up. The gearing is completely enclosed in a cast-iron casing, which also forms the support for the bearings of the gear wheel shaft and pinion spindles. Jets of lubricating oil from the forced lubrication system impinge on the teeth at the point of contact of the pinion and wheel, and the spindles and shaft bearings are also connected to the forced lubrication system. The double helical gear ensures the transmission of pressure from one tooth to another without shock or vibration, and as the teeth are of true involute form complete line contact exists during the time they are in mesh. The pinions are made of nickel steel, having a tensile strength of about 40 tons per sq. in. with 25 per cent. elongation, and the gear wheels of carbon steel having a tensile strength of about 32 tons per sq. in. with

* See also pp. 930, 1818.

PARSONS' GEARING.

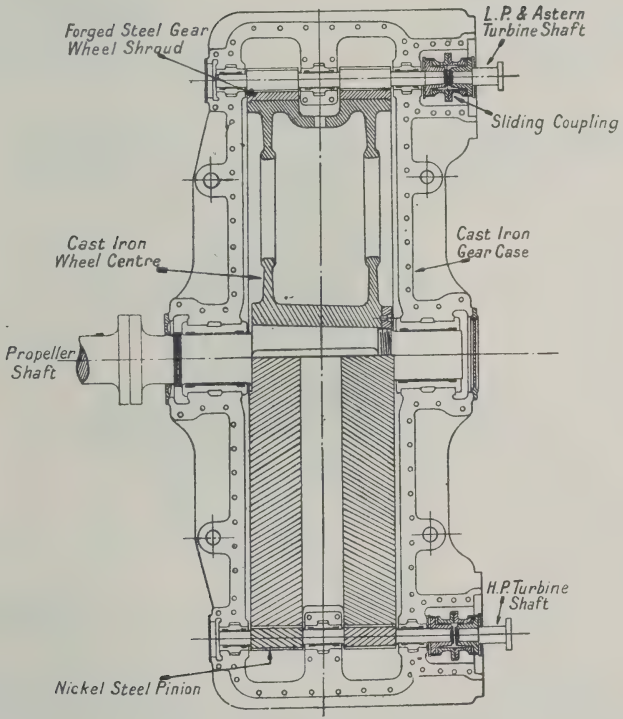


FIG. 19.—Plan with top casing removed.

26 per cent. elongation. Absolute accuracy in cutting the gearing is essential, and any corrective process or hand work after the gear is cut is inadmissible. The machines employed are the result of extensive and costly experiments, both in design and construction. In the double reduction gears the pitch of the teeth in the intermediate pinion and large wheel varies from $\frac{7}{16}$ to 1 inch.

With regard to efficiency, single reductions will go up to 98½ per cent., and double reductions up to 97 per cent.

The first steamer to be equipped with reduction gearing was the 'Vespasian,' a single-screw steamer of 2,150 tons. This gearing, as tested by Sir Charles Parsons, gave 98½ per cent. efficiency, and a fuel economy of about 13 per cent. was obtained. The cross-channel steamers, 'Normannia' and 'Hantonia' (with gearing made by The Parsons Marine Turbine Co., Ltd.), with a gear ratio of about 7 : 1, consumed only 43 tons of coal per trip, as compared with 70 tons per trip required by direct-driven turbine vessels of similar dimensions and speed, engaged in the same service.

Mechanical gearing for turbines has developed enormously in recent years, and is now being used extensively for ships of all powers and classes. Double reduction gearing makes it possible to run the turbines twenty or thirty times as fast as the propellers, with a consequent gain in overall efficiency.

Fottinger's Hydraulic Transformer.

The hydraulic transformer (fig. 20) consists of two centrifugal pump wheels, A, keyed to the primary shaft, which is driven by the turbine. On the secondary, or propeller, shaft three impulse reaction water turbine wheels are mounted, the after two, B and C, for ahead driving,

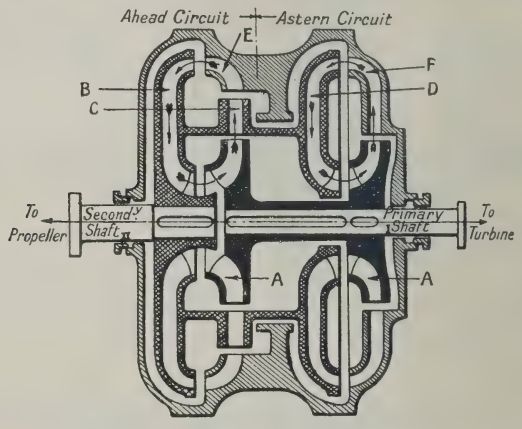


Fig. 20.—Fottinger's Hydraulic Transformer.

and the forward one, D, for astern driving. The wheels for ahead driving absorb the energy delivered to them from the water discharged by the pump wheel in two stages, the primary stage, C, being connected to the astern water turbine wheel, D. Having given up part of its energy in the primary ahead stage, the water moves in continuous compact flow through guide vanes, E, in the casing to the second stage ahead wheel, B, which absorbs the remaining energy in the water. The discharge from the second stage wheel flows back to the beginning of the circuit in the pump wheel, giving up to it the remaining energy of discharge. In the same way the astern driving wheel, D, receives its energy from the water discharged by the forward pump wheel guided into it by the stationary vanes, F, in the casing.

The vanes in the ahead and astern turbine wheels are spaced in opposite directions to each other, thereby giving the reversed motion.

A change valve, controlled by a lever, admits the water from either pump wheel to the corresponding turbine wheel to give the desired direction of rotation.

The wheels and vanes of the transformer are made of bronze, and the casings of cast iron.

The efficiency of the transformer decreases with an increase of speed reduction ratio on account of the higher velocities causing increased fluid friction.

Fresh water is used in the system, and on account of the work done upon it by friction during its passage through the vanes heat is generated. Part of this heat, which would otherwise be lost, is saved by using the water for feeding the boilers, the quantity of water in the transformer system being maintained by filling up with colder water from the air-pump discharge.

The Vulcan Diesel Hydraulic Drive.

The Vulcan gearing, fig. 21, which transmits the power of two Diesel engines to a common propeller shaft, consists of a large helical gear wheel mounted on the propeller shaft, and in which two pinions engage with gear ratio of about 3 : 1. These pinions are connected through liquid clutches to the crankshafts of two Diesel engines. The clutches, built under the Vulcan-Föttinger patents of the Vulcan Werke, each consist of a compartment for running ahead and a compartment for running astern.

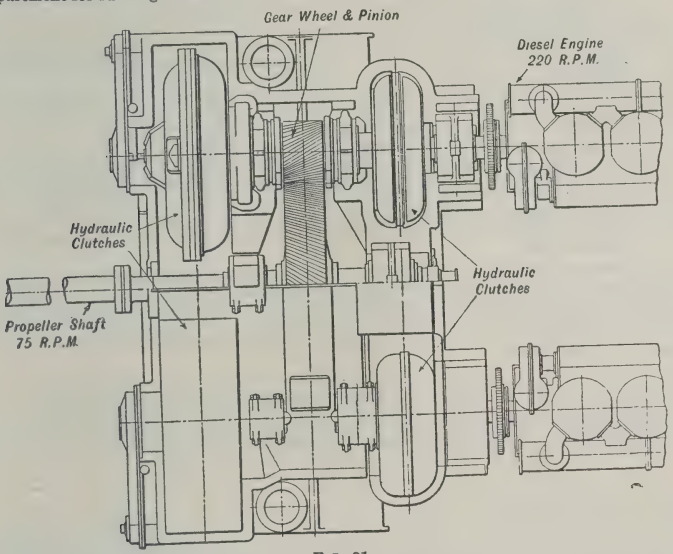


Fig. 21.

The engines are reversed by alternately filling and emptying the ahead and astern compartments by means of a reversing gear operated from the engineer's platform controlling the centrifugal pumps driven from the primary shafts. The driven pinions, and consequently the propeller shaft, run in the same or in the opposite direction to that in which the engine shafts rotate, according to whether the clutches for forward or for astern running are filled with liquid. When the propeller is reversing the Diesel engines require no attention as they run permanently in one direction.

The controlling mechanism is so arranged that the liquid or hydraulic clutches of both the Diesel engines are operated simultaneously. In the event of one of the engines requiring to be stopped during working, an arrangement is provided whereby the two compartments of its respective clutch are emptied and the engine cut out.

The efficiency of the Vulcan gearing is in normal ahead running 97 per cent. The power available for astern running is 65 per cent. of the ahead.

The gearing and the clutches are built together, forming one complete unit in a common cast-iron casing. The hydraulic clutches are of cast iron. The pinions and the wheel rim, which is shrunk on to the cast-iron body of the large gear wheel, are of special steel. The teeth are cut on a precision milling machine by the involute process. The shafts are of Siemens-Martin steel; all bearings are lined with white metal. The front bearing of the large gear wheel is constructed as a thrust bearing and takes up the thrust of the propeller shaft so that no additional propeller thrust block is required.

The liquid used for operating the clutches is lubricating oil. The bearings and teeth of the gear wheels are lubricated with the same oil by which the liquid clutches are operated. The lubricating oil and the oil which drips from the bearings of the gearing and from the clutches is caught in a collecting reservoir placed underneath the gearing. It is drawn from this collecting reservoir by a lubricating oil pump which pumps the oil through a filter and a collector back to the bearings of the gearing and to the teeth of the gear wheels. Oil-circulating pumps attached to the Diesel engines also draw the oil from this reservoir and deliver the oil to the compartments of the clutches as required.

Driving by high-speed Diesel engines and Vulcan gearing possesses *inter alia* the following advantages over direct drive :—

(1) Less weight, lower first cost and less space, particularly in height, with consequently better utilisation of the available space in the ship.

(2) As two Diesel engines drive on to each propeller shaft, the reliability at sea is increased in consequence of this engine standby, and it is therefore a practical proposition to build ships of 5,000 s.h.p. with a single propeller shaft.

(3) Each of the engines can be disconnected from the helical gear wheel by emptying its corresponding liquid clutch. The engine is started working again by simply refilling the liquid clutch so that it is picked up and carried on by the engine, which remains running.

(4) The engines are of the single-acting, non-reversible, four-cycle type. The thermal strains due to admission of cold air to the hot cylinders are obviated, as the engines run continuously in one direction when manœuvring.

(5) The liquid clutches have the property of completely eliminating all irregularities in the turning moment of the Diesel engines not only during normal working, but also at the critical torsional speeds, as the clutches transmit a uniform turning moment, otherwise only attained by turbine drive.

(6) The vessel may be run with as low a number of revolutions of the propeller as may be desired (five to twenty revolutions per minute), which is of importance when manœuvring.

Installations are being constructed by the Vulcan Company for two 10,000-ton d.w. single-screw vessels of 4,000 b.h.p. for the Hansa and the Hamburg Australian Lines. In these instances the propelling units will be of the four-stroke cycle, single-acting type, conforming to marine practice in having crossheads, but utilising steel castings for the main structural parts of the engines, such as bedplates, columns, and cylinders. Four engines can be applied very conveniently to a single shaft.

Electric Transmission of Power for Propelling Machinery.

The application of electricity for transmitting power between the prime mover and the propeller may be divided into two classes—viz. Turbo-electric and Diesel-electric.

TURBO-ELECTRIC DRIVE.

The electrical equipment required is an alternating current generator suitable for direct coupling to a high-speed turbine, a motor of suitable speed for direct coupling to the propeller shaft, a direct current exciter or an auxiliary generator, from which direct current can be obtained, and suitable control gear. The whole can be regarded as a reduction gear having a ratio of reduction determined by the relative number of poles on the motor and generator. Thus with a two-pole generator and a sixty-pole motor the ratio of reduction would be 30 to 1, and, as with mechanical gearing, any variation in speed is obtained by varying the speed of the prime mover.

Two types of alternating current motors have been used for driving propellers :—

(*a*) Induction machines, both slip-ring and squirrel-cage types.
(*b*) Self-starting synchronous motors.

All things considered, the self-starting synchronous motor is the most suitable for this class of work on account of the large air gap, simplicity of construction, accessibility of windings, high power factor, and efficiency. When starting from rest the synchronous motor functions as a squirrel-cage motor with its field unexcited.

s.h.p. account of its greater flexibility direct current is preferable for vessels up to about 3,000 On on one shaft ; beyond this power, owing to the limitations of voltage, the cables become excessively large.

Electrically driven ships in service have demonstrated their unquestionable reliability; the main driving power is transmitted magnetically across large air gaps, giving an elastic medium for absorbing shocks, and thus making it the simplest and most reliable form of speed reduction.

The turbine is of simple construction, being built in one casing and directly connected to the generator, thereby reducing the size of the seatings and eliminating distortion and misalignment troubles due to local straining of the ship. Bearings, shafts, shaft packings, valves and piping are reduced to a minimum, thus decreasing the friction and leakage losses. The astern turbine

is eliminated entirely in an electrically driven vessel, full power astern being available without reversing the direction of rotation of the turbine.

There is an absence of vibration and noise, as the machinery is perfectly balanced, with no rubbing contacts, and the quietness of the running is a feature of great importance in a passenger vessel. Manœuvring is very efficient with this type of power transmission, so that perfect speed control in heavy seas is maintained without danger of racing.

The cost of upkeep, as has been proved by ten years' experience with electrically driven vessels, is much less than with any other type of reduction gear.

The propeller efficiency may be kept very high as the electric drive lends itself practically to any desired speed reduction between the turbine and propeller, and each may therefore be designed to operate at a speed corresponding to its highest efficiency.

DIESEL-ELECTRIC DRIVE.

The Diesel-electric drive will be most suitable for double purpose vessels such as fruit and meat carriers, where a large amount of power is required for operating refrigerating plant, etc., and for dredgers, cable-laying ships, self-propelled floating cranes, and vessels of similar types.

In such vessels it is possible, by the application of electricity, to use the same prime mover for both propelling and auxiliary purposes, also to have a number of comparatively small engines each driving a generator, and a single motor direct coupled to the propeller shaft, which can thus be run at the most suitable and efficient speed irrespective of the speed of the prime mover, the electrical losses being largely offset by the more efficient propeller.

The Diesel engines for electric transmission are simpler in design than those which are coupled directly to the propeller shaft, as the engine speed has no relation to the propeller speed, and as the engines run in one direction only. The large air storage tanks, also, can be eliminated.

With the electric drive accurate and constant knowledge of the power output is always obtainable from meters on the control board; and in this manner the power required at any draft or trim under all weather conditions, the effect of dirty hull on the ship's resistance, etc., can be ascertained and studied.

Rules for the Dimensions of Steamship Shafts.
See p. 1005.

BOILERS.

In determining the principal dimensions of a boiler, or boiler installation, consideration must in the first instance be given to the system of draught employed, whether it is to be forced or natural draught, and since the quantity of steam which a boiler is capable of generating depends on the weight of coal consumed, it follows that the dimension which must first be decided is that of the grate area, the heating surface being proportioned from the grate area depending upon the type of boiler employed. The efficiency of the boiler largely depends on the rate of evaporation per square foot of heating surface, which should be kept as low as possible.

Coal Combustion.

The weight of coal which can be burnt per square foot of grate area per hour under varying conditions of draught is approximately:—

Natural draught . . .	16 to 20 lbs.	Forced draught 2 ins. air pressure	45 to 50 lbs
Forced draught ½ in. air pressure	20 to 25 lbs.	„ 3 ins. „	55 to 60 lbs
„ 1 „	30 to 35 lbs.	„ 4 ins. „	60 to 70 lbs

With cylindrical boilers and water-tube boilers for cruisers and battleships the maximum combustion per square foot of grate rarely exceeds 40 lbs. when working with closed stokeholds and 30 lbs. under Howden's system of forced draught; with the express type of water-tube boiler as fitted in torpedo boats and torpedo boat destroyers the combustion varies from 60 to 80 lbs. of coal per square foot of grate.

Boiler Surfaces.

The heating surface and grate area for a required boiler is determined by the method of draught employed, whether natural or forced, the kind of coal used on account of the difference in calorific values of different varieties of coal, and the type of engine to which steam is being supplied, quadruple expansion engines, for instance, requiring less boiler surface than compound engines on account of the greater economy in steam consumption per I.H.P. of the former.

Economy in coal consumption is obtained by working the boilers easily at low rates of evaporation. Where considerations of weight and space are of importance, as in naval and fast merchant vessels, the boilers are reduced in size, and forced draught is used to increase the rate of evaporation, with corresponding reduction in efficiency. In Howden forced draught the economiser effect of

the air tubes in the uptakes increases the efficiency so that this system compares favourably with natural draught and is much superior to the closed stokehold system.

The following table gives the average H.P. developed per square foot of grate, and the ratio of heating surface to grate area of different types of boilers and average consumptions of coal for various types of engines.

TABLE SHOWING PARTICULARS OF BOILERS OF VARIOUS TYPES OF STEAMSHIPS.

Type of Ship.	Type of Engine.	Type of Boiler.	Draught.	Coal per H.P. per Hour in Lbs.	H.P. per sq. foot of Grate.	Ratio Heating Surf. Grate Area.
Merchant	Compound	Cylindrical	Natural	2·0 to 2·3	8 to 7	30 to 25
,,	Triple	,,	,,	1·6 ,, 1·8	10 ,, 9	35 ,, 30
,,	,,	,,	Howden	1·5 ,, 1·7	15 ,, 13·5	40
,,	Quadruple	,,	Natural	1·5 ,, 1·6	11 ,, 10	35 to 32
,,	,,	,,	Howden	1·45 ,, 1·55	16·5 ,, 15	40
,,	Combination	,,	Natural	1·3 ,, 1·4	12·5 ,, 11·5	35 to 32
,,	,,	,,	Howden	1·25 ,, 1·35	17·5 ,, 16·5	40
Fast Passenger and Mail Steamers	4 Crank Triple	,,	Closed Stokeholds	1·7 ,, 1·9	20 ,, 15	35 to 30
,,	,,	,,	Howden	1·6 ,, 1·8	18 ,, 15	40 ,, 36
,,	Quadruple	,,	Closed Stokeholds	1·55 ,, 1·7	17 ,, 15·5	35 ,, 30
,,	,,	,,	Howden	1·45 ,, 1·6	17 ,, 15·5	40 ,, 36
,,	Turbine, Direct	,,	Closed Stokeholds	1·4 ,, 1·6	19 ,, 16·5	35 ,, 30
,,	,,	,,	Howden	1·3 ,, 1·5	19 ,, 16·5	40 ,, 36
,,	Turbine Single Gear	,,	Natural	1·35 ,, 1·45	12·5 ,, 11·5	35 ,, 32
,,		,,	Howden	1·3 ,, 1·4	17 ,, 15·5	40
,,	Turbine Double Gear	,,	Natural	1·3 ,, 1·4	13 ,, 12	35 to 32
,,		,,	Howden	1·25 ,, 1·35	17·5 ,, 16	40
Paddle	Compound	,,	Natural	2·0 ,, 2·3	8 ,, 7	32 to 28
,,	,,	,,	Howden	2·0 ,, 2·3	17 ,, 15	40 ,, 36
,,	,,	,,	Closed Stokeholds	2·2 ,, 2·5	14·0 ,, 11	32 ,, 27
Battleships and Cruisers	Turbine	Water Tube (Babcock-Wilcox)	Closed Stokeholds Moderate forced	1·3 ,, 1·45	16 ,, 14·5	35 ,, 33
,,	,,	Water Tube (Yarrow)	Closed Stokeholds	1·35 ,, 1·5	25 ,, 22·5	60 ,, 55
Torpedo Boat Destroyers	,,	Water Tube (Express)	Closed Stokeholds	1·9 ,, 2·1	35 ,, 32	60 ,, 55
,, ,,	,,	,,	,,	Oil fuel 1·0 ,, 1·1	H.P. per cub. foot of combust'n chamber 9 to 7·5	Heating Surf. cub. ft. of combust'n chamber 9 to 8

Draught.

The two systems of forced draught most commonly adopted are closed stokeholds and Howden's system (page 2124) with closed ashpits. In the former an air pressure is maintained in the boiler rooms by forcing in air with fans, all openings in the boiler room are closed down with doors, and air locks fitted for access purposes. With Howden's system the boiler room is open, no air locks being required, and the air is forced by fans over the surface of a nest of tubes, or Corrugated Plate Air Heater situated in the uptakes, through which the products of combustion pass on the way to the funnel. The air is raised to a temperature of about 250° F., or over, by the air heater, and passes through passages on either side of the uptake into a reservoir surrounding the furnace fronts, from which reservoir the air is admitted to the furnaces by valves in the furnace fronts, at the sides and the top, to the ashpit and above the fire respectively. Valves are fitted to control the admission of air above and below the fires: these must be shut before opening the furnace door to fire the boiler, otherwise the flame would be blown out into the stokehold. The proper use of the top valve prevents excessive smoke without reducing the boiler efficiency or the amount of evaporation per lb. of coal. Special strips of twisted steel

HOWDEN FRONTS for OIL FUEL

HOWDEN FANS and ENGINES or MOTORS

OVER 28,000,000 I.H.P. NOW FITTED

Specify

HOWDEN'S

PATENT FORCED DRAUGHT

To Howden's Drawings

Fitted in the

Leading Steamships of the World

JAMES HOWDEN & CO., LTD
SCOTLAND STREET, GLASGOW

HOWDEN DRAWINGS for BEST RESULTS

HOWDEN SUPER-VISION to ENSURE SUCCESS

F. 2392

TURBOVANE
FANS FOR ALL PURPOSES

The illustration shows a turbovane fan (without the vane control previously shown in these pages).

These fans are made in an extensive range, of sturdy construction and high static efficiency.

The majority of installations as supplied to date in this country are naturally for mechanical draught in boiler plant, but in addition we have supplied fans for railway tunnel ventilation, road tunnel ventilation, and also for gas circulation.

HOWDEN-LJUNGSTROM PREHEATERS (LAND) LIMITED
CAXTON HOUSE, WESTMINSTER, LONDON, S.W.1

HOWDEN
LJUNGSTROM
AIR PREHEATERS

HOWDEN
STURTEVANT
TURBOVANE FANS

ribbon called 'retarders' are placed in each boiler tube, causing the hot gases traversing the tubes to take a spiral course and be retained longer in close contact with the tubes than they otherwise would.

The latest type of furnace fronts has the side valves controlled by one handle, and is also fitted with a safety guard, so that the furnace doors can only be opened when the air valves are shut, thereby preventing the possibility of any flame coming back through furnace door when it is opened.

Fronts have also been developed for firing powdered coal, in conjunction with various patentees of powdered fuel firing systems. A specialty of the Howden type of furnace front for this class of work is the double ashpit door reducing the radiation to a small figure and giving much pleasanter stokehold conditions, while the circulation of a regulated subsidiary air supply around and through the inner door eliminates 'panting' and has a cooling effect, thus maintaining its length of service under otherwise onerous conditions.

On account of the abstraction by the air of what would otherwise be waste heat passing up the funnel a considerable measure of increased boiler efficiency is attained compared with the closed stokehold system, and the uniform condition of furnace temperatures which Howden's system ensures tends to reduce the wear and tear of the boiler.

A more uniform efficiency of furnace can be maintained with forced draught and it may be readily controlled to suit any desired rate of combustion, being quite independent in this respect of prevailing atmospheric conditions.

The Howden-Ljungström Air Preheater.

One of the most important inventions for the reduction of fuel consumption is the air preheater, which may be considered as a development of the Howden air-tube box. By this means 70 per cent. of the heat in the waste gases from the boiler can be recovered, so that a very considerable increase in efficiency is obtained without the disadvantages of excessive weight or corrosion as in the case of feed-water heaters.

The principle involved is exceedingly simple and effective. Briefly described, a large transmission surface is assembled in a rotating part in such a way that the elements are alternately traversed by the waste gases from the boiler and by the cold air to be used for combustion. The gases give up their heat to the elements, and, as these move round into the current of cold air, the heat previously absorbed from the gases is given up to the air. The process is continuous.

In actual construction, the elements consist of thin plates of metal placed close together and corrugated in order to obtain a very large surface. They are fixed in a rotor in such a way that the passages for the gas and the air between the places are parallel with the axis of rotation. The rotor is supported upon disc rollers fixed in a stationary envelope or casing, so that the power required to drive it is very small.

The casing is provided with inlets and outlets for the gases and for the air, which are separated from each other by diametral partitions above and below the rotor. In the upper section of the casing two fans are provided, one to force the cold air through the elements which are at the moment travelling through the air side of the heater, and the other fan to suck the gases through the elements on the gas side. The two fans are mounted on one shaft which passes through the upper partition and which can be driven by an electric motor or in any convenient way, the motor drive being taken by suitable gearing.

A most important feature of the heater is the ease with which the surfaces of the elements can be kept clean. A radial pipe with steam jets is fitted below the rotor, on the gas side, and steam is discharged while the rotor is revolving, so that every portion of the elements must pass through the jets, and no part of the surface can escape the scouring action of the steam. Every particle of soot is blown away, and the efficiency of heat transmission is thus maintained constant. It is necessary to operate the jets for fifteen seconds only every few hours, so that the consumption of steam for this purpose is negligible.

Figs. 22 and 23 show the arrangement of the air preheater with two cylindrical boilers, but it is also suitable for using with water tube boilers and furnaces of various types.

An arrangement in which the axis of the preheater rotor is horizontal and separate centrifugal fans are fitted has now been introduced as an alternative. In this case the fans may be situated remote from the preheater, possibly in the engine room or in a separate fan room as circumstances may decide.

The Howden Corrugated Plate Air Heater.

(Patent.)

The Howden Corrugated Plate Air Heater is built of a number of elements each, of which is composed of two corrugated steel sheets. The sheets are welded together at the top and bottom and thus brought to sharp edges, and at the ends the plates are offset to give the required spacing between each pair of elements. The ridges of the corrugations of the two plates forming an element are set opposite each other, the air passing through the tubes thus formed while the flue gases pass through sinuous paths between the various pairs of elements. On the air side the heater may be arranged multiflow by re-passing the air through the tubes on the counter flow system. On the gas side the heaters are generally single flow, and the confined tortuous paths through which the gases pass have been proved to give an exceedingly high rate of heat transmission.

THE HOWDEN-LJUNGSTRÖM PATENT AIR PREHEATER.

FIG. 23.

FIG 22.

For pulverised fuel firing, the heaters are adapted to pass the primary and secondary air, respectively, through a different number of passes so that any desired temperature can be obtained for either and, also, the heaters may be subdivided to furnish primary and secondary air at different pressures, or furnish primary air under suction and secondary air under forced draught, or *vice versa*.

The Howden Cellular Air Heater.

The cellular air heater embodies all the features of the tubular air heater with the following additional advantages :—

1. The heating surface for similar space occupied is approximately double that of a tubular heater.
2. The gases and air travel substantially in a contra-flow direction.
3. Due to 1 and 2, a higher transfer of heat from the flue gases is obtained.

The heating elements or cells are made of mild steel plate. Each element is carefully manufactured and gauged before assembling. Special precaution is taken to prevent warping of plates and each cell is made truly parallel.

The guide vanes in the air passages are so arranged to divide the incoming air into an equal number of streams which flow directly opposite to the path of the gases on the opposite sides of the elements. These guide vanes are made of steel strips hung on to ferrules secured to the heating elements. The vanes are designed to give the air a turbulent flow. By distributing the air equally over the surface of the plates in the manner mentioned, it ensures that there can be no short circuit of air and thus avoids any part of the plate becoming overheated.

The complete heater when assembled consists of a number of portable sections securely bolted together and which can at any time be removed for repair if necessary.

Furnaces and Firebars.

If forced draught is employed it is not desirable to have firebars longer than 6 ft. on account of the tendency for large quantities of air to pass through the firebars if the fire should burn into holes. With Howden's system 5 ft. 6 ins. is the maximum length of firebar generally adopted. In naval practice with modern water-tube boilers working at full power under a moderate forced draught of 0·5 in. to 0·75 in. of air pressure, grate bars having a maximum length of 7 ft. are frequently employed ; this, however, requires skilful firing, and extreme care to ensure the boiler efficiency being maintained at a maximum. For natural draught the length of grate bar should not exceed 6 ft. 6 ins., but increased boiler efficiency will be attained by using shorter bars, on account of the greater control over the fires. The diameter of furnaces for cylindrical boilers (figs. 26 and 27) should not be less than 32 ins., nor greater than 48 ins., and the number of furnaces which can be fitted in boilers of the following dimensions is :—

Boilers up to 9 ft. 0 ins. diameter	1 furnace	Boilers up to 15 ft. 6 ins. diameter	3 furnaces
„ „ „ 13 ft. 0 ins. „	2 furnaces	exceeding 15 ft. 6 ins. „	4 furnaces

The grate area of a cylindrical boiler is equal to :—Length of firebars × mean diameter of furnace × number of furnaces.

Length of Cylindrical Boilers.

The length of a cylindrical boiler is determined by the length of combustion chamber tubes or the furnace which is practically the same length as the tubes, the length of the combustion chamber and the water spaces at their backs. The combustion chamber measured horizontally should be as long as possible, and usually is about 30 ins. in single-ended boilers and from 50 to 57 ins. in double-ended boilers. The water space between the back of the combustion chamber and the back end-plate of the boiler should be from 5 to 7 ins. at the bottom and wider at the top, the taper being about $\frac{1}{2}$ in. per foot on the combustion chamber back.

Heating Surface.

The heating surface of cylindrical boilers consists of the external surface of the tubes, the length being measured between the tube plates, the surface of the furnaces above the firebar level, and the surface of the combustion chambers. The surface of the front tube plate on the boiler end, although in contact with the hot gases, is not usually reckoned as effective heating surface, and should not be included in the total heating surface of the boiler. The heating surface of a water-tube boiler is the total external surface of the tubes in contact with the fire and hot gases.

Diameter of Tubes for Cylindrical Boilers.

In forced-draught boilers the combustion chamber tubes are generally $2\frac{1}{4}$ or $2\frac{3}{4}$ ins. diameter, and in natural-draught boilers the diameter is 3 ins. to $3\frac{1}{2}$ ins., the increased sectional area being necessary to reduce the resistance to flow of the products of combustion.

Cylindrical Marine Boiler.

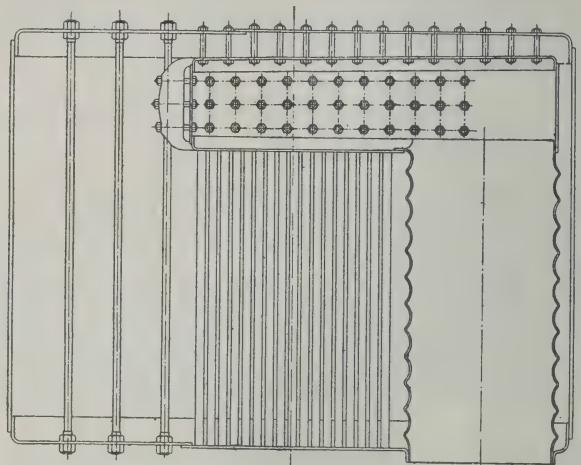


Fig. 25.

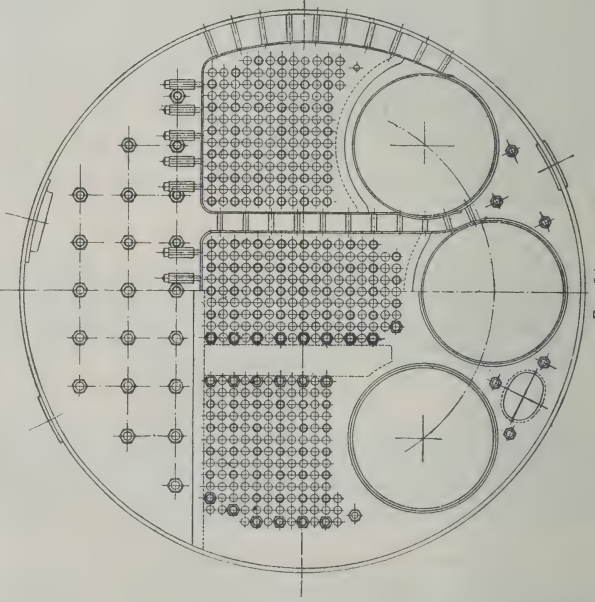


Fig. 24.

Thickness of Cylindrical Boiler Shells.

The shell plates of cylindrical boilers must be of the following thickness to satisfy the requirements of the revised rules of the Board of Trade, Lloyd's, British Corporation, and Bureau Veritas.

The thicknesses shown assume the steel to have a minimum ultimate tensile strength of 28 tons per sq. in., and the longitudinal joint to have double butt straps with an efficiency of riveting equal to 84 per cent.

THICKNESS OF SHELL PLATES IN INCHES.

Internal Diameter.	Working Pressure, lbs. per sq. in.							
	220	210	200	190	180	170	160	150
10 feet	1·027	0·983	0·940	0·896	0·852	0·808	0·765	0·721
11 „	1·124	1·076	1·028	0·980	0·931	0·883	0·835	0·787
12 „	1·220	1·168	1·115	1·063	1·010	0·958	0·905	0·853
13 „	1·316	1·260	1·203	1·146	1·089	1·023	0·975	0·918
14 „	1·413	1·352	1·291	1·229	1·167	1·106	1·045	0·984
15 „	1·509	1·444	1·378	1·312	1·246	1·181	1·116	1·050
16 „	1·606	1·536	1·466	1·396	1·326	1·255	1·186	1·116
17 „	1·703	1·629	1·554	1·479	1·404	1·330	1·256	1·182

WATER-TUBE BOILERS.

Water-tube boilers are divided into two distinct classes, viz. :—Large-tube boilers as fitted in merchant vessels, and small-tube boilers as fitted in cruisers, battleships, torpedo boat destroyers, and launches.

The principal advantages of water-tube boilers as compared with cylindrical boilers are :—

(1) Ability to use higher steam pressures (above 250 lbs.).
(2) Rapidity with which steam can be raised.
(3) Capability to endure forcing to a great extent.
(4) Reduced weight.
(5) Reduced space occupied.

The large-tube boilers are of the Babcock & Wilcox type (figs. 26, 27) and the Yarrow type, both of which have straight tubes, the former varying from 4 ins. to $1\frac{13}{16}$ in. outside diameter, according to requirements, and the latter $1\frac{3}{4}$ in. diameter. The tubes in the Babcock & Wilcox boiler are expanded into front and rear headers, which are sinuous in form and constructed of forged steel. Opposite each tube in the headers handholes are fitted, giving access for expanding, cleaning, and renewal when necessary. The front headers are connected at the top by short tube nipples to the steam drum, and by similar nipples at the bottom to the mud drum. Return tubes about 4 ins. diameter connect the rear headers to the steam space in the steam drum.

The Yarrow large-tube boiler is a development of the Yarrow express type, and is very similar in design and construction. It consists essentially of a central upper steam drum, and two smaller water drums, the latter being situated about the level of the firebars. The generating tubes are expanded into the steam drum and each of the water drums, a manhole being fitted in the front end of each drum for access when expanding or cleaning of the tubes is necessary.

WATER-TUBE BOILERS—EXPRESS TYPE.

The small tube or express type of water-tube boilers principally adopted in British naval practice are the Yarrow, the White Foster, and the Thornycroft type (fig. 28). The tubes of all these boilers are from 1 in. to $1\frac{1}{2}$ in. external diameter, and generally the two rows next the fire are at least $\frac{1}{8}$ in. larger in diameter and one wire gauge thicker.

The White Foster boiler has a similar arrangement of drums, casings, and furnace to that of the Yarrow type. The tubes, however, are bent in such a way that any one can be withdrawn and received through the manholes in the upper steam drum when required, without disturbing other tubes.

THE BABCOCK AND WILCOX MARINE-TYPE BOILER.
(As fitted to Vessels in the Mercantile Marine.)

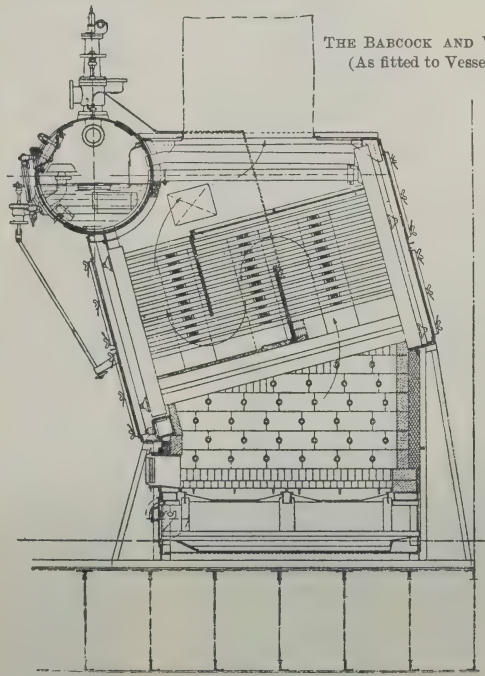


FIG. 26.

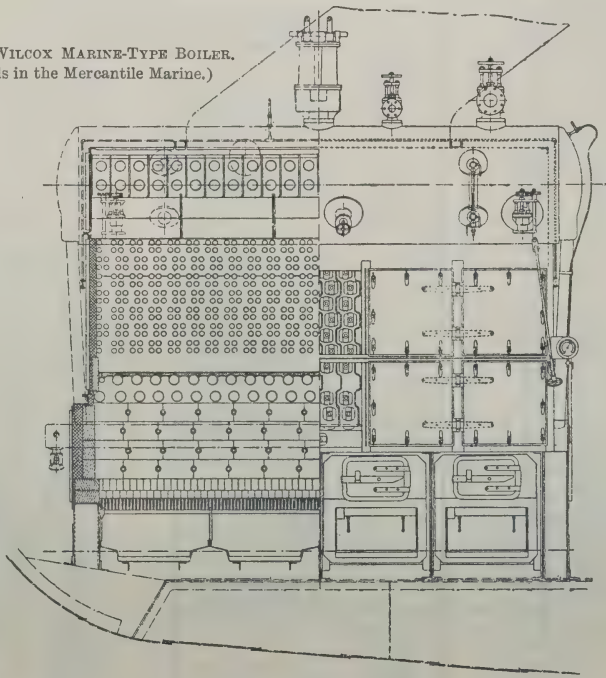
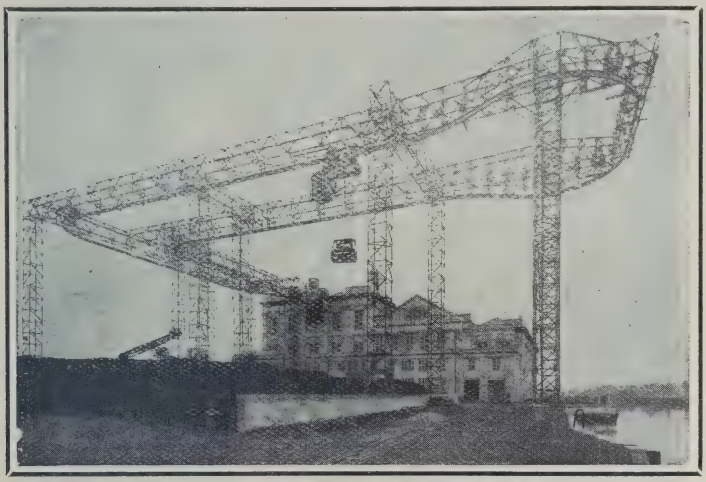


FIG. 27.

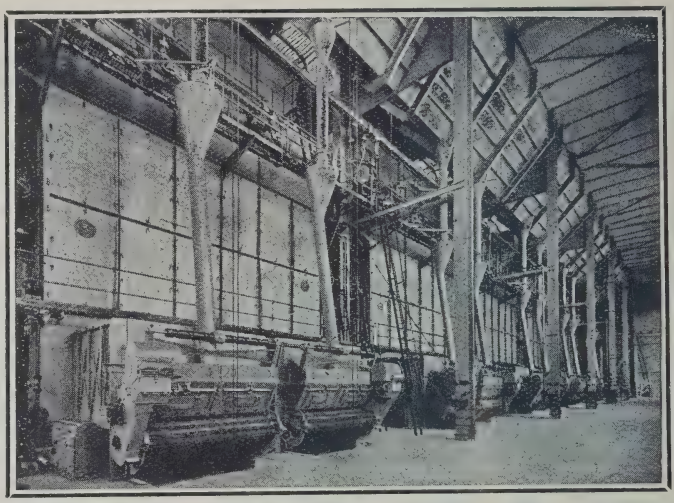


'BABCOCK'
CONVEYING
PLANT

FOR EFFICIENCY & ECONOMY

BABCOCK & WILCOX
LIMITED

BABCOCK HOUSE, FARRINGDON STREET, E.C.4



'BABCOCK'
STEAM RAISING
PLANT

FOR CONTINUOUS RELIABILITY

BABCOCK & WILCOX
LIMITED

BABCOCK HOUSE, FARRINGDON STREET, E.C.4

The modern Thornycroft boilers have one upper steam drum and two lower water drums, and are suitable for burning oil and coal, separately or combined.

The tubes are straight except for a slight curve at the lower end where they enter the water drums. The tubes can be inspected throughout the whole of their length by holding a light in the lower barrel. The lower water pockets being cylindrical can be made of solid drawn steel,

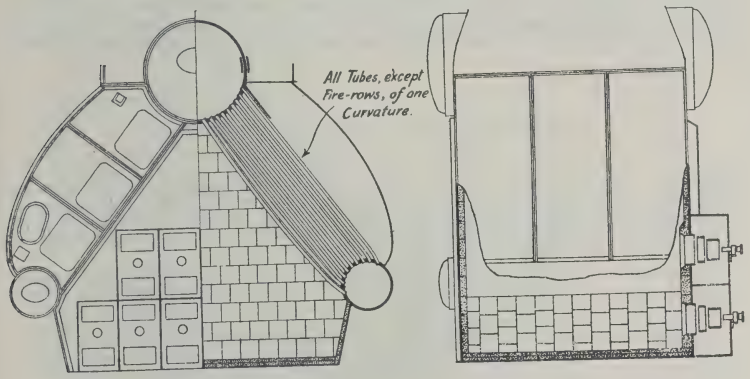


FIG. 28.

The Thornycroft Boiler.

(Oil fuel Burning.)

thus eliminating possibilities of leakage and also making a lighter and stronger construction. The tubes of any one diameter are all curved to the same radius, so that any tube will do for either row, which greatly simplifies the spares. This also allows of the lower drums being made smaller, thus saving weight.

THE YARROW WATER-TUBE BOILER WITH SUPERHEATER.

Figs. 29, 30 (page 2401) show this boiler, which consists of a top steam collector, as usual, and two lower water pockets. On the left-hand side is the superheater, and on this side of the boiler there are fewer rows of generator tubes than on the other side, where there is no superheater, it being thought desirable that the total heating surface and the resistance to the gases on both sides of the boiler should be approximately the same. In one of these boilers with a total heating surface of 6,700 sq. ft., 1,265 sq. ft. consisted of superheating surface; the total heating surface on the superheater side of the boiler being 3,453 sq. ft., and on the other side 3,247 sq. ft.

The superheater consists of a number of U tubes expanded into two longitudinal collectors, small doors being fitted so that access can be obtained to the tubes when required. The leading feature of the arrangement is that the superheater is placed on one side of the boiler only, and a damper is fitted in the up-take on the same side. If this damper is closed the whole of the gases are deflected towards the opposite side of the boiler, and no heated gases pass the superheater, the object being that if the main engines should be suddenly eased or stopped, or when raising steam, the superheater may be shut off, so as to prevent the tubes being damaged, or the steam being superheated to an excessive extent, owing to there not being sufficient circulation of steam. In this way one objection to the introduction of superheating for marine installations is overcome.

A further advantage of the arrangement is that, when the consumption of steam is suddenly reduced or stopped, not only does the damper prevent the superheater tubes from being burnt, but it also greatly diminishes the output of the boiler at the time when a reduced supply of steam

is wanted, because only about one-half of the heating surface comes into contact with the hot gases. To avoid the possibility of the damper getting distorted through over-heating, it is provided with a hollow spindle, to which air is admitted and which passes from thence between the two plates of the damper, escaping at the edge, and thus keeping the damper cool. This arrangement of damper has proved thoroughly successful under the most trying conditions.

At the maximum rate of evaporation with the damper open, namely, when burning 1·237 lb. of oil per square foot of heating surface per hour, the degree of superheat was 93·5° F.; 14·6 lbs. of water were evaporated per lb. of oil per hour, with a steam pressure of 242 lbs.

The results of other experiments indicate that in a properly designed boiler of the type described, it is possible to burn, without injury to the boiler, 2 lbs. of oil per square foot of heating surface per hour.

The 'Johnson' Marine Type Water-tube Boiler.

(Clarke, Chapman & Co. Ltd.)

The design of marine water-tube boilers is following the recent striking developments in land practice where, by the use of water-cooled furnaces, outputs have been doubled and even trebled without sacrifice of efficiency or durability. Similar progress in conjunction with the increased working pressures, superheat and high efficiencies already established in marine installations will effect remarkable economies in space, weight and cost.

The 'Johnson' Boiler (fig. 31, p. 2402) incorporates in a simple and compact form all features which modern practice and investigation has shown to be desirable. The furnace is a large cylinder, carefully proportioned to give the requisite travel and time for complete combustion of the fuel before egress of the gases, preheated air up to 500° F. being employed. As the furnace is practically enveloped by water tubes, the large surface exposed to radiation is prolific in steam generation, while avoiding the undesirable heat stresses inevitable in conventional types of water-tube boilers working at high ratings. Depreciation of brickwork is prevented and the rapid cooling and contraction of the furnace gases enables the largest amount of fuel to be burned per unit of furnace volume.

The drums are supported so as to permit of free expansion of the tubes, which are thus relieved of all strain at their attachments. The tubes have a simple curvature and can be easily brushed and sighted for a sufficient distance to enable the condition of their interior to be ascertained.

The boiler is suitable for all classes of naval and merchant vessels and can be constructed in units ranging from 1,000 horse power to 20,000 horse power capacity, single or double ended, side flow or double flow saturated or superheated steam and of special dimensions to suit shallow draught or other vessels of unusual proportions. It is expressly designed to burn Injected Fuel, solid, liquid, or gaseous, under natural or forced draught. It is extremely elastic in performance and responds immediately to sudden demands for steam, as there is not a mass of refractory to heat. It is suitable for high pressure and its capacity per ton of weight is unrivalled. Its efficiency is equal to that of orthodox types of water-tube boilers, while yielding a much greater output, and while the average rating on the heating surface is higher, the maximum rating of the furnace tubes is lower. Its construction is simple, and careful attention has been given to all details affecting maintenance and replacement.

Waste Heat Boilers.

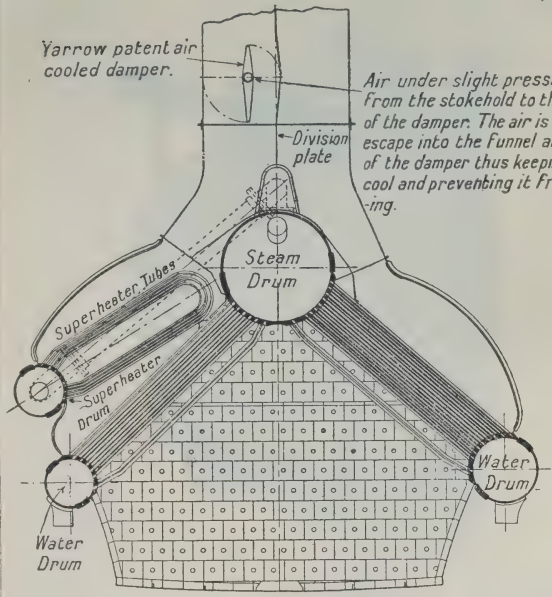
See p. 2194.

Life of Marine Boilers.

The life of a marine boiler depends to a certain extent on the care and attention bestowed on it during the first few months of its existence, provided that the plates have been properly treated at the building of the boiler—that is, all mill scale removed and the plates kept dry and free from atmospheric corrosion. The mill scale, if not removed, has a very irritant action on the clean metal of the boiler, and is most hurtful to the plates if allowed to remain on them. The greatest care and attention is needed for the first period of working of new boilers to keep the new metal free from corrosion. It is advisable on the first voyage to put into the boilers, before closing them up, 5 lbs. of lime per 1,000 i.h.p., and afterwards for a period of about six days 2 lbs. of lime per day per 1,000 i.h.p. should be passed through the hotwell as milk of lime, and 1 lb. of lime per day per 1,000 i.h.p. during the remainder of the voyage.

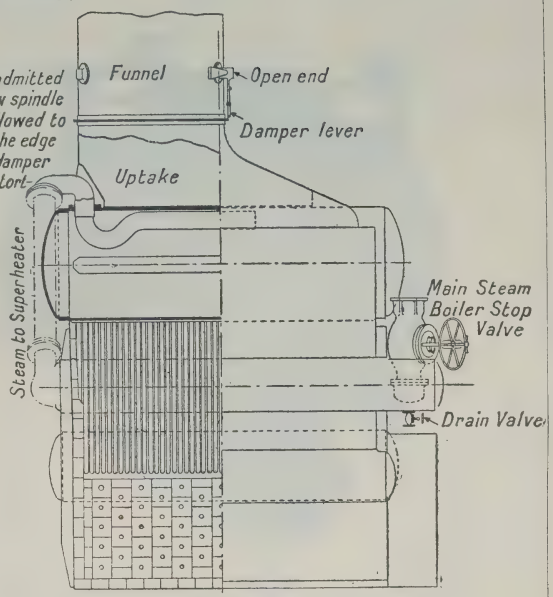
THE YARROW PATENT WATER-TUBE BOILER WITH SUPERHEATER AND CONTROL DAMPER.

(Oil Fuel Burning.)



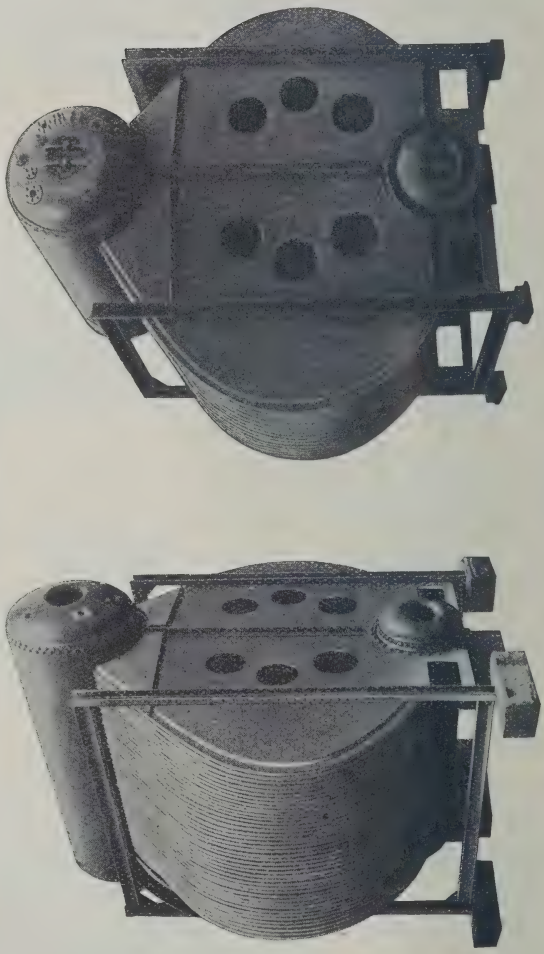
Transverse Sectional Elevation.

FIG. 29.



Half Side Section. Half Side Elevation.

FIG. 30.



Two Illustrations of 'Johnson 'Marine Type Patent Water-Tube Boiler.

Fig. 31.

Rules and Regulations with Reference to Boilers of Steamships.

STANDARD CONDITIONS FOR THE DESIGN AND CONSTRUCTION OF MARINE BOILERS.

A set of rules has been framed and agreed to by representatives of the Board of Trade, Lloyd's Register of Shipping, the British Corporation for the Survey and Registry of Shipping, and the Bureau Véritas International Registry of Shipping, and recommended for favourable consideration and adoption. An abstract of these rules is given on pp. 1563–1571.

Boiler Casings.

The casings of both the Babcock & Wilcox and Yarrow boilers are constructed of light steel plates and angles, and designed to avoid distortion due to the high temperature of the gases with which they are in contact. In way of the furnaces the casings are lined with brickwork, other parts of the casings are lined with asbestos millboard or some other non-conductor, to prevent radiation of heat to the stokeholds. In the Babcock & Wilcox type of boiler, wide use is made of 'Cellinsulate,' a special form of non-conductor which possesses greater resistance to heat transmission per unit of weight than any other form of lagging.

MARINE SUPERHEATERS.

SMOKE-TUBE TYPE.

The types of superheaters manufactured for marine boilers consist in principle of a series of loops or elements of solid cold-drawn steel tubing inserted in the smoke tubes of the cylindrical boiler and connected to suitable headers or collectors placed vertically in front of the boiler between the nests of smoke tubes. The saturated steam from the boiler stop valve enters the saturated

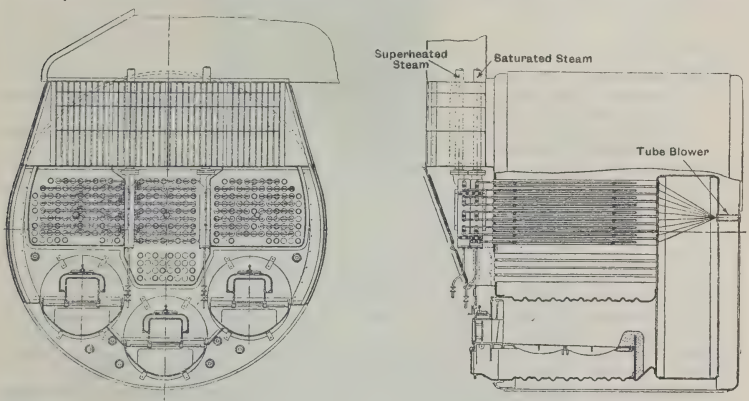


FIG. 32.—Three-Furnace Boiler, Forced Draught, with Smoke-tube Superheater.

steam header, the steam circulates through the elements and returns superheated to the superheated steam collector, and thence to the prime mover. The elements are, therefore, in the direct path of the gases flowing from combustion chamber of boiler to smoke-box. (See figs. 32 and 33.)

The elements are made from solid cold-drawn steel tubing, and the return bends are made integral with the tubing itself providing a clean, smooth surface, both internally and externally.

THE 'NORTH EASTERN' SUPERHEATER.

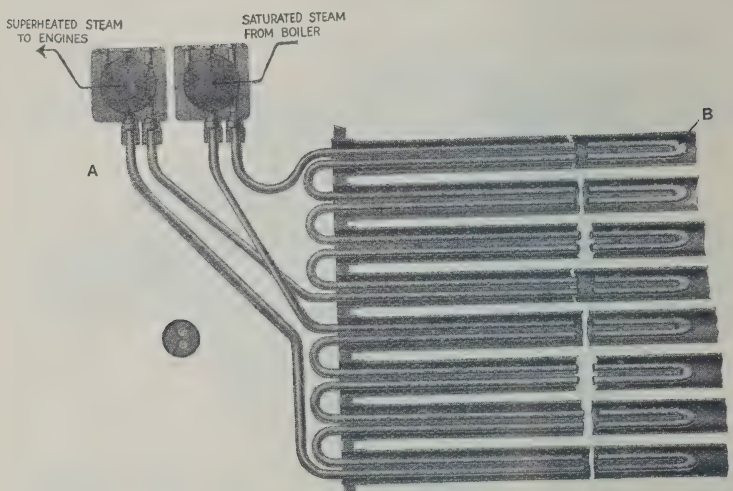


FIG. 33.—The 'North Eastern' Superheater.

Oil Fuel.

The advantages of fuel oil as compared with coal are as follows :—

(1) Reduction in weight and space occupied by the boiler installation.

(2) Increased evaporation of water per lb. of fuel in the ratio of about 14 : 9 as compared with good quality coal.

(3) Utilisation of remote parts of the ship and of constricted spaces, such as the double bottoms, for stowage of fuel.

(4) Less manual labour and greater rapidity in stowage.

(5) No reduction in speed due to cleaning fires, difficulty in trimming coal from remote bunkers, or exhaustion of stokers.

(6) Reduced wages bill for boiler room staff, no trimmers, and comparatively few attendants for the burners being required.

(7) A prompt and effective control of the steam output due to the mechanical supply of the fuel, permitting of more rapid changes of speed than is the case with coal.

(8) Absence of smoke if air supply is properly controlled.

(9) No trouble due to accumulation of ashes ; ash expellers or ejectors are dispensed with.

(10) In the case of naval vessels, fueling can readily be carried on at sea.

Both water-tube and cylindrical boilers are now adapted for burning oil fuel, the direct-pressure system being the one most generally in use for naval and mercantile service.

Pressure pumps fitted in the boiler rooms have suctions from the oil tanks, and discharge at a pressure of from 50 to 150 lbs. per sq. in. through cold filters having fine mesh wire gauze grids ; from thence the oil passes through a tubular heater, the oil being inside the tubes and heating steam outside. The temperature of the oil is raised to about 150° Fah. in the heater, and before finally reaching the burner it is passed through a second, or so-called hot, filter.

It is necessary to heat the oil in order to reduce its viscosity so that it can be atomised and forced through the fine passages of the burner, which are so designed as to give it a rapid whirling effect due to centrifugal force when liberated through tangential passages at the lip of the burner, and thus reduce it to a fine spray of conical formation. The air supply to the burner passes through directing slots cut in a steel truncated cone, 12 to 15 ins. diameter and about 16 ins. long. The burner is fitted at the centre of the entrance to the cone, and the admixture of the air passing through the slots with the oil spray from the burner produces almost perfect combustion.

In the case of naval vessels the closed stokehold system is adopted, the air pressure in the boiler rooms being controlled to suit the output of the burners. For cylindrical boilers in merchant steamers the Wallsend-Howden system (page 2163) with open stokeholds is now in extensive use. The arrangement of burners and air cones is similar to that described above, but the air supply is heated by the waste gases, as in the ordinary Howden forced draught system (page 2124) applied to coal burning. Each burner can pass from 400 to 600 lbs. of oil per hour, and it is possible to burn about 13 lbs. per cubic foot of combustion chamber. The area of the furnace front limits the number of burners which can be fitted, and it is desirable that the volume of the combustion chamber should be as large as possible.

(See also pp. 2149, 2438.)

EVAPORATORS.

For Fresh Feed Make-up Water for Boilers.*

With the high pressures of steam (usually from 150 to 200 lbs. per square inch) now carried in steam boilers for use with modern triple-expansion and other engines, it is found necessary to

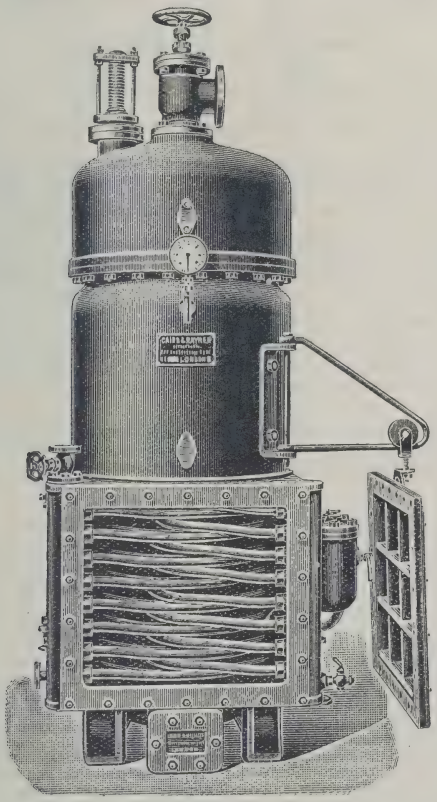


Fig. 34.

* Also used in connection with distilling condensers for producing fresh drinking water instead of the old way of distilling from the donkey boiler.

provide fresh feed-water for the make-up, in order to prevent the great deposit of scale on the heating surfaces which results from these high temperatures.

The quantity of auxiliary feed-water required amounts to from 1 to 5 tons per 24 hours per 1,000 h.p.

To provide this, evaporators have come into general use.

In these evaporators the sea water is boiled by the heat from steam passing through tubes immersed in the salt water. The condensed steam from these tubes is usually drained back to the condenser or hot-well, and so back to the boiler, and the secondary steam produced from the sea water is led to the main condenser, and is there condensed along with the exhaust steam from the engines, and thus delivered to the boiler as fresh feed-water.

The scale formerly left in the boilers is deposited in the evaporators, which are arranged so as to be easily cleaned when necessary.

Condensers and Water Heaters.

The Weir 'Uniflux' Condenser.

(No longer made. Replaced by 'Weir' Regenerative Condenser—see p. 1837.)

This design of condenser, introduced in 1909, has since become extensively adopted in this country for naval and high-class merchant steamers, and the high vacua attainable by its adoption when working in conjunction with the Weir 'Dual' type of combined wet and dry air pump has been an important factor in the improved efficiency of steam turbines.

The construction of the condenser introduces no novel features except as regards the pear-shape of shell which has been designed with the object of causing the exhaust steam to flow over the tubes at a uniform velocity without short circuiting, the steam inlet being arranged as far from the air-pump suction as possible.

The 'Dual' air-pump is of the beam type having one steam cylinder and two pump barrels, one pump for air and vapour and the other for water, air and vapour. Cold injection water is introduced at each stroke into the dry-air pump barrel to condense the vapour and cool the air, and in order to avoid the expansion of the air present in the water filling the clearance space of the dry pump at the moment of discharge the dry-air pump is arranged to discharge below the head valves of the wet-air pumps.

The combination of improved design of air-pump and condenser in conjunction with an increased supply of circulating water has resulted in a reduction of the condensing surface from 20 to 25 per cent. as compared with the condensing plant previously fitted.

The Weir Direct Contact Feed-Water Heater.

This system consists in taking a portion of the steam from the low-pressure receiver of the engine and utilising it to heat the feed water on its passage to the boiler. In actual practice on board ship, this steam is also supplemented by the exhaust from the auxiliary engines.

Fig. 35 shows a sectional view of the heater. The heating steam is taken from the low-pressure casing of the main engine, and the exhaust of the auxiliary engines is also led into the heater through the non-return valve B on the side of the apparatus. A perforated cylindrical waist piece forming an annular steam space round the heater, and surmounted by a conical spray piece, also perforated, is fitted to ensure the uniform mixing of the steam and water. A Weir direct-acting feed pump, or the main engine feed-pump, delivers the feed water into the heater through the spring-loaded valve D on the cover. The feed water is forced through this valve in a thin sheet, and is instantly heated by contact with the steam. The water is further broken up into fine spray by passing through the conical spray piece, thereby bringing the greatest possible surface into contact with the steam. As the pressure in the heater is generally much less than that of the entering water, the effect of this lowering of the pressure, and sudden heating of the water, is to liberate the air in the water, and this is removed to the condenser, or to the atmosphere, by a small cock K on the air vessel placed on the top of the heater. (In the case of twin-screw vessels fitted with one feed heater, the lever cock on the heater air dome is coupled up to both condensers by a T piece.) The feed water is thus rendered non-corrosive, and falls to the bottom of the heater at the boiling temperature corresponding to the pressure.

The combination, fig. 35, of the automatic regulating gear with the heater is a special feature of the Weir apparatus. The float E shown in the lower part of the heater is a pan, with water-tight bottom and sides, but open on the top. It is suspended on two levers, so as to move up and down with a parallel motion ; the top lever spindle is carried through the door at one end, and is

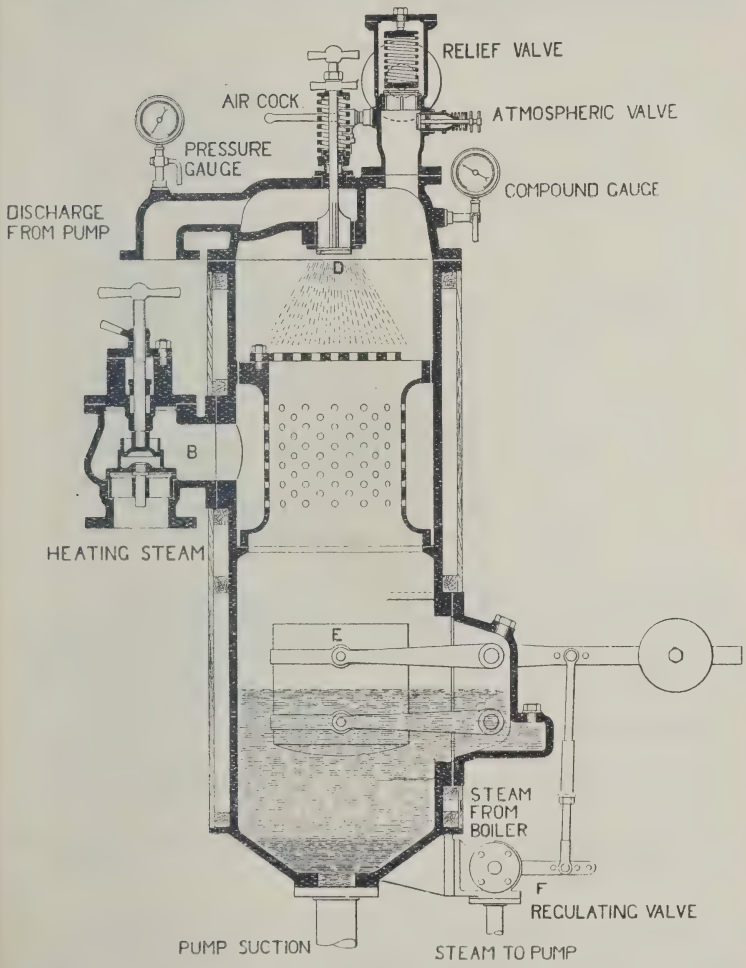


FIG. 35.

balanced by a lever and weight. The float is always full of water, and the weight is adjusted to balance when one-half is immersed in water. The weight lever is connected by a rod to another lever, which actuates the throttle valve F and controls the supply of steam to the pump drawing from the heater. When the water in the heater rises, the float is raised, and the throttle valve

opened, and, when the water level is lowered, the float follows, and the valve is closed ; the level of the water is thus kept constant in the heater, and the pumps are completely filled with water.

When feed heaters are placed on the discharge side of the feed pump they are necessarily subjected to the boiler pressure, and must therefore be of a very substantial character. Such heaters are always of the surface type.

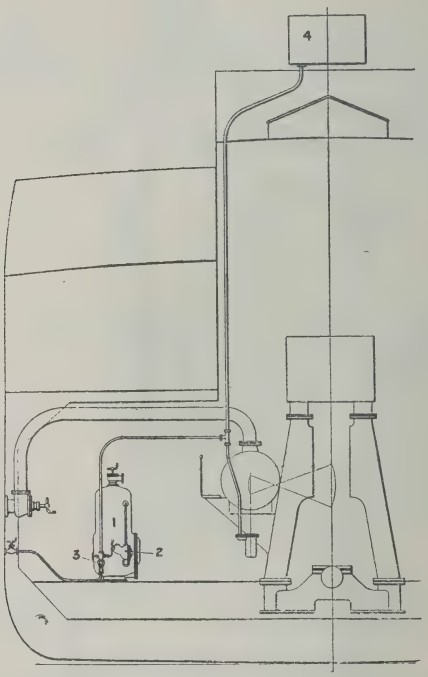


FIG. 36.—Arrangement on board ship of Evaporator with 'Weir' Feed Regulator and Brine Ejector.

1. Evaporator.
2. Feed Regulator.
3. Brine Ejector.
4. Sanitary Tank.

As affecting the engines, it is found that this system secures a fuel economy of from 5 to 8 per cent., according to the steam pressure carried, and the number of expansions in the engine.

Air Pumps.

PARAGON AIR-PUMP.

(G. & J. Weir, Ltd., Cathcart, Glasgow.)

This air-pump is a development of the well-known Dual Air-Pump, and is giving the same steadfast reliability in marine work that was characteristic of the Dual type of pump. It has been adopted by the leading steamship companies, and is shown in fig. 37.

The normal three-valve twin-barrel arrangement has been adopted, driven by a single steam cylinder. Both barrels deal with the water, *i.e.* the suctions are common, and each barrel draws its quota of condensate through the same suction pipe. On the side of each barrel and high up on the barrel is arranged a port, so positioned that when the bucket is at the top of its stroke, the port is in communication with the space under the bucket, and when the bucket has travelled on its down stroke a distance equal to its own depth plus the depth of the port, the port is in communication with the space above the bucket. Communicating with this port is a chamber which

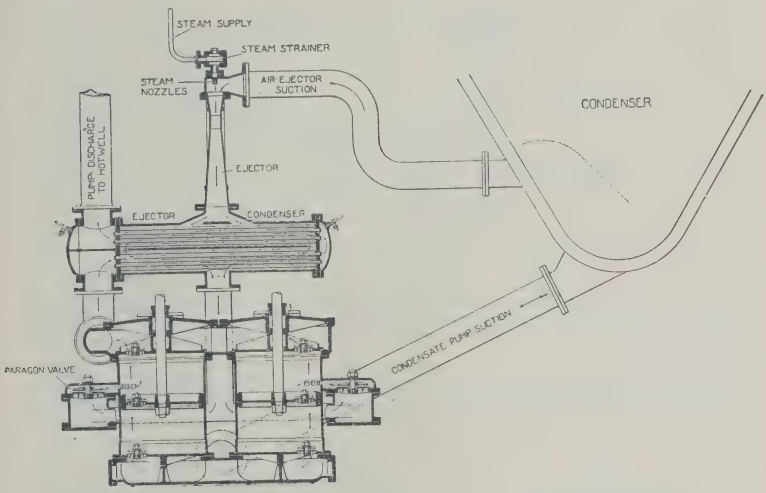


FIG. 37.

houses a non-return valve of a design similar to the pump valves. A branch pipe connects the two chambers with the augmentor condenser outlet. The augmentor condenser is of the tubular type with brass tubes ferruled into brass tube plates. The augmentor is fitted to the top of the augmentor condenser, and discharges into same. The steam discharged by the augmentor is condensed on the outside of the tubes by the condensate, which is discharged through the tubes by the air-pump; the heat in the augmentor discharge therefore is conserved in the condensate, which results in the economy of the apparatus being good. The steam nozzles are of the convergent divergent type, similar to those fitted to the air ejector.

This pump is capable of maintaining the highest vacua, and has a large air handling capacity for the size of the pump.

Experiment has shown that on a 19 in. × 15 in. pump the fall in vacuum was only 0·4 in. Hg, when a ¼ in. diameter air leak nozzle was opened admitting air to the condenser.

Bearings.

THE MICHELL BEARING.

Michell Bearings (thrust and journal) are largely used in modern marine practice, and are manufactured by Michell Bearings, Ltd., Newcastle-on-Tyne. There are in the British Navy alone ships aggregating over ten million horse-power fitted with main thrust blocks of this type, and in passenger and merchant ships the Michell Thrust Block is commonly used, both for reciprocating engines and steam turbines.

A standard thrust block used for light craft of all kinds is shown in fig. 38. This is self-lubricating, and can be arranged upon an extension of the engine-bed or upon a separate seating immediately aft of the crankshaft coupling. The oil level in the thrust block is usually kept just above the under-side of the shaft. The bearing surfaces are lubricated by the thrust collar

carrying oil upon its edge, and the oil is scraped from this by the scraper at top, thus giving a continuous cascade of oil on the pads. The pads are so pivoted that a film of oil is led into the space between pad and thrust collar, and this oil film can withstand a heavy pressure. A feature of the bearing is that as the speed increases, the pressure per square inch can be increased.

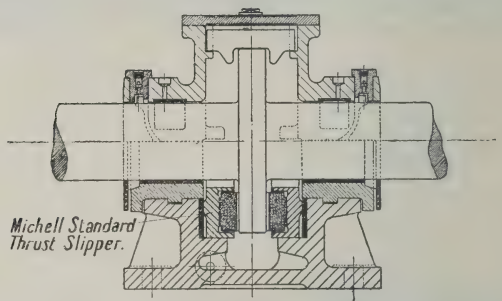


FIG. 38. Michell Slipper Type Thrust Bearing.

Michell thrust blocks have a single collar shaft with a series of pivoted pads arranged forward and aft, and it will be noticed from the illustration that the pads need only be in the bottom half of the bearing. Thrust blocks, however, can be designed to utilise an existing multi-collar thrust shaft.

Large engines are usually fitted with a thrust block arranged for continuous oil circulation, and in these blocks the upper as well as the lower portion of the body sustains thrust load. Oil is introduced into the space behind the pads and compelled to travel radially across the thrust faces. This oil can be cooled and used over and over. In these blocks the upper and

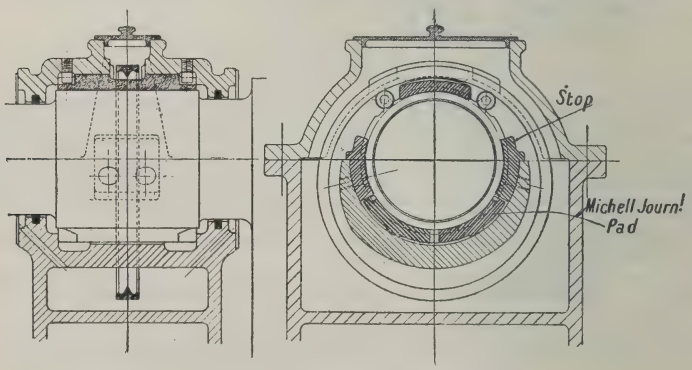


FIG. 39.—Michell Journal Bearing for Marine Oil Engine.

lower parts of the body are keyed together, and liners for adjustment are fitted behind each of the rings which carry the pads.

The Michell principle is also applied to journal bearings, and fig. 39 illustrates one as fitted to a marine oil engine. When speeds are high the ring lubrication is replaced by a continuous supply of oil from a pump. It will be noticed that the oil is taken on to the shaft by means of a ring, and this ring is always kept in a central position, even if the ship pitches, as the central art of the oiling ring is guided by grooves in rollers arranged upon the shaft.

Considerable economy in length is effected by this bearing, as a pressure of about 500 lbs. per square inch projected journal area is satisfactorily carried at any speed.

The pivoting principle, upon which all Michell bearings work, ensures, perhaps, the nearest approach possible to perfection in lubrication. Astonishingly high pressures are quite safely carried (under test as much as 5 tons per square inch has been successfully imposed), whilst speeds of any magnitude may be run by regulating the quantity of oil supplied. High speed is an advantage, inasmuch as oil film thickness increases with the speed.

Temperature Limits of the Michell Thrust Bearing.

The working limits of the Michell thrust bearing are fixed mainly by the temperature of the single thrust collar used. In designs prepared in 1908 lubrication was effected by simple jets of oil. So long as the rubbing speed did not exceed 30 ft. per second, these seldom gave trouble, but when the same plan was adopted for a longer bearing, in which the mean speed was 40 ft. a second, the bearing had to be scrapped. The plan was then adopted of running the bearing in a casing filled with oil, and through which oil was constantly pumped. This plan proved satisfactory as fitted to marine turbines provided with single-reduction gearing, where the rubbing speed seldom exceeded some 80 ft. a second. With the advent of double-reduction gears and consequent higher rubbing speeds, trouble was experienced. With the arrangement described, the immersion of the bearing in oil was relied upon both for lubrication and for cooling, but it was found that the temperature of the oil escaping from the casing gave very little indication of the temperature attained in the lubricating film. It was therefore decided to separate the functions of lubrication and cooling, by forming oil-circulating ports in the bearing through which a free flow of oil was maintained by the rotation of the collar. This not only provided unlimited lubrication for the pads, but, owing to the thorough circulation thus ensured, the temperature of the oil leaving the casing gave a fairly correct idea of the temperature of the collar. This was the vital factor in the whole problem. If the collar temperature exceeded about 150° F., there was a distinct danger of failure. At high rubbing speeds, a supply of oil ample for lubrication might be insufficient to keep the collar cool, and much more oil had to be supplied accordingly.

(*H. T. Newbigin, paper on 'The Crankless Engine,' The North-East Coast Institution of Engineers and Shipbuilders, February* 24, 1928.)

MARINE DIESEL ENGINES.

(Revised by James Thomson, M.I.N.A.)

The Diesel engine as designed for marine work differs in no essential principle from the stationary type; with each manufacturing firm there are certain modifications and peculiarities of construction. On the whole the two-stroke-cycle single-acting engine is considered to have best prospects for the future, but for the very large engines for marine purposes the double-acting two-stroke-cycle motor is expected to give best results, whilst a number of firms still adhere to the four-stroke-cycle single-acting principle for moderate powers.

The superiority of the Diesel engine for marine work in certain cases is not entirely due to its fuel economy. This must always be an uncertain matter, dependent on the relative prices of the fuels. The consumption of oil with a Diesel marine engine is practically the same as for the stationary type, and, generally speaking, with oil of a net calorific value of about 18,000 B.Th.U. per lb. or 10,000 calories per kg. the fuel consumption for marine work may be taken as 0·40lb. per b.h.p. hour for the main propelling engines. Unlike reciprocating steam engines, and steam turbines, the larger Diesel engines possess practically no advantage over smaller ones in this respect.

The fuel consumption of large steam turbine plants with superheat and modern appliances has been reduced so much that fuel rates and first cost, etc., must be carefully considered before deciding whether Diesel or steam turbine machinery is the most suitable for the required service.

Among the most important advantages of the Diesel engine for marine work is the reduced space occupied by the propelling machinery and bunkers. This permits of greater cargo-carrying capacity, or a smaller vessel for the same cargo, with a corresponding reduction of the cost of the hull, although at the present time the cost of a Diesel engine is in excess of a steam engine and boiler plant. A vessel equipped with Diesel engines has from 10 to 15 per cent. extra cargo-carrying capacity as compared with a steam engine, and this is materially increased if the amount of fuel carried is reckoned on the basis of the same radius in both cases, when considerable extra bunker space is rendered available for cargo in the Diesel ship. With special precautions, moreover, the fuel oil can easily be carried in the double bottom of the ship, which allows a further gain in space. In war vessels the advantage of the economy in fuel with Diesel engines will no doubt be utilised by increasing the possible radius, which may be taken at 2 to 3 times that of a steam vessel with geared turbines owing to the relative high economy at lower powers, this being a feature of the Diesel engine as compared particularly with the steam turbine.

In a Diesel ship the number of attendants in the engine room required is less than in a steam vessel, being usually in the neighbourhood of $\frac{2}{3}$. Firemen are entirely dispensed with, the handling of fuel is easy, rapid and clean, whilst the astern power is equal to the ahead, which is not the case in turbine vessels. A list of engine room staff for various motor ships is given on page 2438.

For the same speed a single screw vessel equipped with Diesel engines usually requires slightly more power than when propelled by a steam engine for similar conditions, as the revolutions are too high for the most efficient propeller, but this does not apply to multiple screw ships when the speed exceeds 12 knots.

The weight of Diesel machinery for large powers is usually greater than steam turbine machinery.

Types of Marine Diesel Engines.

There are three distinct types of Diesel marine engines, the first two are the single-acting engines of the four-stroke-cycle and two-stroke-cycle type respectively, whilst the third is the double-acting engine.

The four-stroke-cycle motor is slightly more efficient, but the two-stroke-cycle single-acting engine has the advantage of reduced weight and space, simplicity in reversing arrangements and cheapness of manufacture. It necessitates the employment of scavenge pumps which reduce the overall efficiency of the motor. Larger powers can be developed in single cylinders, and although the limit is not yet certain, it is anticipated that 3,000 h.p. can be produced from a single cylinder. Up to the present 2,000 B.H.P. is the highest power that has been attained in a single-acting two-stroke-cycle motor per cylinder.

The reason for the employment of the double-acting two-stroke-cycle engines is to obtain the maximum power per cylinder. The chief difficulty encountered with the double-acting engine is to get rid of the heat in the cylinder, and there is also the question of the most convenient method of operating the valves in the bottom cover of the cylinder. It is stated that these troubles have now been satisfactorily overcome.

As regards the two-stroke-cycle single-acting type, the chief difference in the design of different manufacturers, apart from details of the reversing gear, is in the method of the provision of the scavenge air. In some engines scavenge valves are employed, fitted in the cylinder cover, and through these the scavenge air enters the cylinder, whilst in other types ports are provided at the bottom of the cylinder usually extending half-way round the circumference, opposite the ports for the discharge of the exhaust. Scavenging through ports presents some difficulties in regard to cleaning out all parts of the cylinders thoroughly, and various arrangements are employed to make it more efficacious, some of these being described later. With valves, the scavenging is excellent, but the valves must be large, and they add to the complication of reversing. The scavenging pumps usually work at a pressure of $1\frac{1}{2}$ to 6 lbs. per square inch, above atmosphere, and the power they absorb is from about 5 to 10 per cent. of the output of the engine. The power developed by a single-acting two-stroke cycle Diesel engine is about twice that of a four-stroke cycle engine of equal cylinder dimensions.

The question of reversing Diesel engines is not one of particular difficulty, the only requirement being a variation in the time of opening of the different valves. In the four-stroke-cycle engine this includes the exhaust valve, which is omitted in the two-stroke-cycle type, whilst if in the latter scavenge ports are provided, the matter is further simplified by the absence of scavenge valves. There are various methods of reversing, which, however, practically do not differ at all in principle, the arrangement being to alter the times of openings of the various valves relative to the crank shaft. In one arrangement which is commonly employed, a single cam shaft is provided on which are fixed two separate sets of cams, one for the ahead motion and one for the astern, each valve thus being capable of operation by either of the two cams. For reverse running the cam shaft can slide in a longitudinal direction, and this motion brings the astern set of cams underneath the levers operating the valves, so that the valves are thus set for going astern. Before the cam shaft can be moved, the rollers of the valve levers have to be moved clear of the cams in order to let them pass under.

A second method of reversing consists in having two separate cam shafts, on one of which the cams for ahead running are mounted, and on the other those for the reverse, and either cam shaft is set in a position for its cams to actuate the valve levers, according as it is required to run ahead or astern. There are various other means adopted for reversing, such as by a rotation of the cam shaft relative to the crank shaft, and utilising the same cams for both directions which necessitates special setting of these cams. In another design a combination of this method and the first is adopted, whilst there are certain other special arrangements, one of which is described later.

The method of injection of the fuel into the cylinders is receiving great attention at present. In the past injection by high pressure air was almost universally adopted, but 'solid injection,' by which is meant the injection of the oil under very high oil pressure through a nozzle which breaks the oil into a fine spray capable of great penetration, has been very successfully fitted on many types of engines. The efficiency of oil burning may be slightly lower than with air injection, but the injection air compressor is dispensed with, and the power saved compensates for any such decrease in the efficiency of burning the fuel.

In the earlier designs of solid injection the fuel valve was opened and closed by a cam, as with air injection, but in the later designs the oil-pump for each cylinder is set so that the discharge pressure automatically opens the fuel valve and injects the fuel into the cylinder when the piston is in the proper position. With this arrangement on a two-stroke port-scavenging engine the only valves operated by cams are the air-starting valves, and automatic valves with remote control can also be employed for this purpose, so that the design of the engine is considerably simplified, and accessibility for overhaul much improved.

Burmeister and Wain Type.

The engine designed and constructed by Messrs. Burmeister & Wain, of Copenhagen, and by Messrs. Harland & Wolff, may be taken as a typical four-stroke-cycle marine Diesel motor. It is built with six or eight cylinders, according to the power, the engines of the 'Glenapp' and several other boats having each eight cylinders, and developing about 2,600 i.h.p. at 115 revolutions per minute.

As is now common in most Diesel engines, and will, no doubt, be generally adopted in the future, external crossheads with water-cooled guides are employed, which, although increasing the height, are of considerable advantage in dismantling and for facility of inspection. An enclosed crank chamber is used, but at front and back there are readily removable covers.

The engine may be considered as arranged in two sets of four cylinders, which is convenient for the placing of the valve gear and manœuvring mechanism described later. The valves are fitted in the cylinder cover, as is generally the case, and each cylinder has a starting air valve, fuel inlet valve, exhaust valve, air inlet valve, and relief valve. The cam shaft carrying the cams which operate the levers controlling the valves is not situated on a level with the top of the cylinders, as is common in many four-cycle engines, but at about half the height of the engine. Each of the valves, except the relief valve, has two separate cams for its control on the cam shaft, one for ahead and one for astern running. In order to reverse the engine, the valve levers are lifted and the cam shaft moved longitudinally a distance equal to the cam width, so that, on dropping the valve levers once more, they are now immediately above the astern cams and are in a position for running in the opposite direction. The sliding of the cam shaft is accomplished by means of a disc carried on it, which moves in a slotted drum on an auxiliary shaft, the slot slanting from left to right. The auxiliary shaft can be turned by means of an air motor controlled from the starting platform, and in turning the cam shaft moves to the left or right, according to the direction of rotation of the auxiliary shaft. The actual operation of the valves from the cams is carried out by having long hollow vertical rods, whose upward and downward motion causes the rocking levers at the top to actuate the valves. It may be mentioned that, contrary to usual practice, the injection valves open inwards towards the cylinders. The drive of the cam shaft is through helical gearing and a vertical intermediate shaft, direct from the crank shaft. Sometimes a direct gear drive is adopted, and in some cases a chain drive is used.

The question of the number of fuel pumps for pumping the oil to the injection valves is somewhat controversial, the majority of manufacturers preferring to have one pump per cylinder, whereas others think it desirable to minimise the number as far as possible. A small hand pump is employed for pumping the oil up to the valves when required, the usual overflow cocks being fitted.

In Diesel engines driving generators the governor controls the amount of fuel admitted to the cylinder by acting upon the suction valves of the fuel pump, and holding them open a longer period during the pressure stroke, if it is required to reduce the oil supply, while in marine engines hand-control is effected in the same way. In this engine the control is from the starting platform and a governor is fitted, which is only for the purpose of preventing the speed rising above the designed revolutions per minute, by cutting off the fuel altogether in all cylinders but one. Forced lubrication is adopted for the engine, and the oil is water-cooled and circulated continuously.

High-pressure air for injecting the fuel is provided by a vertical three-stage air-compressor, driven direct by a crank at the forward end of the crankshaft. These compressors are of ample capacity, and their delivery is controlled to suit the immediate requirements of the engine by throttling the first-stage suction, or by leaking off at the end of the first compression.

The cylinder covers and jackets are cooled by fresh or salt water, and the pistons by fresh water, salt water, or oil. It may be mentioned here that some designers and owners prefer to avoid the use of sea-water for cooling purposes, owing to the possibility of deposit of scale, etc., but the matter is largely an open question.

In a number of the later B. and W. engines mechanical injection of fuel has replaced the air-injection system. The fuel valves are automatic, and the quantity of oil delivered is varied by altering the stroke of the pump plunger, which is cam operated and spring returned.

Fig. 40 shows a section of a 2,000 b.h.p. single-acting Burmeister and Wain motor of the latest type, and figs. 41 and 42 (pages 2415 and 2416) give illustrations of the control arrangements for this motor.

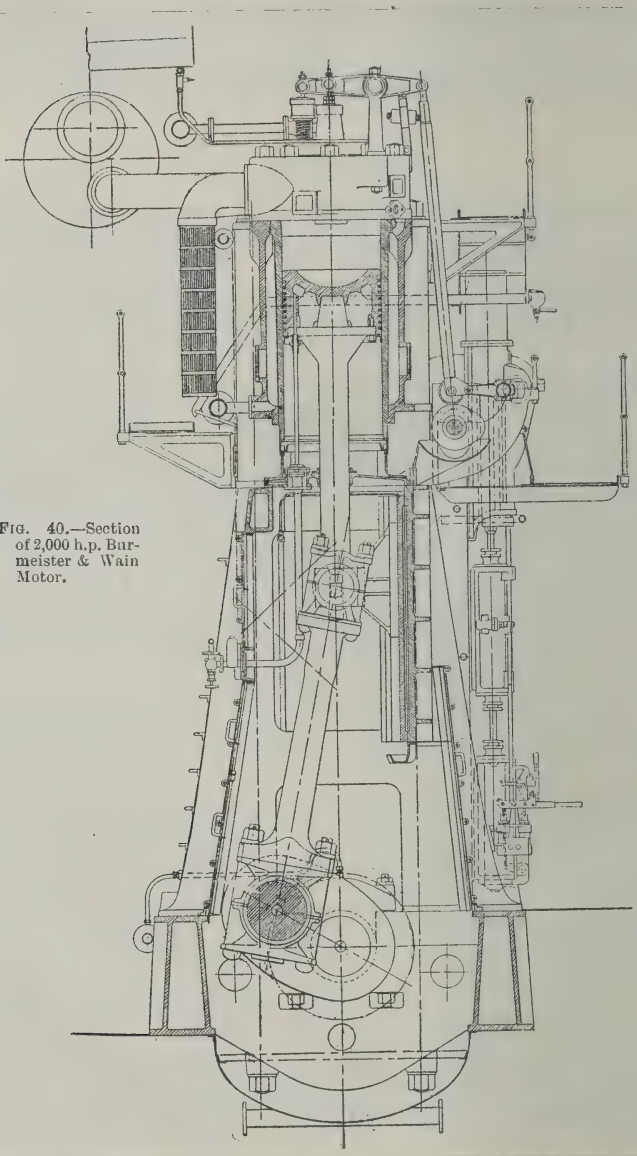


FIG. 40.—Section of 2,000 h.p. Burmeister & Wain Motor.

BURMEISTER & WAIN MARINE DIESEL ENGINE.

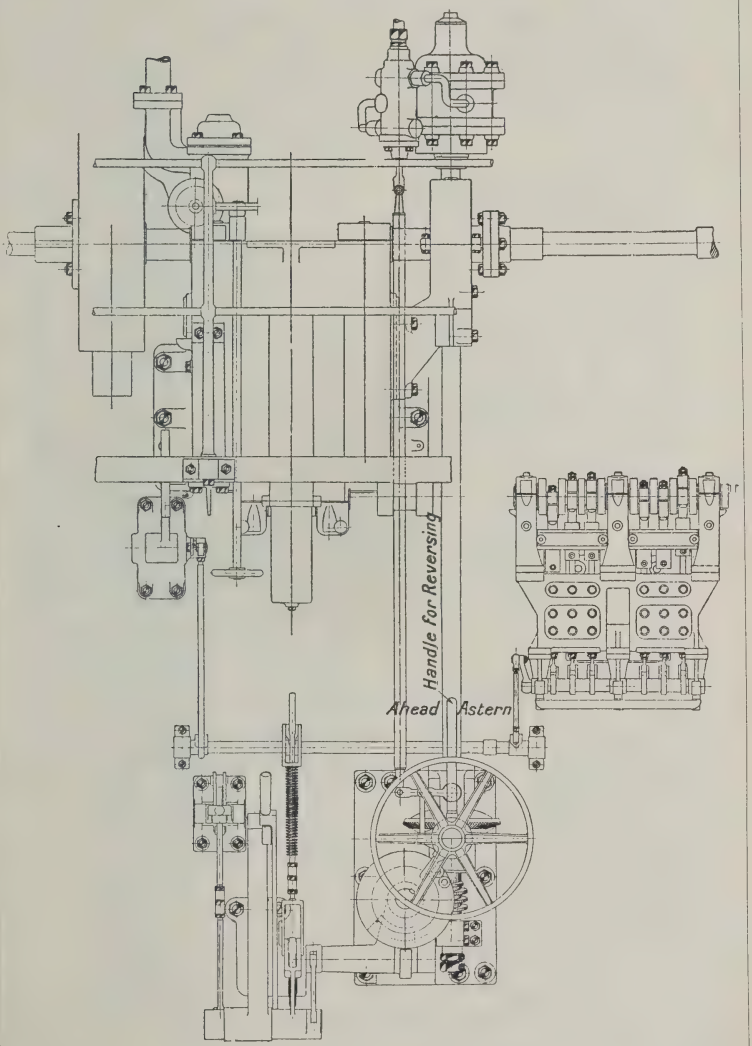


FIG. 41.—Details of Control Arrangements. Front View.

BURMEISTER & WAIN MARINE DIESEL ENGINE.

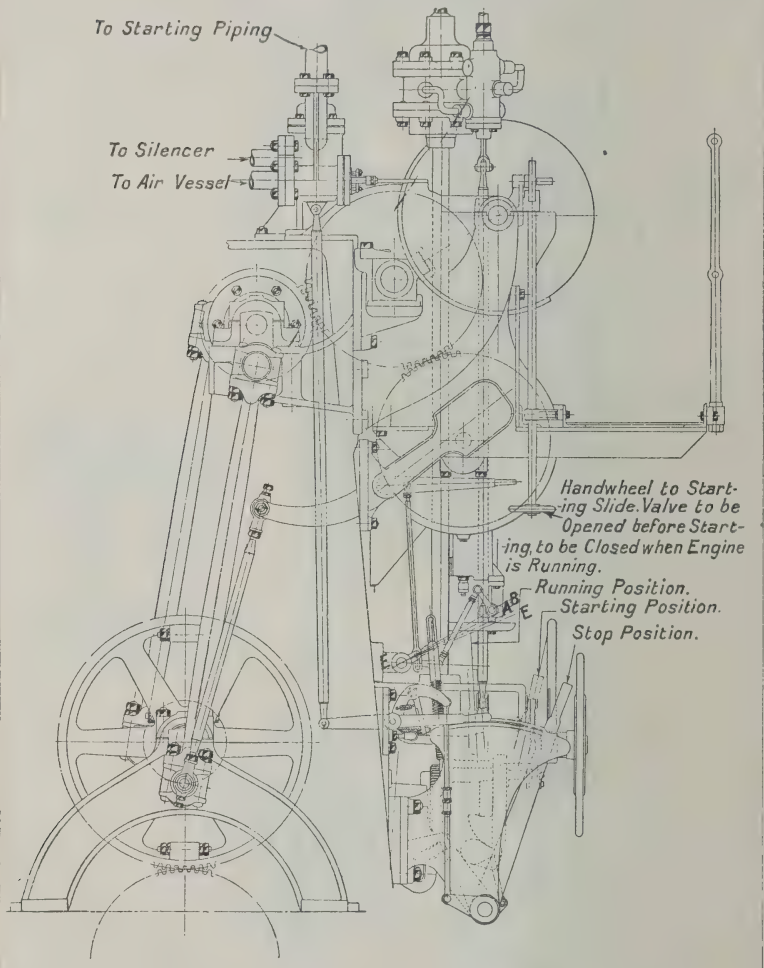
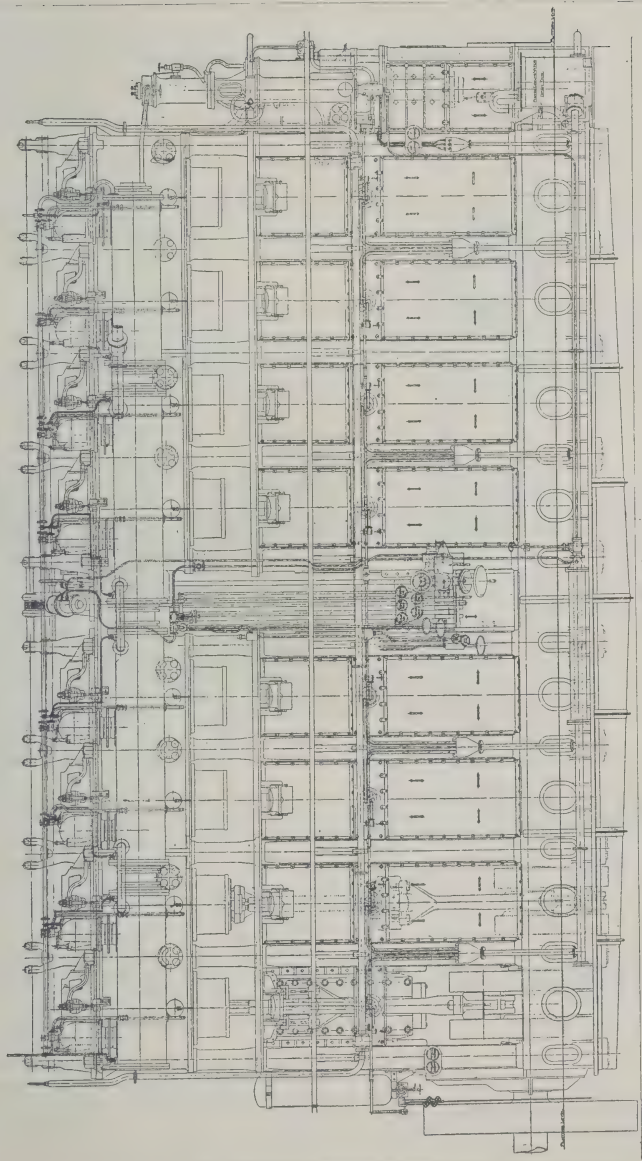


FIG. 42.—Details of Control Arrangements. Side View.



NORTH-EASTERN FOUR-STROKE SINGLE-ACTING DIESEL ENGINE.
(3 cyl. 730 dia. × 1,300 stroke.)

Fig. 43.

NORTH-EASTERN MARINE DIESEL ENGINE.

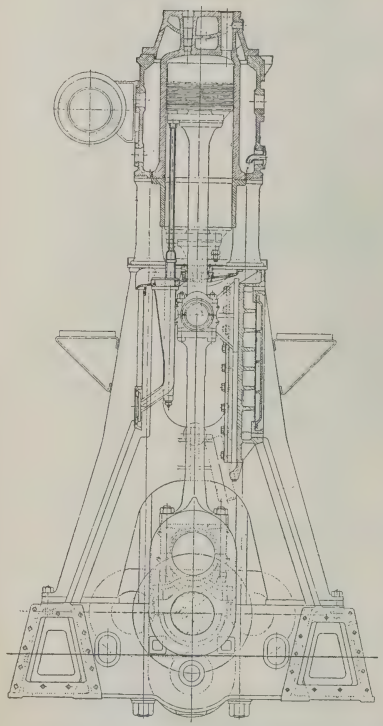


FIG. 44.—Sectional elevation of North-Eastern-Werkspoor, Single-Acting Engine (Four-stroke).

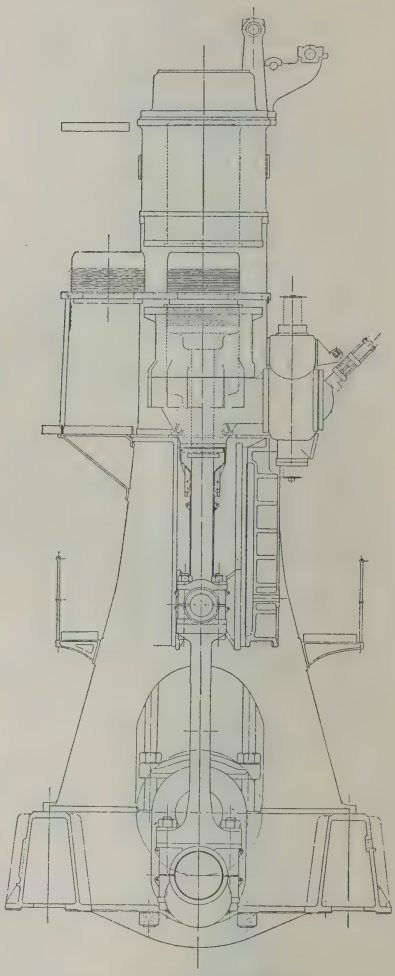


FIG. 45.—North-Eastern-Werkspoor Four-Stroke Double-Acting Engine: method of withdrawing piston.

Double-acting Type.

These engines have been built in sizes up to 10,000 b.h.p. in eight cylinders at 125 r.p.m. The air compressors are independent of the main engine and driven by separate Diesel engines.

The pistons are cooled by oil, and there is a gland between the bottom cover and the crank chamber.

The camshaft is chain-driven, and one camshaft operates the valves in the top and bottom cylinders.

Burmeister and Wain Two-stroke Engines.

Messrs. Burmeister and Wain have recently developed two-stroke engines both double- and single-acting. The outstanding feature of the design is that the exhaust ports are situated at the cover end and are controlled by a smaller diameter auxiliary piston, having a stroke about one-fifth that of the main piston. These auxiliary pistons, which develop about 10 per cent. of the power of the engine, operate through links and levers to an auxiliary crankshaft, which drives on to the main shaft by chain.

North-Eastern Diesel Engine—Werkspoor Type.

The first engine of this type was constructed by the Werkspoor Company of Amsterdam, and installed in the *Vulcanus* in 1910. North-Eastern Marine Engineering Co. was the first of British marine engine-builders to take out a licence to build this type of engine, since when engines varying from 1,000 b.h.p. up to 5,000 b.h.p. have been built, both for British and Continental owners, with very successful results.

Single-Acting Engines.

The motor, figs. 43, p. 2417, and 44, p. 2418, is usually of six cylinders, and, except for a few special features, is of the standard single-acting four-stroke type of engine.

The chief point of difference lies in the cylinder construction, the North-Eastern-Werkspoor engine having a continuous cooling surface from liner to crown of cylinder, this construction giving highly efficient cooling to the combustion chamber, there being no liner joint to interfere with the circulation. The arrangement also allows the valves to be spread further apart, giving more space for circulation between the valve ports.

Another point of difference lies in the arrangements for operating the valves and the method of reversing. There is one camshaft on which the cams for ahead and astern working respectively are mounted in pairs for air starting, fuel injection, induction, and exhaust. This shaft does not move in a fore-and-aft direction during reversal, but the lever which transmits the motion from the cam to the valve is mounted eccentrically on the reversing shaft. The axis of the eccentrics and the shaft on which they are mounted are not parallel, so that when the shaft is rotated through an angle of 180° the rollers are lifted from one set of cams and placed on the other. This is a totally distinct method from that already described, in which the camshaft is moved longitudinally, and illustrates the diversity of the arrangements which may be employed for reversing Diesel engines.

The camshaft is driven by connecting rods from cranks on a countershaft to cranks on the camshaft, the countershaft being driven at half engine speed by spur gearing from the engine crankshaft.

A light type of engine is built with the cylinders entirely supported by steel columns and diagonals, but the standard design has the cylinders supported on cast iron 'A' frames, the cylinder beam, columns, and bedplates being held together by heavy through-bolts running from top to bottom of the engine.

The main running gear has forced lubrication, and is totally enclosed in an oil-tight casing with large doors, particularly easy to remove.

The compressor is of the three-stage type, usually direct driven from the forward end of crank shaft.

A special feature of the engine is the arrangements made for inspecting the pistons. This consists of the lower part of the liner being readily detachable, so that the piston when lowered is exposed for inspection and rings can be changed in a few minutes. In addition, the cylinder bores can be examined without disturbing any of the main running gear in the crank case or the valve gear.

This company has recently installed the Buchi system of exhaust turbo-charging in the M.V. *Raby Castle*.

Double-acting Engines.

The North-Eastern-Werkspoor double-acting Diesel engine has been jointly developed by Werkspoor and North-Eastern Marine Engineering Co., Ltd., the experimental work being carried out on a large single-cylinder engine at Wallsend.

The engine, which is illustrated in fig. 45 (page 2418), is of normal design as regards the main running gear and framing. The bottom combustion space has the important feature that combustion takes place in a separate chamber away from the piston rod, which is therefore not subjected to the extreme combustion temperature. This chamber is also utilised to house the valves, which are thereby rendered more accessible for operation and overhaul.

The lower portion of the cylinder is capable of being lowered by means of the main crosshead, thus exposing the piston (see fig. 45).

Sulzer Marine Diesel Engines.

Messrs. Sulzers' experience of both four-cycle and two-cycle engines has led them to adopt the two-cycle engine for all marine work, and they have standard designs of 4, 6, 8, and 10 cylinder marine engines developing from 300 to 7,200 b.h.p., the latter being a unit of 8 cylinders, or 900 b.h.p. per cylinder.

The Sulzer two-cycle engine has important differences from the other engines described. Chief amongst these is the method of scavenging through a double row of ports, the upper row of which is above the exhaust port opening and is controlled by means of a valve.

On the downward stroke of the piston, the upper row of scavenging ports is first uncovered, but the valve controlling these ports remains shut and the exhaust commences to take place only when the piston has descended further and uncovered the exhaust row of ports

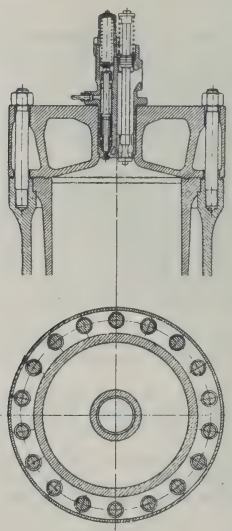


Fig. 46.—Sulzer Cylinder Cover.

Scavenging commences as soon as the lower row of scavenging ports is uncovered and continues through both rows of ports, the control valve for the upper ports having been opened by the pressure of the scavenging air, and air is admitted through the upper ports to the cylinder, after the exhaust ports are closed, as long as the scavenge air pressure exceeds the pressure in the cylinder.

Not only is the scavenging much improved in this manner, but it will be seen that it is possible to obtain a supercharge of air, *i.e.* a slight increase above the atmospheric pressure in the cylinder at the commencement of compression.

Messrs. Sulzers have found that the good combustion obtained by this method of scavenging, and also the simplicity of the cylinder cover design (fig. 46) enables them to maintain with safety a higher mean pressure in their two-cycle engines than in four-cycle engines.

The power for a given size of cylinder on the Sulzer two-cycle type is more than double the power of a similar size cylinder on the four-cycle type, both being assumed to run at the same piston speed.

The fuel consumption is also practically the same as the best four-cycle engine when working under service conditions, *viz.* 0·42 lb. per b.h.p. hour, including scavenge blowers.

For some engines a direct driven scavenging pump is fitted between the h.p. air compressor at the end of the engine and the main engine cylinders.

In the latest type of Sulzer engine the scavenging pumps are in the form of independent rotary blowers electrically driven. This is an advantage from the constructional point of view, and the arrangement has the further gain that the scavenging pump may be speeded up independently of the main engine when it is desired to increase the power. This does not involve any increase in the size of the electric generating sets where electrically driven winches are installed, as these latter are only running in port when the scavenging pump is idle.

An engine, when made with separate turbo-blowers, develops 3,200 b.h.p., working at approximately the same indicated mean pressure and piston speed as the 2,900 b.h.p. engine with reciprocating scavenge pumps.

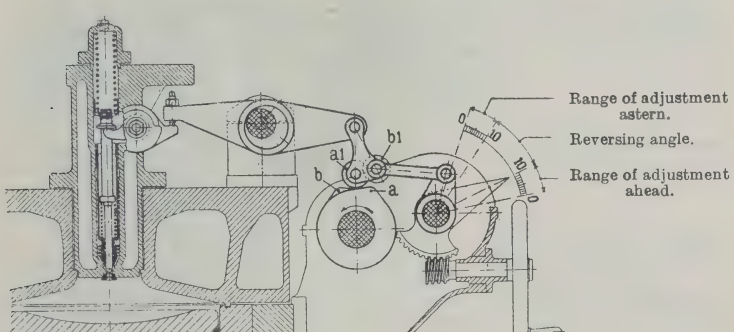


FIG. 47.—Sulzer Reversing Mechanism.

Reversing is carried out with a minimum requirement of power, and is effected by bringing a second set of rollers and cams into operation (fig. 47). Each starting and fuel valve therefore has one cam and roller, a and a^1, for running ahead, and another, b and b^1, for running astern. The hand wheel shown in the above illustration brings a and a^1 or b and b^1 into operation in accordance with the required direction of running.

The weight complete of Sulzer Diesel Marine Engines is approximately 2 cwt. per h.p., *i.e.* 10 h.p. per ton.

Submarine engines of Sulzer two-cycle type have been built developing 4,000 h.p. in 8 cylinders, these engines being the most powerful marine Diesel engines in the world of the light type.

The first large passenger liner driven by Diesel engines is the *Aorangi*, which has engines of the Sulzer type, manufactured by the Fairfield Shipbuilding and Engineering Co., Ltd., at Govan.

THE DOXFORD OPPOSED-PISTON MARINE OIL ENGINE.

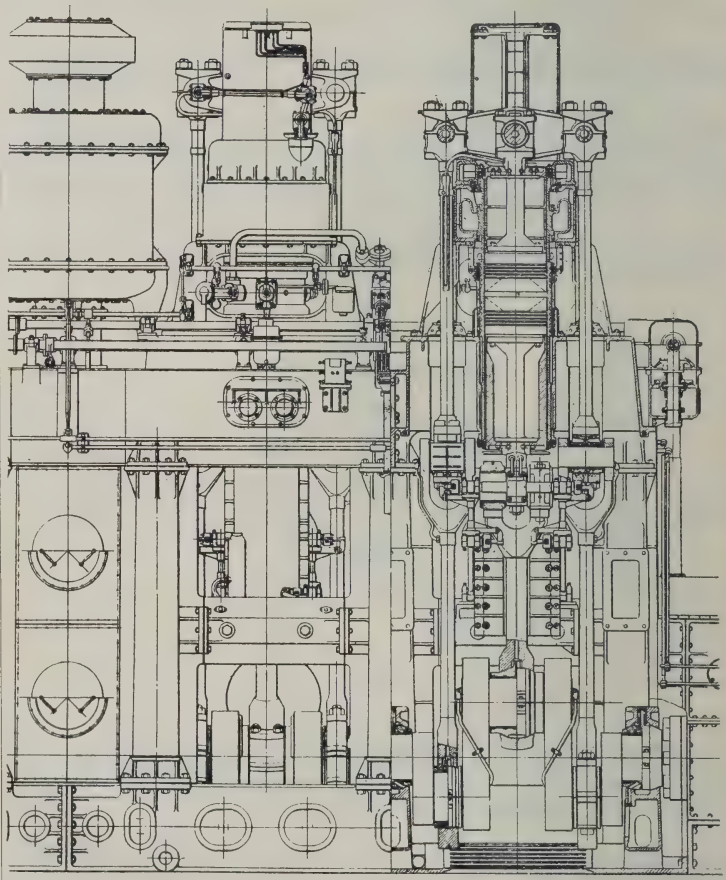


FIG. 48.

The Doxford Engine.

Messrs. Wm. Doxford & Sons, Ltd., have completed a number of motor vessels, and have now built and are building an aggregate h.p. of 400,000, consisting of four cylinder types, single screws, 6,000 i.h.p. at 95 revs.; twin screws, 12,000 i.h.p. at 120 revs.; quadruple screws, 15,000 i.h.p. at 125 revs.; and three cylinder types, single screw, 2,800 i.h.p. at 93 revs., all of the opposed piston two-stroke type operating on the Doxford system of airless injection.

Fig. 48 (page 2422) shows part of the four-cylinder type engine, four of which were fitted into M.S. *Bermuda*, in each cylinder of which there are two pistons working. The lower piston is coupled by crosshead and connecting-rod to a crank below the centre line of the cylinder, and the upper piston acts upwards to a transverse beam extending fore and aft beyond the walls of the cylinder, and coupled by two side connecting-rods to two crankshafts, with the two side cranks arranged at 180° to the centre crankpin.

The stroke of the lower piston is more than the stroke of the upper piston, being inversely proportional to the respective reciprocating weights, whereby a complete balance is obtained.

The fuel is injected between the pistons when they are together, from two fuel valves diametrically opposite each other, and when combustion takes place the pistons are driven apart, thus exerting an almost perfect turning moment on the crankshaft without re-actional load on the main bearings or framework of the engine.

The products of combustion are exhausted through ports round the whole circumference at the top of the liner, which ports are controlled by the upper piston. At the end of the stroke, corresponding ports are uncovered by the lower piston in the bottom end of the liner, to control the scavenging air, thus the entire cylinder is scavenged on the 'Uniflow' principle; the covering of these ports at the beginning of the return stroke traps the remaining air, which is thus compressed ready for the injection of the fuel and repetition of the cycle.

Fuel injection is on the Doxford 'Solid Injection' principle, and the fuel oil is injected transversely across the cylinder in the form of a fan, from each valve placed diametrically opposite, the fan forming a curtain right across the combustion space. The air which is entrapped between the two pistons receives through suitably arranged ports a rotating motion whereby the air is fed through the curtain of fuel oil, so that very efficient combustion is obtained without the oil impinging on to the piston or cylinder walls. The two pistons are, moreover, hollow, and form between them practically a complete spherical space, the surfaces of which are kept at 1000° F., thus greatly assisting the burning of the heaviest grades of boiler oil.

The elimination of exhaust and induction valves, with their liability to defects, is a great advantage, and in dispensing with the cylinder cover many defects of jointing and cooling are obviated. Also, the absence of the high-pressure compressors removes the most delicate part of a Diesel engine, and the danger from possible explosions in the injection air line is completely avoided.

The valve gear is very simple, consisting of one camshaft in front and one at the back of the engine. Both are driven by spur and bevel wheels direct from the thrust shaft. The front camshaft has two sets of cams for the fuel and air starting valves for ahead and astern running, either of which is brought into operation by sliding the cam shaft by means of a hand lever. When going ahead, the front and back fuel valves are in operation, but when going astern only the front fuel valves come into use, giving approximately 60 per cent. of full power, which is equal to the resistance obtainable from the propeller. A hand-wheel controls the air-starting valves and the fuel valves. By rotating the hand-wheel, the air-starting valves are brought into contact with their cams, and air is admitted to the respective cylinders. By a further rotating of this hand-wheel, the air-starting valves automatically come out of action and the fuel valves are thrown into use, and by further rotating the hand-wheel, the speed and power of the engine is regulated. The small hand-wheel controls the suction valves of the fuel pump for regulating the required fuel oil pressure.

An Aspinall governor operates the injection fuel valves, and at the same time acts upon the fuel pump suction valves; thereby racing under any condition is completely prevented.

The cylinder liners and pistons are all cooled with fresh water, and forced lubrication is employed in the crank chamber throughout.

Messrs. Doxford have run extensive trials on one of their four-cylinder 3,400 i.h.p. engines, submitted to the Trial Committee appointed by the Naval Arch. and Inst. of Mech. Engrs., and have proved a consumption of 0·4 lbs. per b.h.p. on 0·96 British Mexican fuel oil, and a mechanical efficiency of 89 per cent. Recently further improvements have been made, and an engine which was exhibited at the North-East Coast Exhibition was submitted to Professor C. J. Hawkes, M.Sc., who proved the consumption on Anglo-Persian oil to be 0·35 lbs. per b.h.p. per hour.

Messrs. Doxford claim to be the first to introduce aboard vessels centrifugal treatment of fuel oil, thus enabling shipowners to burn any brand of fuel oil that may have to be taken aboard.

Scotts' Two-Stroke Marine Oil Engines.

One of the outstanding features of the two-stroke marine oil engine, manufactured by Messrs. Scotts' Shipbuilding and Engineering Co., Ltd., of Greenock, is that means are provided for recovering a portion of the heat contained in the exhaust gases, and also for utilising the whole of the heat passing through the cylinder jackets for the generation of steam. The steam thus raised may be utilised in various ways. For example, in the machinery of the *Dolius* it is admitted to the underside of the combustion cylinder pistons to augment the power, and assists also in driving the scavenge air-blower, which is a separate unit. In the case of the larger engines fitted in the *Eurybates*, the steam is used in separate cylinders, contributing power to the main crankshaft and also supplying the whole of the power necessary for driving the turbo-blowers which deliver the scavenge air to the main engines. In both of these cases starting of the main engines is accomplished by steam pressure, and the arrangements have been very successful in service. To meet the demand for a cheaper engine, however, a modified design is available in which the steam generated by waste heat is used solely for driving the turbo-blowers, and the main engines are started by means of compressed air, as usual on Diesel engines. A cross-section of part of a six-cylinder engine of this type to develop 3,000 b.h.p. at 105 r.p.m. is shown in fig. 49. Each cylinder is 27 ins. diam, with a 45-in. stroke, and works on the two-stroke cycle with port scavenging and solid injection of fuel. A rotary valve of simple form is fitted at the exhaust gas outlet from each cylinder, having a supercharging effect. This permits the use of a short-skirted piston, which minimises weight, friction, and height of engine, and allows inspection or cleaning of the ports from the space below. These valves are driven by spiral gears from the crankshaft, have provision for expansion, and the spindles run in bearings suitable for the temperatures prevailing. The fuel pumps are grouped together at a convenient position and driven by a train of wheels from the crankshaft. One pump is provided for each cylinder. The plunger is actuated by a crank and delivers oil at 4,000 to 6,000 lbs. per sq. in. pressure to automatic jets situated at the cylinder head. An automatic air starting valve is also fitted on each cylinder, the manœuvring gear being grouped at the control panel eblow. No camshafts with overhead driving gear are required, so that the engine is extremely free from gear, with consequent ease of operation and overhaul. Fresh-water cooling arrangements are applied to the pistons, the water being introduced and taken away through the medium of trombone pipes attached to the cross-heads, with an independent pump for circulating under pressure. For the top ends of the main connecting rods a special type of bearing is fitted, originated by the makers, which renders it unnecessary to supply high-pressure oil to this part. All working parts inside the crankcase are supplied with forced lubrication, and mechanical lubricators are fitted for the cylinder liners. Starting air is stored in receivers, replenished by independently driven air compressors. The engine is of rigid construction, and through bolts are provided to transmit the combustion load from the cylinders to the bedplate.

Possessing a high mechanical efficiency, this type of engine is economical in fuel consumption, and requires no auxiliary power for driving the scavenge air-blowers.

In cases where no waste-heat recovery is required, a similar engine is supplied, with scavenge pumps direct or independently driven, and water circulation for the cylinder jackets, retaining the same general construction.

Messrs. Scotts also manufacture four-stroke engines of large powers of a type developed by them in association with Messrs. The North-Eastern Marine Co., and Messrs. Alfred Holt & Co., the well-known Liverpool shipowners. These engines follow usual four-stroke practice and are of very substantial construction.

Vickers' Engine.

Messrs. Vickers, Ltd., of Barrow (now part of Vickers-Armstrongs, Ltd.), were the pioneers of the solid or airless injection type of Diesel engine, and have built these engines in a wide range of powers for submarines and for commercial marine purposes.

The engines are of the four-stroke single-acting type, and the commercial engines have external crossheads, water-cooled pistons, and the crankcase is separated from the cylinders by an oil-tight diaphragm.

The fuel pumps have four or more units discharging to a common main from which the spray valves derive their fuel supply. The spray valves are mechanically operated, the duration of opening being variable according to the load.

Reversing of Vickers engines is usually arranged by means of a sliding shaft with two sets of cams, one set for ' ahead ' and one for ' astern ' running. The reversing gear first lifts the valve actuating levers clear of the cams, then slides the camshaft along, and finally lowers the levers to contact with the second set of cams.

SCOTT TWO-STROKE MARINE OIL ENGINE.

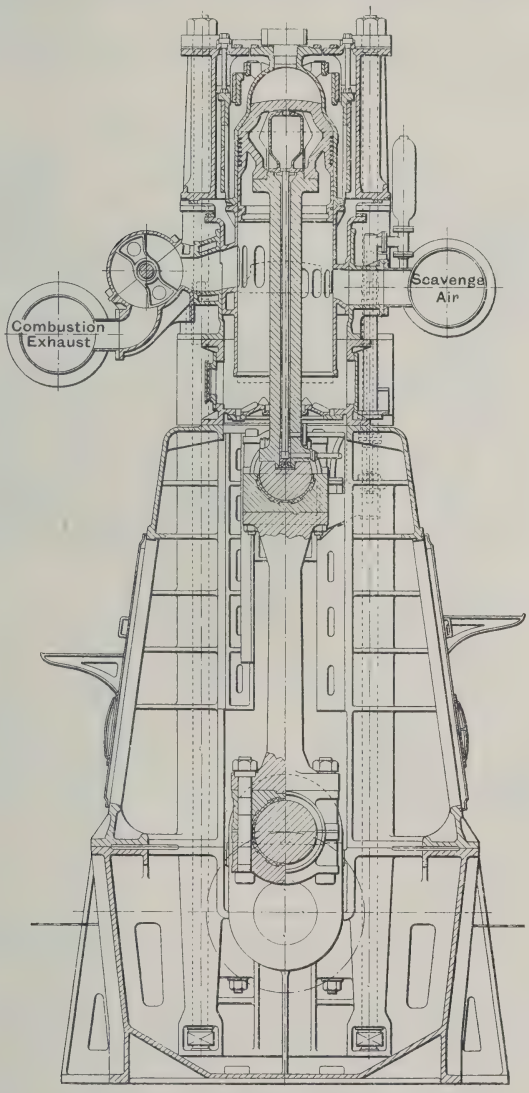


FIG. 49.

VICKERS' FOUR-STROKE MARINE OIL ENGINE.

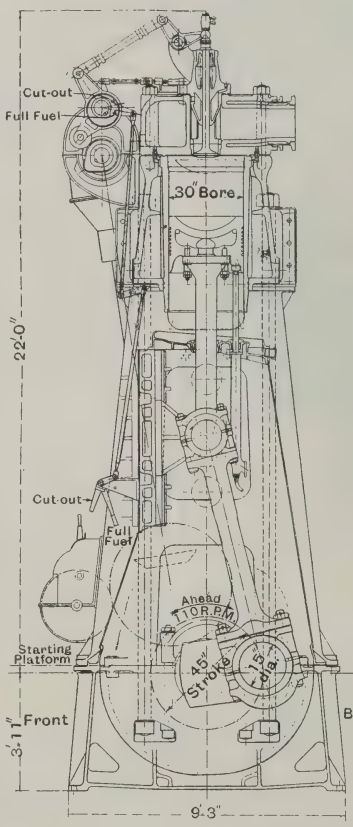


FIG. 50.

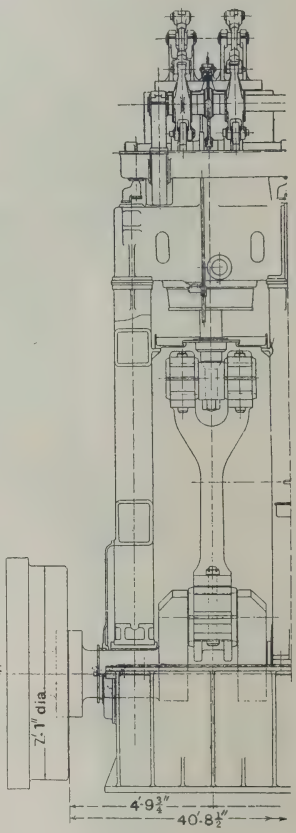


FIG. 51.

The cylinder covers for the larger engines are made with a detachable steel plate on the combustion side, this plate having clearance in the securing holes to allow for expansion relative to the main body of the cover.

The illustration (figs. 50 and 51) shows a portion of the largest engine of this type, which has six cylinders, 30-in bore and 45-in. stroke, and develops 2,000 b.h.p. at 110 r.p.m.

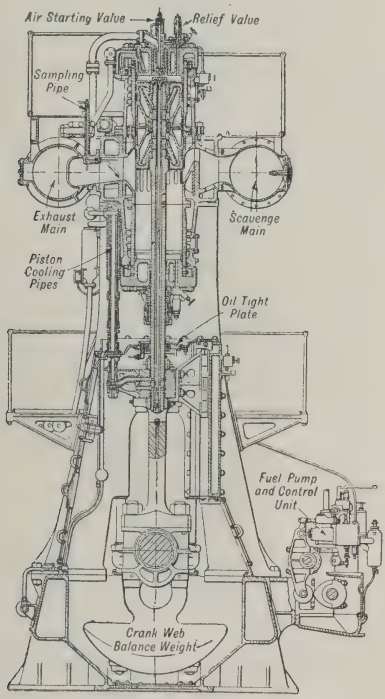


FIG. 52.—Richardsons-Westgarth Double-acting Engine.

M.A.N. ENGINE.

The engine designed and constructed by Messrs. Maschinenfabrik Augsburg-Nürnberg, and also built under licence by Messrs. Vickers-Armstrongs, Barrow, is of the two-stroke cycle type, both single- and double-acting.

The outstanding feature of these engines is the patented scavenge system. The exhaust and scavenge ports are above one another. The scavenge ports are so shaped and directed that the air current is directed towards the piston and then flows along the opposite side of the cylinder to the cover. It is there deflected and follows the exhaust gases to the exhaust ports.

The efficiency of this system is demonstrated by the fact that the scavenge pressure required is very low, about 1¼ lbs. per sq. in.

These engines are supplied either with air-blast or solid injection.

The illustration fig. 53 shows a typical section view of the M.A.N. double-acting engine.

The highest-powered Diesel engine in service is a non-reversing M.A.N. double-acting engine, built by Messrs. Blohm and Voss, and installed in the Hamburg Electricity Works.

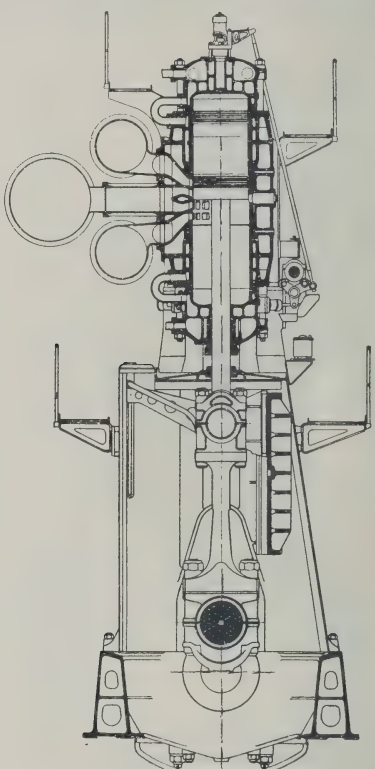


FIG. 53.—M.A.N. Double-acting Engine.

This engine has nine cylinders 33·9 in. in diam., and 59-in. stroke, and develops 15,000 b.h.p. at 94 r.p.m.

THE RICHARDSONS-WESTGARTH ENGINE.

Messrs. Richardsons, Westgarth & Co., Ltd., of Hartlepool, have produced a double-acting two-stroke cycle engine, which is noteworthy in many respects, and the first unit built for sea service was fitted in the oil tanker *Irania*, which sailed on its first voyage in April 1929.

The engine is all-British in design, and is worked on the airless injection principle, with controlled fuel pump delivery and automatic fuel valves.

The principal features of the engine are shown in the view of the *Irania's* engine (fig. 52).

The liner is fixed at its centre and free to expand at both ends, the cover-joint being a spring loaded lead alloy wedge ring. A special heat-insulating ring extends into the combustion chamber a few inches and fits closely against the liner walls up to the point reached by the piston. This

protects the walls of the liner in way of the fuel jets and also reduces the heat rejection to the cooling water. The top and bottom cylinder covers are interchangeable, the passage for the piston rod in the bottom cover is arranged to house the starting air valve in the top cover.

The top and bottom pistons are interchangeable and are secured to the piston rod by an external nut. Cooling water enters the piston rod at the crosshead by way of telescopic pipes, and passes in series through the lower and upper pistons and thence back outside the conveyor tube.

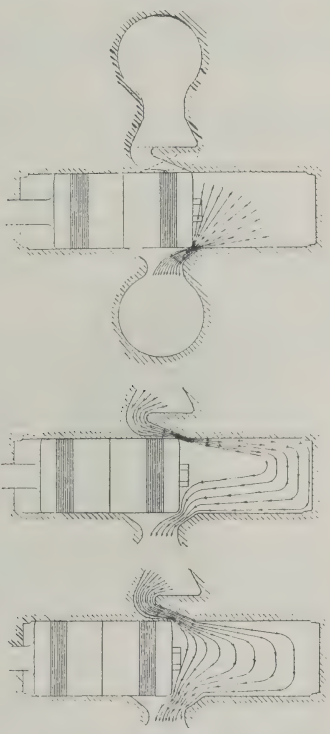


FIG. 54.—Scavenging Arrangement.

Fig. 54, shows a diagrammatic arrangement of the scavenging, in which alternate ports deliver air to top and bottom cylinders respectively. The air is directed towards the combustion space, and the length of liner is kept to a minimum.

The scavenge pump is of original design with two cylinders driven at three times the engine speed by spur gearings from the thrust shaft. The scavenge valves are chain-driven rotary type.

The fuel pump is cam operated and spring returned, and is driven from a camshaft at the crankshaft level. The time of opening of the injection valves is controlled by altering the time of closing of the pump suction valve, while their rapid closing is effected by the opening of a spill valve, the timing of which is also under control.

Two injection valves are fitted vertically into each cylinder cover. Each valve has five jets, four of which spray horizontally—two each side of the piston rod or nut, and the fifth one vertically. the oil from this fifth jet being finely atomised by a rotary motion, while the four horizontal jets have greater penetrating power.

The starting valves in the cylinders are automatic, the opening of the valves being controlled by timing valves operated by cams on the fuel pump camshaft.

F.I.A.T. Engines.

These engines are of Italian design, and generator engines of this type are constructed by the Fiat British Auxiliaries Ltd.

A wide range of single-acting engines have been manufactured to meet the requirements of generators, submarines, and the marine service; and an experimental double-acting has been exhaustively tested.

The scavenge ports which are uncovered at the same time as the exhaust ports are controlled by multiple plate automatic valves, which have replaced the earlier piston type. The convex piston top helps to direct the scavenge air to the top of the cylinder.

The piston heads and cylinder covers are of cast steel instead of cast iron, and the cylinder cover extends downwards into the liner, so protecting the heavy liner flange from the heat of combustion.

The cylinders are entirely separated from the crank chamber by a diaphragm plate, the piston rods passing through light shifting-boxes into the crank chamber. The object of this is to prevent any carbonised oil from the cylinder walls or any water leakage from the telescopic tubes from mixing with the oil in the crank chamber, but it has the disadvantage of adding considerably to the height of the engine.

Krupp Marine Diesel Engines.

Two-Cycle Single-Acting Marine Engines.

The Krupp two-cycle marine Diesel engine is built for powers from 1,300 b.h.p. up to 6,500 b.h.p. Two complete ranges are constructed. The first with a stroke bore ratio of about 1·85 and the second with a stroke bore ratio of about 1·5, both working at a piston speed of 810 ft. per minute. A section through a 2,500 s.h.p. engine having six cylinders 26·7 ins. diam. by 51·1-in. stroke running at 90 r.p.m. is shown in fig. 55.

A special feature of the engine is the scavenging arrangement. The working cylinders are cast in pairs, each block being rigidly bolted together and to the columns. There is one double-acting scavenging pump for each pair of cylinders. The pumps are mounted directly to the side of the cylinder blocks and in line with one of the cylinders. The casting incorporates the scavenging trunk, and the receivers of the several pumps are interconnected. Large suction discharge valves are fitted in the pump housing. This arrangement permits extremely short air passages from the pump to the working cylinders, so that fluctuations in pressure usually experienced in long scavenging trunks are avoided. For efficient scavenging only 28 per cent. in excess of the swept volume of the cylinder is necessary, and at full load the scavenging air pressure is about 1·4 lbs. per sq. in.

The arrangement of the ports in the cylinder liner is such that the scavenging air enters at the back, and the exhaust ports are arranged on either side, which gives good scavenging as there is less chance of the scavenging air blowing straight through the exhaust ports without first scavenging the cylinder.

The drive of the scavenge pump is taken from the crosshead by means of a heavy forged steel bracket. This bracket also carries the piston cooling inlet and outlet telescopic pipes. These two pipes work in a chamber outside the crank case and are connected with fixed piping to the piston head. The piston itself is made in two pieces, the top portion being of forged steel without internal ribbing, and the load is transmitted through the walls to a heavy flange at the top of the piston head. The piston skirt, which is of special cast iron, is secured at the bottom to a similar flange on the piston rod.

The cylinder cover is a simple and light casting, and is secured to the cylinder heads by means of a cast-steel ring which bears on a stepped flange, so that the stresses from the holding-down bolts are evenly distributed. It is provided with a central opening to accommodate the valve housing, which carries only a fuel, starting air, and relief valve. The cylinder covers, valve housings, and pistons are cooled by sea-water, a special device being arranged to ensure intensive cooling at the top of the liners and in the covers. At the forward end of the engine is a three-stage air compressor, providing the air for injection and for filling the starting air storage bottles.

The camshaft is on a level with the top of the cylinders and is driven by a vertical shaft. Ahead and astern cams are provided, and the operation of reversing is accomplished by a longitudinal displacement of the camshaft. The starting air valve is so arranged that air can only enter the cylinder provided the pressure in the latter is not greater than 20 atm., so that both fuel and starting air can be admitted during the same stroke. The starting-air cam only controls the closing of the valve, and this arrangement allows for quick starting and economy in the consumption of starting air.

The reversing gear is connected to the engine-room telegraph answering lever in such a manner that a movement of the lever from the ahead to the astern side of the dial sets the reversing control gear in the astern position. The operation of starting the engine by the controls automatically moves the camshaft to the correct position and opens up the supply of both starting and injection air.

The pressure of the injection air is automatically controlled to suit the engine speed, as is also the air compressor suction, so that when charging the storage air bottles the compressor adapts itself to the increased load and at the same time maintains the requisite injection air pressure. Each cylinder has its own individual fuel pump, the pumps being accommodated in a common housing, and are driven from the vertical shaft. The engine is lubricated throughout with forced

lubrication. The crosshead bearings are supplied with oil at high pressure by a small pump mounted directly to each crosshead, the pumps being worked by bell-crank levers operated by the movement of the connecting-rod top end.

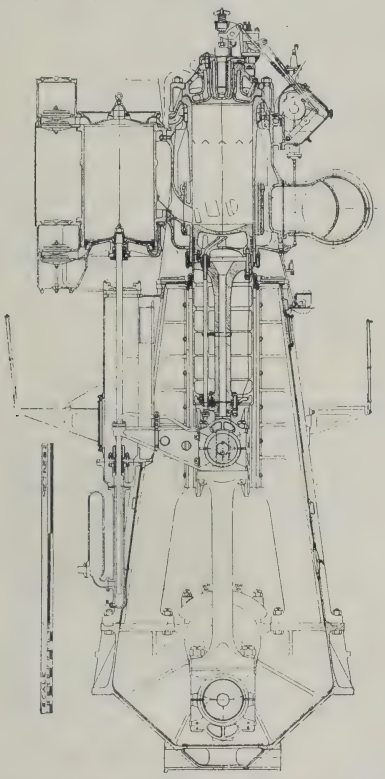


FIG. 55.—Section through Working Cylinder of Krupp Two-Cycle Engine.

On trial the Krupp two-cycle engine shows a fuel consumption of about 0·40 lb. per b.h.p. hour at full load, and a mechanical efficiency of about 78 per cent. The engines work at a normal m.i.p. of 82 lbs. per sq. in.

KRUPP FOUR-CYCLE MARINE ENGINES.

These engines are of the trunk piston type and work on the airless injection principle. They are designed for powers ranging from about 100 b.h.p. to 3,000 b.h.p., and are directly reversible. As before, two ranges of standard engines are constructed, the first with a stroke bore ratio of about 1·75, and the second with a stroke bore ratio of about 1·5, the piston speed in each case being about 1,000 ft. per minute.

The engines are of the totally enclosed type, the cylinders and crank cases of the box pattern being mounted on an oil-tight bed-plate. Forced lubrication is employed throughout, and in engines of the larger sizes pistons are cooled by oil. In the smaller types the pistons are fitted with a special cast-iron mushroom insertion piece, which is insulated from the remainder of the piston and acts as a heat accumulator, thus ensuring reliable ignition.

As in the two-cycle engine the camshaft is on a level with the top of the cylinders and reversing is accomplished by a longitudinal displacement of the shaft. Reversing mechanism is controlled

as before by the engine-room telegraph answering lever. The same type of automatic starting air valve is employed, so that the fuel and starting air can be admitted during the same stroke, thus facilitating very quick manœuvring.

The Krupp four-cycle marine engine shows on trial a fuel consumption of about 0·39 lb. per b.h.p. hour and a mechanical efficiency of 80 per cent. at full load. The engines normally work at a mean indicated pressure of 94 lbs. per sq. in.

The Tosi Diesel Engine.

These engines are of the enclosed design four-stroke cycle type, and are noteworthy for the elegance of their appearance.

A large number of engines have been supplied for submarines and other work in sizes up to 4,000 b.h.p., and they give a fuel consumption of 0·4 lb. per b.h.p. at full power.

Before standardising the four-stroke cycle engine, Messrs. Franco Tosi constructed a two-stroke cycle engine, so that they could take comparative tests. The latter engine had six cylinders and developed 1,300 b.h.p. when running at 300 r.p.m. The results of the tests showed that their eight-cylinder 1,450 b.h.p. engine was much more economical in fuel.

The outstanding feature of the Tosi design is a director valve fitted in conjunction with a combined valve in the cylinder head, which alternately admits air to the cylinder and allows the exhaust gases to escape from it, depending on the position of the director valve. In this way the cylinder valve is kept cool and free from carbon.

The latest Tosi engines are arranged for supercharging.

Double-acting Beardmore-Tosi engines have been installed in the converted steamer *Wulsty Castle*, driving the propeller shaft at 80 r.p.m. through Vulcan hydro-mechanical gearing, the engine speed being 250 r.p.m.

See also p. 1898.

The 'White' Diesel Engine.

The Diesel engines made by Messrs. J. Samuel White & Co., Cowes, were first used for submarines, and they are now supplied for all marine purposes.

One of the White 450 b.h.p. two-stroke cycle engines was subjected to severe tests in order to prove its capabilities as regards power, flexibility, and endurance. A 72-hour full load trial was satisfactory, and it was proved that the engine did not require vigilant attention. The exhaust was practically invisible. At the end of this run the engine was slowed down to two-thirds speed and manœuvring trials were conducted. It was seen that 14 cubic feet of air at 900 lbs. per sq. in. was sufficient for starting and reversing the engine 42 times, the pressure of air going down to 90 lbs., which was insufficient for starting.

As an example of flexibility, the tests to determine the minimum speed consistent with smooth running of the engine are interesting. The speed was gradually reduced from 400 down to 150 r.p.m. with suitable power and without alteration to fuel pumps, etc. It was found that 150 r.p.m. could be maintained without fluctuations, and with a steady blast pressure of 630 lbs. per sq. in.

The White engines are vertical, with stepped pistons, the working piston having a clearance of about 0·024 in. and relying for gas-tightness wholly upon its rings, while the lower portion fits the cylinder liner and takes the side thrust of the connecting-rods; this lower portion is always comparatively cool, and hence is easily kept effectively lubricated. An important feature of the latest design is the rotating scavenge valves, the sleeve within which they work forming part of the reversing mechanism of the engine.

Other Types of Marine Diesel Engines.

Among the other British makers of Diesel engines for marine work are Messrs. Harland and Wolff of Glasgow and Belfast, and John Kincaid & Co., Greenock, on Burmeister and Wain design; Messrs. Barclay Curle, Glasgow, and Richardson, Westgarth, Hartlepool, on the Doxford design; and Messrs. Armstrong-Whitworth, Newcastle; John Brown & Co., Glasgow; Wm. Denny Bros., Dumbarton; Fairfield Shipbuilding and Engineering Co., Glasgow; Alexander Stephen & Sons, Glasgow; Wallsend Slipway, Newcastle; and Workman Clark, all on the Sulzer design—all of these makers are well known in marine engineering, and in many instances embody in the engines they manufacture modifications from the patentees practice, to meet the special requirements of a marine installation, and in this way uphold the high traditions of British marine practice.

Diesel Engines for Submarines.

Practically all the European and American navies are now solely relying upon Diesel engines for the propulsion of submarines on the surface. These motors are of totally distinct type from those of the ordinary marine engine owing to the necessary high speed in order to reduce the weight sufficiently. They usually run at 400 to 500 r.p.m., and they are now constructed in powers up to 1,300 b.h.p. In the German and several other European Navies the Krupp two-cycle engine was employed, and also that of the M.A.N. Co., which is likewise of the two-cycle single-acting type. The former is a six-cylinder machine with two scavenge cylinders in the

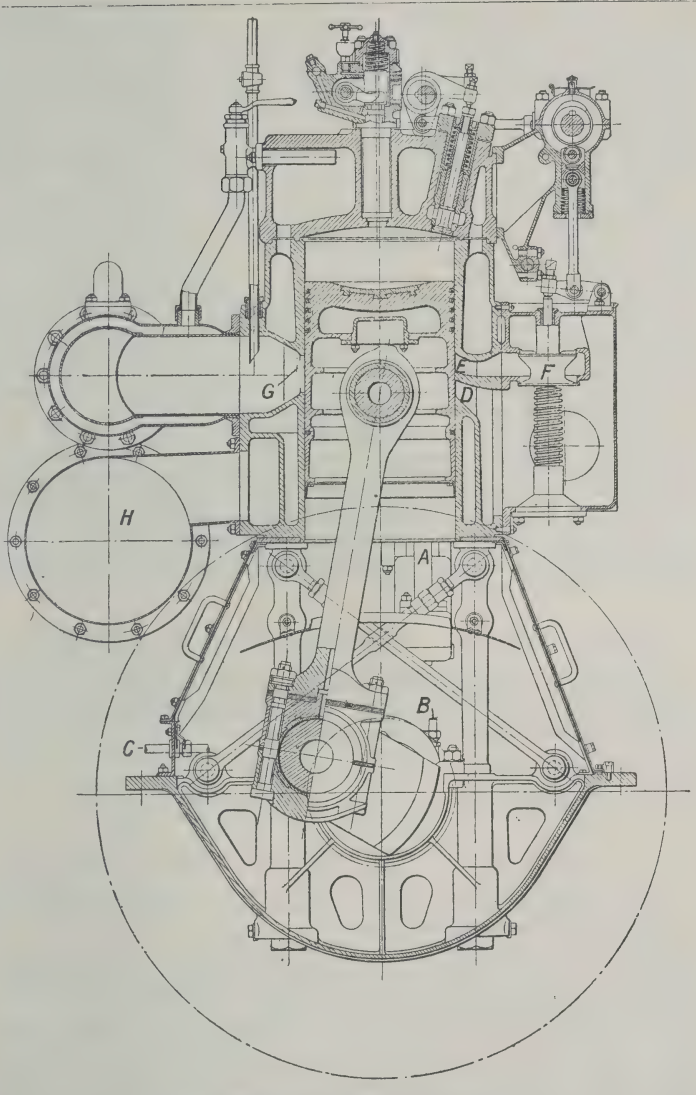


FIG. 56.—SULZER SUBMARINE DIESEL ENGINE.

centre driven direct off the crankshaft, whilst in the latter the scavenge pumps are arranged directly beneath each working cylinder. The air pumps for air injection are also driven off the engine, so that it is practically self-contained. The weights of these motors are low, and usually somewhere in the neighbourhood of 40 to 60 lbs. per b.h.p. For the British Navy the Vickers type is usually employed, which is a four-cycle twelve-cylinder motor of about 1,200 b.h.p. In this engine, solid injection of fuel is always adopted—that is, the injection by mechanical means instead of compressed air. The advantage of this is that no air compressor is required, which is very desirable in a submarine. Although the engine as a Diesel engine is not quite so efficient as with air injection, the absence of air compressor renders the overall economy about the same as in other types of motors. It is doubtful, however, whether such heavy fuel may be satisfactorily employed as with the other engines, and Broxburn shale oil is generally used in these submarines. Two-cycle engines have also been designed for operation with mechanical injection.

Other types of Diesel engines are now used in British submarines, and one of the first of the two-cycle types to be employed for the purpose is the F.I.A.T., which was installed in the first submersible built by Scott's Shipbuilding and Engineering Co. In this engine the scavenge pumps are arranged beneath the working cylinders as with the M.A.N. engine previously mentioned, the pistons being stepped in the ordinary manner. Other types of two-stroke Diesel motors will be installed in British submarines, if they have not already been so fitted.

In order to keep the piston speed of submarine engines within the limits which it is not desirable to exceed (from 900 to 1,100 ft. per min.), it is usual to make the ratio of the stroke to the bore very much less than with the ordinary mercantile type of motor. Frequently the stroke is made equal to the diameter of the cylinders, and is very seldom more than 10 per cent. greater. In this way it is possible to construct Diesel for submarine purposes almost equal from the point of view of reliability to the slow-running types. High-speed engines of 1,300 h.p. have been constructed for installation to submarines, and others of over 2,500 b.h.p. in eight cylinders have been designed, and are possibly now being built for the same purpose.

A particularly important type of submarine Diesel engine is illustrated in fig. 56. This is a Sulzer motor as employed in several navies, and mainly follows along the lines of the slow speed mercantile Diesel motors built by the same firm and described in the preceding pages. The most notable feature is that port scavenging is employed and the method of introduction of an extra supply of scavenging air through valve controlled auxiliary ports is the same as in the larger sets. It had been thought that the high speed of rotation (350–500 r.p.m.) might render this system inefficient for submarine engines, but this does not appear to be the case. Motors of this design are being built in powers up to 4,000 b.h.p. per engine.

In their latest submarine motors the F.I.A.T. Company have also adopted the port scavenging arrangement, and engines of 1,300 h.p. in eight cylinders built according to this principle have already been installed in Italian submarines.

Output, Dimensions, and Weights of Diesel Marine Engines.

The design of Diesel engines in regard to the main dimensions of cylinders, etc., is not yet on a very definite or exact basis, mainly owing to the variations of mean effective pressure in the cylinders, and the uncertainty regarding the best speed of revolution. The output of a cylinder varies with the diameter, the stroke, the mean effective pressure, and the speed of revolutions. In general terms it may be expressed as follows:

B.H.P. = KD^2LNn, where K = a constant; D = diameter of cylinder; L = stroke of piston; N = revolutions per min.; n = number of cylinders.

Taking the dimensions of the cylinders in inches it is found that for four-cycle engines the value of K varies between 0·00007 and 0·00009, a good average figure being

$$K = 0{\cdot}00008.$$

For two-cycle engines the value is between 0·000125 and 0·000155, the average being

$$K = 0{\cdot}00014.$$

As the mean effective pressure varies between 70 and 105 lbs. per square inch, and is sometimes higher, it is evident that cylinder dimensions are no absolute criterion of the exact output of a Diesel engine at a constant speed of revolution.

The piston speed in two- and four-cycle engines is approximately the same, namely, about 3 to 4 metres per sec., or 580 to 780 ft. per minute, with the ordinary marine motors. It does not often exceed these limits, except in high-speed engines of the type employed in submarines, when it may be as much as 1,100 ft. per minute. In the Burmeister and Wain double-acting four-cycle type the piston speed is about 1,200 ft. per minute.

The ratio of stroke to bore is a matter of wide variation, and in some cases is as low as 1·3 and others as high as 1·8. This latter figure is common in two-cycle motors with valve scavenging. Some types of four-cycle motors have a ratio of 1·4 and others 1·8.

The weight of Diesel marine engines varies as a rule between 200–300 lbs. per b.h.p., but is reduced to 60–70 lbs. per b.h.p. in submarine motors. Four-cycle engines are heavier than two-cycle, but the difference is not very marked, except in special cases, average figures being 300 lbs. and 250 lbs. respectively. This includes air compressors, scavenge pumps (in two-cycle motors) and various bilge, oil, and circulating pumps driven direct by the engine.

Power of Diesel Marine Engines.

The following tables give details of the dimensions and powers of some of the types of Diesel marine engines as now built :—

Type of Engine.	B.H.P.	Diam. Ins.	Stroke. Ins.	R.p.m.	No. of Cylinders.
Two-Cycle Single-Acting Engines.					
Doxford opposed piston	1,850	22·8	45·5 + 2	83	3
	2,900	23·6	30 + 41	112	4
Fiat	1,600	20·5	32·3	150	6
	2,250	29·5	49·5	110	4
Krupp	1,750	26·7	51·1	92	4
	2,500	26·7	51·1	90	6
M.A.N.	1,400	21·3	35·4	110	6
	2,150	27·5	47·2	90	6
Nobel	500	13·4	22·4	210	4
	2,000	26·5	47·2	100	4
Polar	2,000	23·6	47·2	100	6
Still	2,500	24·0	45·0	110	5 + 2
Sulzer	675	13·4	21·2	220	6
	4,000	28·0	39·0	110	8
	5,000	35·4	63·0	90	5
	7,200	30·0	52·8	100	10
Two-Cycle Double-Acting Engines.					
Burmeister and Wain	7,000	24·4	55·0	100	6
Deutsche Werke	4,200	25·6	41·3	115	6
M.A.N.	3,700	23·6	35·4	135	6
	15,000	33·9	59·0	94	9
Richardsons-Westgarth	1,250	21·5	38·0	90	3
Worthington	4,000	28·0	40·0	128	4
Four-Stroke Single-Acting.					
Burmeister and Wain	1,100	20·5	28·5	140	8
	3,300	29·1	59·0	115	8
Deutsche Werke	2,500	29·5	47·2	115	6
Krupp	1,400	25·5	39·4	125	6
M.A.N.	1,750	23·6	27·6	215	6
McIntosh and Seymour, B. & W.	2,700	32·0	60·0	95	6
North-Eastern, Werkspoor	1,100	22·0	39·4	125	6
	2,300	28·7	51·2	93	8
Tosi	1,255	24·4	38·4	120	6
Vickers	1,250	24·5	39·0	118	6
	2,700	30·0	45·0	110	8
Four-Stroke Double-Acting.					
Burmeister and Wain	6,750	31·5	55·1	125	6
	10,000	33·1	59·0	125	8
Beardmore-Tosi	800	20·0	24·5	240	3
McIntosh and Seymour	2,700	32·0	52·0	95	4
North-Eastern, Werkspoor	4,000	32·3	59·0	90	6

The Büchi System of Exhaust Turbo-Charging.

The Büchi system of Exhaust Turbo-Charging (which is the invention of Dip. Ing. Alfred Büchi of Winterthur, Switzerland) is a pressure intake system which is a recent considerable advance in four-stroke Diesel engine practice ; it has so far found application principally to marine propelling engines, but stationary engines of as little as 350 b.h.p. and as much as 3,000 b.h.p. have been successfully turbo charged.

The principal advantages of the Büchi system are :—

1. For given cylinder dimensions, a normal power 50 per cent. greater than that of the ordinary Diesel engine with the same maximum cylinder pressures and therefore the same maximum loading of the working parts.

2. Greatly increased safe overload up to 30 per cent. or more, reckoned on the increased power.

3. Equal or lower combustion and exhaust gas temperatures and less heat carried away in the cooling water, due to the special method of passing excess scavenging air over the piston head through the cylinder and valves.

4. Complete combustion at all loads.

5. Increased mechanical efficiency ; reduced fuel consumption of from 4 to 5 per cent. at normal load, up to 20 per cent. at low load, and lower lubricating oil consumption, due to smaller pistons, bearings, etc.

6. The extraordinary flexibility due to the automatic increase or decrease of the quantity and pressure of the charging air with rising or falling load.

As the cylinder of the ordinary Diesel engine is limited in its power output by the quantity of air aspired and the amount of oil which can be effectively taken up by that fixed quantity of air, it has always been of necessity infra-charged ; the Büchi system has for its primary object the overcoming of this weakness, which it effects to the extent indicated above, while at the same time reducing the cylinder heat stresses, the fuel consumption, and generally improving the working and behaviour of the engine.

The complete turbo-charged internal combustion engine set comprises the internal combustion engine, an exhaust gas turbine, and a pump or blower to supply air to the cylinders of the engine for the triple purpose of charging, cooling, and scavenging. In the arrangement so far found most practicable, the Diesel engine is treated as an independent prime mover of normal speed of revolution, and the turbine is also arranged as an independent unit directly driving a turbo-blower ; thus engine, turbine, and blower run at their most efficient speed of revolution, no gearing is necessary, and the turbine and blower can be placed in any suitable position subject only to a suitable lead for the exhaust piping from the engine to the turbine.

The quantity of air delivered by the turbo-blower to the cylinders is sufficient, and is proportioned for perfect combustion at all loads, even up to 100 per cent. over the power rating of ordinary Diesel engines of the same bore, stroke, and revolutions, and the present normal rating of the Büchi charged engine for continuous running is 50 per cent. above that of the ordinary Diesel of equal bore, stroke, and revolutions. The maximum pressures in the cylinders are the same as in the ordinary Diesel but, combustion lasting longer, the increased power is shown by the flattening of the diagram, indicating higher mean pressures.

The turbo-charged engine effectively responds to demands for increased power over a wide range, and this remarkable flexibility depends on the fact that as, at higher loads, the pressure in the exhaust pipe becomes greater, the revolutions of the turbine rise, thus increasing the quantity and pressure of the air delivered by the blower. As a consequence combustion is always complete, and even at 30 per cent. overload (calculated on the increased output due to charging) the exhaust is absolutely clear.

As the maximum cylinder pressures are not augmented, with the exception of a very slight increase of certain crankshaft dimensions, the bearings and other frictional surfaces of the Büchi turbo-charged engine of 50 per cent. higher rating are the same as those of the ordinary Diesel. Professor Stodola in his report on the test of a four-cycle Diesel engine normally rated at 850 b.h.p. (Büchi charged at 1,275 b.h.p.), put at his disposal by the Swiss Locomotive & Machine Works, Winterthur, Switzerland, says :

' The fuel consumption was 0·391 lbs. per b.h.p. hour with fuel with the low heat value of 18,240 B.T.U. per lb.'

' The same type and size of engine running as an ordinary Diesel without turbo-charging has a fuel consumption of 0·408 lb. per b.h.p. hour, or in other words, turbo-charging lowered the fuel consumption of the engine tested by approximately 4 per cent., due to the improvement of the mechanical efficiency from 72 per cent. to 80·2 per cent.'

Notable as are the increase of power of flexibility, and of mechanical efficiency attained by the Büchi system, it is remarkable, and indicates the success of the system, that the increases should be accompanied by lower heat transference to the cooling water, and reduced heat stresses, due to the temperatures of the combustion gases being no more, and those of the exhaust even less, than in ordinary Diesel engines.

Professor Stodola in the report referred to above remarks that :

' Another remarkable effect of the system is the reduction of the amount of heat passing through the walls of the piston, cylinder, and cylinder head, *i.e.* 1,620 B.Th.U. per b.h.p. hour at full load. Assuming that the whole of the heat carried away in the piston cooling water passes through the piston head, the B.T.U. per sq. ft. per hour are :

$$Q_1 = \frac{530 \times 1,275}{15 \cdot 89} = 42,500 \text{ B.Th.U. per sq. ft. per hour.}$$

' The average amount of heat transmitted through the cylinder liners is only 27,300 B.T.U. per sq.ft. per hour, which is far less than the figure for ordinary four-cycle Diesel engines, and in two-cycle engines the heat transmission is naturally much higher still.'

These results are due to the passing of a large quantity of air through the valves and over the piston head to exhaust which thoroughly scavenges and is highly effective as a cooling medium.

For the explanation of how it is possible to get the increase of output which characterises the Büchi system, with the relatively low-charging air pressure, the change of state of the exhaust gases and the intake and exhaust strokes must be considered.

The change of state of the exhaust gases from the cylinder to their state at the turbine inlet may be explained in the following simplified way. The pressure in the cylinder may be conceived as dropping instantaneously to the pressure at the turbine inlet. Hence there is a drop in pressure at constant volume, thereby producing a decrease in temperature and a high velocity, with which the gases exhaust into the exhaust manifold, where the kinetic energy is transformed at constant pressure again into heat.

The suitable and necessary degree of pressure fluctuation in the piping between the engine and the turbine is attained by correctly proportioning the inlet area of the turbine, the above-mentioned piping, and by special timing of the inlet and outlet valves. To avoid interference of the exhaust and scavenging periods of the different cylinders in a multi-cylindered engine more than one exhaust pipe to the turbine is provided.

The intake and exhaust strokes are arranged in such manner that effective exhausting, scavenging, and filling of the combustion cylinders shall occur. This is made possible by the artificial pressure variation, referred to above, in the exhaust piping between the Diesel engine and the turbine, which is not, however, noticeable on the ordinary indicator diagram, owing to the small spring scale. The charging pressure before the inlet valve remains nearly constant. The exhaust pressure on the other hand shows marked fluctuations due to the special conditions provided ; a big drop of pressure occurs at the end of the exhaust stroke when the cylinders empty themselves to a high degree, the minimum pressure being about 2–3 lbs. per sq. in. gauge. During that period the charging air pressure is higher than the pressure in the exhaust piping to the turbine, when it is possible therefore to scavenge the cylinders with pre-compressed cold air which enters through the inlet valve and exhausts through the outlet valve. The compression space is thereby swept absolutely clear of exhaust gases, and the hot cylinder walls and the valves are thus effectively cooled, so rendering possible the low temperature at the beginning of the compression stroke. As this action occurs at the t.d.c. when the cylinder volume is very small, being only the compression space or about 8 per cent. of the swept volume, and because the exhaust pressure has much decreased, the scavenging just described is especially effective. Tests have shown that it is possible to blow 25 to 30 per cent. and even more of the air delivered by the blower, in the scavenging period, through that space, amounting in effect to refilling it three or more successive times with fresh and relatively cold air. In its influence on the temperatures of the working cycle and the possible increase of power this scavenging and cooling effect is of decisive importance.

Compared with the ordinary Diesel engine the exhaust turbo-charged engine shows a relatively low mean temperature, as at the end of the exhaust stroke and at the beginning of the suction stroke an effective cooling of the cylinder walls occurs.

As the compression ratio is less than in the ordinary Diesel engine the rise of temperature during compression is much less also. Then if, for combustion, the amount of fuel introduced is also in the same proportion to the volume of air in the cylinder for both types of engine it is obvious that the rise of temperature during combustion must be the same in both cases, whence it follows that the combustion temperature of the turbo-charged engine is lower. During expansion the temperatures in both types of engine are about the same because combustion in the turbo-charged engine lasts longer, and at the beginning of the exhaust stroke the temperatures are equal, but again at the start of the scavenging period the temperature of the turbo-charged engine drops far below that common in ordinary four-cycle Diesel engines.

These conditions of the working cycle of the exhaust turbo-charged engine result in a lowering of the mean temperatures of the working cycle by as much as 194° F. (90° C.). This lower temperature in its relation to the heat stresses in the walls of the cylinders is very important, but its value is increased by the fact that the Büchi charged engine, power for power, is much smaller than the ordinary Diesel, and heat tension in engines of small dimensions for a given power is lower, and less dangerous than in those of larger size.

It has been thought that in exhaust turbo-charged Diesel engines, as a greater quantity of fuel is injected corresponding to the greater amount of charging air, there follows an increase of the heat stresses in the walls of the cylinder covers, pistons, and cylinder liners ; but tests such as that by Professor Stodola already referred to, prove that the heat transfer through the cylinder walls is less than in the ordinary four-cycle Diesel engine.

The improvement in the power-weight ratio of the exhaust turbo-charged engine led to its early application to marine work, and its obvious value for work at high altitudes and its general advantages have brought it into use for land installations.

Diesel Marine Engines in Service.

The progress of the development of marine engines was considerably retarded by the war period, and at the end of 1918 the largest Diesel engines fitted in any merchant vessel were the twin-screw engines made by Messrs. Harland and Wolff of Glasgow for the motor vessel *Glenapp*, a vessel of 7,374 tons gross.

The Fairfield Shipbuilding and Engineering Co. completed a large passenger and cargo vessel in 1924, with four screw engines of 12,000 b.h.p. of the Fairfield Sulzer type. The *Aorangi* is 600 ft. long, with a service speed of 17 to 18 knots. This vessel is on service between Canada and Australia, and has conclusively demonstrated that the Fairfield Sulzer Diesel engine is entirely satisfactory for propelling ocean liners.

According to Lloyds Daily Index, dated July 30, 1929, there is a total of 812 motor ships above 2,000 tons in service, of which 545 ships, aggregating 3,316,955 tons gross, are driven by four-stroke engines, and 267 ships, aggregating 1,870,129 tons gross, by two-stroke engines.

Of the four-stroke engined ships 398, aggregating 3,437,019 tons gross, have Burmeister and Wain type engines, and 147 other types of four-stroke engines. Of the two-stroke vessels 105, aggregating 761,731 tons gross, have Sulzer engines, 56 of 571,059 tons gross Doxford engines, 28 of 251,966 M.A.N. engines, and 78 of 485,373 have other types of two-stroke engines.

The following table of typical vessels illustrates the advance which has taken place in the construction of Diesel engines :—

Name.	Tons.	Twin or Single Screw.	Date of Construction.	Cylinders.		Stroke.
				No.	Diam.	
'Vulcanus'	1,179	Single	1910	6	15½	23
'Selandia'	4,950	Twin	1912	16	20⅞	28
'Juno'	2,345	Single	1912	6	22	39⅜
'Siam'	5,296	Twin	1913	16	23½	31½
'Fiona'	5,219	,,	1914	12	29½	43 5/16
'Panama'	5,239	,,	1915	12	24 13/16	37 13/16
'Glenamoy'	7,269	,,	1916	12	26⅜	39⅜
'Glenauy'	5,075	,,	1917	12	24 3/16	33½
'Glenapp'	7,374	,,	1918	16	29½	43 5/16
'Aorangi'	18,500	Four	1922	24	27½	39
'Asturias'	22,000	Twin	1925	16D.A.	33	59
'Augustus'	32,000	Four	1927	24D.A.	27½	47¼

Oil Fuel for Diesel Engines.

(See also FUELS Section, page 2149.)

The fuel most commonly used in Diesel engines is that known as the residual oil, which is obtained by the process of distilling the crude oil when the lighter spirits are carried over, and the residual remains. Some crude oils are, however, directly used in Diesel motors, and in particular Texas oil, which does not contain sufficient proportion of petrol and other lighter oils to warrant its distillation. The only oil fuel which is unsuitable for employment is that from an asphaltum base, and it is for this reason that Mexican fuel oil is not as a rule employed for these motors without special treatment. Besides Texas oil, that from the Californian, Persian, Trinidad, Galician, Roumanian, and Russian oilfields is satisfactory for use in Diesel motors. Tar oil, which is obtained from the distillation of tar, is also very largely utilised in Germany, where the duties on imported oils are excessive, and often render the employment of Diesel engines unprofitable. Tar oil has a higher specific gravity than the petroleum fuel (usually in the neighbourhood of unity or slightly over), and has a lower calorific value, so that the consumption of an engine using this fuel is greater than when running on the ordinary oil. It is very viscous and, therefore, often requires heating, whilst some slight modifications have usually to be made in the fuel injection arrangements in order that an engine should run quite satisfactorily on tar oil. The usual method is to allow the injection of a small quantity of a lighter fuel oil before the main body of the tar oil passes into the cylinders through the pulveriser. It is claimed, however, that some motors run without any modification whatever, but this should not be taken as a general statement.

For information relating to Diesel engines for land installations, see page 1893.

ENGINE ROOM STAFF FOR MOTOR SHIPS

Vessel.	Engine Room Staff.	
	Engineers and Electricians.	Greasers, Mechanics, Donkeymen, etc.
'Conde de Churruca'	4	5
'Handicap'	8	6
'Glenapp'	11	6
'Glenluce'	10	(?)
'Lobos'	11	9
'William Penn'	3	9
'Asia'	9	4
'Ossag'	7	4
'Magvana' and 'Melma'	12	17
'Seminole'	8	(?)
'Domala'	12	24 (Natives)
'Aorangi'	19	—

COMPARISON OF MACHINERY FOR A CARGO SHIP DRIVEN BY A TRIPLE-EXPANSION RECIPROCATING STEAM ENGINE AND BY AN OIL-DRIVEN DIESEL ENGINE.

Dimensions of ship 360 ft. by 48 ft. by 29 ft.

Speed . . . . . 13 knots.
E.H.P. 1,500

Diesel Engine—		
Revolutions	100	
Propulsive efficiency	·51	
B.H.P.	2,950	
Weight of 6-cylinder single-acting 2-cycle engine	290	tons
Weight of boiler 5 ft. by 10 ft. high, with water uptakes and funnel	10	,,
Diesel electric generators	40	,,
Electrically-driven auxiliaries	30	,,
Silencers, piping and other fittings in engine-room	100	,,
Shafting propellers and other tunnel weights	70	,,
Tons	540	
Steam Engines—		
Revolutions	60	
Propulsive efficiency	·50	
I.H.P.	3,000	
Weight of engines and boilers, etc.—		
(*a*) with Howden draft at 5·5 I.H.P. per ton	545	tons
(*b*) ,, natural ,, 5 ,,	600	,,
Length of machinery spaces except bunkers—		
Oil engine	60	ft.
Steam engine, Howden draft	50	,,
,, natural ,,	65	,,

			Per h.p. per hour. Lbs.	Per day. Tons.
Fuel Consumption	Oil Plant	Main engine . . . . . oil	·43	13·6
		Auxiliary machinery, electrically driven	·04	1·4
		Total	·47	15·0
	Steam Plant	Main and auxiliary machinery . . . coal	1·7	55

Engine Room Complement	Oil Engines:	Engineers	6 } 14
	,,	Greasers	8 }
	Steam Engine:	Engineers	4 } 30
	,,	Greasers	6 }
	,,	Stokers and Trimmers	20 }

LLOYD'S RULES FOR MARINE INTERNAL-COMBUSTION ENGINES. See page 1926.

HIGH PRESSURE AIR COMPRESSORS FOR MOTOR SHIP AUXILIARY DUTIES.

Fig. 57 shows a typical Weir three-stage, single-acting, twin-crank, motor-driven air compressor, for a delivery pressure of 1,000 lbs. per sq. in. In this machine the first and second stages are formed by means of a differential type piston, the air passing from the first stage of compression through the first stage intercooler and thence to the annular space forming the second stage. After second stage compression and cooling, the air passes to the third stage situated at top of the machine, where, when third stage compression is completed, the air is delivered through the after-cooler and oil and water separator.

The coolers are of the tubular type, the air passing through copper tubes expanded into rolled brass tube plates. Arrangements for the longitudinal expansion of the tube units are provided, and all troubles associated with tube packings are completely eliminated.

The pistons are of high-grade cast iron fitted with cast-iron rings of the Ramsbottom type. With the perfected lubricating and cooling arrangements in the Weir compressor, the wear on these cast-iron rings over long periods of running is practically negligible, the tool marks being visible after years of running.

The air valves are of special design, the valve plates being made in a steel alloy, heat-treated to obtain the correct degree of hardness. The lift of the valve is exceedingly small, preventing undue shock and noise and obtaining a high degree of efficiency. As a special precaution the valve seats are made in mild steel perforated with small holes, this type being superior to the usual

cast seat with annular slots, as the former design makes it practically impossible for fragments of a broken valve to enter the cylinder with destructive effect. The assembled valve units are arranged round the cylinder heads in such a manner as to reduce the cylinder clearance to a minimum, and give a resultant high volumetric efficiency.

The motion work generally is robustly designed to meet the high stresses generated in reciprocating air compressor machinery. The crank webs and connecting rod jaws are cut from solid mild steel forgings and machined all over. The crosshead and piston rod are forged in one piece, and tapers are provided on the rod to take the pistons.

As in best modern practice, forced lubrication is applied to all bearing surfaces, the crankcase bearings being served through a special system of oil ducts by an oil pump of the oscillating disc type. Cylinder lubrication is effected by a mechanical sight feed lubricator, feeding minute quantities of oil at the correct points on the cylinder walls, and maintaining the thin even oil film so necessary with comparatively high-temperature conditions.

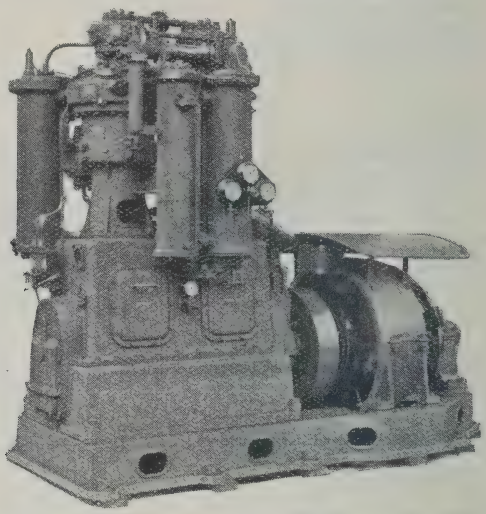


FIG. 57.

The balancing of these compressors has received careful consideration, the weights in all case being designed to effectively balance the forces set up by the reciprocating parts.

The circulation of the water spaces is carried out by a plunger type water-pump driven from the compressor crankshaft. This pump is of a very compact and positive design, and has a cast-iron body and gun-metal plunger with valves of the Kinghorn type.

The foregoing description is typical of compressors largely supplied for motor ship auxiliary duties, *viz.* Diesel engine air-blast injection, manœuvring, etc.

Compressors dealing with oxygen and other industrial gases for delivery pressure ranging to 6,000 lbs. per sq. in. are constructed, and may be arranged for drive either by electric motor, independent steam engine, tandem steam engine, or oil engine.

Metals for Propellers.

A properly designed bronze screw will give a very marked increase of speed (as much as *half a knot*) as compared with an iron or steel propeller, this being due to reduced scantlings, smoother surfaces, etc. Whilst the efficiency of a bronze screw continues unimpaired, that of an iron or steel propeller diminishes, through the pitting and corroding of its surface. In the case of solid screws the reduction in weight as compared with a cast-iron screw ranges from 10 to 20 per cent.

Nearly all vessels of any size or pretension are now fitted with bronze propellers. The metal has a tensile breaking strength of about 32 tons per square inch with about 20 per cent. elongation, and a transverse strength of over 7,500 lbs. per square inch. All warships as well as many merchant steamers are fitted with bronze propellers. Cast-steel propellers cost about twice as much, and bronze propellers about six times as much, as those of cast iron.

The *Bull's Metal Company* have with great success combined the study of Bronze Propeller designs with that of material, manufacture, and finish, large numbers of ordinary Bronze Propellers having been replaced with Bull's Metal propellers, giving, it is stated, a very material increase in speed on a reduced consumption of coal.

The *Delta Metal Co. Ltd.*, supply their special corrosion-resisting 'Delta' propeller bronze for casting propellers and propeller blades.

(See also Section XI., METALLURGY, p. 2659.)

Elements of Propulsion.

If a vessel (without propeller) moving at a speed of V feet per minute experiences a total resistance of R pounds, a force R must be applied continuously by means of a thrust or tow-rope pull in order to overcome the resistance and maintain the speed. The power required is RV foot-pounds per second and is called the Tow Rope Power or Effective Power.

$$\text{Effective Horse Power, E.H.P.} = \text{R.V.} \div 33000 = \cdot 00\dot{3}0\dot{7}\ \text{RK.}$$

If the thrust is delivered by a screw propeller working behind the ship, it is found that the action of the propeller increases the resistance. This increase is called the 'Augment of Resistance.' T, the total thrust of the propeller, must therefore be equal to R + Augment = R + A, and T − A = R. The augment of resistance can therefore be regarded as a deduction from the thrust: it is alternatively called the 'Thrust Deduction' and is written A = tT, where t is the 'Thrust Deduction Fraction.'

$$\text{T} - t\text{T} = \text{T}(1 - t) = \text{R}$$

$(1 - t)$ is called the Thrust Deduction Factor.

If a vessel is being towed at a speed of K knots, the skin friction sets the adjacent water in motion, and at the stern there will be found a 'wake' current moving in the same direction as the ship at some speed W knots. The stern, and with it the propeller, will therefore be moving at a speed K − W relatively to the wake water, and this speed is called the 'Speed of Advance.' Froude compares wake speed with speed of advance, and writes $\text{W} = \omega\,(\text{K} - \text{W}) = \omega\text{K}'$.

$$\text{Wake fraction} \quad \omega = \frac{\text{W}}{\text{K} - \text{W}} = \frac{\text{W}}{\text{K}'}$$

$$\text{Wake factor} \quad 1 + \omega = \frac{\text{K}}{\text{K} - \text{W}} = \frac{\text{K}}{\text{K}'}$$

$$\text{Wake per cent.} \quad 100\omega.$$

Taylor compares wake speed with ship speed, and writes—

$$\text{W} = \omega\text{K}$$

$$\text{Wake fraction} \quad \omega = \frac{\text{W}}{\text{K}}$$

$$\text{Wake factor} \quad 1 - \omega = \frac{\text{K} - \text{W}}{\text{K}'} = \frac{\text{K}'}{\text{K}}$$

$$\text{Wake per cent} \quad 100\omega.$$

If a propeller having a pitch P and making N revolutions per minute is working behind a ship and producing a thrust T with a torque Q in the shafting just forward of the propeller, the ship's speed being K and the speed of advance of the propeller K′ = K − W, the work obtained will be RK (E.H.P.) and the work expended 2πNQ, which we can call the 'Received Horse Power Behind' R.H.P.$_b$. The efficiency η is therefore $\frac{\text{E.H.P.}}{\text{R.H.P.}_b} = \frac{\text{RK}}{\text{R.H.P.}_b} = \frac{\text{T}(1 - t) \times \text{K}'\,(1 + \omega)}{\text{R.H.P.}_b}$.

TK′ is called the 'Thrust Horse Power' (T.H.P.) and the product $(1 - t)\,(1 + \omega)$ the 'Hull Efficiency,' being the product of one factor $(1 - t)$ representing the effect of the propeller on the hull and another $(1 + \omega)$ representing the effect of the hull on the propeller. If we write—

$$\text{Hull Efficiency} = h = (1 - t)\,(1 + \omega)$$

we have

$$\text{Screw Efficiency Behind} = \eta = \frac{\text{T.H.P.}}{\text{R.H.P.}_b}\,h.$$

Now if the same propeller be mounted on a shaft alone, without ship or model, and run in open water at the same revolutions N, and its speed of advance adjusted until it gives the same thrust T, we can fairly assume that the speed of advance so found must be the same as the speed of advance when in place behind the ship. We are thus able to determine the wake speed. The torque is not necessarily the same as when behind the ship, though it is usually not very different, so we call the power in the shaft 'Received Horse Power in the Open,' R.H.P.$_o$, and the 'Efficiency in the Open,' $\odot$, is $\frac{\text{TK}'}{\text{R.H.P.}_o} = \frac{\text{T.H.P.}}{\text{R.H.P.}_o}$.

The ratio $\frac{\text{R.H.P.}_o}{\text{R.H.P.}_b}$ is called the 'Relative Rotative Efficiency,' r.

Finally therefore, we can write—

$$\eta = \frac{\text{T.H.P.}}{\text{R.H.P.}_b} h = \frac{\text{T.H.P.}}{\text{R.H.P.}_o} \times \frac{\text{R.H.P.}_o}{\text{R.H.P.}_b} h = \odot rh,$$

or screw efficiency behind = screw efficiency in open × relative rotative efficiency × hull efficiency.

S.H.P., the 'Shaft Horse Power' (or B.H.P., the 'Brake Horse Power'), is the power in the shafting just abaft the engine. The shaft line efficiency l is the ratio $\frac{\text{R.H.P.}_b}{\text{S.H.P.}}$. The mechanical efficiency of the engine m is the ratio of S.H.P. to the indicated horse power (I.H.P.).

$$\therefore \quad \text{S.H.P.} = m \times \text{I.H.P.}$$

$$\text{R.H.P.}_b = l \times \text{S.H.P.} = l \,.\, m \,.\, \text{I.H.P.}$$

$l \times m$ is the 'Transmission Efficiency,' $f = \frac{\text{R.H.P.}_b}{\text{I.H.P.}}$

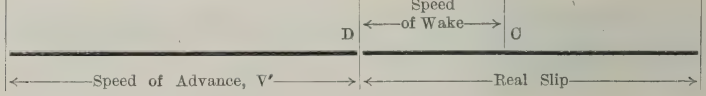
The overall propulsive efficiency or propulsive coefficient, e,

$$= \frac{\text{E.H.P.}}{\text{I.H.P.}} = \frac{\text{E.H.P.}}{\text{R.H.P.}_b} \times \frac{\text{R.H.P.}_b}{\text{I.H.P.}} = \eta f = \odot rhf.$$

η is sometimes called the 'Quasi-Propulsive Coefficient.'

A propeller having a pitch P and making N revolutions per minute would advance a distance PN if there were no slip. On account of slip, however, it actually advances a lesser distance V′ called the 'speed of advance.' The difference is called slip, so that slip = PN − V′, and the 'slip ratio' $s = \frac{\text{PN} - \text{V}'}{\text{PN}} = 1 - \frac{\text{V}'}{\text{PN}}$. If the thrust is T and the torque Q, the efficiency $= \frac{\text{TV}'}{2\pi\text{NQ}} = \frac{\text{TP}\,(1-s)}{2\pi\text{Q}}$.

When attached to a ship the apparent speed of advance is the ship's speed K knots or V feet per minute, where $\text{V} = 101\frac{1}{3}\text{K}$. The apparent slip is PN − V. On account of the presence of the wake stream, however, the true speed of advance of the propeller through the water surrounding it is K − W(= K′) knots or V′ (= $101\frac{1}{3}$K′) feet per minute, and the real slip is PN − V′. Let AB represent to scale the 'Speed of the Propeller':



AC the speed of the ship V, and CD the speed of the wake.

$$\text{Then apparent slip ratio} = \frac{\text{CB}}{\text{AB}}$$

$$\text{Real slip ratio} = \frac{\text{DB}}{\text{AB}}$$

$$\text{Froude's wake fraction} = \frac{\text{DC}}{\text{AD}}$$

$$\text{Taylor's wake fraction} = \frac{\text{DC}}{\text{AC}}$$

(*W. J. Luke.*)

Law of Comparison for Model Propeller Experiments.

If a model propeller $\frac{1}{n}$th of the size of the ship's propeller is run at $\sqrt{n}$ times the revolutions and its speed of advance adjusted so that both screws are working with the same slip ratio, the thrust

of the ship's propeller will be n^3 times that of the model, the torque n^4 times, the powers $n^{3\frac{1}{2}}$ times, and the efficiencies equal.

	Model Propeller	*Ship's Propeller*
Diameter	d	$D = nd$
Pitch	p	$P = np$
Revolutions	r	$R = n^{-\frac{1}{2}} r$
Speed of Advance	v'	$V' = n^{\frac{1}{2}} v'$
Slip Ratio, s	$1 - \frac{v}{pr}$ =	$1 - \frac{V'}{PR}$
Thrust	t	$T = n^3 t$
Torque	q	$Q = n^4 q$
THP = U	$\frac{tv'}{33000}$	$\frac{TV'}{33000} = n^{3\frac{1}{2}} \frac{tv'}{33000}$
RHP = P	$\frac{2\pi vq}{33000}$	$\frac{2\pi RQ}{33000} = n^{3\frac{1}{2}} \frac{2\pi vq}{33000}$
Efficiency	$\frac{tv'}{2\pi rq}$ =	$\frac{TV'}{2\pi RQ}$
ρ_U (Rho-U)	$\frac{ru^{\frac{1}{2}}}{v'^{2\frac{1}{2}}}$ =	$\frac{RU^{\frac{1}{2}}}{V'^{2\frac{1}{2}}}$
ρ_P (Rho-P)	$\frac{rp^{\frac{1}{2}}}{v'^{2\frac{1}{2}}}$ =	$\frac{RP^{\frac{1}{2}}}{V'^{2\frac{1}{2}}}$
δ (Delta)	$\frac{dr}{v'}$ =	$\frac{DR}{V'}$

The last three functions are the same for all similar propellers working at the same slip ratio. ρ_U and ρ_P depend on revolutions, power and speed, which are given quantities in the initial stage of propeller design. The results of systematic propeller experiments are usually plotted on some function of Rho as base, since this is independent of diameter. For each of a series of pitch ratios, a curve of efficiency and a curve of some function (such as δ) involving the diameter are plotted, from which the most suitable diameter and pitch ratio can be found for the given power, revolutions, and speed of advance. Taylor has published four 'Rho-delta' diagrams obtained by combining and averaging his own propeller experiments with those of Froude, Durand, and Schaffran. These were reproduced in *The Shipbulder*, January 1925.

Metals for Marine Engines.

Delta Bronze Nos. I. and IV. alloys (see Index), Phosphor bronze, and Phosphor white lining metal No. II. are especially suitable for various parts of marine engines: the latter metal (phosphor white) has a higher fusing point than good Babbitt metal, thus allowing a bearing to heat without causing the metal to melt, a matter of some importance at sea.

Delta Bronze No. IV., a malleable bronze of great strength and tenacity, is specially adapted to replace steel in cases where a material of the highest corrosion-resisting qualities is desired. For this reason it is employed on a very large scale in the form of forgings or extruded or rolled rods, etc., for air and circulating pump rods, shafts, plungers, spindles, thrust block bolts and other parts of marine engines. For particulars of corrosion-tests, see p. 174.

Phosphor-bronze sheet is superior to copper in withstanding the action of sea-water. In a comparative experiment made at Elankenberghe, lasting over a period of six months, between the best English copper and phosphor bronze, the following results were arrived at: the loss in weight due to the oxidising action of the sea-water averaged for copper 3·058 per cent., while that of the phosphor bronze was only 1·158 per cent.

For lining bearings, crank-pin bushes, and other parts, exclusive of crosshead bushes, the following alloy has been found to give good results after extended experience in hard ocean steaming :—12 parts tin, 1 part copper, 1 part antimony, mixed in the following way : 6 parts tin alloyed with 1 part of antimony, and 6 parts tin with 1 of copper, and the two mixed together. For lining crosshead brasses the following harder alloy has been found to give good results : 85 per cent. tin, 5 per cent. copper, and 10 per cent. antimony.

The *Eyre Smelting Co., Ltd.*, who have made a special study of bearing metals for marine engines, supply a particular alloy of white metal which is stated to give good results in the big end bearings of Diesel engines. For the small end bearings the company supply a mixture of phosphor bronze, which is largely used in this country and on the Continent.

Painting Ships' Bottoms.

Ordinary oil paints are not suitable for this purpose, particularly because the drying of each coat of oil paint takes a longer time than owners can afford to allow vessels to remain in dock ; besides, these paints possess little or no power to prevent fouling. The practice, therefore, is to use what are termed ' compositions,' which are usually applied in two coats.

The composition paints have to be used in combination. The anti-corrosive, as a preservative and insulating coating, should be applied when the bottom is well scraped and dried. The anti-fouling composition contains powerful poisons and can be applied almost immediately after the primary coating, and the vessel can be floated as soon as the painting is completed, although it is desirable to allow the compositions a few hours to harden before the vessel enters the water.

It is very important to remove all rust or scale on the plates, as when such is not done corrosion is likely to continue beneath the surface and will eventually throw off any paint that can be applied. The ingredients in the best compositions are very expensive and are consequently not used to any extent in the manufacture of the cheaper brands.

The careful *stirring* up of these compositions is an essential point too often overlooked. The more valuable ingredients are heavy, and tend to form a sediment at the bottom of the drums, which requires to be carefully worked up. Much also depends upon careful painting—a thing which is unfortunately very difficult to arrange for in many of the dry docks owing to hurried and inefficient work.

Concrete Vessels.

A concrete weighing but 110 lbs. per cub. ft. and having a crushing strength after 28 days of 4,000 lbs. per sq. in., has been employed in the construction of reinforced concrete vessels of large tonnage, in America, for the United States Government. The aggregate itself, although light and cellular in structure, is not porous. The lighter quality will actually float on water, and as a test the United States Shipping Board authorities kept an amount of aggregate floating in water for some months, and it was demonstrated that the absorption was very small.

The aggregate is made from clay, calcined in kilns in a similar manner to that employed in the manufacture of bricks, the fundamental difference being that the heat (about 2,000° F.) is applied in a comparatively short time of 30 hours. This relatively sudden application of heat causes the material to swell, forming a clinker. The clay may be pressed into bricks and partially dried and stacked in kilns in the ordinary manner, care being taken to provide sufficient room around the bricks for expansion during burning. The size of the brick adopted in the United States is 8½ ins. by 4½ ins. by 2¾ ins. During burning, the bricks run together and form a solid mass of clinker, which is allowed to cool for about 3 days, in order to ensure a high degree of toughness. The cooling of the material is then completed by water, which causes the mass to split into pieces suitable for handling, and it is also found that this greatly facilitates the unloading of the kilns. The material may also be calcined in cement rotary kilns, the clay in this case being pressed into lumps of suitable size.

After burning, the product is crushed to the desired size. During the operation a proportion of the lightest aggregate is reduced to dust. This, however, is used to take the place of the sand or crushed stone in ordinary concrete, the constituent parts being carefully selected and graded. Light pneumatic hammers are used to vibrate the shuttering during ramming, materially assisting the flow of the concrete around the steel reinforcement, and producing a dense and impermeable material.

A number of reinforced light concrete oil tankers and cargo steamers of 7,500 tons deadweight have been under construction in various parts of the United States. These vessels are 434 ft. long. (*W. L. Scott, ' Engineering.'*)

When wood is used as a reinforcing material for concrete it should be impregnated with magnesium chloride, and the wood must be so placed that any change in the cross-sectional area does not affect the durability of the structure. If the latter precaution is overlooked the concrete may crack. Small strips of wood are better than larger ones, and rushes appear to be excellent for the purpose, especially for thin-walled structures such as ships.

(*Beton u. Eisen.*)

Lloyd's Rules for Ferro-Concrete Vessels.

No definite rules have as yet been formulated, but in regard to the quality and testing of steel to be used for reinforcements these are the same as given in Section 3 of the Rules of the Society, for Steel Ships. The cement should comply with the requirements of a recognised standard specification for Portland cement or the equivalent thereto. In the matter of the preparation of the concrete, the proportions of the cement, sand, and aggregate have been left to the individual constructors to submit their proposals for the consideration of the Committee, as these depend to some extent on the local facilities for obtaining sand and gravel, the quality of which varies in different districts.

In the design of vessels to be built of ferro-concrete, it is considered that concrete is incapable of withstanding any of the main tensional stresses, and the steel reinforcements embedded in the concrete must therefore be of sufficient strength to take up all the tensional stresses produced in the structure due to strain from longitudinal bending.

Lloyd's Tentative Regulations for the Application of Electric Arc Welding to Ship Construction.

(*Abstract.*)

Before welding, the surfaces to be joined must be fitted close to each other.

All butt and edge connections are to be lapped or strapped.

With lapped connections the breadths of overlaps of butts and seams and the profiles of the welds are to be in accordance with the following table :—

Thickness of plate. Inches.	Width of overlap seam and butt. Inches.	Throat thickness. Inches.
·40 and under	$2\frac{1}{4}$	·28
·60	$2\frac{1}{2}$	·38
·80	$2\frac{3}{4}$	·48
1·00	3	·50

Intermediate values may be obtained by direct interpolation, and for thicknesses below ·40 the throat thickness is to be about 70 per cent. of the thickness of the plate.

See also ' Electric Welding,' p. 2779.

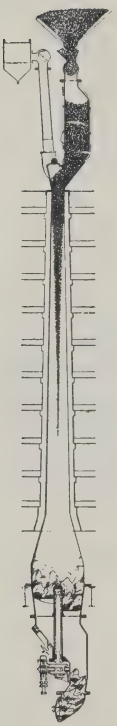
SECTION XXXVII

GAS AND GASWORKS

GENERAL PRINCIPLES GOVERNING THE SIZE AND CAPACITY OF APPARATUS—COMPOSITION OF VARIOUS GASES — CONDENSATION, PURIFICATION, STORAGE — RETORTS—TAR DISTILLATION—WATER GAS—COMPLETE GASIFICATION — CYANOGEN — BENZOL RECOVERY — SULPHATE OF AMMONIA—FLOW OF GAS IN MAINS—GAS TESTS —ACETYLENE GAS—PETROL AIR GAS.

(Contributed by Alwyne Meade, Assoc.M.Inst.C.E., author of 'Modern Gasworks Practice.')

Glover-West Vertical Retorts for all kinds of Coal



The following figures are extracted from results published by the Fuel Research Board of the carbonization of eight different coals in the Glover-West Vertical Retorts at the Fuel Research Station:—

COAL	STEAM Used % of Coal	GAS C. ft. /ton	GAS Therms /ton	GAS Gained by steaming (Therms)	GAS Cal. Val. B.Th.U.	COKE Made per ton
						cwts.
Consett (Durham)	5%	13,590	70	2	518	15.0
	21%	17,080	79	11	463	14.8
Mitchell Main Gas Nuts (Barnsley)	5%	16,430	85	14	517	13.9
	12%	20,060	96	25	477	13.3
	21%	22,580	104	33	460	13.2
Arley (Wigan)	5%	15,980	80	—	502	13.6
	10%	16,980	84	4	492	13.4
	15%	18,950	89	—	469	13.3
	20%	19,980	91	11	457	13.1
Holmside (Durham)	5%	13,200	69	9	524	14.8
	20%	15,940	77	8	480	14.5
Meiros (S. Wales)	5%	15,070	74	9	493	13.8
	20%	19,210	88	23	458	13.3
Ravine (Lancs.)	5%	14,530	71	10	488	13.2
	20%	18,960	86	25	450	12.8
Wearmouth (Durham)	5%	15,355	80	—	521	—
	20%	17,840	86	6	482	—
Bristol and Somerset	5%	16,030	82	—	511	13.7
	15%	19,200	90	8.5	471	13.2

WEST'S GAS IMPROVEMENT Co. Ltd.

Telephones— COLLYHURST 2961 (3 lines) **Miles Platting, Manchester** Telegraphic Address— "STOKER, MANCHESTER"

LONDON
Regent House, Kingsway, W.C.2.
Telegraphic Address— "Imvertret, Westcent, London."
Telephone— Holborn 4108.

PARIS
Cie. Gle. de Construction de Fours, 8, Place des Etats-Unis, Montrouge (Seine).
Telegraphic Address— "Cormiques-Montrouge (Seine)."
Telephone— Vaugirard 27-95 & 27-96.

NEW YORK
West Gas Improvement Co. of America Incorporated.
441 Lexington Avenue, New York.
Telegraphic Address— "Verticals, New York."
Telephone— Murray Hill 2-7242.

SYDNEY
Union House, 243 George Street, Sydney, N.S.W.
Telegraphic Address— "Vertical, Sydney."
Telephone— B5865.

SECTION XXXVII

GAS AND GASWORKS—GENERAL PRINCIPLES GOVERNING THE SIZE AND CAPACITY OF APPARATUS—COMPOSITION OF VARIOUS GASES—CONDENSATION, PURIFICATION, STORAGE—RETORTS—TAR DISTILLATION—WATER GAS—COMPLETE GASIFICATION—CYANOGEN—BENZOL RECOVERY—SULPHATE OF AMMONIA—FLOW OF GAS IN MAINS—GAS TESTS—ACETYLENE GAS—PETROL AIR GAS.

(Contributed and revised by Alwyne Meade, Assoc.M.Inst.C.E., author of 'Modern Gasworks Practice.')

Parliamentary.

The public supply of gas in this country is governed largely by local Acts, but in general by the Gasworks Clauses Acts of 1847 and 1871, the Sale of Gas Act of 1859, and the Gas and Water Works Facilities Acts of 1870 and 1873, and most recently by the Gas Regulation Act (1920). London companies are, in addition, under the regulations of the Metropolitan Gas Act (1860). Under the Public Health Act of 1875, the local authority may become its own gas supplier, or may purchase an existing undertaking by agreement. Thus gasworks are controlled by—

(*a*) Statutory companies.

(*b*) Non-statutory companies.

(*c*) Local authorities.

The supply of gas in Scotland is, for the most part, in the hands of the local authorities. In England local authorities predominate as owners in the counties of Yorkshire and Lancashire, and in the North generally, but in the Home and Southern counties private ownership is the general rule. In addition to the Acts named above, the supply of gas in Scotland is governed by the Gas Acts of 1864 and the Burghs Gas Supply Acts of 1876 and 1893.

The obligations of the gas undertakings in the way of standards of quality were somewhat modified by an order made by the Minister of Munitions in June, 1918, as to the manufacture and production of gas, compliance with which order by the gas undertaking protected the undertaking from any forfeiture for failing to comply with its statutory obligations with regard to the illuminating power or the calorific power of the gas supplied, provided that in any even gas of a lower calorific power than 450 B.Th.U. gross was not supplied to the consumers.

It may be said that a new era in the supply of gas was marked by the introduction of The Gas Regulation Act of 1920. The commodity is no longer sold on a basis of cubic feet, but in terms of 'Therms.' In other words, the consumer now pays for the useful energy only, i.e. British Thermal Units; accordingly, the incombustible constituents in gas have no longer any commercial value. The standard 'Therm' is equivalent to 100,000 B.Th.U. Thus, if 10,000 cubic feet of gas are registered on the consumer's meter, and the 'declared' or normal calorific power of the gas is 500 B.Th.U. per cubic foot, then the number of B.Th.U. consumed is 5,000,000, or 50 Therms.

The main provisions of the Act are as follows :—

(*a*) The obligation to charge consumers on the heat unit basis.

(*b*) The adjustment of the 'standard' or 'maximum' price (on the heat-unit basis) to meet the additional expenses of manufacture and distribution arising from post-war conditions.

(*c*) Powers are given to the Board of Trade to limit the proportion of inert constituents in the gas, if thought fit; but the committee appointed to examine the question of incombustibles recommended that no limit should yet be imposed.

(*d*) There is an obligation to supply gas at 2 ins. water pressure in mains or services of 2 ins. diameter and upwards.

One of the most noteworthy provisions of the Act is the latitude which is given to gas undertakings in connection with the quality of gas supplied. The old standard of illuminating power has completely given way to calorific power, and the producer is permitted a free choice in the type of gas he distributes. He must, however, declare at what calorific power he is prepared to supply, consumers' appliances must be adjusted by him to consume the 'declared' variety of gas to the best advantage, and, in order to preclude excessive fluctuations in quality, certain penalties are imposed when the heating value falls below a prescribed limit. As regards the machinery for carrying out the Act, powers are given to the Board of Trade to grant facilities which in the past have entailed formal application to Parliament.

An important feature of the Act is that it empowers the Board of Trade to make amending orders revising the standard or maximum price if at any time it is shown to the satisfaction of the Board that the costs incidental to the production and supply of gas have substantially altered by reason of circumstances beyond the control of the gas undertakings. The effect of this was experienced some few years ago in the London area, when on the application of the London County Council the standard prices of certain undertakings were substantially reduced. More recently, however, on the application of certain gas undertakings themselves, increases in standard prices have been approved, and in addition Bills have been introduced which permit of the payment of a minimum dividend irrespective of the price of gas.

The Act prescribes that the quality of the gas distributed shall be determined by a calorimeter of the recording type which gives a continuous record of heating value. For some years recording calorimeters were not insisted on, and the system of periodical spot testing is still generally in force. The Gas Referees, however, commenced to prescribe the installation of recording calorimeters in certain cases as from the commencement of 1928, the two calorimeters specified for the purpose being the Fairweather and the Boys.

A non-statutory company operates as the result of established practice, or under the goodwill of the local authority. Its position is in many respects similar to that of a statutory company, but before opening roads for the laying of mains permission must be obtained from the district or county authority.

Statistics of Gas Undertakings in Great Britain and Ireland.

The following table shows the total number of gas undertakings which have existed in the United Kingdom during various years since 1901. It should be noted that the total number of separate undertakings has undergone decrease during the last twenty years, this being due to the policy of amalgamation which still continues. It is noticeable, too, that while undertakings owned by private companies and individuals have undergone a decrease, those owned by local authorities have increased, showing the tendency for the town authorities to take over undertakings from private ownership.

Year.	Owned by Companies & Private individuals.	Owned by Corporations & Local Authorities.	Total Number of Undertakings.
1901	1,349	265	1,614
1905	1,330	285	1,615
1907	1,322	297	1,629
1908	1,322	296	1,628
1909	1,330	296	1,626
1910	1,339	302	1,641
1911	1,320	308	1,628
1912	1,316	311	1,627
1915	1,281	320	1,601
1916	1,263	323	1,586
1917	1,256	322	1,578
1918	1,252	325	1,577
1919	1,252	325	1,577
1921	1,229	330	1,559
1922	1,220	321	1,551
1924	1,155	331	1,487
1926	1,134	332	1,466
1931	1,058	338	1,396

The largest gas undertaking in the world is the Gas Light & Coke Co. of London. This Company distributes more energy per annum than all the public electricity supply stations in Great Britain added together. The third annual report of the Electricity Commissioners shows that the authorised electricity undertakings of the United Kingdom generated, in the year ending March 1923, 4,600 million Board of Trade units of electricity, or 15,690,600 million B.Th.U. (156·906 million therms), while the Gas Light & Coke Co. supplies its consumers annually with over 237 million therms.

The largest gasworks in the world is the Beckton station of the above company. During a year, over a million and a quarter tons of coal are carbonised at this works, and the maximum daily output of gas amounts to over 70 million cu. ft. The Beckton works were opened in 1870, and have a total area of 475 acres. There are 14 retort houses on the works, and the water-gas plant is capable of an output of 27 million cu. ft. per day. There are upwards of 50 miles of standard gauge railway on the works, and the rolling stock consists of some 14,000 wagons and 40 locomotives.

Number of Gasworks in Great Britain.

In 1931 the total number of gasworks in operation in Great Britain and Ireland was 1,396. The total is made up as follows :

In England and Wales—	
Companies and privately owned	839
Local authorities	244
In Scotland—	
Companies and privately owned	158
Local authorities	74
In Ireland—	
Companies and privately owned	61
Local authorities	20
	1,396

Gas Made in Great Britain.

The quantity of gas made in Great Britain during 1927 was 314,582 million cu. ft. This was 3·1 per cent. in excess of that made in 1926. It is a remarkable fact that in the past twenty years the gas consumed in Great Britain has increased by nearly 67 per cent., while the total number of consumers (which now number 9,200,000) has increased by 58 per cent. The total amount of coal carbonised for the production of towns' gas now amounts to 16,500,000 tons per annum, in addition to which some 1,500,000 tons of coke and 100,000,000 gallons of oil are utilised for the manufacture of water gas. The gas industry also produces approximately 200,000,000 galls. of tar per annum, the greater proportion of which is dehydrated and used for road purposes.

The capital employed by the statutory gas undertakings alone amounts to some 150,000,000*l.*, and the total length of street mains employed runs to some 45,000 miles.

Consumption of Gas per Head of Population.

In the larger towns the average consumption of gas per head of population varies from 6,000 to 10,000 cu. ft. per annum. In smaller towns it ranges between 2,500 and 7,500 cu. ft. Gasworks statistics are, however, generally based on the consumption per consumer, which figure, for consumers through ordinary meters, approximates to 42,000 cu. ft. per annum, and 15,000 cu. ft. for those taking the gas through 'slot' meters. In addition, there will be an appreciable consumption for public lighting, and about 5 per cent. of the whole output will be written off as 'unaccounted for.' In straggling districts with long lengths of mains the latter figure may reach so much as 10 per cent. Taking the country as a whole, the average consumption per consumer at the present time works out at almost exactly 30,000 cu. ft. (say 150 therms) per annum.

Capital Expenditure on Gasworks.

This item varies considerably in accordance with the size of the installation and the price of labour and materials. In pre-war times, however, a concern promoted for the erection of a modern works with the accompanying distribution plant found it possible to complete the scheme for 600*l.* per million cu. ft. sold per annum. For smaller country works, the outlay was very much greater, varying from 750*l.* to 1,000*l.* per million cu. ft., or even more.

The following figures are the latest available for the undertakings named :

	Per million cu. ft. per annum.		Per million cu. ft. per annum.
Gas Light & Coke Company	433*l.*	Bristol Gas Company	433*l.*
Croydon Gas Company	562*l.*	Birmingham (Corporation)	329*l.*
Brighton & Hove Gas Company	437*l.*		

It is, perhaps, more accurate nowadays to rate capital expenditure on the therm basis in lieu of the cubic foot basis. As an example of how this method compares with the figures given on p. 2451 it may be mentioned that the capital of the Gas Light & Coke Company is now 1*s.* 9*d.* per therm, and that of the Croydon Gas Company 2*s.* 2*d.* per therm.

It is a somewhat surprising fact that at the present day the expenditure on the district mains, etc., is generally rather greater than that laid out in the erection of the works. Of the total outlay approximately 55 per cent. will be expended on the district, 38 per cent. on the works, and the remainder on Parliamentary, promotion, and engineers' expenses.

General Rules relating to Capacity of the Apparatus, etc.

LAND.

The area of land required for the erection of the works will amount to at least 2½ acres per million cubic feet of gas per maximum diem.

(NOTE.—The make of gas per maximum diem is calculated by dividing the total annual production by the factor of 240.)

RETORT HOUSE.

Dependent upon the size of the unit, i.e. the number of retorts set in each arch or section. The make per retort per 24 hours for different types of units is given in the following table :—

STOP-ENDED OR 'SINGLE' RETORTS.

	Therms per diem.
Length 9 to 10 ft., section 15 ins. round	15 to 20
,, 10 ft., section 16 ins. round	20 to 27
,, 10 ft., section 21 ins. by 15 ins. dee	35 to 45
,, 10 ft., section 22 ins. by 16 ins. dee	

'THROUGH' RETORTS.

	Therms per diem.
Length 20 ft., section 24 ins. by 16 ins. dee	90 to 100
,, 20 ft., section 22 ins. by 18 ins. oval	90 to 105

A later development is the short 'through' retort, 12 to 16 ft. in length, while a 21-ft. retort is now sometimes employed.

VERTICAL RETORTS.

	Therms per diem.
Woodall-Duckham type	150 to 450
Glover-West type	120 to 300
Chamber ovens	500 to 600

The gas yield from vertical retorts is extremely flexible, very much depending upon the proportion of steam admitted and the class of coal carbonised.

COAL STORAGE.

In the present day, owing to uncertainty of delivery and such causes, it is advisable for gasworks to keep in hand much larger stocks than were thought necessary some years ago. No undertaking can now afford to risk three weeks' storage, although this is a usual and sufficient allowance for that kept under cover. When ground area is available large stocks may be arranged for out in the open. The calculation of the stock of coal in hand may be performed by assuming that one ton of average gas coal occupies from 42 to 43 cub. ft.

Coke Storage.

One ton of coke occupies 85 cub. ft. (average).

For the convenience of dealing with stocks of other materials used on gasworks the following figures are given :—

One ton of	broken coal occupies	40 cub. ft.
,, ,,	cannel coal occupies	35 to 45 cub. ft. (varies considerably).
,, ,,	anthracite coal occupies	35 to 39 ,,
,, ,,	breeze occupies	60 to 65 ,,
,, ,,	lime occupies	42 ,,
,, ,,	oxide of iron (Dutch)	45 ,,
,, ,,	ditto (prepared)	41 ,,
,, ,,	sulphate of ammonia	47 to 48 ,,
,, ,,	ashes	50 ,, (about).
,, ,,	Portland cement	23 to 26 ,,
,, ,,	loose earth	28 to 30 ,,

CONDENSERS.

Water condensers are now largely used in preference to atmospheric types. Many of the latter kind are, however, still in use, and are not without their recommendations. Rules for condenser capacity vary to some considerable extent, and there is by no means agreement on the subject. The following are given as representing the average practice :—

Atmospheric condensers.—Allow 5 sq. ft. of atmospheric condensing surface per 1,000 cub. ft. of gas made per diem.

Water condensers.—Allow 3 sq. ft. of surface per 1,000 cub. ft. per diem.

In using the first of these two rules it must be borne in mind that the figure arrived at includes (in addition to the areas exposed by the actual condenser) the area of all piping between the outlet of the hydraulic main and the inlet of the condenser. In making calculations the actual surface to be considered is that with which the gas is in contact. Thus, when the gas travels inside a tube the interior surface must be taken, and when outside the exterior surface is considered.

Water condensers are of two distinct types :—

(*a*) Multitubular condensers.

(*b*) Water-tube condensers.

In the first type the gas passes through the tubes, which are surrounded by a water jacket. In the second type the water is within the tubes and the gas around them. As to which type is chosen depends entirely upon the conditions of working. When any deposit is likely to take place it is preferable to have this within the tubes which may be readily cleaned. Thus, when muddy water alone is available there should be no hesitation about adopting the water-tube type. On the other hand, when the gas dealt with is likely to be particularly dirty or to contain a large quantity of heavy tar vesicles the multitubular type is to be preferred. Condensers for carburetted water-gas are invariably of the water-tube type. As regards general dimensions the height of the apparatus should be from two to five times the diameter ; 3-in. tubes are preferable for the larger condensers, but 2-in. pipes will suffice for those of smaller design. The ratio of the length of the tube to its diameter should be within the limits of 80 to 1, otherwise the tube loses in structural stability. In the direction of the number of tubes to be employed a good basis to work upon is that of arranging the tubes so that they occupy about 18 per cent. of the total area of the tube-plate. For water-gas condensers this may be increased to 25 per cent.

EXHAUSTERS.

In a works of from 4 to 6 million cub. ft. capacity per annum the saving to be effected by the installation of exhausters is problematical, after consideration has been given to interest on capital and running costs. Over six millions capacity the advantages of the machine are unquestionable. In the smaller works (from 15 to 20 million cub. ft. per annum) driving is nearly always carried out by means of a gas engine, the same engine providing power for operating tar, liquor, and other pumps. The effective hourly capacity of an exhauster is guaranteed by the makers. For the capacity of a four-blade machine, however, the following formula may be taken ; it is not to be recommended when dealing with very large or very small exhausters :—

$$Q = 65D^2LN ;$$

where,

Q = cub. ft. passed by the exhauster per hour ; D = inside diameter of outer drum in ft.; L = length of drum in ft. : N = number of revolutions per minute.

This gives the gross capacity. To arrive at the nett capacity, about 20 per cent. must be deducted (for 'slip') from the figure obtained.

On the larger works it is usual to maintain a vacuum of about 4 ins. of water at the exhauster inlet. At the outlet, the pressure to be overcome varies in accordance with the resistance offered by the apparatus following and the quantity of gas passing at the time. As a general rule the total back-pressure may vary between 8 and 10 ins. on a small country works to as much as

50 ins. of water on the largest plants, but under average conditions the following pressures will approximate to those found at the different portions of the apparatus :—

Retorts	Level gauge.
Hydraulic main	½ in. to 1 in. vacuum.
Exhauster inlet	4 ins. vacuum.
,, outlet	30 ins. pressure.
Tar extractor, outlet	26 ins. ,,
Washers, outlet	24 ins. ,,
Scrubbers, outlet	22 ins. ,,
Purifiers, outlet	9 ins. ,,
Station meter, outlet	8·5 ins. ,,
Holder, inlet and outlet	8·5 ins. ,,

It will be seen from the above that the forcing of the gas through the dry purification material and the raising of the holder bell account for by far the greatest portion of the work against back-pressure which the exhauster is called upon to perform.

Running Speed.

The speed at which the exhauster operates should be gauged, not by a consideration of the revolutions performed per minute, but by the peripheral speed of the tips of the blades. This speed should be approximately constant for all sizes of exhausters, and should lie within the limits of 650 to 850 ft. per minute.

Lubrication.

This is a point requiring the most careful attention. Much depends upon the type of lubricant made use of, the best being a mixture of refined creosote and a mineral oil. These should be admixed in the proportion of one part of creosote to three parts of mineral oil. A vegetable oil should not be used, as this will tend to thicken the tar inside the drum.

In modern practice there is a tendency for the ordinary rotary blade type of machine to give way to the newer high speed turbo-exhauster working on the fan principle and providing large clearances. An advantage attached to this type of machine is that it has some considerable volumetric capacity per unit of floor-space occupied. The construction consists of an outer case provided internally with two or more impellers which rotate on a common axis, but in distinct cells. The gas is passed from cell to cell in series until it is discharged from the machine at the requisite pressure, the pressure obtained being dependent upon the size and speed of the impellers.

WASHERS AND SCRUBBERS.

Washers.

The capacity of the wet purification plant depends very largely upon the type of apparatus employed, whether mechanical or merely static. Mechanical washer-scrubbers are now largely used, while of the static washers the well-known Livesey type has few rivals.

Rules for Capacity.—For washers of the Livesey and similar types allow one-seventh to one-fifth of a cubic foot capacity per 1,000 cub. ft. of gas passed per maximum diem.

For mechanical washer-scrubbers an allowance of one-half to three-quarters of a cubic foot capacity per 1,000 cub. ft. of gas per diem may be made.

Scrubbers.

Some works are provided with scrubbers of the tower type only, and have no separate washing vessels. In such cases the size of the scrubbers and their number may be based on the following rule :—

Allow 10 cub. ft. of capacity per 1,000 cub. ft. of gas dealt with per maximum diem. When the scrubbers are operated in conjunction with mechanical washer-scrubbers the above allowance may be reduced to from 5 to 6 cub. ft. When Livesey and such washers are employed prior to the scrubbers, the latter should have a capacity based on 6 cub. ft. of volume per 1,000 cub. ft. of gas per diem.

Washing and scrubbing capacity is greatly dependent upon the manner in which condensation is carried out. Extremely effective condensation will be the means towards reducing the usual wet purification allowances by nearly 50 per cent.

Horse-Power Required to Drive Washer-Scrubbers.

The following figures refer to the power necessary for the modern vertical type of machine. The horizontal machines of similar capacity will usually require one or two horse-power less :—

500,000 cub. ft. capacity per diem	4 H.P.
1,000,000 ,, ,, ,,	5 H.P.
2,000,000 ,, ,, ,,	7 H.P.
3,000,000 ,, ,,	10 H.P.

It is advisable to allow a small margin (about 20 per cent.) above these figures for emergencies.

If the power employed is electricity, and this is obtainable for a halfpenny per unit, the cost of driving amounts to about 0·033*d.* per 1,000 cub. ft.

The ratio of the height to the diameter of tower scrubbers varies between 2½ and 5 to 1. In exceptional instances, however, it may be so much as 7 to 1, particularly in the case of scrubbers of small capacity. When the diameter becomes greater than 30 ft. the flat cast-iron top is inclined to sag. In such cases it may be found advisable to provide a domed top having a rise equivalent to one-sixth of the diameter, and made from steel.

As regards the material employed for packing, Livesey has given the following figures relating to the area of wetted surface exposed to the gas :—

	Sq. ft. per cub. ft. of Scrubber Capacity.
Coke	8½
Drain pipes, 3 ins. diameter	17
,, 2 ins. diameter	21
Boards on edge	31

Boards on edge, although expensive in the first instance, are to be preferred to any other type of packing. They conform to no hard and fast rule as regards size, but the following will afford satisfactory results :—

6-in. boards, ¼ in. thick, laid ¾ in. apart.
11-in. boards, ⅛ in. thick, laid ⅛ in. to ¾ in. apart.

WATER REQUIRED.

The amount of water required to effect wet purification depends very largely on the type of apparatus, but very variable results are found. As an average it may be taken that the water admitted to the final scrubber will vary between 10 and 13 gallons per ton of coal carbonised. It has been found that with a modern form of vertical washer-scrubber the whole of the ammonia can be removed by admitting only 3¾ gallons of water per ton of coal carbonised.

DRY PURIFICATION PLANT.

Capacity of the dry purifiers depends upon the extent to which purification is carried. Nowadays it is customary to remove sulphuretted hydrogen alone, but if lime purification (for the removal of carbon disulphide) is resorted to it is advisable to employ rather larger vessels. It is always as well, however, that the vessels should be too large rather than too small, for the first cost increases but slowly as compared with the capacity, and purification charges undergo considerable reduction as the unit capacity becomes greater. For oxide boxes the following rule may be taken :—

For a set of four purifiers allow *for each box* an area of 0·5 sq. ft. per 1,000 cub. ft. of gas to be dealt with per maximum day. If lime is to be used this figure should be increased to 0·6 sq. ft., or 0·65 to 0·7 if no catch boxes are available.

Hunt's rule for the total amount of purification space required, i.e. the combined area of all boxes, says that 20 to 30 sq. ft. per ton of coal carbonised per maximum day should be allowed.

As regards the purifying capacity of oxide of iron it may be taken that one ton will purify about two million cubic feet of gas before it is finally spent.

In order that the oxide of iron in the dry purifiers may be retained in an alkaline condition it is not advisable to remove the whole of the ammonia from the gas. The usual practice is to permit 1 grain of ammonia per 100 cu. ft. of gas to remain as a minimum.

Purifiers are invariably square or rectangular in shape and are constructed from cast-iron plates bolted together. In one or two isolated instances reinforced concrete vessels have lately been experimented with.

As regards the necessary thickness of the cast-iron plates the following figures may be taken as indicative of the general practice :—

Purifiers up to 10 ft. square	⅜ in. plates.
,, ,, 20 ,, ,,	11/16 in. ,,
,, above 20 ,, ,,	½ in. ,,

The flanges are usually cast 1/16 to ⅛ in. thicker than the body. The smallest vessels are usually 3 ft. deep, while the largest sizes vary from 6 to 8 ft. in depth. The following rule is suggested for ascertaining the required depth :—

$$\text{Depth in feet} = \sqrt[4]{\text{Total superficial area}} \times 1{\cdot}1.$$

The depth as given may be taken to the nearest foot. For smaller purifiers, i.e. up to 300 sq. ft. in area, multiply by 1·3 instead of 1·1.

PURIFIER CONNECTIONS.

The diameter of the connecting mains is most readily and accurately computed from the following formula :—

$$\text{Diameter of pipe in ins.} = \sqrt{\text{area of each box in sq. ft.}}$$

For small purifiers (up to 60 sq. ft. area) the result can be taken as it stands, but for medium size boxes deduct 16 per cent., and for large vessels 25 per cent.

CONSTRUCTIONAL NOTES.

From the point of view of economy water-lute purifiers should be so constructed that the lute is portion of the side plate itself and not a distinct unit. This method of construction facilitates erection and is less costly in the first place. The depth of the lute is usually half the depth of the side plate, but the depth of water obtainable must in all events be sufficient to withstand the greatest pressure which is likely to prevail within the box. The covers of purifiers are invariably made from wrought iron or steel plates, usually about $\frac{1}{4}$ in. in thickness. They are either flat or slightly curved, but in either case some form of trussing or stiffening is necessary, and strong curbs should be provided when the vessel is of any size. In modern purifiers the water-lute principle is largely abandoned, and dry-luted covers are provided. These possess many advantages over the older method, chiefly in the direction of initial outlay, whilst they combine greater durability with increased safety. Moreover, owing to the lighter construction of the covers the lifting apparatus may be less cumbersome and costly. A gastight joint between cover and box is usually ensured by inserting a strip of rubber or tallowed hemp between the parallel flanges, the two flanges being afterwards pulled together by means of special fasteners or ordinary bolts.

For carburetted water gas purifiers the capacity allowed should be rather greater than is the case with purifiers used in connection with coal gas. A satisfactory allowance is 0·8 sq. ft. per 1,000 cub. ft. of water gas per maximum day.

STORAGE CAPACITY.

The holder capacity necessary is somewhat dependent upon whether or not the works is provided with a water gas plant. If such a plant is available the storage allowed for need not exceed 18 hours maximum make, but if there is no water gas to fall back upon provision should be made for 21 to 24 hours maximum production. In years gone by it was customary to provide considerably greater storage chiefly on account of the heavy fluctuations in output. The levelling up of the hourly load-curve has, however, had a marked effect in the direction of diminishing the quantity of gas kept in hand.

PROPORTIONS OF GASHOLDERS.

Single lift height = 0·3 to 0·4 of diameter.
Multiple lift height = 0·6 to 1·0 of diameter.

In computing the capacity of a holder the volume enclosed by the crown is neglected, as the gas contained therein is not available for distribution. Accordingly

$$\text{Capacity in cub. ft.} = 0.7854 \times \text{diameter in ft.}^2 \times \text{height in ft.}$$

With multiple lift holders the capacity of each lift should be estimated separately by applying the above formula to each section.

DEPTH OF LIFTS.

Each lift will be approximately of equal depth.

RISE OF CROWN.

The greater the rise the stronger will be the crown. The dome should conform to a segment of a sphere. Normally, the rise will range between $\frac{1}{15}$th to $\frac{1}{10}$th of the diameter, but when trussing is employed it may be less than when the plain crown only is used.

The cubic capacity of the crown space may be found from the following :—

$$\text{Volume in cub. ft.} = 0.5236\,R\left(R^2 + \frac{3D^2}{4}\right),$$

where,

R = rise of crown in ft.;
D = diameter of top lift in ft.

The depth of each lift should never be less than one-seventh of its diameter, otherwise locking or tilting may occur. In general, the proportion varies between one-fourth to one-fifth of the diameter.

The crown sheeting of gasholders varies from 14 gauge to 10 gauge (about $\frac{5}{64}$ in. to $\frac{1}{8}$ in.), but much depends upon the size of the holder.

The side sheeting varies from 12 gauge to 10 gauge, but curb or cup and grip rows are thicker, $\frac{1}{4}$ in. to $\frac{3}{8}$ in. Top lift sheets are usually thicker than those of succeeding lifts, often about 8 gauge.

The thinner sheets of the bell are generally riveted up cold with $\frac{1}{4}$ in. rivets at a pitch of 1 in. to $1\frac{1}{8}$ in. All rivets above $\frac{1}{4}$ in. must be driven hot.

In the gas holder recently built for the Gartson station of the Liverpool Gas Co., wrought-iron plates are used, as these are less liable to corrode than plates of mild steel. The gasholder, which is of the spirally guided type, is 150 ft. in diameter and has a total height of 148 ft. The iron plates have a tensile strength of 21 tons per sq. in. and an area reduction of 18 per cent. with the fibre, and 18 tons per sq. in. with an area reduction of 8 per cent. across the fibre.

(*Machinery*, *April* 2, 1925.)

GASHOLDER TANKS.

Steel tanks have many advantages over the masonry variety, and cast-iron as a material for this purpose may now be looked upon as obsolete.

The thickness of plate required may be calculated as follows:—

$$T = \frac{PD}{2fe},$$

where,

T = thickness of the plate in ins.;

D = diameter of the tank in ins.;

f = allowable working stress (usually $5\frac{1}{2}$ tons per sq. in.);

e = the efficiency of the vertical riveted seams. This will range up to 90 per cent. for large holders. For small holders it will vary between 75 and 85 per cent.;

P = the pressure due to the water in lbs. per sq. in.

The latter may readily be found from the following:—

$$P = WD,$$

where,

W = the weight of a cub. ft. of water = 62·4 lbs.;

D = depth below the surface of the section under consideration.

As regards the apportionment of the cost of gasholders, the following figures represent normal conditions:—

	Per Cent. of Total Outlay.
Bell, complete	35 to 45
Guide framing	18 to 23
Tank (steel)	35 to 45

Single lift holders (up to about 50,000 cub. ft.) cost in pre-war days from 25*l.* to 28*l.* per 1,000 cub. ft. If of the spiral type the outlay would be reduced by 25 per cent. Two lift holders (about 250,000 cub. ft. capacity) ranged from 16*l.* to 20*l.*; or, if spirally guided, about 10*l.* per 1,000 cub. ft. capacity. Holders varying from 2 to 3 million cub. ft. capacity cost from 9*l.* to 12*l.* per 1,000 cub. ft. The large 12-million holder at the East Greenwich Works cost five guineas per 1,000 cub. ft. capacity, whilst a $10\frac{1}{2}$-million holder at Manchester worked out at 5*l.* per 1,000 cub. ft.

CALCULATION OF PRESSURE THROWN.

The pressure thrown is dependent upon the weight of the floating portion and its diameter. It may be found as follows:—

$$\text{Maximum pressure (ins. of water)} = \frac{549W}{D^2},$$

where,

W = weight of bell and water in lutes, in tons;

D = diameter of holder in ft.

When dealing with a multiple lift holder take the mean diameter.

Size of Works Mains.

No hard and fast rules are adhered to, but the following will form a guide as to the average conditions :—

Size of Works.	Suitable Size for Mains. Ins. Diam.
3 to 4 million cub. ft. per annum	5
6 to 10 " "	6
15 to 25 " "	8
30 to 45 " "	10
55 to 90 " "	12
100 to 150 " "	15
200 to 300 " "	18
350 to 450 " "	20 to 21
500 to 600 " "	24
600 to 900 " "	30

Composition of Various Gases.

Per Cent. by Volume.	Coal Gas.	Carburetted Water Gas.	'Blue' Water Gas.	Mond Gas.	Suction Gas.	Dowson Gas.
Hydrogen	49·0	35·0	52·0	26·0	16·0	20·0
Carbon Monoxide	9·0	32·0	38·0	15·0	34·0	24·0
Carbon Dioxide	2·5	4·5	4·5	12·0	3·0	6·0
Methane	27·0	13·0	1·0	3·0	nil	1·0
Heavy Hydrocarbons	4·0	10·0	nil	nil	nil	nil
Nitrogen	8·5	5·5	4·5	44·0	47·0	49·0
Candle Power	14·0	18·0	nil	nil	nil	nil
Calorific Power, B.Th.U. gross	550	580	300	165	160	155

Calorific Value of Various Fuels and their Cost per Therm.

The following table (compiled by G. Evetts) gives the calorific value of various fuels, and the lost per therm (100,000 B.Th.U.). The cost must only be regarded as the prime cost of a therm used in such a manner that the heating power can be developed :—

	Calorific Value. B.Th.U.	Cost.	Cost per Therm Pence.
Coal	13,000–15,000 per lb.	36*s.* per ton	1·4
Coke	11,500–13,500 per lb.	30*s.* per ton	1·35
Coal Gas	400–560 per cu. ft.	10*d.* per therm	10·0
Electricity	3,440 per unit	2*d.* per unit	58·2
Petrol	19,500 per lb.	2*s.* per gall.	16·6
Fuel Oil	18,000 "	5*l.* per ton	3·0
Motor benzole	19,000 "	2*s.* per gall.	14·2
Alcohol	12,000 "	5*s.* "	62·0

Composition of Coal and Coke.

Per Cent.	Anthracite Coal.	Average Gas Coal.	Average Gas Coke.
Carbon	94·0	80·0	88·0
Hydrogen	2·5	5·5	0·2
Oxygen	0·25	8·8	traces to 3·0 (varies considerably)
Nitrogen	0·5	1·5	2·0
Sulphur	0·5	0·8	1·0
Ash	2·25	3·4	5·8
Calorific Power, B.Th.U.	15,500 per lb.	14,800 per lb.	13,500 per lb.
Volatile Matter	8 per cent.	30 per cent.	1 per cent.

Products of Distillation.

One ton of average gas coal will yield (approximately) the following products :—

Gas	14,000 cub. ft., or approximately 70 therms.
Coke	12 to 13 cwts., excluding breeze.
Breeze	1 cwt.
Tar	9 to 11 gallons.
Ammoniacal liquor	32 to 40 gallons at 8 ozs.
Sulphate of ammonia	24 to 28 lbs.
Sulphur	5 to 7 lbs.
Cyanogen (HCN)	4 lbs.
Virgin liquor	10 to 12 gallons.

The yield of gas as given above is the normal quantity of straight coal gas obtained, but it should be borne in mind that the modern vertical retort in which 'steaming' is employed will give a yield of from 85 to over 100 therms per ton of coal.

(The sulphate of ammonia is obtained from the ammoniacal liquor, and not in addition to it. The virgin liquor is included in the figure given for total yield of ammoniacal liquor. The figure given for sulphur is that which may be recovered during purification of the gas, and does not include the quantity remaining behind the coke.)

The approximate distribution of the elements of the original coal, and the manner in which each exists after the process of carbonisation, is illustrated in the following table :—

One ton of coal, consisting of :—

	Lbs.
Carbon, 1,792 lbs., distributed as—	
Coke	1,385
Breeze	107
In gas, as CO and CO_2	30
In gas, as hydrocarbons	130
In tar	122
As scurf, cyanide, etc.	18
Hydrogen, 123 lbs., distributed—	
In gas and tar	98·5
As ammonia	1·0
As water	18·5
In coke	5
Oxygen, 197 lbs., distributed—	
In gas, as CO, CO_2, and combined in tar	49
As water	148
Nitrogen, 34 lbs., distributed—	
In gas (free)	11·8
In coke	14·6
In tar (combined)	1·0
As ammonia	5·8
As cyanogen	0·8
Sulphur, 18 lbs., distributed—	
As gaseous impurities and in liquor	6
In coke	12
Ash remaining in coke and breeze	76
Total	2,240

Amount of Coke Used as Fuel.

The quantity of coke used for heating the retort benches varies in accordance with the system of firing adopted. There are three types of firing now in common use, namely :—

(*a*) Direct-fired settings.
(*b*) Semi-gaseous settings.
(*c*) Gaseous-fired settings.

In addition, some works employ external producers, in which the producer gas is generated in a producer of special design situated away from the retorts, and forming a central source from which all the settings are supplied. At the Fuel Research Station at East Greenwich blue water gas is employed for heating a setting of vertical retorts.

The following shows the amount used :—

	Per cent. of the original weight of coal carbonised.
Direct-fired settings	28
Generator settings	15 to 18
Regenerator settings	11 to 14
Vertical retorts :—	
(*a*) Intermittent types	16
(*b*) Continuous types	11½
Outside producers	13 to 16

With modern systems it should be possible actually to sell 10 cwts. of coke per ton of coal.

See also ' Coke and Coking,' p. 2096, and ' Coke as Boiler Fuel,' p. 2101.

Composition of the Producer Gas.

The following is the average composition under good conditions :—

	Per cent. by volume.
Carbon monoxide	25
Carbon dioxide	5
Hydrogen	6
Methane	1
Nitrogen	63

(Oxygen should be absent, for its presence indicates an excess of primary air.)

The analysis of the waste gases emitted from the retort setting should approximate to the following :—

	Per cent. by volume.
Carbon dioxide	19
Oxygen	0·4
Nitrogen	80·6

(The presence of carbon monoxide in the waste gases is indicative of a deficiency of secondary air, and is wasteful in firing. The oxygen, if above 0·5 per cent., denotes an excess of secondary air.)

Gasworks Temperatures.

	F.°
Retorts	1800 to 2100
Ascension pipes	200 to 300
Foul main (inlet)	140
Condensers (inlet)	125
,, (outlet)	75 to 85
Washers and scrubbers (outlet)	60
Purifiers (inlet)	60
,, (outlet)	65
Holders	60

The modern combustion chamber temperature rarely, if ever, exceeds 2800 F.° The prevailing temperature of the fuel-bed is in the neighbourhood of 1800 to 1900 F.°

Before the introduction of the regenerative system, one of the chief causes of the loss of heat arose from the escape of the waste gases of the producer at unnecessarily high temperatures. In such cases the normal temperature at which the gases are allowed to leave the settings is in the neighbourhood of 1700 to 1800 F.°, but by the transference of the sensible heat from these gases to the inflowing secondary air the temperature of the waste gases has, in the modern regenerator, been reduced to 600–800 F.° before their escape is permitted. In the ordinary way, the secondary air approaches a temperature of 1800 F.° before combustion.

Grate and Chimney Area.

The grate area of the retort bench producer must in all cases be ample, so as to ensure a steady travel for the air passing through the fuel. When the ordinary horizontal fire-bar grate is employed the recognised modern rule for grate area is as follows :

Allow one-tenth square foot of grate area for every lineal foot of retort in the setting. Thus, for a through setting of ten 20-ft. retorts, this gives a total grate area of 20 sq. ft.. Another,

but less common rule, is that which gives the allowance as 1 sq. ft. of area for every 8 lbs. of coke consumed by the furnace per hour. As the actual consumption of coke (which is shot into the producer hot) is somewhat difficult to compute, the rule is not readily applied.

The ordinary horizontal form of bar grate is now rapidly giving way to the sloping form of grate. The original sloping arrangement consisted of a number of flat plates placed progressively in the manner of a stairway, cooling being effected by a small stream of water which was permitted to drop from step to step. The latest development is the Grille grate, which is composed of a number of independent components set together so as to form an interlocking grille. The grate is made entirely from special cast iron, and embraces a series of hanger bars, which are hung from specially shaped cross-bars. When in the working position each hanger is automatically locked, so that it cannot be displaced during the process of cleaning the fire, while at the same time, should it be necessary to remove any one of the hangers, this may readily be done by applying a slight forward movement to the lower end. The grate is employed in conjunction with a supply of steam, and the formation of clinker is practically avoided, the inert matter coming down in the form of a fine dust.

The rule given above for grate area does not apply when grates of the sloping type are employed, for, owing to the more favourable conditions of the base of the fuel bed, the circulation of primary air is uninterrupted by layers of dead matter. With the Grille type of grate the following rule may be employed :—Allow 0·6 sq. ft. of sloping superficial area per 10 lineal ft. of retort in the setting.

Flue and chimney area is calculated by allowing $1\frac{1}{2}$ sq. ins. of area per lineal foot of retort in all those settings which depend upon the flues and chimney under question. Thus, if a retort bench consists of eight settings, each containing ten 20-ft. retorts, the area required would be $16\frac{1}{2}$ sq. ft.

In large retort houses the height of the chimneys varies from 50 to 70 ft. The chief considerations to bear in mind are the heights of adjoining buildings, also the avoidance of down draughts and the effect of shifting winds. It is not advisable to connect a chimney to a greater number of retort beds than ten, for above this number its dimensions become unwieldy. Circular chimneys, except for vertical retort installations, are now almost things of the past. The rectangular chimney with an external batter is most commonly employed. The batter usually varies between the limits of 1 in 36 to 1 in 40. Retort house chimneys are usually braced for their entire height; the general methods being to run angle-iron up each corner, the angles being held in position by tie bands placed at intervals of about six feet. An inspection door should be left at the base of the shaft for cleaning purposes. As regards material of construction, stock bricks may be employed for the external courses, but a lining of fire bricks in the interior is essential. Very frequently the whole chimney is constructed from fire bricks, in which case no lining is, of course, necessary. The volume of gas drawn up a chimney increases in proportion to the square root of its height; thus any increase in height produces only a comparatively small change in this respect. On the Continent it is occasionally found that a separate chimney is installed for each retort setting, but the idea is confined almost wholly to stop-ended settings. The construction of retort bench shafts should be such that the cross sectional area, in spite of the batter, is constant throughout the whole height. This may readily be arranged for by reducing the thickness of the brickwork from the bottom upwards.

Proportions of Horizontal Retort Benches.

The following normal dimensions are given as exemplifying the usual proportions of various types of retort houses and benches :—

(*a*) Single retort bench, 10-ft. retorts :—	ft.	ins.
Width of bench	12	6
At back of bench	1	6
In front of bench	18	0
Total width inside houses	32	0

The height of the retort house to the eaves would be 20 ft., and the height of the retort bench from ground level 11 ft.

(*b*) Through retort bench, hand charging and discharging—	ft.	ins.
Width of bench	20	0
Each side of bench, 20 ft.	40	0
Total width of house	60	0

The height of the retort houses to eaves, if house of the stage type, would be about 35 ft.; if of the subway type 8 to 10 ft. less. The height of the retort bench from base of producers to top of brickwork about 20 ft., if retorts set in three tiers.

(*c*) Through bench, mechanical charging and discharging—	ft.	ins.
Width of bench	20	0
Charging side of bench	23	0
Discharging side	17	0
Total width of house	60	0

Retort House Governors.

The retort house governor is a more or less essential auxiliary of the modern retort house. It differs from the station governor in that, whilst the latter is employed with the object of reducing pressure, the retort house governor is interposed for the purpose of reducing the intensity of vacuum created by the exhauster. Thus the governor ensures a constant vacuum in the hydraulic main, whatever the volume of gas coming away from the coal charges may be. The intensity of vacuum required is obtained by loading the bell on similar lines to those employed with the station governor. The best place for the governor is on the top of the retort bench, as close up to the end of each section of the foul main as possible, for it is not so sensitive when interposed at the inlet or outlet of the condenser as is occasionally found. By-passes should be fitted to the governor, for it will require cleaning out about every four months.

Cost of Manufacturing Coal Gas.

The cost of manufacturing towns' gas depends primarily upon the capacity of the works, its geographical position relative to suitable coalfields, and the character of the plant in use. It must be borne in mind, too, that the actual cost of production (*i.e.* cost into gasholder) represents only about one-half of the ultimate selling price, the remainder being accounted for by the cost of distribution, overhead charges, dividends, etc. In considering a normal case of an undertaking supplying about 1,000 million cu. ft. of 500 B.Th.U. gas per annum, the manufacturing costs would be approximately as follows, assuming that coal gas only was produced :—

	Cost per 1,000 cu. ft.	Cost per Therm.
Coal	26*d.*	5·2*d.*
Less Residuals	17*d.*	3·4*d.*
Net cost of coal	9*d.*	1·8*d.*
Purification	0·5*d.*	0·1*d.*
Salaries and wages	3·5*d.*	0·7*d.*
Wear and tear	7·5*d.*	1·5*d.*
Net cost into gasholder	20·5*d.*	4·1*d.*

Gasworks Conveyors and Elevators.

The formula given below may be taken as giving a useful guide as to the capacity of bucket conveyors :

$$\text{Capacity} = \frac{S \times B \times W}{96{,}768 \times P} \text{ tons per hour,}$$

where,

B = capacity of buckets in cub. ins. ;
S = speed of conveyor in ft. per minute ;
W = weight per cub. ft. of substance carried (see page 2452) ;
P = pitch of buckets in ft.

The horse-power required to operate elevators and conveyors may be readily arrived at by adopting a coefficient of friction in accordance with the conditions under which the machine is working.

Let M = the weight of material carried in lbs.
W = the weight of the moving parts in lbs.
K = the coefficient of friction.
S = the speed of the conveyor or elevator in ft. per minute,

then, for

(*a*) Vertical elevator :

$$\text{H.P.} = \frac{M \times S}{33{,}000} + \frac{K \times S(M+W)}{33{,}000};$$

(*b*) Inclined elevator :

Let α° = the angle of inclination with the horizontal, then,

$$\text{H.P.} = \frac{M \times S \times \sin\alpha^\circ}{33{,}000} + \frac{K \times S(M+W)\cos\alpha^\circ}{33{,}000};$$

(*c*) Horizontal conveyor :

$$\text{H.P.} = \frac{K \times S(M+W)}{33{,}000}.$$

As regards the value of K, the following figures may be taken :

0·15 for metal sliding on metal ;
0·07 to 0·8 ,, ,, (well lubricated) ;
0·25 for wood sliding on wood.

Purification of Coal Gas.

The purification of coal gas is carried out in two stages, namely, by means of wet purification in the washers and scrubbers and afterwards in the dry purifiers. During the wet purification of the gas the alkaline ammonia is made use of for removing a portion of the acidic impurities, sulphuretted hydrogen and carbon dioxide. The two last-named are, therefore, absorbed with the formation of certain salts, which are themselves soluble in water. The resultant solution consists of water containing a certain proportion of these salts, and is known as ammoniacal liquor. In crude coal gas the ammonia content is usually in the neighbourhood of 1·5 per cent. by volume, and of this amount from about one-third to one-half will have been deposited prior to the gas reaching the wet purification plant. Hunt has given the following figures for the distribution of ammonia :

	Per cent. of total.
Ammonia removed by condensation . . .	42·7
„ „ first scrubber . . .	43·3
„ „ second scrubber . . .	**14·0**

When gas containing carbon dioxide and sulphuretted hydrogen comes in contact with a solution of ammonia both acid and normal salts are formed. Thus we have :

(*a*) $2NH_3+H_2O+CO_2=(NH_4)_2CO_3$ (normal) (*c*) $2NH_4OH+SH_2=(NH_4)_2S$ (normal) $+2H_2O$.
(*b*) $NH_3+H_2O+CO_2=NH_4HCO_3$ (acid) (*d*) $NH_4OH+SH_2=NH_4HS$ (acid) $+H_2O$.

Composition of Gas Liquor.

The ammonia present in gas liquor is combined in two distinct ways. In the first place it is present as easily decomposable compounds, such as carbonates and sulphides, and secondly, as more stable compounds which will only undergo decomposition by heating in the presence of a caustic alkali. In general, the final mixture of gas liquor obtained consists of from 75 to 80 per cent. of the more unstable salts, i.e. 'free' ammonia, whereas the remaining 20 to 25 per cent. is made up of the more stable salts, i.e. 'fixed' ammonia. The possible constituents of gas liquor are enumerated below, although it must be borne in mind that any one sample of liquor will seldom contain all of the compounds given. In fact, some of those included are only met with on rare occasions.

Water, containing in solution:

Ammonium carbonate „ bicarbonate „ carbamate „ sulphide „ hydrosulphide „ cyanide „ polysulphide	'Free' ammonia.	Ammonium chloride „ sulphocyanide „ sulphite „ thiosulphate „ sulphate „ ferrocyanide „ acetate	'Fixed' ammonia.

Also phenols, amines, and other nitrogenous bodies.

The amount of ammonia in gas liquor is in reality quite small, and varies between 1·5 and 2 per cent. In general, it will be apportioned as follows :

	Per cent.
'Free' ammonia	1·2 to 1·6
'Fixed' ammonia	0·3 to 0·4

The following figures relate to sulphuretted hydrogen and carbon dioxide :

	Per cent.
Carbon dioxide in gas liquor .	1·3 to 2
Sulphuretted hydrogen . .	0·1 to 0·6

The distribution of the total ammonia in gas liquor varies, of course, to some considerable extent. The following table, however, refers to the average solution :

	Per cent. of total ammonia.
Ammonium sulphides	6
„ carbonates	74
„ chloride	15
„ sulphocyanide	1·75
„ thiosulphate, sulphite, etc. . . .	3
„ sulphate	0·25

The most effective manner of operating the washing and scrubbing plant is to ensure that the gas on first entering the apparatus is treated with the strongest liquor available. In each succeeding vessel the washing liquor employed becomes gradually weaker and weaker until, in the final scrubber, clean water should be employed. Where circumstances permit, it is preferable that the first washer should be operated with the liquor from the hydraulic main, for in this way both CO_2 and SH_2 may be considerably reduced.

The amount of water which must be added for the purpose of wet purification seems to vary to some considerable extent in accordance with the conditions prevailing. The writer has

found with Durham coal that the ammonia may be completely removed in ordinary coke-filled tower scrubbers by adding from 10 to 13 gallons per ton of coal carbonised. When mechanical washer-scrubbers are in use the quantity of water required will be considerably less, probably no more than 4 gallons per ton of coal.

Mr. Thos. Glover states that the liquor distributed per sq. ft. of cross sectional area of the scrubber should be 0·145 gallon per minute. This is less the quantity of clean water added to the final scrubber.

Ammonia Lost.

The quantity of ammonia lost in the direction of that permitted to go forward from the final scrubber may be conveniently calculated as so many gallons of 10 oz. liquor.

$$\text{Quantity of 10-oz. liquor lost per week (in gallons)} = \frac{\text{Total grains of ammonia passing forward from scrubber per week.}}{1{,}522}$$

Impurities in the Gas at various stages.

	Grains per 100 cub. ft.
Outlet of condensers—	
Ammonia	200 to 300
Sulphuretted hydrogen	600 „ 900
Carbon dioxide	900 „ 1,400
Sulphur compounds	35 „ 45
Hydrocyanic acid	60 „ 80
Naphthalene	20 „ 25
Inlet of purifiers—	Grains per 100 cub. ft.
Ammonia	nil to 1·5
Sulphuretted hydrogen	450 „ 700
Sulphur compounds	35 „ 45
Hydrocyanic acid	50 „ 70
Carbon dioxide	750 „ 1,150
Outlet of purifiers—	Grains per 100 cub. ft.
Ammonia	nil
Sulphuretted hydrogen	nil
Carbon dioxide	750 to 1,150
Sulphur compounds	35 „ 45
Hydrocyanic acid	nil „ 10

(The last three named above may all be removed, or partially removed, by the employment of special processes, but in the majority of gasworks such processes are not resorted to.)

Removal of Sulphuretted Hydrogen.

Sulphuretted hydrogen is now almost universally removed by means of oxide of iron in one of its forms. Natural ferric oxide (bog-ore) is still largely employed for the purpose, although in recent years various artificial preparations have found a good deal of favour, chiefly owing to their lower cost. The natural bog-ore is chiefly obtained from deposits in Holland, Belgium, and Ireland. In the dried state its composition varies between the following limits :

	Per cent.
Hydrated ferric oxide (Fe_2O_3, H_2O)	60 to 65
Organic matter	15 „ 25
Silica	3 „ 6
Alumina	1

The active material in bog-ore and artificial preparations is not actually the iron oxide, but the oxide plus a certain quantity of water of combination, i.e. a hydrated ferric oxide. In addition to this moisture the material as used should contain a further quantity of water, usually from 20 to 30 per cent. The artificial oxides are employed with a similar degree of moisture. The made-up or artificial material, if of good quality, is usually rather more active than the natural product, owing to a rather higher content of ferric hydrate, which may reach so much as 70 to 75 per cent.

Reactions during the Removal of Sulphuretted Hydrogen.

The absorption of SH_2 by the active oxide occurs in two distinct ways :

(*a*) Fe_2O_3, $H_2O + 3H_2S = Fe_2S_3 + 4H_2O$.

(*b*) Fe_2O_3, $H_2O + 3H_2S = 2FeS + S + 4H_2O$.

In the first instance ferric sulphide is formed, whilst secondly the result is ferrous sulphide and some free sulphur.

When spent and inactive the material is removed from the purifier and subjected to the process of revivification, or exposure to the air, when the sulphides are oxidised to oxides and the material is ready for further use. The following are the revivification reactions:

$$(a)\ 2Fe_2S_3 + 3O_2 + 2H_2O = 2Fe_2O_3,\ H_2O + 3S_2.$$
$$(b)\ 4FeS + 3O_2 + 2H_2O = 2Fe_2O_3,\ H_2O + 2S_2.$$

Oxide is generally laid in the purifiers on grids to a depth of from 8 to 10 ins. In some works, however, it is still customary to employ deep layers of as much as 30 ins. This is particularly the case in those purifiers in which hurdle grids are made use of.

A space of at least two inches should be left between the top of one layer and the base of the sieves or grids of the layer immediately above. Oxide as it becomes fouled increases in bulk, hence this precaution. Partial revivification *in situ* is nearly always arranged for in present-day systems. This is effected by admitting a small measured quantity of air to the purifiers along with the gas. The air may be taken from a small compressor and forced into the box, or may be injected by means of steam. In other cases, it is sucked in with the gas through a pipe attached to the main on the inlet side of the exhausters. When air is admitted at some point prior to the washers and scrubbers it should not be overlooked that undesirable reactions may be set up in the wet purification plant.

It is generally understood that the material in any one purifier cannot simultaneously extract sulphuretted hydrogen and undergo revivification. The assumption is not entirely accurate, but the process of sulphiding takes place at a very much more rapid rate than does the oxidation effect.

If the capacity of the purification plant is not ample the full effect of the admitted air will not be realised. For the most effective results a slight excess of air above that theoretically required (as determined from the SH_2 present) should be admitted.

When air is admitted to a purifier a rise in temperature of about 5 degrees Fahr. will usually be found as the gas passes from inlet to outlet. The air acts, of course, as a diluent, and therefore a careful check must be kept on the quantity admitted with the gas. As regards the effect of the air in this direction the following may be quoted:

Audouin and Berard found that 1 per cent. of air reduces illuminating power by 6 per cent.

	Per cent.
2 per cent. of air reduces illuminating power by . .	11
3 " " " " . .	18
4 " " " " . .	26

Würtz has stated that

3 per cent. of air causes 15·69 per cent. loss in light.
5 " " 24·0 " "

Dr. W. B. Davidson, in more recent experiments, has found the following results:

				Per cent.
1 per cent. of	oxygen	reduces	candle power by .	3
1 "	"	"	calorific power by .	1
1 "	nitrogen	"	candle power by .	2·8
1 "	"	"	calorific power by .	1
1 "	carbon dioxide	"	candle power by .	3·5

Lime Purification.

Prior to the year 1905 lime purification for the removal of carbon dioxide and sulphur compounds other than sulphuretted hydrogen was in almost universal use. To-day, purification by this method is met with only in isolated instances.

Lime in the slaked condition is an active absorbent of sulphuretted hydrogen and carbon dioxide, and—after special treatment—is capable of removing carbon disulphide. The following reactions occur during the process of purifying by means of lime:

(*a*) **Removal of carbon dioxide—**

$$Ca(OH)_2 + CO_2 = CaCO_3 + H_2O.$$

(*b*) **Removal of sulphuretted hydrogen—**

$$Ca(OH)_2 + 2SH_2 = Ca(SH)_2 + 2H_2O.$$

(*c*) **Removal of CS_2.**

It must be borne in mind that slaked lime in itself has no action upon carbon disulphide. The active medium which accounts for the removal of the CS_2 is now generally supposed to be calcium hydroxyhydrosulphide. This is prepared by sulphiding the slaked lime, in other words by admitting gas containing sulphuretted hydrogen to the vessels which are destined to

remove the CS_2. Carbon disulphide certainly cannot be absorbed according to the following equation, which has been so often set down as illustrating the reaction:

$$CaS + CS_2 = CaCS_3.$$

Veley has laid down the following theory as to the actual reactions which occur:

(1) Calcium hydrosulphide is formed in the first place—

$$Ca(OH)_2 + 2H_2S = Ca(SH)_2 + 2H_2O.$$

(2) By the addition of water or oxygen the hydrosulphide is converted into hydroxyhydrosulphide—

(*a*) $Ca(SH)_2 + H_2O = CaOH,SH + SH_2$

or

(*b*) $2Ca(SH)_2 + O_2 = 2CaOH,SH + 2S.$

(3) The carbon disulphide and hydroxyhydrosulphide then react as follows:

(*a*) $2CaOH,SH + CS_2 = SH_2 + Ca(OH)_2,CaCS_3$

or

(*b*) $3CaOH,SH + CS_2 + H_2O = 2SH_2 + (Ca(OH)_2)_2,CaCS_3.$

When gas is manufactured and is to be freed from the whole of its impurities it is usual to instal a system in which oxide of iron and lime are used in conjunction. Many methods of operating for this purpose are employed, but one of the most common systems is that generally known as the 'Beckton' method. It consists in the main of four sets of purifiers operated in rotation:

First Set, Lime.—These remove the whole of the CO_2 and some SH_2, which is afterwards displaced by CO_2. Some CS_2 is also absorbed.

Second Set, Oxide.—These remove the whole of the SH_2, including any which may be displaced from the lime in the previous set.

Third Set, Lime.—This lime has been previously 'sulphided' and removes the bulk of the CS_2. It will not remove the whole of this impurity.

Fourth Set, Oxide.—These vessels remove any SH_2 which is disengaged from the third set.

Cost of Purifiers.

It is difficult to give any reliable figure, based on each 1,000 cub. ft. capacity, for the cost of purifiers, for everything is dependent upon the size of the vessel. The larger the box, the lower the cost per unit of capacity. As regards the distribution of the cost, the following is illustrative of normal conditions with a medium size installation:

	Per cent. of total expenditure.
Purifying boxes	50
Covers (dry lute)	15
Lifting apparatus	4 to 6
Grids	6 to 10
Connections and valves	20 to 30

Castings for the boxes, in normal times, cost from 10*l.* to 11*l.* per ton. The covers cost from 17*l.* to 19*l.* per ton. As regards valves, it is usual to base these per inch of diameter. The following figures are approximate pre-war costs:

	Per inch of diameter.
Ordinary rack and pinion valve	1*l.*
Centre valves	9*l.*
Weck's valves, and such types	10*l.* 10*s.*

Removal of CS_2 by Hot Purification.

Owing to the uncertainty and expense of lime purification for the removal of sulphur compounds other methods of effecting the same end have recently been introduced. The most important of these is the Carpenter-Evans system of hot purification, which is now in everyday use at some works. The principle of the process consists in converting the CS_2 into sulphuretted hydrogen, and then removing the latter by means of ordinary oxide of iron purification. The gas, after being purified from SH_2 in the usual way, is passed into special chambers containing a catalyst heated to a temperature of about 800°F. The CS_2 is thereby decomposed according to the following reaction:

$$CS_2 + 2H_2 = 2H_2S + C.$$

The catalytic agent employed consists of fireclay balls about 1 in. in diameter impregnated with nickel reduced from the chloride. The free carbon resulting from the reaction is left behind in the catalysing apparatus, and after about thirty days' run it is necessary to remove it. This is carried out by the process known as 'aeration.' Air is pumped through the catalyser tubes at a definite rate, thus burning out the carbon and disposing of it as carbon dioxide. The aeration

process on the larger plants usually occupies about one week, after which the apparatus is again ready for work. When starting and stopping special precautions have to be taken to ensure immunity from explosion. The effect of the process on the general composition of the gas seems to be but slight; as an instance the following figures, given by Dr. Carpenter, are quoted:

Analysis of Gas.	Before Treatment.	After Treatment.
CO_2, per cent. by volume	1·44	1·45
Heavy hydrocarbons	3·55	3·63
Oxygen	0·33	0·06
Carbon monoxide	8·16	8·03
Methane	26·85	27·26
Hydrogen	54·19	54·25
Nitrogen	5·48	5·32
Illuminating power	14·70	14·70
Calorific power (Gross, B.Th.U.)	590·0	594·40

Cost of Purification.

The present-day cost of operating the Carpenter-Evans system of purification is shown in the following table:—

	Pence per 1,000 cubic feet.
Fuel (coke @ 42*s.* 6*d.* per ton)	0·2550
Conveying coke and cleaning producer	0·0240
Power	0·0129
Labour and chemical supervision	0·1180
Repairs and maintenance	0·1649
Capital charges	0·1967
Total	0·7715*d.*

Purification by means of Charcoal.

Mr. W. Gordon Adam has introduced a method for removing carbon disulphide and thiophen by passing the gas, after washing and scrubbing, through animal or wood charcoal in a granulated state. The impurities (including naphthalene) are afterwards removed from the charcoal in an unchanged condition by the application of heat, a temperature of about 300° C. being sufficient. After this treatment the charcoal is ready for further use, and is employed in a purifying vessel of ordinary construction. The prevention of ethylene absorption is effected by curtailing the quantity of charcoal used—in other words, by increasing the space-velocity above that at which ethylene is extracted.

Removal of Naphthalene.

During recent years it has become usual to employ special naphthalene solvents, to the action of which the gas is subjected before travelling to the district mains. Paraffin, water-gas, tar, and anthracene oil have all proved effective, particularly the latter. In general the gas will contain, if no special treatment is accorded it, from 6 to 12 grains of naphthalene as it leaves the works. Butterfield has pointed out that a million cubic feet of gas cooled from 68° to 50°F. may deposit about 14 lbs. of naphthalene. The introduction of carburetted water gas has to a certain extent lessened the naphthalene evil, although its effect in this direction is somewhat erratic. As regards the methods of washing with solvents, the most ready plan to adopt is that of treating the gas in a Livesey or mechanical washer just prior to its entering the holders.

Carburation with the aid of solvents was introduced by Botley, who injected, by means of a special spray, about 4 gallons of thoroughly atomised paraffin oil per million cubic feet of gas. Botley found that petroleum oil was more suitable for the purpose than spirits such as benzol. Leather made use of a creosote oil for washing the gas, but owing to the 'stripping' properties of this oil he took the precaution to pre-benzolise it first. The practice of Ferguson Bell is somewhat uncommon. The gas whilst in a warm state is washed with warm tar, and at the outlet of the wet purification plant it is further treated with a naphtha oil.

Recovery of Cyanogen on Gasworks.

Cyanogen is a subsidiary by-product which is only recovered on the largest of works. The maximum amount which can be obtained from a ton of coal is about 1½ lbs. of hydrocyanic acids or about 10,000 grains. The processes now employed recover the hydrocyanic acid either as ammonium sulphocyanide or as sodium ferrocyanide (prussiate of soda). The most readily applied method is that operating on the polysulphide principle. In this system the gas, prior to washing and scrubbing, is passed through a special vessel (of the nature of a small purifier)

containing a layer of spent oxide of iron about 2 ft. in depth. Above the oxide material sprays are arranged around the box, and the oxide is periodically sprayed with water or weak cyanide liquor. The gas, containing hydrocyanic acid, passes upwards through the layer of oxide and comes in contact with the polysulphide solution trickling through the mass. The hydrocyanic acid combines with the ammonium polysulphide and yields ammonium sulphocyanide according to the following equation :

$$(NH_4)_2S_2+HCN=NH_4SCN+NH_4SH.$$

The spent oxide used in the process should contain about 50 per cent. of sulphur, and it is a surprising fact that although sulphur is required for the completion of the reaction the spent oxide, when removed from the vessel after working for about three months shows little or no loss in sulphur, while occasionally some slight increase may be found. Accordingly, the process would be of a continuous nature were it not for the fact that the physical nature of the oxide gives rise, after a time, to excessive back-pressure, when the only remedy is the substitution of a fresh layer. Figures obtained in practice show that one ton of spent oxide is capable of removing the cyanide evolved from 350 tons of coal.

The Recovery of Crude Benzol.

While the 'stripping' of gas for the recovery of the aromatic series of hydrocarbons has been carried out in the case of coal gas produced on coke-oven works for some years, the practice has only recently been introduced in gasworks, primarily as a means of meeting the demand for certain essential war products. The coal gas, either in the semi-crude state (*i.e.* after the wet purification plant) or when completely purified, is subjected to washing with a solvent which absorbs the benzene hydrocarbons but which has no effect on the remainder of the constituents. The solvent employed consists usually of a coal-tar distillate known as 'green oil,' although other substances, such as creosote, blast-furnace oil, or crude petroleum, may also be employed. The oil, after having been in contact with the gas, is said to be benzolised, when it may contain up to 5 per cent. of benzene hydrocarbons, according to the extent to which 'stripping' is carried. The benzolised oil is then passed through a series of vessels comprising the distilling plant in which the benzene hydrocarbons, with some light naphthas and a small proportion of naphthalene, are driven off and separated from the wash-oil. The benzene vapours thus obtained are condensed and collected as a crude spirit, while the wash-oil, having been 'debenzolised,' is cooled and once more employed for treating the gas. In general, it is customary to recover from normal coal gas from 1¼ to 2½ gallons per ton of coal carbonised, depending upon the amount of aromatic hydrocarbons contained in the original gas and the proportion of these which it is deemed advisable to extract. The crude spirit will vary in composition, but in general about 65 to 70 per cent. will distil over up to 120° C., and 80 per cent. up to 160° C. The portion which distils over will consist of about 80 per cent. of benzol and 15 per cent. of toluol, the remainder being xylol. The amount of wash-oil employed for extraction per ton of coal carbonised undergoes considerable variation in accordance with the taste of the responsible engineer. It may be said, however, that it should never be less than 60 gallons per ton of coal, while it may be so much as 100 gallons. Water in the wash-oil is strongly to be guarded against, and should never exceed 1 per cent. by volume if effective results are to be obtained. Temperatures play an important rôle in the process, and it may be said that the scrubber should never exceed about 70° F., and as regards the temperature of the debenzolised oil a good rule to observe is that of maintaining the oil about 5° F. above the temperature of the gas. Now that the demand for explosives has ceased many gas undertakings have discontinued the recovery of benzol.

There can be no question, however, that experience with benzol stripping has shown that a valuable effect results so far as naphthalene is concerned. On quite a number of works it has, therefore, become a common practice to wash the gas with oil, the primary object being, not the recovery of benzol, but the reduction of naphthalene. Green oil of the usual kind may be utilised for the purpose, but gas oil has in many cases been found to be superior.

Tar Distillation.

(Contributed by Harold Moore, M.Sc., A.I.C., F.C.S.)

The yield of coal tar depends very largely upon the volatile content of the coal : but, in addition, it is considerably affected by the design of the retort. Retorts in which heating is conducted gradually, and which ensure the rapid removal of the volatile products from the heat zone, yield the largest quantities of tar. The nature of the tar depends, to a slight extent, upon the type of coal used, but is much more seriously affected by the carbonising conditions. A rapid rise in temperature, with a high maximum temperature, tends to produce tar high in specific gravity, high in viscosity, rich in free carbon, and mainly consisting of aromatic hydrocarbons. Gradual heating, low maximum temperature, and rapid removal of the volatile products result in brown mobile tars, comparatively devoid of free carbon and rich in aliphatic hydrocarbons.

Horizontal retorts subject the coal to a very rapid heating with a high maximum temperature, whilst the volatile products have to pass along the heated retort walls, whereby they undergo cracking. The result is a viscous tar, rich in free carbon. Horizontal retort tars frequently

contain over 20 per cent. They possess a specific gravity of about 1·2. In continuous vertical retorts the coal is slowly passed from the cool zone into a hotter zone, undergoing gradual heating, whilst the volatiles are removed from the cooler end of the retort, and, therefore, are not so liable to undergo cracking. The resulting tar possesses a specific gravity of about 1·08, and about 1 to 8 per cent. of free carbon. Inclined retorts give products of intermediate composition; whilst coke ovens yield tars of very varied composition, depending upon the design of the ovens.

Special processes for the low-temperature carbonisation of coal, such as the Coalite and Premier Tarless systems, yield brown low-gravity tars in which free carbon is almost entirely absent. Gas works and coke ovens supply nearly all the tar produced in this country. Tar produced in gas works is accompanied by ammoniacal liquor, and requires settling out previously to distillation. The tar in distillation is usually divided into the following fractions in English distilleries:—

	Temperature.	Approx. Yield.
1. Crude naphtha	Up to 110° C.	3 per cent.
2. Light oil	110° to 200° C.	7 ,,
3. Crude carbolic oil . . .	200° to 240° C.	7 ,,
4. Creosote oil	240° to 270° C.	15 ,,
5. Anthracene oil	270° to finish	10 ,,
6. Residuum of pitch . . .	—	58 ,,

The finishing-point of the distillation is decided by the quality of pitch desired. The yields of the above products vary, of course, with the nature of the tars, horizontal retort tars yielding a residue of about 65 per cent. of pitch, whilst vertical retort tars give only about 50 per cent. of pitch. The benzene and toluene are mostly present in the crude naphtha fraction, the yield of benzene being about 2 gallons to the ton of tar. They are separated by distillation combined with washing with dilute alkali, which removes phenols, cresols, and other acidic bodies, followed by washing with dilute acid, which removes pyridine, quinoline, and other basic substances, and are finally refined by treatment with strong sulphuric acid, which removes certain sulphur-containing compounds and a portion of the unsaturated hydrocarbons originally present in the tar. Similar processes of refining are adopted for other fractions, though the strong acid treatment is usually applied only to the benzene, toluene, and solvent naphtha-containing fractions. The most common grades of these three distillates are sold under the names of 90*s.* benzol, 90*s.* toluol, and 90*s.* xylol. The 90*s.* benzol distils 90 per cent. of its volume at 100° C. by the tar distillers' retort test; 90*s.* toluol yields 90 per cent. at 120° C. by the same test; whilst 90*s.* solvent yields 90 per cent. at 160° C. The solvent fraction consists mainly of the three isomeric xylols. Naphthalene and anthracene are removed from the higher boiling fractions (naphthalene mainly from the carbolic oil fraction, anthracene from anthracene oil) by cooling and decantation, or by cooling and filtration.

The limiting temperatures of the fractions are liable to alteration, depending upon market conditions, whilst the distillation is frequently stopped at an earlier stage than that indicated in the above table, in order to yield a soft residue known as treated or refined tar, which is used for tar spraying on the roads. A soft finished residue is also frequently sold as pitch-creosote mixture, and used for the fuel market.

Benzol is mainly used for motor fuel, together with a considerable quantity of the toluol, though smaller amounts are employed in the synthetic chemical industry. Carbolic acid (phenol) and cresylic acid (cresols), obtained by alkali washing, both find their main market for the manufacture of disinfectants. Creosote oil is used for timber preservation and as fuel. The use of creosote as fuel is somewhat intermittent, depending upon the price of petroleum. Pitch is mainly used for road repairing, but also for many other purposes, a considerable amount being consumed for electric insulation.

WATER GAS.

Manufacture of Water Gas.

Water gas, which is employed as an auxiliary to coal gas in gasworks, is now usually of the 'blue' type, although in some instances oil enrichment has again been resorted to owing to the very considerable drop in the price of oil. The composition of these two gases has already been given on pp. 2189, 2458. The reactions occurring in the generator of the water gas plant are as follows:

$$(a)\ C + 2H_2O = CO_2 + 2H_2.$$
$$(b)\ C + H_2O = CO + H_2.$$

The carbon dioxide produced in the lower portion of the fuel bed according to equation (*a*) is then reduced to carbon monoxide on passing through the remainder of the fuel bed :

$$(c)\ CO_2 + C = 2CO.$$

The extent to which the reaction (*b*) takes place depends largely upon the temperature prevailing in the lower part of the fuel bed.

As the above reactions are of an endothermic nature the heat of the fuel bed is gradually absorbed, accordingly this is replaced at intervals by the process of blowing. The reaction occurring during the 'blow' is of an exothermic nature, being simply as follows :

$$C + O_2 = CO_2.$$

All practical water gas plants are now operated on the intermittent principle of providing for short periods of a gasmaking 'run' and a 'blow.' In the more common carburetting plants the 'run' continues for 5 or 6 minutes, and is followed by a 'blow' extending for 3 or 4 minutes. In certain forms of plant intended entirely for the production of 'blue' gas the blowing period is generally of much shorter duration, lasting for about 70 to 90 seconds. The primary object of the 'blow' is to convert the coke direct into CO_2, thereby liberating the maximum quantity of heat, but this condition is never attained in practice, some CO being always present. This CO, in the enriching plants, is made use of for heating up the subsequent apparatus in which enriching takes place. In some of the later forms of 'blue' plants (in which no enriching vessels are required) the CO is employed for the purpose of raising the steam required during the 'run.' The average composition of the 'blow' gases issuing from the top of the generator is as follows :

Carbon monoxide	17 per cent. by volume.
Carbon dioxide	10 „ „ „
Nitrogen	73 „ „ „

Traces of hydrogen, methane and sulphuretted hydrogen may also be found. The CO is at a maximum, and the CO_2 at a minimum, during the first minute of the 'blow,' and as the period proceeds the CO undergoes reduction while the CO_2 increases in quantity.

As regards temperatures the generator should, at the commencement of the 'run,' be worked up to 2,000F.° By the time that the 'run' is completed the temperature will have dropped by from 300° to 500°. Owing to this fact the quality of the water gas undergoes gradual deterioration from the commencement of the 'run' onwards ; thus at the end of the first minute of the 'run' the CO_2 content will only amount to some 2 per cent., whilst at the end of the fifth minute it may amount to from 7 to 8 per cent. The temperature of the carburettor and superheater varies in accordance with the type and quantity of oil in use at the time. Under normal conditions the following will apply :

Carburettor	1350 to 1400 F.°
Superheater	1400 to 1450 F.°

For the manufacture of carburetted water gas the materials required are as follows :

Coke . .	35 to 41 lbs. per 1,000 cub. ft. of gas ;
Oil . . .	about 2½ gallons per 1,000 cub. ft. of 18 candle gas ;
Steam . .	27 to 30 lbs. per 1,000 cub. ft. of gas.

The cost of carburetted water gas is mainly influenced by the current value of the enriching oil utilised. To-day, blue water gas is a far more common and important product, so that many plants formerly utilised for producing carburetted gas are now employed for the production of blue gas only. The present-day cost of manufacturing blue gas is as follows :—

BLUE WATER GAS : COST OF MANUFACTURE (1925).

	Per 1,000 cubic feet. *d.*	Per therm. *d.*
Coke, generator	6·0	2·0
Coke, for steam	3·0	1·0
Water, for steam	0·15	0·05
Wages	3·5	1·16
Maintenance	2·5	0·84
Purification	0·5	0·16
Establishment charges . . .	0·75	0·25
Total	16·40	5·46

The impurities in water gas consist of carbon disulphide and sulphuretted hydrogen, which are present in the following quantities:

	Grains per 1,000 cub. ft.
Sulphuretted hydrogen	110 to 120
Carbon disulphide	10 to 15

As regards water gas tar, S. Carter has given the following analysis:

Specific gravity	1·0665
	Per cent.
Benzol	1·11
Naphtha	2·33
Creosote oil	21·35
Naphthalene	4·09
Anthracene oil	15·91
Pitch	56·70

The Complete Gasification of Coal.

Now that the choice of the quality of gas to be distributed rests with the gas engineer instead of being stipulated by Parliament, a certain amount of interest has been displayed in the process known as the complete gasification of coal. By this is meant the treatment of the coal under such conditions that no solid residue other than ash remains. Complete gasification may, of course, be carried out in two ways, namely, by treating the coal in two stages for the production, firstly, of straight coal gas, and secondly, of blue water gas in distinct plants, the two gases being subsequently mixed either in the crude or purified state. The alternative is to employ plant of special design in which the above two processes are carried out simultaneously.

From the standpoint of gas production, the two methods may be contrasted as follows:—

A. Dual Process.

	B.Th.U.
1 ton of coal yields—	
13,000 cubic feet of straight coal gas @ 520 B.Th.U and 11 cwts. surplus coke.	= 6,760,000
11 cwts. coke yields—	
20,500 cubic feet of blue gas @ 300 B.Th.U.	= 6,150,000
Total yield in gaseous form from 1 ton of coal.	= 12,910,000

B. Single Process.

	B.Th.U.
1 ton of coal yields—	
50,000 cubic feet of mixed gas @ 370 B.Th.U.	= 18,500,000

The composition of the gas obtained from the modern single-stage complete gasification plant is as follows:

	Per Cent.
Hydrogen	51·0
Carbon monoxide	30·0
Methane	8·0
Unsaturated hydrocarbons	1·4
Carbon dioxide	4·6
Nitrogen	5·0
Oxygen	trace
Yield per ton of coal (cubic feet)	50,000
Calculated calorific power (gross)	370
Therms per ton of coal	185

SULPHATE OF AMMONIA.

Average commercial salt contains:

Ammonia	from 24½ to 25½ per cent.
Equal to	
Nitrogen	20·2 to 20·8 per cent.
Free acid	0·4 per cent.
Moisture	2 to 3 per cent.

Materials required for the manufacture of 1 ton of sulphate of ammonia:

Liquor, approximately	3,400 gallons of 8 oz. strength.
Acid ,,	18 cwts. at 80 per cent. strength.
Lime ,,	3 cwts.

The sulphate of ammonia recovered per ton of coal varies in this country between 24 and 28 lbs.

Equation for formation :

$$2NH_3 + H_2SO_4 = (NH_4)_2SO_4.$$

Gallons of liquor × ounce strength × 0·0823 = lbs. of sulphate.

The total quantity of ammonia products (mainly sulphate of ammonia) produced in England, Scotland, and Wales in 1923, was 425,590 tons.

CONCENTRATED LIQUOR.

The exigencies of the war were responsible for a new gasworks product in the form of concentrated and purified ammoniacal liquor. This product contains about 25 per cent. of ammonia, while the sulphuretted hydrogen is reduced to 0·5 per cent. by volume. The concentrated liquor is prepared by passing the ordinary liquor from the storage well through a desulphurising column in which the carbon dioxide and sulphuretted hydrogen are removed, the purified liquor being afterwards concentrated in a special still. With the war ended the majority of these plants have been shut down. During 1923 the quantity of concentrated liquor, made in Great Britain was equivalent to 39,071 tons of sulphate of ammonia.

USEFUL MEMORANDA.

1 in. of mercury = 0·490 lbs. per sq. in. ; 1 in. of water = 0·0361 lbs. per sq. in.; 1 lb. per sq. in. = 27·7 ins. of water.

A British Thermal Unit is the amount of heat required to raise 1 lb. of pure water 1° Fahr. correctly, from 39·1° Fahr. to 40·1° Fahr.

A Calorie is the amount of heat required to raise 1 kilogramme of water 1° Cent. ; correctly from 0° to 1° Cent. This is the *large* calorie, which is the unit employed on gasworks.

1 British Thermal Unit	= 0·252 calories.
1 Calorie	= 3·968 B.Th.U.
1 Therm	= 100,000 B.Th.U.
Therms	= cu. ft. × calorific value per cu. ft. ÷ 100,000.
Cu. ft.	= therms × 100,000 ÷ calorific value per cu. ft.

CALORIFIC VALUE OF VARIOUS GASES.

South Met. Co. . . .	560 B.Th.U. per cu. ft.
Gas Light & Coke Co. .	500 „ „
Commercial Gas Co. . .	500 „ „

200 cu. ft. of Gas Light and Coke Co.'s gas = 1 therm.

CALORIFIC POWER OF VARIOUS SUBSTANCES.

	B.Th.U. per lb.		B.Th.U. per lb.
Coke-oven coke	14,436	Petrol, specific gravity 0·684 . .	20,923
Peat, 5 per cent. water . . .	9,900	Shale oil	18,217
Wood, pine	9,153	Blast-furnace oil	16,080
„ oak	8,316	Heavy tar oil	16,050
Petroleum, American . . .	19,627	Coal gas tar	15,500
„ Texas	19,242	Alcohol, absolute	12,931
„ Russian	19,440		(*Lewes.*)

At normal temperature and pressure 32 cu. ft. of coal gas (14 candle power) weigh approximately 1 lb.

1 cu. ft. of coal gas (14 candle power) at N.T.P. weighs ·03125 lb.
1 „ blue water gas „ „ ·0376 lb.

GASWORKS TERMS.

A barrel of tar	25 gallons
A butt of liquor	108 „
(Usually 8 oz. strength)	
A bushel of coal	80 lbs.
A bushel of coke	45 lbs.
A chaldron of coal	25½ cwts.
(Varies according to locality)	
A chaldron of coke	12½ cwts.
(Varies according to locality)	
A cord of wood	128 cub. ft.
A fodder of lead . .	19½ cwts.
A hundred of deals . .	120 in number
A load of lime	32 bushels
(A bushel = 70 lbs.)	
A long thousand of slates .	1,200 in number
A seam of glass . . .	120 lbs.
A short ton (American) .	2,000 lbs.
A tonne (French) . .	2,204·6 lbs.
A ton of freight . . .	40 cub. ft.
A truckload of bricks	About 3,300 in number

VOLUMES OF OXYGEN AND AIR REQUIRED FOR COMBUSTION OF ONE VOLUME OF COMBUSTIBLE GAS.

Combustible Gas.	Oxygen.	Air containing 21 per cent. Oxygen.	Volume of Products of Combustion.
Hydrogen . .	0·5	2·38	1 vol. H_2O
Carbon Monoxide	0·5	2·38	1 vol. CO_2
Marsh Gas . .	2·0	9·52	1 vol. CO_2, 2 vols. H_2O
Ethylene . .	3·0	14·28	2 vols. CO_2, 2 vols. H_2O
		Incomplete Combustion.	
Marsh Gas . .	1·5	7·14	1 vol. CO, 2 vols. H_2O
Ethylene . .	2·0	9·25	2 vols. CO, 2 vols. H_2O

High Pressure Lighting.

For high-pressure lighting the cost of compressing the gas only amounts to from 2½ per cent. to 4 per cent. of the total cost of the gas consumed, which is very small when it is compared with the extra light obtained.

With a water-driven compressor, from 250 to 300 gals. water are required per 1,000 cubic feet gas compressed to, say, 9 ins. pressure; with a gas-driven plant, 12·4 to 13·4 cubic feet per 1,000 cubic feet compressed to same pressure.

Compressing coal gas reduces the illuminating value until, with a pressure of 200 lbs. per sq. in., the loss is 16 per cent.

(See also 'LIGHTING,' Section XXXVIII., Part III., p. 2525.)

AUTOMATIC CONTROL OF INDUSTRIAL GAS BURNERS.

When gas is used for industrial heating processes, in many cases automatic control is desirable; this can be effected efficiently by a 'Thermo-gas Regulator' such as is made by The Drayton Regulator and Instrument Co., Ltd., of West Drayton.

Incandescent Burners.

The Bunsen burners used for incandescent lighting should give a flame with a green inner cone and not a violet one, so as to show that the proportion of air used for primary combustion is as large as possible. This ensures an intense heat in the flame, which intensity adds greatly to the light emitted by the burner. This intensity of flame is obtained in the high pressure systems, which are now becoming so general for street and shop lighting, by increasing the pressure of this gas to about 2 lbs. per sq. in., and when this is used with an inverted mantle of a much longer shape than the ordinary low pressure ones, the rush of gas at this high pressure at the nipple causes the full quantity of air required for combustion to be drawn in at the base of the burner mixing tube and so give a flame of the highest intensity. This flame only gives off the same heat per cubic ft. of gas as an ordinary Bunsen burner, but the heat is concentrated and the flame is much smaller. If the quantity of air induced by the current of gas into the mixing tube is 2½ times that of the gas there will be a lilac inner cone, and the hottest part of the flame will be about 2,800°F., and the mantle will then show about 20 candles per cubic ft. of gas consumed. If 3½ times of air are induced, a green inner cone will be noticed, and then the flame will be about 3,000°F., and 30 candle power per cubic ft. of gas obtained.

Products of combustion must be conveyed away and not allowed to enter at the primary air hole of the Bunsen burner. The air used there should contain the full proportion of oxygen to ensure proper combustion. Mantles must fit the flame, or, inversely, the flame must fit the mantle, as then it will be in the hottest part of the flame which is at the outside where the C in the gas is combining with the O_2 of the air to form CO_2 with the evolution of the heat. As the flame is hollow, if the mantle is smaller than the flame it will be in the cooler region of the flame where the gas is not being fully burned. Upright burners give their greatest light horizontally, inverted burners downwards. Shades for gas lights should be at 45° with the horizontal, not 60° or 70° as usually adopted, when very little light is reflected.

The best effects are obtained with burners which permit of the adjustment of both air and gas. These burners should be adjusted from time to time. The burner should be thoroughly cleaned out each time a mantle is fixed.

Incandescent Mantles.

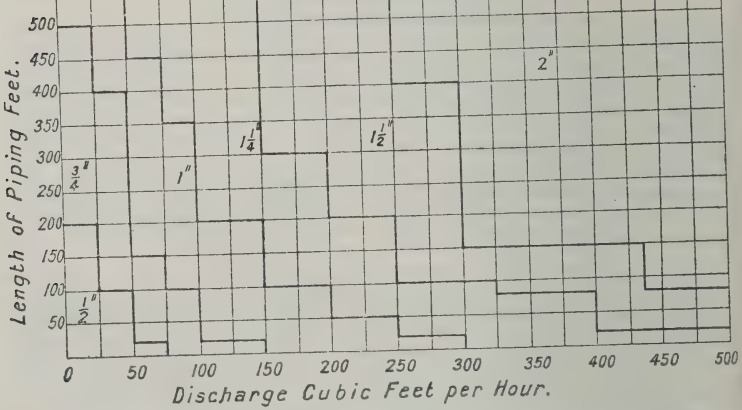
Mantles are made from cotton, ramie, or artificial silk knitted into a 'stocking' or hose about 2 ins. in diameter. This stocking is thoroughly washed and dried. Then it is saturated with a solution of nitrate of thorium and 1 per cent. of nitrate of cerium. After the stocking has been washed and dried, it is either cut to proper size and impregnated with the thorium solution, or is first impregnated and then cut to the required lengths. The impregnated stockings, after being cut and dried, are sewed at one end with asbestos thread so as to form a head provided with a loop, which serves to hold the mantle in process of manufacturing and when in use on the burner. Then the impregnated stocking is hung on a wire by the asbestos loop, and heated in a gas flame of the Bunsen type. This is done in order to burn out the thread. The next process is called the shaping. The mantle now consists only of ashes of thorium (oxide of thorium), and is carefully held over a high pressure Bunsen flame, and gradually given the right shape, at the same time being hardened. As soon as the mantle is shaped and hardened it is practically completed, but in order to protect it from breakage it is 'dipped' in a stiffening solution. There are many forms of 'dip,' but the one most used is made of soluble cotton (gun cotton) dissolved in good alcohol and acetone. To this mixture castor oil and shellac are added.

Gas versus Electricity.

The inverted incandescent gas lamp requires 47 B.Th.U. per candle-power hour obtained. The one-watt electric lamp requires 5·4 B.Th.U., and the ½-watt lamp 3·1 B.Th.U. A considerable proportion of gas burners consist of the upright type, which are less efficient, requiring 82 B.Th.U. per candle-power hour. The average may be taken at 64 B.Th.U. per candle-power hour. High-pressure incandescent burners reduced the heat consumption to 23 B.Th.U. per candle-power hour and flame arcs in electric lighting use only 1·5 B.Th.U. At about 75 per cent. of full load a gas engine of the ordinary type would deliver to the consumer as actual or brake horse-power 11 per cent. of the heat used at the gasworks, while an electric motor would deliver, also as brake horse-power, only 6·8 per cent. of the heat used at the electricity generating station. The efficiency of the gas power production and distribution was thus over 1·6 times that of the electricity system. (*Sir Dugald Clerk.*)

Gas Piping.

In the following diagram the point of intersection of consumption in cub. ft. per hour and length of pipe falls into an area indicating size of pipe for the work.



The diagram shows the size of pipe for the conveyance of a given quantity of gas. It is based on a fall of pressure between the extreme ends of the pipe of $\frac{3}{10}$-inch water gauge, and for

ordinary low-pressure gas supply it would not be desirable to have a drop greater than this. The diagram does not provide for piping smaller than ¼ in. because generally sizes smaller than this are only used for the branches to pendants or other apparatus consuming less than 25 cub. ft. per hour. Up to 10 cub. ft. per hour ¼ in. pipe is sufficient and for 10 to 25 cub. ft. ⅜ in.

WEIGHT OF WROUGHT-IRON GAS PIPES.

Bore in inches .	¼	⅜	½	¾	1	1¼	1½	1¾	2	2¼
	2½	2¾	3	3½	4	4½	5	5½	6	
Outside diameter .	17/32	11/16	27/32	1 3/16	1 11/16	1 11/16	2 9/32	2 5/8	2⅞	2⅞
	3	3¼	3½	4	4⅛	5	5½	6	6½	
Threads per inch .	19	19	14	14	11	11	11	11	11	11
	11	11	11	11	11	11	11	11	11	
Weight in lbs. per ft.	0·38	0·58	0·81	1·16	1·60	2·36	3·00	3·60	4·10	4·60
	5·33	5·84	6·33	7·16	8·05	9·08	10·17	11·49	12·57	

WEIGHT OF CAST-IRON GAS PIPES.

Bore, in inches.	Thickness of Metal.							
	⅜	½	⅝	¾	⅞	1	1⅛	1¼
	Lbs. per Lineal Foot.							
2	8·7	12·3	16·1	—	—	—	—	—
3	12·4	17·1	22·2	—	—	—	—	—
4	16·1	22·1	28·3	—	—	—	—	—
5	19·8	26·9	34·4	42·3	—	—	—	—
6	23·4	31·9	40·6	49·7	—	—	—	—
7	27·1	36·8	46·7	56·8	—	—	—	—
8	30·8	41·6	52·8	64·3	—	—	—	—
9	34·4	46·0	58·9	71·7	—	—	—	—
10	—	51·4	65·1	79·0	93·3	—	—	—
11	—	56·4	71·0	86·4	101·8	—	—	—
12	—	—	77·3	93·7	110·4	127·4	—	—
14	—	—	89·6	108·4	127·5	147·0	—	—
15	—	—	—	115·7	136·1	156·8	—	—
16	—	—	—	123·1	144·7	166·6	—	—
18	—	—	—	139·9	161·8	186·2	—	—
20	—	—	—	—	178·9	205·8	260·3	—
22	—	—	—	—	—	225·4	284·8	—
24	—	—	—	—	—	245·0	309·3	—

The weight of two flanges or one socket may be reckoned weight of 1 foot.

AVERAGE WEIGHTS OF GAS PIPE CONNECTIONS.

Internal Diameter.	Tees.			Collars.			Syphons.			Caps.		
	Cwts.	Qrs.	Lbs.	Cwts.	Qrs.	Lbs.	Cwts.	Qrs.	Lbs.	Cwts.	Qrs.	Lbs.
2	0	1	17	0	0	12	2	0	14	0	0	9
3	0	2	11	0	0	25	2	0	25	0	0	16
4	0	3	9	0	1	5	2	1	4	0	0	21
5	1	1	0	0	1	22	4	0	14	0	1	2
6	1	2	0	0	2	0	4	1	7	0	1	13
7	1	3	21	0	2	20	4	1	25	0	1	21
8	2	1	21	0	3	7	4	2	7	0	2	3
9	2	3	14	1	0	11	4	2	14	0	2	24
10	3	2	11	1	0	14	4	3	25	0	3	5
12	4	2	7	1	2	7	6	1	0	1	0	14
14	6	3	7	2	0	0	7	0	7	1	1	25
15	7	0	18	2	1	0	7	0	7	1	3	7
16	8	1	7	2	2	14	7	2	25	1	3	14
18	9	1	21	3	0	14	11	1	0	2	1	11
20	10	1	14	3	1	4	12	2	14	2	1	25
24	16	3	0	5	0	0	13	0	0	3	1	7

Flow of Gas in Mains.

Pole's Formula.

Let Q = the discharge of gas in cub. ft. per hour.
d = the diameter of pipe in inches.
p = pressure of gas in inches of water,
s = specific gravity of gas, air being 1.
l = length of pipe in yards.

then,

$$Q = 1350d^2\sqrt{\frac{p\,d}{s\,l}};\quad d = \sqrt[5]{\frac{Q^2 s\,l}{(1350)^2 P}};\quad p = \frac{Q^2 s\,l}{(1350)^2 d^5};\quad l = \frac{(1350)^2 d^5 p}{Q^2 s}.$$

Unwin's Formula for High Pressure Gas Mains.

Let D = diameter of pipe in ft.
S = specific gravity of gas (air being 1).
L = Length of pipe in ft.
P_1 = initial pressure in lbs. per sq. inch absolute.
P_2 = final pressure in lbs. per sq. inch absolute.
C = coefficient of friction $= 0{\cdot}0044\left(1+\frac{1}{7D}\right)$.
U_1 = initial velocity of gas in feet per second.
U_2 = final velocity of gas in feet per second.
W = area of pipe in sq. ft.
Q = discharge in cub. ft. per hour,

then,

$$Q = 1{,}323{,}250\sqrt{\frac{P_1^2 - P_2^2}{P_1^2}\cdot\frac{P_1^2}{P_2^2}\cdot\frac{D^5}{CSL}};\quad U_1 = 468\sqrt{\frac{D}{CSL}\cdot\frac{P_1^2 - P_2^2}{P_1^2}};\quad U_2 = \frac{P_1}{P_2}U_1.$$

Cement Joints for Gas Mains.

An excellent method of making cement joints for gas mains is as follows :—

The inside of the bell and spigot end of the pipes should be thoroughly cleaned with a wire brush before being placed in the trench, and the open joint kept covered with sacking to keep out earth and dirt.

In making the cement joint, two caulkers work on the joint, one on each side. Air is first blown into the open joint at 50-lb. pressure through an air hose to clean out all particles that may have accumulated there.

Eight strands of untarred yarn are next saturated in neat cement-grout and tamped home with a hand tool. About 1 in. of common paste is then tamped in with a hand tool and the joint allowed to set at least 12 hours. This backing of cement keeps the moisture from being forced into the pipe and drying out the mixture by caulking too much.

The joint is then filled with the cement mixture (two cement to one sand), which is carefully and thoroughly mixed to a stiff paste.

The packing yarn, consisting of 25 strands of untarred yarn, is coiled and tamped into the joint with pneumatic caulking hammers working at 46–60-lb. pressure, commencing at the bottom and working around to the top.

The packing yarn is then pulled out and the joint again filled with cement paste, and another coil of 25 strands of yarn is caulked in and allowed to remain.

The joint is finished off smooth with a collar of cement.

Careful supervision must be exercised on the caulkers, and careless habits corrected at once.

Each caulker should mark his own joint with a distinguishing letter, so as to facilitate correcting carelessness in any of the men.

After the joint has been made, the sacking covering the joint should be watered daily until backfilled. All joints should be tested with soapy water at 75 lb. per sq.i n. air pressure. A joint that leaks at the first test should be cut back 4 in. and remade.

A 4-in. cement joint should cost less than 1*l.*, including all labour and material.

TABLE OF CONSTANTS FOR FRICTION IN GAS MAINS.

Diameter of Main in Ins.	Diameter of Main in Ft. = D.	Area of Main in Sq. Ft. = W.	Value of C = Coefficient of Friction $= 0{\cdot}0044\left(1+\frac{1}{7D}\right)$
2	0·166	0·022	0·0082
3	0·250	0·049	0·007
4	0·333	0·087	0·0063
5	0·417	0·136	0·006
6	0·500	0·196	0·0057
7	0·583	0·267	0·00554
8	0·666	0·349	0·00537
9	0·750	0·442	0·00527
10	0·833	0·545	0·00515
12	1·000	0·785	0·005
14	1·166	1·069	0·00492
15	1·250	1·227	0·00488
16	1·333	1·396	0·00485
18	1·500	1·767	0·0048
20	1·666	2·182	0·00475
21	1·750	2·405	0·00473
24	2·000	3·142	0·0047

(*Professor Unwin.*)

EVERITE GAS PIPING.

(*Asbestos Cement Building Products, Ltd., Trafford Park Manchester.*)

Everite piping is specially manufactured for the gas industry, and all the usual accessories can be obtained, including flue pipes, bends, baffiers, cone caps, etc., etc.

Gas Pressures.

Generally, gas pressures (in a factory, for example) are sufficiently constant for the correct working of gas consuming apparatus, but occasionally owing to piping being of insufficient size a variation in the consumption will cause a variation of pressure. The best remedy is to relay or supplement the piping in the building, but, as a temporary measure, governors can be installed on any apparatus which needs a fairly constant supply of gas. The Stott, Peebles, Tilley, Sugg, and Williams governors are typical of this appliance. They maintain a constant pressure on their outlet regardless of wide variation at the inlet.

Gas Meters.

Size of Meter.	Normal Capacity.	Safe Maximum Capacity.
Lights.	Cub. ft. per hour.	Cub. ft. per hour.
5	30	60
10	60	100
20	120	200
30	180	300
50	300	450
60	360	480
80	480	600
100	600	750
150	900	1,100
200	1,200	1,500
250	1,500	1,900
300	1,800	2,300
400	2,400	3,000
500	3,000	3,700
600	3,600	4,500
800	4,400	6,000
1,000	5,000	7,500

It is desirable that the normal capacity should not be exceeded for continuous working, as there will be some risk of the meter breaking down and the supply of gas temporarily stopped. If additional gas-consuming apparatus is installed a larger meter should, if necessary, be used.

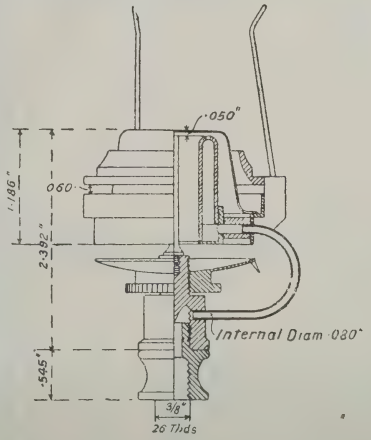


FIG. 1.

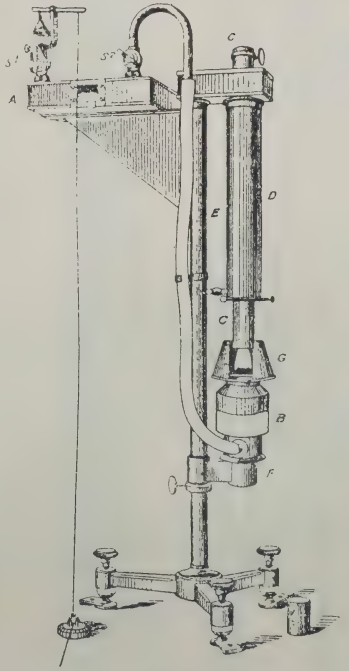


FIG. 2.

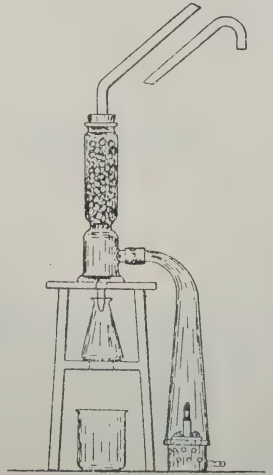


FIG. 3.

Gas Tests.

Gas Tests as Prescribed by the Gas Referees.

As gas is now sold on the 'Therm' basis, the determination of calorific power is the most important of the official tests. Illuminating power, however, is frequently required as a matter of interest rather than importance, and for this reason the test is given below.

(1) *Illuminating Power.*

Although calorific power is taking the place of illuminating power as a standard of quality, the latter test is still carried out in many instances.

This test is made by a photometric comparison between (*a*) a standard Argand burner called the Metropolitan Argand Burner No. 2, of which a part section drawing is given in fig. 1; (*b*) a standard flat-flame wing burner $\left(\text{Bray's } \frac{G}{L} \text{ 5 ft. } \frac{15}{10}\right)$ having a slit 0·028 in. wide, and a 10-candle power standard lamp burning a mixture of pentane and air. The standard lamp is shown in fig. 2. A is a receptacle subdivided by seven alternate partitions which do not extend the full width of the chamber; so that air, drawn in through the cock S', by means of which the supply is regulated, is caused to traverse the surface of the liquid, with which the chamber is filled to about two-thirds of its height, eight times. The mixture of gas and air passes through the cock S^2 to a burner B, consisting of a steatite ring pierced with thirty holes between 1·5 and 1·25 mm. in diameter. The flame is viewed through a mica window in G. Careful adjustments are made to regulate the height of the flame and the concentricity of the chimney D.

The pentane for the lamp is prepared and tested in the following manner:—Light American petroleum, such as is used for making air-gas, is rectified by three successive distillations, at 55° C., 50°, and 45°. The distillate is shaken up from time to time, during two periods of not less than three hours each, with one-tenth its bulk of (1) strong sulphuric acid, (2) solution of caustic soda. After this it is again distilled, and that portion which comes over between the temperatures of 25° and 40° is collected. This consists chiefly of pentane, with small quantities of other homologues the presence of which does not affect the light of the lamp.

The density of the liquid pentane at 15° C. should lie between 0·6235 and 0·626 compared with water of maximum density.

Any admixture with pentane of hydrocarbons having a higher photogenic value, such as benzene or amylene, must be avoided. The presence of these may be detected by the following test:—Pour into a stoppered 4-oz. bottle of white glass 10 c.c. of dilute nitric acid, specific gravity 1·32; add 1 c.c. of a dilute solution of potassium permanganate (1 gr. in 200 c.c.). Pour into the bottle 50 c.c. of the sample of pentane, and shake strongly during five successive periods of 20 seconds. If there is an admixture of as much as $\frac{1}{2}$ per cent. of amylene or benzene, the pink colour will disappear.

The photometric comparison is made by producing equality in the illumination of a white paper screen by shifting the test burner nearer to or further from the screen; the result, after applying barometric and temperature corrections, is expressed to the nearest hundredth of a candle.

(2) *Test for Sulphuretted Hydrogen.*

The gas is passed through a glass vessel in which are suspended six strips of bibulous paper moistened with a solution of 6·5 grams acetate of lead crystals in 100 c.c. water. The gas having been turned on full for three minutes, one of the slips is compared with a slip similarly moistened but not exposed to the gas. The gas is taken as exhibiting a trace of H_2S if the exposed is unmistakably darker than the unexposed slip.

(3) *Tests for other Sulphur Compounds.*

Ten cubic feet of gas is burned in a Bunsen burner at the rate of ·5 cubic foot of gas per hour. The stem of the burner is surrounded with about 2 ounces of sesquicarbonate of ammonia, and the combined products of combustion and volatilisation are collected by means of a trumpet glass fitted over the burner and salt, an air-way being provided for the burner in the stand upon which it is placed (see fig. 3). The products are condensed as far as possible in the upper glass vessel, which, to aid the condensation, is filled with a number of glass balls about 15 mm. in diameter. The distillate, collected in a beaker placed beneath the bottom orifice of the condenser as shown, is poured, together with the washings of the trumpet and condenser, into a measure glass. A sample $\left(\frac{1}{n}\text{th part}\right)$ of the liquid is treated with hydrochloric acid in a covered beaker until there is an excess of acid, and then raised to boiling point; an excess of a barium

chloride solution is added, and the boiling continued for five minutes. After standing for a time, barium sulphate settles; the latter is filtered off and washed with hot distilled water, in order to remove all traces of ammonium and barium chlorides. (A sample of the washings is tested for this in a test tube with silver nitrate, the washing being continued until no cloudiness is produced by the nitrate.) The filter paper with the barium sulphate is heated in a weighed platinum

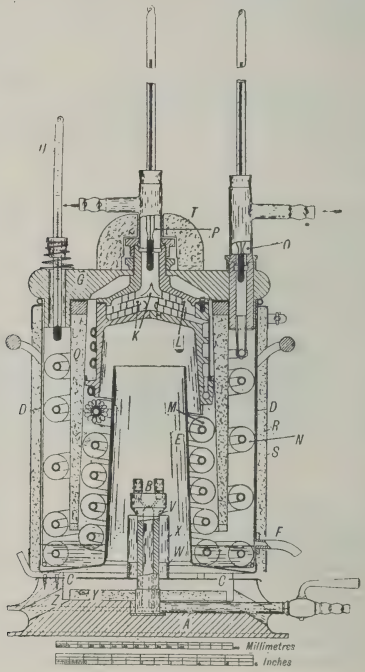


FIG. 4.

crucible until the paper is removed. When cool the crucible is placed in a desiccator over strong sulphuric acid and again weighed. The number of grains of barium sulphate thus obtained, multiplied by $\frac{11n}{8}$ (n being the quantity of liquid under test as above), gives the grains of sulphur per 100 cub. ft. of gas. This figure is corrected for barometric and temperature variations as in the case of the illuminating power test.

(4) *Tests for Calorific Power.*

These are carried out by observations taken by means of a calorimeter illustrated in fig. 4, in which water flowing is heated by a double burner consuming about $\frac{1}{10}$ cu. ft. per minute. The calorimeter having been set up, the burners are lighted and the water flow is so adjusted that the difference between the inlet and outlet temperatures is about 20° C. The heating of the calorimeter is allowed to proceed for not less than 45 mins. While the hand of the meter is making four revolutions, four readings of the inlet thermometer and fifteen readings of the outlet thermometer are taken at intervals, during which period the discharged water is collected in a measuring vessel. The temperature of the surrounding air, of the gas in the meter, and of the effluent gas are also observed. The mean of the four readings of the inlet temperature is subtracted from the mean of the fifteen readings of the outlet temperature, and the difference is multiplied by 11·976 and by the number of litres of water collected. The product is divided by the tabular

number.* The difference in degrees Fahrenheit of the temperature of the effluent gas and the surrounding air is multiplied by $\frac{2}{5}$, and this is added to the result previously found if the effluent gas is the warmer of the two, or subtracted if the effluent gas is cooler. The result is the gross calorific power of the gas in British thermal units per cu. ft. (*c*).

In addition to the foregoing, the amount of condensed water resulting from the combustion of the gas is measured. For this purpose the condensation water is led into a measuring vessel not less than 20 mins. after the calorimeter has been set up. The amount collected in not less than 30 mins. is measured, and the time of collection accurately noted. The number of cubic centimetres collected is multiplied by the number of seconds taken for four revolutions of the meter and by the constants 1·86 and 3·968 (*a*). The number of seconds during which the water is collected is multiplied by the tabular number (*b*). The first product (*a*) divided by the second (*b*) and subtracted from the gross calorific power (*c*) gives the net calorific power in British thermal units per cubic foot.

The meters used for these tests are of the 'wet' type, having a calibrated capacity of $\frac{1}{10}$ cu. ft. per revolution of the drum. The drum carries an index hand moving over a dial divided into 100 divisions, and the meters are provided with a thermometer for measuring the gas temperature.

ACETYLENE.

(Contributed by the late Professor Vivian B. Lewes, F.C.S., F.I.C.) Revised by Charles Bingham, C.E.

Acetylene, which is a gaseous compound of hydrogen and carbon, represented by the formula C_2H_2, is a colourless gas with a density of 0·92. As ordinarily prepared, it has a strong and very penetrating odour, but when purified from minute traces of other gases, invariably present with it in its crude state, has a not unpleasant ethereal odour. It can be converted into a liquid at a pressure of 21·53 atmospheres, at a temperature of 0° C., and liquefies under ordinary atmospheric pressure when cooled to a temperature of −82° C. The critical point of the gas is 37° C., and at this temperature the pressure has to be raised to 68 atmospheres for its liquefaction.

Acetylene is readily soluble in water, which at normal temperature and pressure takes up about its own volume of the gas; its solubility in other liquids is given in the following table:

100 Volumes of		Volumes of Acetylene.	100 Volumes of		Volumes of Acetylene.
Brine	absorb	5	Fusel oil	absorb	100
Water	,,	110	Benzene	,,	400
Alcohol	,,	600	Chloroform	,,	400
Paraffin	,,	150	Acetic acid	,,	600
Carbon disulphide	,,	100	Acetone	,,	2,500

Extended researches by Bistrow and Liebreich upon the toxic properties of acetylene show that it acts upon the blood in much the same way as carbon monoxide, but exhaustive experiments show that it is less poisonous than ordinary coal gas.

Acetylene forms compounds of an explosive character with copper and silver, and at first this was thought to be a great drawback to its general use; but experience has shown that these fears were unfounded, and that brass gas fittings, when gas-tight, can be used with perfect safety.

Acetylene, when heated, polymerises, like most hydrocarbons, i.e. it is partly or wholly split up, and the resulting gas is usually a mixture of acetylene itself and poorer gases, such as hydrogen, methane, etc., while tarry compounds and even solid carbon are formed, if the heating is carried far enough.

This is a matter of the utmost importance to acetylene users. For lighting purposes, it means that a poor light is given; for oxy-acetylene welders it means that the temperature at the blow-pipe is reduced considerably, indeed in some cases it has proved impossible to make satisfactory welds, owing to the poor quality of the gas.

These troubles arise almost exclusively with water-to-carbide generators, which are often faultily designed, so that overheating is bound to occur.

Experiments lately carried out by the French Acetylene Office have shown that polymerisation starts in an ordinary commercial generator at about 130° C., *i.e.* at a much lower temperature than was originally believed to be the case. Unfortunately many generators sold in this country act by driving the water away from the carbide container, when a certain amount of gas has been generated, leaving the live carbide surrounded by a mass of moist lime sludge, which is again surrounded by a blanket of acetylene. Acetylene, however, is a far poorer conductor of heat than water. At the instance of the writer, the heat conductivity of acetylene was ascertained by the National Physical Laboratory, we believe for the first time. It was found to be 0·000043 c.g.s., or only about one thirty-second of that of water. If, therefore, decomposing carbide is

* The tabular number depends upon the barometer reading and the temperature of the gas passing through the meter.

surrounded by acetylene instead of by water, only about 3 per cent. of the heat is carried off in a given time, which could be carried off by water, and the writer has seen sludge become actually visibly red under such conditions.

It is therefore of the utmost importance that acetylene users should be careful to see that they select acetylene generators in which there is always an ample supply of water in constant contact with the carbide, or constantly surrounding the carbide containers. It may also be pointed out that the better the carbide is, the more quickly it is attacked by water and, in consequence, the greater the quantity of heat liberated in a given time by its decomposition.

A new type of acetylene generator which differs radically in its generating chamber from anything previously used, although of extremely simple construction, has been put on the market in this country. The system has been in considerable use for about two years, and has given complete satisfaction.

In contact with air acetylene inflames at a temperature of 480° C. and burns with a flame which, when properly spread out so as to obtain a large quantity of oxygen, emits more light than any other gaseous compound of carbon and hydrogen.

Gas.	Candle hours per 5 cub. ft.	Carcel hours per cub. metre.	Gas.	Candle hours per 5 cub. ft.	Carcel hours per cub. metre.
Methane	5·2	3·5	Propane	56·2	40·0
Paris gas	13·2	9·6	Ethylene	70·0	49·0
London gas	16	11·5	Butylene	123·0	86·0
Ethane	35·7	25·0	Acetylene	240·0	168·0

Professor Clowes has shown that acetylene has a wider range of explosibility when mixed with air than any other combustible gas, the limits in percentages being as follows:

	Per cent.		Per cent.		Per cent.
Acetylene	3 to 82	Carbon monoxide	13 to 75	Methane	5 to 13
Hydrogen	5 to 72	Ethylene	4 to 22		

Acetylene is made commercially by acting upon a compound known as calcium carbide (CaC_2) with water, when acetylene gas is liberated and calcium hydrate (slaked lime, $Ca(HO)_2$) is formed.

Calcium carbide is made by fusing together very pure coke (or high-carbon anthracite coal) and very pure lime by the intense heat of the electric furnace. No other way of heating the mixture to the required temperature has yet been discovered, although many attempts have been made; for instance, by the use of the oxy-acetylene blowpipe.

Acetylene generators, in which the interaction of the calcium carbide and water to generate the gas takes place, may be either automatic or non-automatic. In the former class, devices are used to stop the generation of the gas automatically, by cutting off the supply of water or the supply of carbide, thus doing away with the need of a large store-holder for the gas, whilst in the non-automatic class the action is allowed to continue until all the carbide is decomposed, the gas being stored in a holder from which it is used as required.

In making acetylene, water may be brought in contact with the carbide either by adding it to the carbide, or by adding the carbide to water.

The overwhelming majority of acetylene generators in practically all countries are now constructed on the water-to-carbide system.

Of late years the automatic type of apparatus for the generation of acetylene has almost entirely superseded the non-automatic type, owing to the very large quantity of acetylene now being required for the oxy-acetylene welding process.

Experience has shown that automatic generators can be constructed with a high factor of efficiency, while the capital outlay per unit of gas generated is considerably less than in the case of non-automatic apparatus.

Owing also to the much smaller space occupied by the automatic type, it has also been found much more convenient for private use.

Until comparatively recent times, the pressure customary in acetylene generators was from 4 to 8 inches water-column, these limits having been found to give the most satisfactory results for lighting purposes. The advent of the oxy-acetylene welding blowpipe, however, soon caused experiments to be made with higher pressures, with the most favourable results. To-day the general tendency in most countries is to have the acetylene under slightly higher pressure than the oxygen in the welding blowpipe, it being claimed that less oxidation of the metal took place under these conditions and that there was less chance of the right proportion of the two gases being interfered with by accidental obstruction in the blowpipe, and, lastly, there was less tendency for the flame to flash back.

In consequence of this movement, the Governments of the United States, Canada, Switzerland, and Denmark now allow a maximum pressure of 1 atm. in acetylene generators for welding, and Germany is on the point of adopting this maximum pressure also. Very interesting details of tests have been published by the Government Bureau of Standards in the United States.

The oxy-acetylene blowpipe has been most successfully used for a considerable time for case-hardening various parts of machinery, and in particular the gear-wheels and other parts of motor cars. Lately a most important development has taken place in this direction, the

blowpipe having come into the most extensive use, not only for cutting and repairing rails, but for applying an exceedingly hard surface, to a depth reaching even the high figure of ¼ inch and more, to the entire rail system of electric tramways.

The improvement shown in the wearing properties of rails is shown by the following figures:

	Wear of rails in 12 months. Untreated.	Treated.	No. of cars carried.
Bristol	0·045	0·0157	300,000
Liverpool	0·030	0·0125	200,000

The rails are hardened *in situ*, without otherwise interfering with the track in any way.

When acetylene first made its appearance, its use was confined almost entirely to lighting detached buildings, motor-cars, and cycles.

The advent of electric sets materially reduced the field for acetylene for lighting purposes, but recently there has been a decided revival in this direction, owing to the enormously increased construction of houses of the bungalow type, large numbers of which are situated so far from a centre of supply of coal-gas or electric current that the cost of laying the necessary supply pipes or cables makes the capital outlay too great, in view of the modest requirements per bungalow.

In such cases the very low capital outlay required for an acetylene outfit, its extreme simplicity, and the beautiful light it gives, have made this form of lighting extremely popular.

The use of the oxy-acetylene blowpipe has continued to increase during the past year; indeed these and other applications have so greatly extended the use of acetylene that whereas the consumption of carbide before the war amounted to about 28,000 tons, it rose in 1925 to 40,000 tons per annum. There was a set-back, owing to the difficulties in the coal industry, but at the time of writing a consumption of approximately 40,000 tons a year has again been reached.

The use of acetylene in other directions than as a combustible again made great strides in the past year. One of the most interesting new derivatives is the 'Meta' block, which has been placed on the market by a Swiss firm. This block resembles in appearance solidified methylated spirit, but in its properties it is greatly superior, inasmuch as the block does not liquefy when burning, nor evaporate even if exposed to the air for many months.

Acetylene as obtained from ordinary generators contains a certain amount of impurities. The use of the acetylene blowpipe for welding high-class steels has shown more and more clearly the necessity for removing these impurities from the gas, a purification which can easily be effected by the use of such purifying materials as catalysol, heratol, etc.

One pound of commercial calcium carbide, when acted upon by water, yields on an average about 4·5 cub. ft. of the gas.

Before the war the price of carbide was about £12 10*s.* per ton, packages included. The price is now about £16 to £18 in the United Kingdom, while it is still higher in many of the countries producing it.

Dissolved Acetylene.

By an Order in Council issued in 1897 the compression of free acetylene was made illegal, but further orders were issued in 1901 and 1912 exempting the process now known as 'dissolved acetylene.' This is acetylene thoroughly purified and washed, and compressed into steel cylinders previously filled with a porous mass and having a fixed quantity of acetone added. Acetone is a liquid hydrocarbon, having the property of absorbing 25 times its own volume of acetylene at one atmosphere and 15° C.

For each additional atmosphere of pressure, the same proportion of gas is absorbed, but an allowance has, of course, to be made not only for the volume occupied by the porous filling of the cylinder, but, in addition, in order to allow for expansion, so that it is customary to charge these cylinders with only 43 per cent. of acetone.

A cylinder will therefore in actual practice contain about 100 times its own volume at 10 atmospheres pressure and about 150 times at 15 atmospheres pressure.

In order to show the enormous capacity for storing light obtained in this manner, it may be mentioned that the standard oil-gas cylinder in use on railways has a capacity of 14 cub. ft. at atmospheric pressure. As the oil-gas is usually compressed to 12 atmospheres, this gives a gas volume of 168 cub. ft., which, if filled with oil-gas, will supply eight burners using 2 ft. per hour and burning eight hours per day, for about a day and a half. If this cylinder were filled with porous material and acetone in the manner described above, and charged with acetylene at 15 atmospheres, its capacity would be 2,100 cub. ft., or sufficient to supply eight acetylene burners of 1 cub. ft. each (giving a most brilliant light), for about thirty-three days (basing on eight hours' lighting per day), before the cylinders would require to be re-charged.

The possibility of obtaining acetylene in compressed form opens up a very wide field for the expansion of its utility.

The tests carried out by the Home Office Authorities before issuing the exemption order of 1901 were exhaustive, and furnished satisfactory proof of the safety attending the process of manufacture and the use of the cylinders.

Storage of Dissolved Acetylene

Acetylene dissolved in acetone and stored, under a pressure not exceeding 225 lbs. per sq. in. in a vessel filled as completely as possible with some porous substance is not deemed an explosive, and can therefore be manufactured, stored, and used without risk of infringing the Explosives Act.

(See also 1919 Order in Council, 'Dissolved Acetylene.')

Porous Material.—Various types of filling are in use and appear to give satisfaction. In this country the Kapok filling, patented by *Allen-Liversidge, Ltd.*, is chiefly in use. The porosity of the material should not exceed 80 per cent. The material should completely fill the receptacle, and should contain no pockets or cavities.

The Solvent.—Acetone is the most suitable commercial solvent, but substitutes for acetone could be permitted. The solvent should not completely fill the porosity of the material under any rise of temperature likely to be met with in practice.

Pressure Allowed.—The pressure can be 225 lbs., provided the cylinders used are of the solid drawn type, the pressure to be measured at 60° F.

The Cylinders.—The cylinders should be made of mild steel containing not more than 0·25 per cent. of carbon and not more than 0·05 per cent. each of phosphorus and sulphur. The test pressure should be four times the working pressure, and should be maintained for not less than fifteen minutes. All cylinders should be annealed, but the use of fusible plugs, at present compulsory on acetylene cylinders, should be discontinued, as they are apt to become loose and, further, in the event of fire occurring, may actually lead to the fire being augmented.

It is not considered necessary to impose any conditions as regards the retesting of acetylene cylinders beyond requiring them to be thoroughly examined visually at intervals. The form of retesting to which oxygen and other cylinders are subjected cannot readily be applied to acetylene cylinders with their content of porous material and solvent.

Acetylene Lighting.

For the illumination of buildings, towns, and large areas by acetylene, generators of some suitable type are in general use.

Dissolved acetylene is used successfully from an economical and efficient point of view for many purposes where a constant supply of acetylene is required, and is particularly adaptable where space available for the apparatus is strictly limited.

In the matter of motor vehicle lighting it has been very extensively adopted. A cylinder containing acetylene absorbed in acetone under pressure and of convenient size is attached to the footboard or other suitable position on the car. Such a cylinder contains a sufficient supply of acetylene to meet the requirements for the lighting of the necessary lamps. The time for which the supply is available is variable according to the rate of consumption; the pressure of gas in the cylinder is reduced by an automatic governor or regulating valve, and the supply in the service pipes is maintained at a constant pressure to the lamps.

The standard automobile cylinders contain 20 cub. ft. and 40 cub. ft. of dissolved acetylene, and by the exchange system originated by the manufacturers it is possible for users to obtain refills at practically all important garages throughout the country.

Larger vehicles, such as motor omnibuses, mail vans, motor lorries, etc., may be equipped with outfits of larger capacity to meet the requirements of any particular case.

Another recent departure is the manufacture and circulation of cylinders containing dissolved acetylene of such capacity to meet the demands of motor cyclists. These cylinders are standardised and contain 5 and 10 cub. ft. of acetylene and, as with the standard cylinders for automobile work, they may be exchanged throughout the country.

For buoy and coast lighting dissolved acetylene has many advantages. The purity of the gas in this form, which is assured before compressing, gives the light increased brilliance and greater penetrating power, hence a shorter light period can be employed than with any other illuminant. Various special apparatus are in use, in which flashes of long or short duration are obtained by automatic means, the light being extinguished between the flashes instead of obscured (a small bypass burner remaining alight), thus economising gas. In some cases a flash of as short duration as one-tenth of a second is obtained, with perfect accuracy and flame. By means of this economical consumption and the great storage capacity of a comparatively small cylinder, a gas supply for periods as long as 12 months can be easily carried on a buoy, no attention being required until the gas is exhausted, when the cylinder is disconnected and replaced by a full one. The cylinders may be arranged in the form of a battery where larger consumption is involved.

Owing to the smallness of space required for acetylene in this form it is particularly adaptable for lighting railway coaches, yachts, and small craft. The effect of atmospheric temperature on the gas in this form is almost negligible, and other advantages are obtained by reason of the purity of gas and the elimination of handling carbide, water, waste and residue, and by the fact that the gas in this form can be stored indefinitely without deterioration.

The applications of this form of acetylene are ever increasing, and it has been recently applied with success to cinematographs, searchlight, signalling and projection work, where a small and compact plant consistent with efficiency is required.

The most recent application for illuminating purposes is that of its adoption for portable hand-lamps for mining and other similar purposes.

Lately acetylene buoys have commenced to displace other forms of coast-lighting to a very considerable extent. Such buoys take a very large charge of calcium carbide, usually one to two tons. They are entirely automatic, require practically no attention, and as they run for six to twelve months before requiring recharging, they form an exceedingly economical means of coast and harbour-lighting. Another great advantage is the fact that the light can be recognised at a much greater distance than that of the oil, or oil-gas, buoys previously used.

(See also pp. **1490, 1497.**)

PETROL AIR GAS.

(Contributed by E. Scott-Snell, A.M.S.E.)

Petrol gas is made by impregnating air with the vapour of petroleum spirit, this mixture being sent through pipes in the usual way for combustion with incandescent mantles. The essential features of an apparatus for producing this gas are an air compressor, a carburettor—in which the air and petrol are brought into contact—and a gas-holder from which the pipe service is fed. In automatic apparatus, *i.e.* machines in which gas is made according to the demand, the holder is also used as a governor for controlling the supply of air and petrol to the carburettor. The relative proportions of air to petrol are very important: in fact, upon the relation of these ingredients, and the constancy of this relation under varying load and climatic conditions, the success or failure of any apparatus must depend. As a mantle is used for light-giving, a flame is required at the burner which shall give the highest possible temperature, *i.e.* the amount of air mixed with the petrol vapour must be such as to completely combust it. With the more volatile spirit, mainly hexane (C_6H_{14}), used for making this gas, complete combustion is assured when approximately two volumes of vapour are mixed with 98 volumes of air. This is calculated from the typical equation $2C_6H_{14}+19O_2=12CO_2+14H_2O$ representing the combustion of the chief constituent of light spirit. At present, opinion seems to be divided as to the advisability of sending such a mixture—which from its composition is obviously an explosive one in a confined space such as a gas-holder—through the system. It is found, however, that when the percentage of vapour in the mixture exceeds 4½ per cent., e.g., a 5 per cent. or 6 per cent. mixture, the gas is incombustible without the addition of further air, and in the better class machines advantage is taken of this fact to make a slightly richer gas than 4½ per cent., diluting it with the extra air required for complete combustion by the aid of Bunsen burners.

The air-compressing apparatus of a petrol gas machine may be actuated by any motive power, but as the work of forcing the gas through the pipes is comparatively light, nothing more powerful than a hot-air motor or wound-up weights is usually required. In a good apparatus, the relative amounts of air and petrol are mechanically measured before passing to the carburettor, so that constancy of mixture is obtained under all conditions of temperature within wide limits. When other means are adopted, such as leading the air across a bath of petrol, selective evaporation takes place, producing erratic results, the make of gas varying with every change of temperature. In fact, attempts to make apparatus of this sort automatic almost inevitably end in failure.

Petrol gas is probably one of the safest lighting mediums known, by reason of the small compass over which mixtures of petrol vapour and air are explosive (1½ per cent. to 4½ per cent., c.f. acetylene 3·3 per cent. to 52·3 per cent., coal-gas 7·9 per cent. to 19·1 per cent.) and the fact that over 90 per cent. of the escaping gas from any leak is air. It is also non-asphyxiating.

The gas varies in calorific value according to the proportions of the mixture and the spirit used, but a 6 per cent. mixture from 0·680 sp. gr. petrol has a gross calorific value of nearly 300 B.Th.U. per cubic foot. Although this is a comparatively low figure, cooking and heating can be efficiently performed with suitably proportioned burners.

One gallon of light spirit gives 5,000 candle-power hours (Lewes). With petrol at 1*s.* a gallon this is equivalent to 2·4*d.* per 1,000 candle-power hours. The Government tax of 3*d.* per gallon is remitted when petrol is used for purposes other than propelling road vehicles.

Petrol gas can be stored almost indefinitely over water, but since a small 'Bijou' mantle requires anything from 4 to 12 cub. ft. per hour (according to whether a 6 per cent. or 2 per cent. mixture is used) the storage of any reasonable quantity of the gas entails the use of very large holders. Hence the advantage of the weight-driven type of automatic generator.

Attempts are being made to effect economy by using the less volatile petrols of 0·720 to 0·760 sp. gr. for lighting purposes, such as are in general use for motor cars. The physical characteristics of these spirits, however, are such that they do not lend themselves readily to cold carburation, for either considerable quantities will pass through the carburettor unvaporised, or if heat is used to evaporate them, heavy condensation is unavoidable in the pipe service, which must necessarily be very carefully laid to provide for it.

An explosive (or 'weak') mixture in the pipes and holder is unavoidable if heavy spirits are used, since they are not sufficiently volatile to allow of the use of a 'rich' mixture. See also Scott-Snell on 'Petrol Air Gas,' Soc. of Eng. Journal and Trans., March 1911; Illuminating Engineers' Soc. Journal and Trans., February 1913.

Heating Value and Candle-Power of Petrol-Air Gas.

According to *The Snercold Engineering Co.*, as the result of tests of a variety of different brands and grades of petrol, the fair average heating value of a gallon of petrol is 143,500 B.Th.U., and the maximum candle-power hours that can be obtained under the best conditions is 6,025. The following table shows different mixtures in percentages of petrol vapour used; for calculation purposes the heating value has been taken as 143,000 B.Th.U., and the candle-power hours as 6,000 per gallon.

TABLE SHOWING VARIOUS MIXTURES OF PETROL-AIR GAS.

Mixture Percentage of Petrol Vapour.	Gallons per 1,000 Cubic Feet of Gas.	Heating Value B.Th.U. per Cubic Foot.	Cubic Feet of Gas to the Gallon.	Candle-power Hours per Cubic Foot.
6	2·184	312	458·3	13·104
5½	2·0	286	500	12·0
5	1·82	260	555	10·92
4½	1·638	234	612	9·828
4	1·4638	208	687·5	8·7828
3½	1·274	182	785·7	7·644
3	1·092	156	916·6	6·552
2½	0·91	130	1100	5·46
2	0·728	104	1375	4·368
1½	0·546	78	1833·2	3·276

The heating value of coal-gas varies from 420 to 550 B.Th.U. per cubic foot according to the source of supply.

THE SNERCOLD PETROL-AIR GAS PLANT.

In this plant a 5½ per cent. mixture is generated. The carburettor will vaporise petrol in very cold weather without the application of warmth; in it, the temperature of evaporation falls as low as 49° of frost (− 17° F.), which is well below the lowest winter temperature in this country; for this reason condensation is impossible with the apparatus. The arrangement extracts the moisture from the air, which is carefully kept from contact with water, and thus the air on carburation is dry; hence it always absorbs the correct proportion of petrol, and a uniform gas is produced without possibility of variation. The air cannot obtain more than the predetermined quantity of petrol, while from the design of the carburettor it must take all that is supplied. When no gas is being consumed the apparatus is entirely inactive.

HEAT TREATMENT OF GAS CYLINDERS.

(Second Report.)*

In view of experiments made by the Gas Cylinders Research Committee the Committee agree that the periodical re-heat treatment of carbon steel gas cylinders which have not been obviously damaged, serves no useful purpose, and they recommend that the practice be discontinued.

ADDENDA.

GAS MADE IN GREAT BRITAIN. (See p. 2451.)

The quantity of gas made in Great Britain is now 330,000,000 cu. ft. per annum. The total number of consumers is now yearly 10,000,000, of which nearly 6,000,000 are slot consumers.

VERTICAL RETORTS. (See pp. 2452 and 2462.)

WOODALL-DUCKHAM SYSTEM.—The retorts are now built in five standard sizes which vary only in the dimensions of the major axis, the minor axis being the same in all cases. The coal carbonised and the average gas output per day of the different sizes of retorts are as follows:

Mean Major Axis.	Coal Carbonised, tons.	Gas Output, cu. ft.
44 ins.	3 to 5½	51,000 to 79,000
53 ins.	3½ to 6½	59,000 to 94,000
62 ins.	4½ to 7½	72,000 to 109,000
80 ins.	5½ to 9½	90,000 to 142,000
103 ins.	6½ to 12	115,000 to 180,000

* 'Periodical Heat Treatment of Gas Cylinders.' Published for the Department of Scientific and Industrial Research by H.M. Stationery Office. Price 2s. 6d. net.

GLOVER-WEST SYSTEM.—The retorts on this design are now standardised in three sizes, as follows:

	Type I.	Type II.	Type III.
Major and minor axis of ellipse, at top	33″ × 10″	40″ × 10″	50″ × 10″
Major and minor axis of ellipse 21 ft. from top	39″ × 18½″	46″ × 18½″	56″ × 18½″
Total length of retort	25 ft.	25 ft.	25 ft.
Capacity per day in cubic feet of gas	60,000	75,000	90,000

SECTION XXXVIII

PART I

HEATING BY HOT WATER, STEAM, GAS, &c.—DRYING
(pp. 2491-2512)

PART II

VENTILATION (pp. 2513-2524)

PART III

LIGHTING (pp. 2525-2546)

PART IV

FIRE EXTINCTION—FIRE-RESISTING BUILDINGS
(pp. 2547-2551)

SECTION XXXVIII

PART I

HEATING.

(Revised by J. Roger Preston, M.I.Mech.E., M.R.S.I., Past-President of the Institution of Heating and Ventilating Engineers.)

General.

The unit of heat or British Thermal Unit (B.Th.U.) is the heat required to raise the temperature of 1 lb. of water 1° F.; the unit of surface is the square foot; the unit of pressure is the pound per square inch; the unit of temperature is the degree Fah. Air may be expressed in cubic feet or in pounds weight, and Table A shows how these are convertible.

TABLE A.

Air Temperatures.	12°	22°	32°	42°	52°	62°	72°
Relative Volume of dry air at given temperatures (Volume at 32° F. = 1) . . .	·960	·980	1·00	1·02	1·04	1·06	1·08
Weight per cubic foot in pounds	·0842	·0824	·0807	·0791	·0776	·0761	·0747

NOTE.—The volume of 1 lb. of dry air at 32° F. = 12·38 cub. ft. For any other of the above temperatures, multiply by the relative volume.

OPEN FIRE GRATES.

It would seem that the open fireplace is not the criminally wasteful contrivance that it is commonly supposed to be, and that the 'well' type of fireplace is open to criticism. 'The pleasant British practice of sitting round the open fire,' says the Fuel Research Board Report, 1918-19, 'is amply justified from the scientific point of view, when we know that so much as one-fifth of the total heat of the coal is immediately available for absorption by the sitters if they constitute themselves into a sufficiently close screen around the fire.' In addition to the directly measured radiant heat, the open fire, it seems, is to be credited with heat conducted, radiated, and convected from the fireplace, chimney, and walls to an extent which is such that all together some 60 to 70 per cent. of the heat of the coal is utilised in warming the room and the general fabric of the building.

The following hints as to the best method of altering ordinary house fire-grates, so that they will be more effective and economical, have been circulated by the Royal Institute of British Architects: 'Line your grates with fire-bricks. See that the fire-bricks are high enough to stand up above the fire. Take care that the back brick leans forward and is not more than 4 ins. from the front bars. Fill in the space at the back with fire-clay. See that your register is open not more than one-fourth of its full capacity. If you have no register, partially close the chimney opening with another fire-brick. Fill in the front space beneath the grate with a movable piece of metal. If you cannot do this, put a small fire-tile in the bottom of the grate.'

It must be remembered, however, that the above efficiencies are only obtainable under the most favourable conditions ensuring the proper combustion of the fuel.

Heat Losses.

When a room is maintained at a higher temperature than the surrounding air, there is always a loss of heat through the windows, walls, air-change, etc. Modern practice demands that these losses shall be estimated as accurately as possible; a few of the most useful factors concerned are given below:—

Brick wall 12″ thick	0·30	B.Th.U. loss per	F. deg. diff.	per sq. ft. per hour
Brick wall 18″ thick	0·24	,,	,,	,, ,,
Without plaster 12″ thick	0·34	,,	,,	,, ,,
Without plaster 18″ thick	0·26	,,	,,	,, ,,
Concrete wall 4″ thick	0·53	,,	,,	,, ,,
Concrete wall 8″ thick	0·40	,,	,,	,, ,,
Single window	1·06	,,	,,	,, ,,
Double window	0·76	,,	,,	,, ,,
Air	0·019	,,	,,	,, cu. ft. ,,

Thus with an inside temperature of 70° F. and an outside temperature of 30° F., the loss per sq. ft. per hour in the case of a single window would be

$$1{\cdot}06 \times (70 - 32) = 40 \text{B.Th.U.}$$

The following are some of the channels through which heat may be lost:—

Gases of high calorific value escaping to the chimney unburnt.
Flue gases passing away at high temperatures.
By radiation and convection from uncovered pipes and fittings in Boiler House.
,, ,, ,, ,, ,, in Trenches.
,, ,, ,, ,, ,, in Basements.
,, ,, ,, ,, ,, on Roof.
Steam blowing away into atmosphere.
Condense water cooling in collecting well, etc.
Leakage from, and the cooling of air in, ducts where heated at intake.
Heated air passing away without doing any useful work.
Excessive heating of rooms.
Normal heating of rooms where not in use.
Leakage from air flues.

Reliable comparison of existing work can only be made after exhaustive tests, as owing to conditions varying so much, comparison can easily be misleading, a careful theoretical test balance would be more reliable than an incomplete one of existing conditions.

Care in design at every point to prevent waste, makes a very important whole in large buildings.

Economy.

The most economical system is that which wastes the least heat and costs the least in attendance and upkeep, providing the initial cost is reasonable.

Direct radiation plants are the most economical. Systems where the air is fully heated at the intake and distributed by ducts throughout the building will obviously have a rather high fuel consumption.

As a distributing medium, hot water comes the first, except where exhaust steam is available; there is not a great deal of difference between water and steam, providing that each is the best of its kind.

Cost of Heating.

The warming of the air costs much more than moving it; where it would take 2*d.* to move 1,000,000 cubic feet, it would cost 12·5*d.* to heat it from 30° to 60° F.

Heating Systems.

The heating medium may be either water, steam, or gas. In the case of water and steam these may be distributed in various ways, as follows:—

Water	Low pressure gravity circulation. Forced or induced circulation by accelerators. Various automatic induced.
Low-pressure steam.	Condense through traps (live-steam or exhaust-steam). Vacuum systems. Automatic vapour systems.

Heating by Hot Water.

Gravity Circulation.

For installations not exceeding, say, 5,000 sq. ft. of radiating surface, where upkeep and attention are serious items, and the building is suitable, this system is very satisfactory, and, if no exhaust steam is available, for economy it cannot be beaten.

Arrangements of Circulation.

The two-pipe method was at one time nearly universal. The single pipe has taken its place to a great extent, but a combination of the two is most general. The 'drop' system gives a very rapid circulation.

The following points require to be borne in mind :—Short circuits, dips and air-pockets, size and lengths of mains.

Size of Mains.

The first thing to decide is the maximum drop in temperature to be allowed between the flow and return ; a small drop is desirable, but not always practicable, 20° may be taken as a standard. The following formula agrees very well with practice :—

$$U = \left[\left(16{\cdot}353 \frac{\left(H - \frac{S_1 H}{S_2}\right) 12D}{L} + 0{\cdot}00665 \right)^{\frac{1}{2}} - 0{\cdot}0816 \right] D^2 1224F,$$

where,

H = height of circulation in feet; S_1 = specific gravity of water in flow pipe ; S_2 = specific gravity of water in return pipe ; L = equivalent length of pipe in feet (bend, say, 12 diameters) ; D = diameter of pipe in inches ; F = drop between flow and return in degrees ; U = British Thermal Units per hour.

The rapidity of circulation in vertical pipes often increases at a greater rate, owing to bubbles of steam forming.

The above formula gives B.Th.U., but the curves on the following diagram, fig. 1, are in sq. ft. of radiation, calculated on an average of 160 B.Th.U. per ft. of radiating surface.

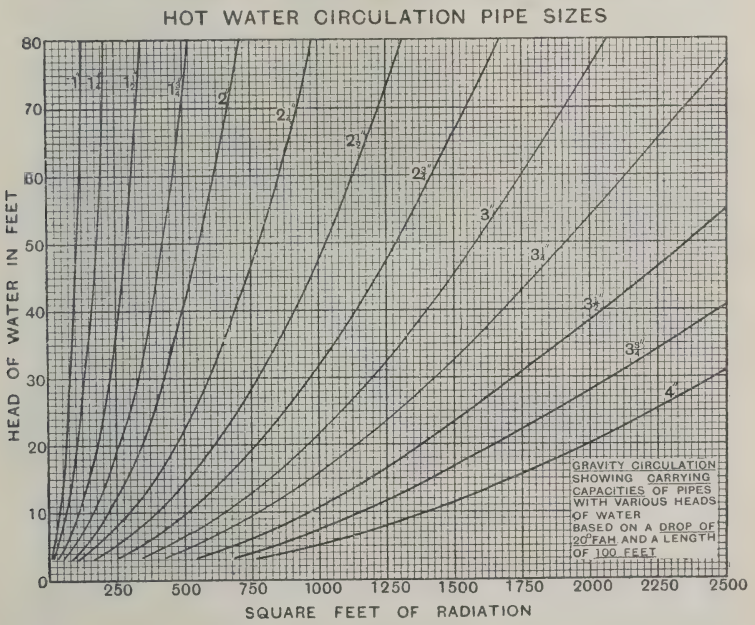


FIG. 1.

For short pipes, such as connections to radiators, a liberal allowance should be made to make up for velocity loss at entry and through connections; these have a considerable effect, so that apart from having a well-designed scheme, it is necessary to have an intelligent fitter.

TABLE OF FACTORS FOR DIFFERENT LENGTHS OF MAINS.

Length of Main. Ft.	Factor.
50	1·5000
100	1·0000
150	0·8125
200	0·7000
250	0·6125
300	0·5375
350	0·4875
400	0·4500
450	0·4375
500	0·3875

For different lengths of mains, multiply the heating surface from fig. 1 by the factor given.

CIRCULATION BY ACCELERATORS.

In large buildings where the radiation is extensive and at many different levels, pumps may be employed to circulate the water; this allows of smaller mains, owing to the higher velocities obtained. The piping may be arranged either on the one-pipe or two-pipe system. With the one-pipe, practically only the mains are under the influence of the pump, though there is a little pull on the branches, due to the loss of head on the main between the point where the flow is taken off and the return back.

The number of gallons of water required to supply the heat can be ascertained as follows:—

$$G = \frac{U}{10D},$$

where,

G = number of gallons per hour; U = heat loss in B.Th.U.; D = drop in temperature (F.°) allowed between flow and return.

The size of mains depends upon their length, the number of gallons, and the maximum loss of head due to friction to be allowed. A fair velocity in the mains is from 60 to 200 ft. per min., according to the size and length, arranging for the frictional head not to exceed about 20 ft.; this is a point that should be carefully watched, as a contractor may cut down the size of mains to save initial cost, but, as a result, increasing the cost of upkeep.

The friction head and power required may be calculated in the usual manner for cold water.

There are several special types of accelerators on the market, though most centrifugal pumps are suitable, also drum or other pumps may be used if desired.

AUTOMATIC INDUCED CIRCULATION.

Where power is available, a simple accelerator is to be preferred; but there are conditions where an automatic arrangement is an advantage. These utilise the power of steam in some form, either by condensation, a partial vacuum being formed, which exerts a powerful pull on the water in the returns, or by the pressure being utilised to lift water from one tank to another at a higher level, the water returning through the circuits by gravity; a separate steam boiler is not always needed, steam being formed in connection with the water boiler.

Apart from the particular apparatus for producing the motive power, the heating system is the same as any other low-pressure system.

Low Pressure Steam Heating.

GRAVITY RETURN.

In this system the condense water finds its way back to the boiler by gravity. The steam may be distributed to the radiators by either the one- or the two-pipe system.

In single-pipe work the steam in the mains travels in the direction of the condense water, but in the risers and branches they meet (except in the drop system), and when starting the heat, noise is created. Where absence of noise is essential at all times, the two-pipe system is preferable; with this arrangement there are two pipes to each radiator, the condense pipes are carried down into a wet return or fitted with an automatic water-seal at the outlet to the radiator.

Size of Pipe.

The size of pipe to supply a given amount of heat at a given point depends upon the length initial pressure, and the maximum loss of pressure to be allowed.

Taking the known formula for steam, and modifying the exponent and value of P, we get the following:—

$$U = \left(\frac{WDP^5}{L\left(1 + \frac{3\,6}{D}\right)} \right)^{\frac{1}{2}} 10^6,$$

where,

U = British Thermal Units per hour; W = weight of a cubic foot of steam; P = loss of pressure in inches of water; D = diameter of pipe in inches; L = length of pipe in feet.

The results shown in Fig. 2 have been worked out for about 1·3 lb. initial gauge pressure and 3 ins. water gauge drop.

For one-pipe work, multiply B.Th.U. by 0·5.

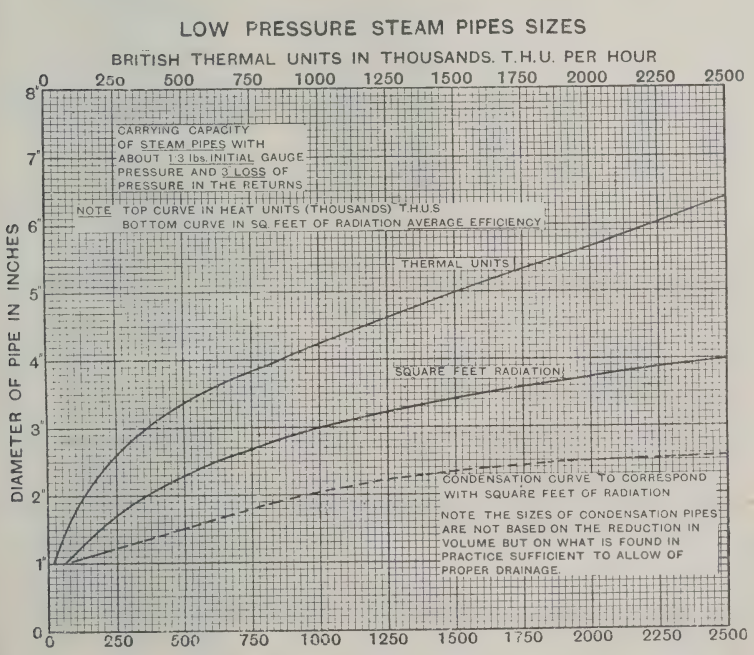


Fig. 2.

Pipe Sizes for Vapour or Low Pressure Steam Systems.

For small work, and ready reference, the following tables may be used; they are based on a conservative drop in pressure in all piping; care should be exercised in ascertaining length of runs:—

Example.—To find size of main required under the following conditions:—Length of run, 150 ft., with six 90° elbows, and carrying 250 sq. ft. of radiation. From Table I a 2 in. main approx. is required. Under 2 in. on Table II it is found that each 90° elbow has the equivalent resistance of 5 ft. of pipe, or 30 ft. for the six elbows. This makes the total length of pipe equivalent to 180 ft. Checking back under 200 ft. of run (the nearest length to the run desired), it is found that a 2 in. pipe will carry 325 sq. ft. of radiation.

Riser sizes are determined in the same way. All lengths to be measured from the boiler.

TABLE I.—MAIN AND RISER SIZES.

Main Sizes.						Riser Sizes.				
Pipe Size in ins.	Length of Run in feet.					Pipe Size in ins.	Length of Run in feet.			
	200	300	400	500	600		100	200	400	600
	Capacities in sq. ft. of direct radiation.						Capacities in sq. ft. of direct radiation.			
2	325	270	220	200	160	¾	25	20	—	—
2½	600	500	420	360	280	1	65	50	35	20
3	1050	850	720	620	560	1¼	120	100	75	45
3½	1560	1250	1050	900	800	1½	200	150	100	70
4	2050	1800	1500	1300	1100	2	400	320	240	175
4½	2800	2400	2000	1725	1500					
5	3800	3100	2600	2300	2000					
6	6000	5000	4300	3700	3300					

Return Main Sizes.		Return Riser Sizes.	
Radiation (sq. ft.).	Size in ins.	Radiation (sq. ft.).	Size in ins.
0–100	¾	0–50	½
100–400	1	50–200	¾
400–1200	1¼	200–400	1
1200–1800	1½	400–900	1¼
1800–3000	2	900–1500	1½
		1500–3000	2

TABLE II.—PIPE LENGTH IN FEET EQUIVALENT TO RESISTANCE IN ELBOWS.

Pipe Size.	45° Elbow.	90° Elbow.	Pipe Size.	45° Elbow.	90° Elbow.
Ins.	Ft.	Ft.	Ins.	Ft.	Ft.
2	3	5	4	6	13
2½	3	6	4½	8	16
3	4	8	5	10	19
3½	5	10	6	13	23

British Standard Low-pressure Heating Pipes.*

(No. 40—1908.) (*Abstract.*)

For working pressures up to 100 ft. head.

Dimensions of Pipes (fig. 3).

			Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
Pipe.	Internal Diameter	A	2	3	4	5	6	7	8	9
	Approximate External Diameter	B	$2\frac{3}{8}$	$3\frac{7}{16}$	$4\frac{15}{32}$	$5\frac{9}{16}$	$6\frac{5}{8}$	$7\frac{11}{16}$	$8\frac{3}{4}$	$9\frac{13}{16}$
	Approximate Thickness	C	$\frac{3}{16}$	$\frac{7}{32}$	$\frac{15}{64}$	$\frac{9}{32}$	$\frac{5}{16}$	$\frac{11}{32}$	$\frac{3}{8}$	$\frac{13}{32}$
	Diameter of Spigot Bead and Astragal	D_1 D_2	$2\frac{5}{8}$	$3\frac{3}{4}$	$4\frac{25}{32}$	$5\frac{15}{16}$	7	$8\frac{1}{8}$	$9\frac{3}{16}$	$10\frac{5}{16}$
	Projection of Spigot Bead and Astragal	E_1 E_2	$\frac{1}{8}$	$\frac{5}{32}$	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{7}{32}$	$\frac{7}{32}$	$\frac{1}{4}$
	Width of Spigot Bead	F	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$
	Radius of Spigot Bead	G	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{25}{64}$	$\frac{15}{32}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{19}{32}$	$\frac{21}{32}$
	Width of Astragal	H	$\frac{9}{16}$	$\frac{5}{8}$	$1\frac{1}{16}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$
	Clearance	I	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{7}{64}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{5}{32}$	$\frac{5}{32}$
	Caulking Space	$E_1 + I$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{17}{64}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{11}{32}$	$\frac{3}{8}$	$\frac{13}{32}$
Socket.	Internal Diameter	J	$2\frac{7}{8}$	$3\frac{15}{16}$	5	$6\frac{3}{16}$	$7\frac{1}{4}$	$8\frac{3}{8}$	$9\frac{1}{2}$	$10\frac{5}{8}$
	External Diameter	K	$3\frac{7}{16}$	$4\frac{9}{16}$	$5\frac{21}{32}$	$6\frac{15}{16}$	$8\frac{1}{16}$	$9\frac{1}{2}$	$10\frac{7}{16}$	$11\frac{5}{8}$
	Thickness	L	$\frac{9}{32}$	$\frac{5}{16}$	$\frac{21}{64}$	$\frac{3}{8}$	$\frac{13}{32}$	$\frac{7}{16}$	$\frac{15}{32}$	$\frac{1}{2}$
	Internal Depth	M	3	$3\frac{1}{2}$	4	4	$4\frac{1}{2}$	$4\frac{1}{2}$	5	5
	Internal Radius	N	$\frac{7}{16}$	$\frac{15}{32}$	$\frac{1}{2}$	$\frac{19}{32}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$
	External Radius	O	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{15}{16}$
	Diameter of Beads	P	$3\frac{15}{16}$	$5\frac{1}{4}$	$6\frac{13}{32}$	$7\frac{11}{16}$	$8\frac{15}{16}$	$10\frac{1}{8}$	$11\frac{7}{16}$	$12\frac{5}{8}$
	Width of Beads	Q	$\frac{9}{16}$	$\frac{25}{32}$	$\frac{13}{16}$	$\frac{7}{8}$	1	1	$1\frac{1}{8}$	$1\frac{1}{8}$
	Projection of Beads	R	$\frac{1}{4}$	$\frac{13}{32}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$
	Distance between Beads	S	$1\frac{3}{8}$	$1\frac{7}{16}$	$1\frac{19}{32}$	$1\frac{15}{32}$	$1\frac{11}{16}$	$1\frac{3}{4}$	$1\frac{15}{16}$	$1\frac{13}{16}$
	Distance of Bead from Edge	T	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{7}{16}$
	Taper on End	U	$\frac{1}{16}$	$\frac{1}{16}$	$\frac{3}{32}$	$\frac{3}{32}$	$\frac{3}{32}$	$\frac{3}{32}$	$\frac{1}{8}$	$\frac{1}{8}$
	Width of Rest at Top	V	$\frac{7}{16}$	$\frac{25}{32}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$
	Width of Rest at Bottom	W	$\frac{9}{16}$	$\frac{25}{32}$	$\frac{13}{16}$	$\frac{7}{8}$	1	1	$1\frac{1}{8}$	$1\frac{1}{8}$

Minimum Weights of Pipes (fig. 4).

Overall Length. Feet.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
3	15	24	32	50	67	86	108	130
6	27	45	61	94	125	160	201	243
9	—	66	90	138	182	234	294	356

All sizes of low-pressure heating-pipes and fittings shall be tested by an internal hydraulic pressure of 100 lbs. per square inch, applied for at least 2 minutes. While under pressure each pipe and fitting shall be struck all over with a hand hammer weighing $\frac{1}{4}$ lb.

* By permission of the British Engineering Standards Association.

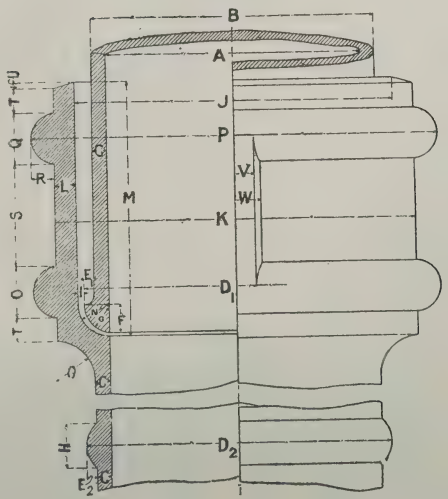


FIG. 3.

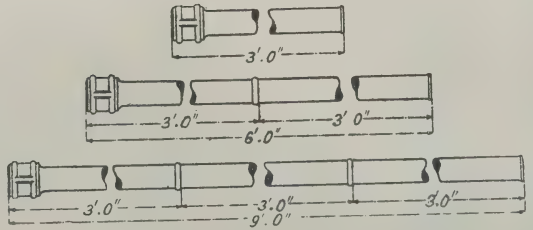


FIG. 4.

Vacuum System.

One of the objections to the ordinary gravity steam work is the difficulty of displacing the air in the radiators, and controlling the heat; this may be overcome by attaching a system of piping to the air valves connected to a small exhausting pump, which draws away the air and forms a partial vacuum; another way is to attach all the condense mains to a pump which will exhaust the air and draw away the condense water, this method is known as the 'Vacuum System.'

The piping is much the same as an ordinary gravity arrangement, except that the mains are rather smaller, and the condense pipes much smaller.

Owing to the thorough circulation of steam, the radiating surface has a high efficiency per sq. ft. per degree difference. It should be noted, however, that this does not materially affect the fuel efficiency.

The success of a vacuum system depends to a large extent on the proper working of the valves which are fixed on the radiators, or wherever condense water is likely to collect.

Plenum System.

'Plenum' is the name given to the system where the air is dealt with at the intake, tempered, washed, heated, and blown into the various rooms in such a quantity as to provide ventilation and make good the heat losses, the size of outlets being so proportioned as to allow of a slight pressure in the building, hence the name 'Plenum.'

The inlets may be from 8 to 10 ft. above the floor level, outlets at floor level on the same side of the room, the initial velocity of the air carrying it across the room, some of it cooling and gradually falling on the way—an extra outlet at ceiling level is advisable, especially if gas is the illuminant. Two ducts are necessary, one for hot and one for tempered or cold air.

NOTE.—Any system where the air is propelled into a building and maintained at a higher pressure than the atmosphere is a Plenum system, though the name is usually associated with downward methods of ventilation only.

Centrifugal fans are the most suitable, as the resistance of the heaters is considerable, apart from the ducts.

There are several special makes, including the multivane types, for which a much higher efficiency is claimed.

For economy a fan should be chosen that will pass the amount of air just running at a suitable speed to overcome the resistance, as the power rises as the cube of the speed, while the output is only directly as the speed.

Heating by Exhaust Steam.

The use of exhaust steam for heating and other purposes such as drying, hot-water supplies, etc., is to be recommended wherever possible, as the adoption of this system enables the latent heat in the steam to be used, whereas this cannot possibly be used for the production of power, and the extension of this system would render many small and inefficient power plants at present in existence more efficient.

For instance, the Technical Advisor to the Coal Controller states that the average thermal efficiency of steam-driven electric power plants at the present time is only 8½ per cent., whereas, if sufficient demand for heat can be found to enable the exhaust steam to be used for heating purposes, etc., the thermal efficiency of these plants might be increased to something like 60 per cent.

The exhaust steam can be either taken direct from the engines and conveyed without back pressure, for considerable distances, by the adoption of a modern vacuum system, or alternatively it can be used to heat water in calorifiers and the water pumped round to the various points where heat is required.

In many institutions, hotels, etc., it is found that there is practically no difference in the coal consumption whether the power is purchased from outside or generated on the premises, as there is a demand for heat all the year round which can be met by the exhaust steam from the power plant, so that the extension of this system to general domestic heating from the existing small power plants would enable these small power plants to run at a greater efficiency than any large condensing plant.

The utilisation of exhaust steam for heating when, as is too frequently the case, there is a back pressure on the engine—does not always result in economy. For assume that at a large works offices are to be warmed, requiring 800 sq. ft. of radiation, which offices are at such a distance from the main engine that a back pressure of 8 to 10 lbs. is required to force the circulation, the engine being 200 h.p. cutting off automatically at one-third of its stroke; the effect of the back pressure in this case would be to decrease the range of expansion to approximately one-half stroke, and presuming the initial pressure to be 80 lbs., and the consumption at one-third stroke to be 25·7 lbs. weight of steam per h.p. per hour, the consumption would be increased to 29 lbs. per h.p. hour, or a total for the 200 h.p., of 200 × 3·3 (the latter figure being the difference in the con-

sumption without and with back pressure) = 660 lbs. of steam per hour. Now as the amount of radiation mentioned would require, at 8 to 10 lbs. pressure, less than 300 lbs. per hour, there is, so far from being an economy, a loss of over 360 lbs. of steam per hour as the result of using the exhaust steam under such conditions of back pressure. Only when it is used without back pressure will the full economy be derivable from the utilisation of exhaust steam for heating.

One square foot of radiation will give off about 250 heat units an hour. One pound of steam an hour will supply about 4 square feet of radiation. A rough-and-ready rule for finding the radiation that may be supplied by an engine is to multiply the horse-power of the engine by 131.

When estimating on exhaust-steam heating, with non-condensing engines, the steam consumption for each indicated horse-power an hour will be about as follows :—

Engine.	Total.	Available for Heating.
Compound automatic	20 lbs.	18 lbs.
Simple Corliss	30 lbs.	27 lbs.
Simple throttling	40 lbs.	36 lbs.
Simple automatic, single valve . . .	35 lbs.	31 lbs.

When exhaust steam is employed as the heating medium it is necessary to extract the grease by means of a separator, so placed as to intercept the grease before the steam enters the heating main. It is also necessary to have an outlet to the atmosphere which should be closed by a valve weighted to open at a pressure of ½ lb. above the atmosphere, so as not to cause any back pressure. So low a pressure can, however, only be used for circulation when a vacuum return line is maintained, as in the vacuum system, for unless the air is first exhausted from the apparatus, a definite pressure is required to force it out through the air valves. With a vacuum return line no air valves are required on the radiation, and no foul air from the radiators is discharged into the rooms.

In steam heating, the temperature depends on the pressure of steam in accordance with the figures given in the following table :—

TABLE OF THE PROPERTIES OF SATURATED STEAM.

Absolute Pressure per Square inch.	Tempera-tures.	Total Heat of One Pound of Steam from Water at 32 deg. F.	Latent Heat of One Pound of Steam from Water at 32 deg. F.	Weight of One Cubic Foot of Steam.	Volume of One Pound of Steam.	Pressure as shown by Gauge.
Lbs.	Deg. Fah.	Units.	Units.	Lbs.	Cub. Ft.	
7	176·9	1135·3	990·0	·01889	52·936	15
7·5	180·0	1136·3	987·8	·02016	49·610	14
8	182·9	1137·2	985·7	·02142	46·686	13
8·5	185·7	1138·0	983·8	·02268	44·097	12
9	188·3	1138·8	981·9	·02394	41·777	11
9·5	190·8	1139·5	980·1	·02547	39·261	10
10	193·3	1140·3	978·4	·02642	37·845	9
10·5	195·6	1141·0	976·7	·02767	36·145	8
11	197·8	1141·7	975·2	·02890	34·599	7
11·5	200·1	1142·4	973·6	·03026	33·045	6
12	202·0	1143·0	972·2	·03137	31·879	5
12·5	204·0	1143·6	970·8	·03260	30·678	4
13	205·9	1144·2	969·4	·03382	29·573	3
13·5	207·8	1144·8	968·1	·03504	28·536	2
14	209·6	1145·3	966·8	·03627	27.573	1
***14·7**	**212·0**	**1146·1**	**965·2**	**·03797**	**26·360**	**0***
15	213·1	1146·4	964·3	·03870	25·843	—
16	216·3	1147·4	962·1	·04112	24·320	1
17	219·6	1148·3	959·8	·04253	23·513	2
18	222·4	1149·2	957·7	·04594	21·766	3
19	225·3	1150·1	955·7	·04834	20·687	4
20	228·0	1150·9	953·8	·05074	19·710	5
21	230·6	1151·7	951·9	·05311	18·828	6
22	233·1	1152·5	950·2	·05549	18·022	7
23	235·5	1153·2	948·5	·05786	17·282	8
24	237·8	1153·9	946·9	·06023	16·603	9
25	240·1	1154·6	945·3	·06259	15·977	10

Approx. inches of Vacuum.

Approx. pounds per sq. inch.

* Atmospheric Pressure.

TABLE FOR CALCULATING HEATING BY EXHAUST STEAM.

Details.	Above Atmosphere.								Below Atmosphere.		
Steam pressure	0	1	2	3	4	6	8	10	2	4	6
Absolute pressure	14·7	15·7	16·7	17·7	13·7	20·7	22·7	24·7	12·7	10·7	8·7
Temperature of steam	212	216	219	222	225	230	235	240	204	201	199
Temperature of air	70	70	70	70	70	70	70	70	70	70	70
Temperature difference	142	146	149	152	155	160	165	170	134	131	129
B.Th.U. per min. from 100 sq. ft. rad.	426	438	447	456	465	480	495	510	402	393	387
Total heat in steam above 212°	966	967	968	969	970	971	973	974	964	961	957
Latent heat in steam	966	963	960	958	957	954	950	947	970	979	986
Cubic feet in 1 lb. steam	26·4	24·6	23·2	21·0	20·6	19·5	17·8	16·4	31·9	38·0	47
Cubic feet steam in $\frac{2}{3}$ lb.	17·6	16·4	15·5	14·0	13·7	13·0	11·8	10·9	21·2	25·3	31
Cubic feet steam per min. to supply 100 sq. ft. radiation—air 70°	11·6	10·9	10·3	9·3	9·0	8·6	7·8	7·2	10·8	16·8	20·8
Weight per cubic foot steam—lb.	0·038	0·0387	0·041	0·437	0·046	0·061	0·056	0·0609	0·031	0·026	0·021
Throttling Engine— Radiating surface supplied—sq. ft.	152	146	143	139	136	131	128	126	162	175	188
Automatic Engine— Radiating surface supplied—sq. ft.	134	129	127	122	121	120	119	117	146	151	168
Simple Corliss— Radiating surface supplied per horse-power	114	110	107	104	101	98	96	95	122	127	146
Compound Engine— Radiating surface supplied per horse-power	95	91	90	87	84	82	80	79	104	108	116

Boilers.

If plenty of draught is available, a boiler with a liberal amount of secondary flues and small grate area may be chosen; if the draught is small, one with less flue resistance would be more suitable. It should be of such a design as to allow of proper combustion in the fire-pot, free circulation in the waterways, the lighting, feeding, and ash doors to be well fitted, so as to stop an inrush of air at undesirable points; the general design to allow of clinkering and flue cleaning to be carried out with a minimum of trouble. The catalogue rating should give the number of B.Th.U. given off with a *six hours' firing interval*; *this should be taken from test, not based upon square feet of boiler heating surface, the value of which varies very much with different makes.* The *fuel efficiency* should be stated, and should *not be less than* 60 *per cent.* On testing a well-known sectional boiler, the highest efficiency was found to be at seven hours' firing interval (78 per cent. evaporative test), with a slight drop to four hours (due to loss from high flue temperature) and a quick drop to ten hours (due to incomplete combustion). While in power boilers combustion has received a deal of attention, heating boilers have been neglected in this respect.

Approximate Minimum Size of Boiler House for Different Sized Heating Installations.

Amount of Radiation in Feet Super.	Dimensions of Boiler House.*		
1,000 . . .	9 ft. long	5 ft. wide	7 ft. 6 ins. high
2,500 . . .	11 ft. ,,	6 ft. ,,	7 ft. 6 ins. ,,
5,000 . . .	13 ft. ,,	9 ft. ,,	8 ft. ,,
10,000 . . .	14 ft. ,,	13 ft. ,,	9 ft. ,,
20,000 . . .	18 ft. ,,	16 ft. 6 ins. ,,	10 ft. ,,
50,000 . . .	18 ft. ,,	32 ft. ,,	10 ft. ,,

* Exclusive of fuel space.

If hot water supply, boiler, etc., are required, extra space must be allowed.

Approximate Size of Chimney for Various Sized Installations.

Radiation in feet super. . . .	1,000	2,500	5,000	10,000	20,000	50,000
Size of chimney in inches . . .	9×9	14×9	14×14	21×18	24×24	36×36

The Cochran Hot-Water Boiler.

This boiler (*Cochran & Co., Annan, Limited*) is manufactured in twenty-two sizes corresponding to the range of standard boilers, the essential difference being the considerably reduced height, diameter for diameter, as compared with the steam units.

The 8 ft. 6 in. diameter boiler can be limited in height to 14 ft. should headroom considerations necessitate this.

The range of duties vary from 390,000 to 6,500,000 B.Th.U. per hour reckoned on a two-hourly stoking interval in the case of boilers with two fireholes.

These boilers retain all the constructional advantages of the standard Cochran boiler.* They can be supplied for a working pressure of 100 lbs. per sq. in. (*i.e.* for use with a head of 230 feet of water), and, with reasonable care, will last indefinitely. The Cochran boiler manufacturers claim that as a hot-water unit the arrangement occupies less floor space than any other type of boiler. As compared with cast-iron sectional types, the high efficiency and durability of this boiler are important considerations, while the prices are below those of steel sectional and wrought welded boilers. Also, owing to the entire absence of brickwork setting, etc., they are cheaper than the built-in types of boiler, a feature of importance in large heating schemes.

A fire-box of any capacity can be supplied, but the standard sizes are fully equal to all normal requirements, and to meet 'banked' conditions two fire-doors can be provided.

With regard to the outlet connection at the top of the boiler, it is necessary to submit details of the available head-room and size of flow-pipe in the system, to enable a suitable flanged bend to be supplied. The diameter of the return pipe must also be indicated, in order that a suitable flange connection may be provided. In the absence of this information standard flow and return connections would be supplied; these are made with flanges to B.S. Table No. 1.

The smoke-box of the Cochran hot-water boiler enables the funnel to be taken off in any desired direction relative to an existing system of flues in the building, thus keeping within the height of the boiler, in cases where headroom is limited.

* See p. 1582.

Fuel.

The average calorific value per pound of fuel may be calculated at 12,000 B.Th.U., but of this it is only possible to obtain in the boiler an actual efficiency of 70 to 75 per cent. In other words, 8,000 B.Th.U. would represent the heat available from the boiler per pound of fuel burnt, the coal being of an average quality.

SIZE OF MINIMUM FUEL SPACE REQUIRED FOR VARIOUS SIZES OF HEATING INSTALLATIONS ON A BASIS OF ONE MONTH'S SUPPLY.

Radiation in feet super . . .	1,000	2,500	5,000	10,000	20,000	50,000
Fuel space in cubic feet . . .	500	1,000	2,000	3,800	7,600	19,000

FUEL SPACE REQUIRED FOR GENERAL OFFICE BUILDINGS.

Cubic feet of fuel space = cubic contents of building $\times \frac{6 \cdot 45}{1\,000}$.

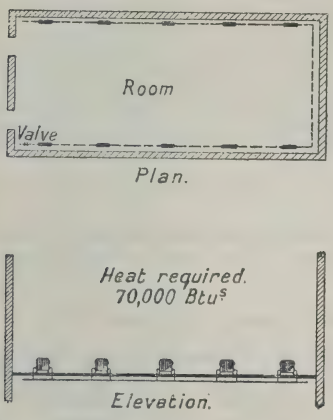
Radiators.

RADIATING SURFACE.

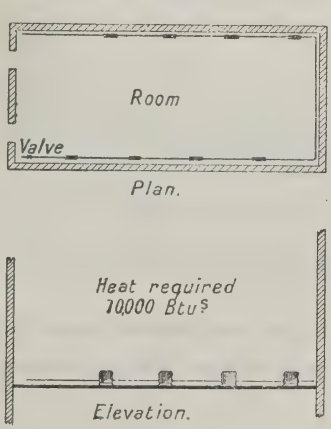
Vertical Loop Radiators are mostly used. Their efficiency varies with the design and size. They should be plain, smooth, and the loops so spaced as to allow of every part being easily cleaned.

Where it is permissible for the pipe supplying the radiators to be above the floor, the initial cost is considerably reduced, as will be seen from fig. 6; this is an important point often lost sight of in considering the cost of a heating installation.

The amount of radiating surface will vary according to the class of surface and its temperature.



Pipes below floor: cost £98.
FIG. 5.



Pipes above floor: cost £73.
FIG. 6.

A recent test carried out by the Research Committee of the Institution of Heating and Ventilating Engineers has shown that a safe figure for modern radiation is 150 B.Th.U. per sq. ft., with a difference of 100° F. between the temperature of the water and surrounding air.

The above figures show the advantage of having the pipe above the floor where permissible. Note.—Prices fair competitive basis (January 1929).

It has also been proved that the emission varies as the 1·3 power of the temperature difference between the heating medium and surrounding air. This is a very useful factor, as within reasonable limits obtained in general heating the transmission can be found at any given temperature difference.

It will be seen, therefore, that the amount of radiation in any room will vary considerably according to the temperature of the heating medium; for instance, if a room has a heat loss of 100,000 B.ThU., the amount of surface required for providing a temperature of 60° F. would be:—

| | | |

RADIATING SURFACE REQUIRED FOR DIFFERENT CLASSES OF BUILDINGS, AND COST OF HEATING.

Type of Building. (Taking the total cube.)	Average Heating Surface in Feet Super. per 1,000.	Approximate Cost of Heating per 1,000 Cubic Feet.
Churches	12	75 shillings
Hospitals	20	140 "
Block of offices	15	98 "
Public buildings	12	84 "
Factories	7 to 10	47 "
Residences	12	84 "
Small detached residences	15	105 "
Retail shops, etc.	8	52 "

The amounts given include the surface in mains for average work.

These costs, which are based upon present-day prices (January 1929), do not include builders' work, cutting away, making good, etc.

For country work, add 10 per cent. to cover additional allowance to workmen, travelling, and incidental expenses.

Where all pipes are concealed, add 25 per cent.

APPROXIMATE HEATING SURFACE REQUIRED FOR HORTICULTURAL HEATING.

Type of House.	Approx. surface in feet super per 1,000 cubic feet.
Houses requiring to be kept at a temperature above 35° in order to prevent frost entering	16
Peach and similar Houses	27
Greenhouses	40
Tomatoes, Vines, etc.	60
Forcing Pits	130

TABLE OF RADIATION.

Height of Radiator in Ins.		46	38	36	32	30	26	24	23	20	18
Type.	Length of One Section. Ins.	Heating Surface per Section in Ft. Super.									
Single Column	3	—	3	—	$2\frac{1}{2}$	—	2	—	$1\frac{3}{4}$	—	—
Two Columns	3	5	4	—	$3\frac{1}{2}$	—	$2\frac{3}{4}$	—	—	—	—
Three Columns	$2\frac{3}{4}$	6	5	—	$4\frac{1}{4}$	—	$3\frac{1}{2}$	—	—	—	$2\frac{1}{4}$
Four Columns	3	—	8	—	$6\frac{1}{2}$	—	5	—	—	—	3
Classic 4 Column	2	—	—	3	—	$2\frac{1}{2}$	—	2	—	—	$1\frac{1}{2}$
Classic 6 Column	2	—	—	$4\frac{1}{2}$	—	$3\frac{3}{4}$	—	3	—	—	$2\frac{1}{4}$

To find the space necessary for radiators, multiply the above figures by the number of sections required, adding 8 ins. in length for valve, elbow, union, and bushing.

WIDTH OF RADIATORS AT FEET.

	Ins.		Ins.
Single column	5½	Three columns	10½
Two columns	8⅝	Four „	11

The kind of metal, and the thickness of the latter, of which the radiator is made, has little effect upon the heat transmission, provided the radiator is not painted.

Experiments have proved that painting affects the heat emission of a radiator. Taking the heat emission from a bare radiator as 100, the painted radiators compare as follows:—Flat-tone brown, 104·8; flat-tone cream, 104; white-gloss enamel, 102·2; ordinary white paint, 100; ordinary green, 96; aluminium, 93·7; liquid gold, 92·6; copper and aluminium bronze, 72.

The *thickness of the coat* makes very little difference, *i.e.* it is only the final coat that influences the heat emission.

THE WESTINGHOUSE PATENT STEAM RADIATOR.

The Westinghouse Patent Radiator shown in fig. 8 was designed to overcome the disadvantages formerly attendant on the use of steam for heating purposes. Steam enters the radiator at A and passes through the opening in the valve seat into a distribution pipe D, in the upper side of which a small hole is drilled opposite each column. It then passes up each column as shown

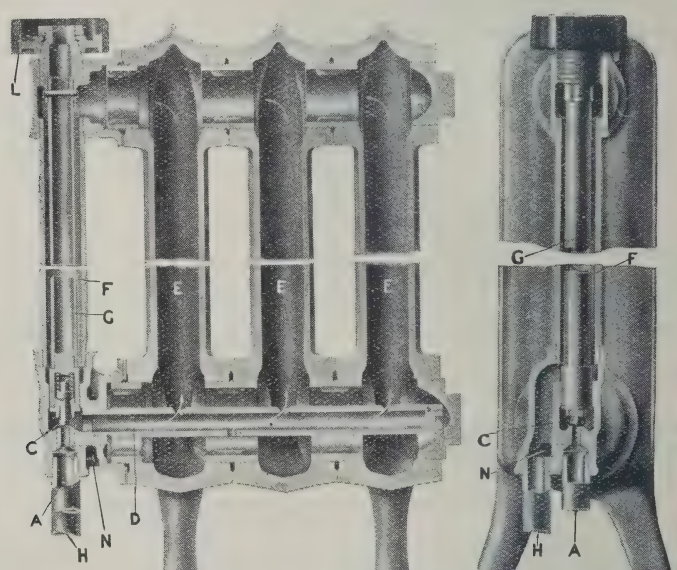


FIG. 8.—The Westinghouse Patent Radiator.

by the arrows and into tube F. This tube contains another tube G of metal with a high coefficient of expansion, which, when it reaches a temperature of about 210° F. or any other lower temperature to which it is set, expands and closes the valve C, thus shutting off steam until the temperature is slightly reduced, when the valve again opens.

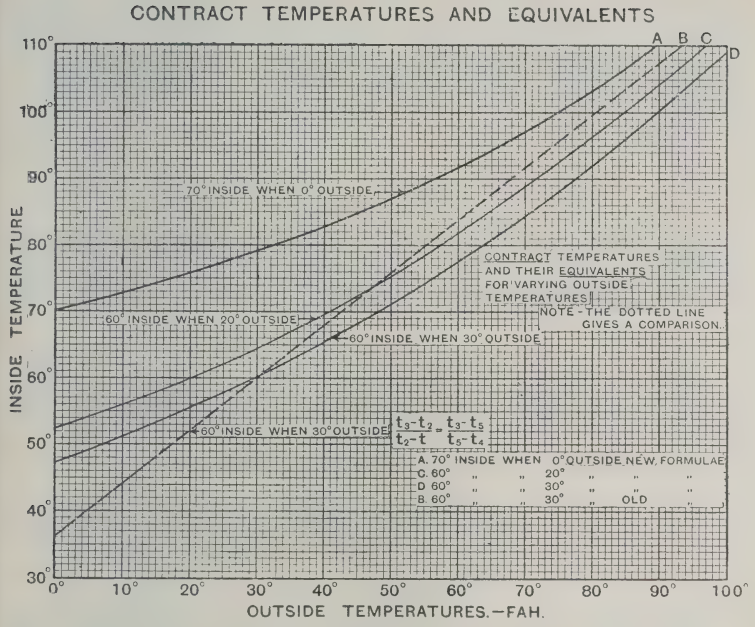

CONTRACT TEMPERATURES AND EQUIVALENTS
A B C D
110°
100°
90°
80°
70°
60°
50°
40°
30°
INSIDE TEMPERATURE
70° INSIDE WHEN 0° OUTSIDE
60° INSIDE WHEN 20° OUTSIDE
60° INSIDE WHEN 30° OUTSIDE
60° INSIDE WHEN 30° OUTSIDE
$\frac{t_3-t_2}{t_2-t} = \frac{t_3-t_5}{t_5-t_4}$
CONTRACT TEMPERATURES AND THEIR EQUIVALENTS FOR VARYING OUTSIDE TEMPERATURES
NOTE - THE DOTTED LINE GIVES A COMPARISON.
A. 70° INSIDE WHEN 0° OUTSIDE NEW FORMULAE
C. 60° " " 20° " " "
D 60° " " 30° " " "
B. 60° " " 30° " OLD "
0° 10° 20° 30° 40° 50° 60° 70° 80° 90° 100°
OUTSIDE TEMPERATURES.—FAH.

FIG. 9.

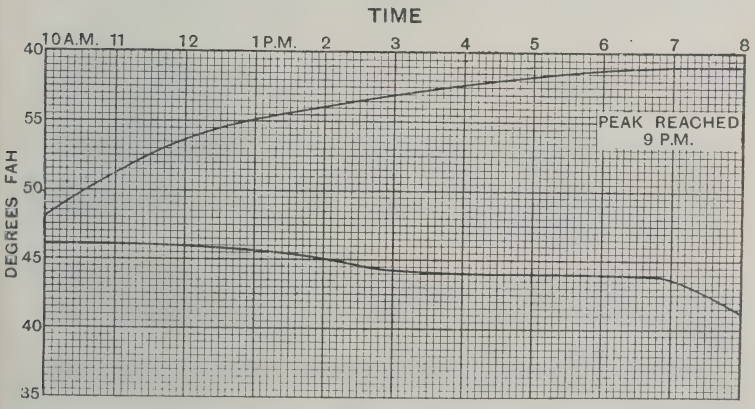

TIME
10 A.M. 11 12 1 P.M. 2 3 4 5 6 7 8
40
55
50
45
40
35
DEGREES FAH
PEAK REACHED 9 P.M.

FIG. 10.

The temperature of the radiator can be varied as desired up to the maximum by turning the handwheel L which, owing to its screwed socket, moves the tube F and valve C nearer to, or farther from its seat B, and thus causes the valve to close at a lower or higher temperature as desired. The temperature of the radiator can never exceed 212° F., as the steam inlet is restricted by the valve while the outlet is fully open to the exhaust, and the steam is therefore entirely condensed in the radiator, only water escaping to the atmosphere. It will therefore be seen that the temperature of the radiator does not depend on the pressure of the steam supply, nor is it affected by the distance of the radiator from the source of supply.

Contract Temperatures and their Equivalents.

(Direct Radiation.)

These, which are of considerable interest to architects and engineers, have been the subject of considerable controversy. Some years ago a Committee of the American Institute of Heating and Ventilating Engineers reported on the subject; in 1903 Mr. Walter Jones gave a paper before the English Institute. Professor Allen's paper on 'The Coefficient of Heat Transmission in Cast Iron Radiation,' June 1908, is one of the most modern.

The expression $$\frac{t_2 - t_3}{t_2 - t} = \frac{t_3 - t_5}{t_5 - t_4}$$

is considerably in error, *even taking into consideration the different coefficients of transmission, extra losses due to increased air changes, etc.* In the discussion on Professor Allan's paper, Mr. C. B. Thompson said he had recently made some tests, but could not get more than 105° F. inside, with an outside temperature of 85° F. and surface that gave 70° inside when 0° outside.

The following empirical formula is found to agree closely with *actual tests* :—

$$t = [t_1^{12} - t_2^{12} + t_3^{12}]^{\frac{1}{12}}$$

where,

t_1 = contract inside absolute temperature F.;
t_2 = contract outside " "
t_3 = existing " "
t_4 = estimated inside " "

The curves shown in fig. 9 are worked out from the above formula for reference under different contract conditions.

The time factor is an important one, as will be seen from fig. 10, which is from an actual test of a large public building warmed by steam-heated accelerated hot water. The upper curve shows the inside temperature and the lower curve the outside temperature.

Automatic Regulation of Heating Apparatus.

It is very desirable that with hot water or steam installations some arrangement should be made for automatic regulation of the room temperatures independent as far as possible of the human element. The chief drawbacks to regulation of this description have been—firstly, expense; secondly, complicated apparatus requiring skilled attention to ensure the proper efficiency of same. Recently, however, the tendency has been for each room to have its independent automatic regulation instead of being part of a complete system of automatic apparatus, and this tendency has gradually evolved a simpler form of apparatus, some of which are being used with great success at the present time.

In addition to the independent room regulation it is also desirable to have further regulation of the boiler independently, not based on the water temperature of the boiler, but on the variation of the outside temperature; and experience proves that in this country it is preferable to regulate the boiler by means of the outside temperature than by the temperature inside the building.

The Thermutator automatic boiler damper regulator combined with outdoor thermostat, made by *The Drayton Regulator and Instrument Co., Ltd., West Drayton*, for hot-water boilers, operates the draught door in accordance with variation of outdoor temperature. The boiler thermostat is arranged above the boiler in a tube, and can be isolated by closing a cock.

The 'Drayton' A.M. Regulator.

(*The Drayton Regulator and Instrument Co., Ltd., West Drayton.*)

This regulator, fig. 11, is operated by water pressure. The pilot valve is actuated by metal expansion.

The fig. shows the regulator applied to the control of a water heater, but in place of the diaphragm operated valve, diaphragm operated dampers or other devices may be substituted.

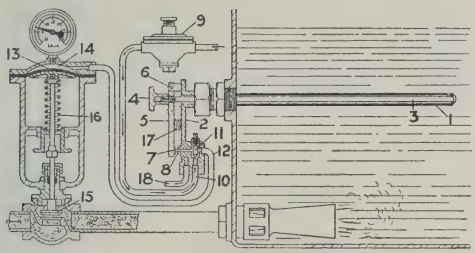


FIG. 11.—'Drayton' A.M. Regulator.

A modified form of this type of regulator is also made for the control of room temperatures.

The great power obtained by the 'Drayton' pilot valve system enables the largest valves or dampers to be operated without loss of accuracy.

HOT WATER SUPPLY.

Hot water supply can be divided into three classes:—

1. Domestic hot water supply suitable for ordinary houses.
2. Hot water supply for institutions, hotels, etc.
3. For industrial work.

Domestic Hot Water Supply.

The general apparatus for this purpose in an ordinary house is a boiler behind the range, a cylinder in a convenient position, and a supply tank. Points which should have attention are as follows:—

Boiler and its flues should be so fixed in the range as to be easily cleaned; also it should have an emptying tap.
Cylinder should be as near the boiler as possible, so as to ensure an easy circulation.
Feed tank should be in an accessible position, and protected from frost.

The fuel efficiency of a range boiler is low, and experience proves that even in quite small houses an independent boiler is more economical. It will be found that the major portion of the fuel used in a range is taken by the boiler, and not by the ovens required for cooking.

Institutions and Hotels Hot Water Supply.

For this class of work where steam is not required for other purposes, independent hot water boilers are the most economical. The type of boiler will depend on the nature of the water; where the water is hard it is desirable to have the boiler of as simple a form as possible, with ample provision for cleaning out, a bolted top being very desirable in this connection. In some cases it may be found necessary to either soften the water or put in a system of secondary heating, in which case the water in the boiler is circulated through a coil which heats the water in a secondary manner. Where the water is not hard, the sectional boilers 'Bower-Barffed' have proved very successful and highly efficient. In domestic supply it is desirable that the cylinder, which would usually be of a horizontal type with dome ends, should be as near the boiler as

possible ; from this cylinder, secondary circulations of considerable length may be taken to the various parts of the building, to supply the different fitments. In arranging hot water supply circulations and draw-offs, it must always be remembered that where there is a branch off the circulation, and the tap on the end of that branch is open, water will flow both ways, both from the flow direction and from the return direction, the return, therefore, should connect to the cylinder at a high point. This is most important, and many failures have been due to neglect of this necessary arrangement.

In many cases steam is required, especially in institutions and hotels, for different purposes, such as cooking, etc. ; in such cases, instead of having a separate hot water supply boiler, it is desirable to heat the water by steam. This is done by means of a calorifier · in principle it is simply a coil of steam pipe enclosed in a vessel of water, the steam being condensed and the latent heat of the same being conveyed through the coil to the water outside. This system also lends itself to thermostatic control. For this class of work it is very necessary that the pipe from the supply tank to the storage cylinder be of ample size. In some installations of a large size it is desirable to do away with the supply tank and simply keep the installation full by means of a pump, the pump automatically starting on a tap being opened. This system has many advantages, but is not so well known as the usual one of having a supply tank.

Westinghouse Lavatory Water Heater.

Where a supply of steam is available this heater affords a convenient means of obtaining hot water instantly without the use of hot-water tanks.

Referring to fig. 12. steam enters by a pipe connection B, which is in direct communication with chamber D. Cold water enters by pipe connection A which communicates directly with chamber C. Valves E and F are kept closed by springs G and H, when the handle J is in the ' OFF ' position.

When the handle is moved to position ' COLD,' the bell crank K pushes open valve F, and cold water only flows through the passage L to the space between the tubes N and M, and thence by pipe connection O to the lavatory basin.

On moving the handle further over to the position 'HOT,' the steam valve E is also opened by the bell crank K pushing against its stem, and steam then flows by passage P to the interior of tube N, and thence to the atmosphere by pipe connection R.

The heat of the steam being transmitted through the walls of the tube N to the water as it flows down the narrow space between the tubes M and N, the water is hot by the time it reaches the outlet O, while the steam in tube N is partially condensed before issuing from outlet R. The spring S returns the handle to the ' OFF ' position.

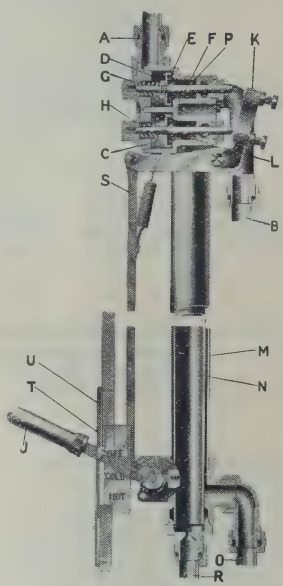


Fig. 12.

Method of ascertaining Heat necessary for a given Supply of Hot Water.

The basis of the calculations is that one British Thermal Unit will raise the temperature of 1 lb. of water 1° F. This figure varies a little according to the temperature of the water, but the variation is so small that for the purpose of hot water supply calculation, it may be neglected.

On the above basis we get the following simple formula—

$$B = \frac{G \times 10 \times T}{H} + R,$$

where,

B = British Thermal Units per hour ;
G = gallons of water ;
T = degrees it is required to raise the temperature of water ;
H = time in hours allowed for obtaining the required temperature ;
R = radiation losses from pipes and cylinder, per hour.

Example.

It is required to raise 100 gallons of water from 40° to 150° F. in 2 hours; then

$$B = \frac{100 \times 10 \times 110}{2} = 55{,}000.$$

With regard to radiation losses, we may assume the cylinder and pipes loss to be, say, a mean of 10,000 B.Th.U. per hour; the total B.Th.U. per hour would then be

$$55{,}000 + 10{,}000 = 65{,}000.$$

It is assumed that the circulating pipes and cylinder would be covered with non-conducting material to conserve the heat.

A boiler will transmit on an average from 2,500 to 7,000 B.Th.U. per hour per foot of boiler surface, 2,500 being for secondary flue surface, and 7,000 for fume firepot surface. An average of 5,000 may be taken for an ordinary independent boiler with normal firing. A tube in a well-designed calorifier will transmit from, say, 8,000 B.Th.U. per foot at 1 lb. pressure to 37,000 B.Th.U. at 50 lbs. pressure. The pressure of the water surrounding the tubes has quite an appreciable effect on transmission, apart from the temperature of the steam.

Apart from the usual hot water supply in a hospital, a special sterilised supply is required for surgical work in the operating theatres; steam is generally used for this purpose, and the water is heated to sterilised temperature, or generally boiled, and the apparatus so arranged that the sterilised water does not come into contact with unsterilised water or air before it is delivered at the taps for the use of the surgeon.

Hot Water by Gas.

Gas is often used for the purpose of heating water, especially in small installations, and a very usual application of this, in an ordinary size house, is to connect up the gas-heating boiler to the circulating pipes, so that during the summer when the range is not required (the gas-cooker being used for cooking) the gas-heater boiler can be used when desired. Apart from the geysers, etc., the systems of heating the water are arranged in a general manner, as regards arrangements of circulation, etc. The amount of gas necessary is on the basis of the calorific value per cubic foot. The average calorific value of town gas is, say, about 500 B.Th.U. per cubic foot. Assuming the boilers of the heaters to have an efficiency of 80 per cent., this would mean that 400 B.Th.U. of the 500 should be actually transmitted to the water, so that, taking the amount of water to be heated as mentioned above, namely, 100 gals., to be raised 110°, which requires 65,000 B.Th.U, this divided by 400 will give 162·50 cub. ft. of gas necessary.

Heating Water by Electricity.

For domestic uses surgical work, industrial, and other purposes, a method of heating water by electricity is very handy; but there is a disadvantage, it is very expensive if a very large quantity of water is required; this will be seen from the following:—

The electrical Board of Trade Unit contains 3,411 B.Th.U., and as the efficiency of electrical heating apparatus is very high, it may be considered that 3,000 B.Th.U. of the 3,411 would be transmitted to the water. Again, taking the quantity mentioned above, the amount of electricity required would be 65,000 B.Th.U. divided by 3,000, which gives us 21·7 units of electricity required. With regard to the type of heater, one very handy form, and one easily replaced, is the 'Immersion Heater.'

Industrial Hot Water Supply.

Hot water is required in a great many industries, and the method of heating the same is generally similar to the foregoing methods, although, very often waste heat of the furnaces in various industrial processes can be utilised for heating the boiler. For some requirements such as bottling, washing, etc., where it is desirable to have the water under considerable pressure the closed pump system lends itself to this class of work. (*J. Roger Preston.*)

Heating and Drying.

Not infrequently rooms and chambers are found provided only with steam pipes, which have to do duty at night and in the day time in order to keep pace with requirements Such rooms

are very bad drying places, and very slow into the bargain because the essential air current is lacking, and failure attends the introduction of a fan under these circumstances, owing to the influx of cold air following its use and lowering the temperature. To be effective the air must be raised to the required temperature before distribution and should act as a vehicle for the conveyance of the heat to and about the drying rooms or chambers.

To derive the maximum benefit from the use of exhaust steam, drying rooms of considerable size should have two or more divisions, as by this means operations may commence in one section soon after starting the engine, and conveniently proceed so that the operators may work in the cool part and finish up within a period of time to coincide with shutting off steam.

The proper temperature to effect rapid and perfect drying varies considerably, and depends upon the nature and texture of the material to be treated. Provided, however, the warm air current is fairly proportioned, then mechanical drying will considerably enhance the value and quality of the work, and vastly expedite and cheapen the process.

Drying Room Temperatures.

Material.	Room Temperature.	Time to dry.
	Mean ° F.	Hours.
Stains and varnish (caskets, pianos, &c.)	110	7
Shirts, flannels, &c. } In drying machine with air circu-	120	½
Blankets } lated internally	120	⅔
Cement sacks, on drying horses	120	1
Candied peel, on trays	165	2
Potatoes (steamed), on trays	170	6½
Cabbage and carrots (raw)	140	4½
Apples (sliced) (raw), on trays	175	6½
Hops, on floor in usual kiln	130	12
Retted flax on racks, lightly spread	120	10
Leaf tobacco on racks, lightly spread	135	12
Stem tobacco	180	12
Cinematograph films, on wooden drums	80	½
Millboard sheets, hung from clips	95	10
Ramie fibre, on perforated table	140	10
Earthenware plates, in revolving stove	110	8
Basils and skivers, on frames	75	48
Heavy hides from liquors, on poles { 1st.	atmospheric	100 to 200
{ 2nd	70	72
Bone glue (thin sheets), on wire trays { 1st	70	100 to 124
{ 2nd	80	
Wool after scouring, on perforated floor	150	4
Toilet soap in ribbons (Cressonnieres' Patent)	90	½
Hanks, hung on poles	120	2
1½-in. oak or hardwood, piled in stacks	90 to 125	20 days of 10 hours
2-in. to 3-in. oak or hardwood, piled in stacks	90 to 125	3 to 4 weeks.
The following are in tunnel dryers:	Tunnel inlet.	Hours to dry.
Common bricks of easy-drying clay, on trucks	170	30
1-in. air-dried, soft woods, piled on trucks	175	72
1-in. green soft woods, piled on trucks	175	120
Casein, on trays	140	8

(*George H. Gill, 'Engineering,' May* 4, 1923.)

SECTION XXXVIII

PART II

VENTILATION.

(Revised by J. Roger Preston, M.I.Mech.E., M.R.S.I., Past President of the Institution of Heating and Ventilating Engineers.)

Amount of Air necessary for Ventilation.

Whenever possible, it is desirable to provide 3,000 cubic feet of air per hour per person for halls, theatres, etc. Owing to the restrictions of cost and building construction, the sizes of the ducts necessary for this amount are rarely attainable, so a more usual figure is 1,800 cubic feet of air per hour per person. The L.C.C. regulations give 1,000 cubic feet of air per hour per person as the minimum amount of air required in buildings licensed by them.

Assuming a building in which there are, say, 3,000 people, and taking the L.C.C. requirements of 1,000 cubic feet per hour, this means about 830 cubic feet of air per second. The velocity permissible in the ducts will depend to some extent on their length, but 15 to 20 feet per second may be taken as a good maximum average. This means that the inlet and distributing ducts should have an area of not less than 50 square feet. As a streamline section must be maintained, it follows that where a change of direction in a duct of this size takes place, considerable space is taken up, and unless provision has been made for same, difficulties may be encountered. Steelwork is often a very important factor, especially in connection with the vertical subsidiary ducts, and is very difficult to modify; this trouble can only be avoided by close collaboration between the various parties concerned in the early stages of the building.

A very important point to bear in mind in the general planning is to allow plenty of room for the engineering equipment and to arrange that every part shall be easily accessible to the engineer in control of the plant. Experience proves that this is most essential, but is often overlooked or sidetracked to make room for something else.

The aim should be to make a system of ventilation as noiseless as possible, though this may not be necessary in industrial work. A good deal can be done by attention to details, but the fundamental factor is to keep low velocities, enabling the fans to run at slow speed, thus saving power. The space required and the initial cost will probably be more than for a system having high velocities, but it cannot be too strongly emphasised that in good class work *low air velocities and slow-running fans are essential to successful ventilation.*

The foregoing statement applies whatever system is adopted, and it must be remembered that *high air velocities and high-speed fans are a source of trouble and expense*, however attractive it may appear at the time to install small trunks that are easily adapted to the building.

Distribution of Air.

Convection currents are a most important factor in ventilation, and should be closely studied. This can conveniently be done either by balanced mica vanes or by blowing air through HCl and NH_4OH, the vapours so formed causing, when mixed, a dense white smoke (not heated). When these convection currents are known it can then be decided whether natural or mechanical methods will be required to give satisfactory ventilation and the best positions for the introduction of the air.

If natural ventilation is utilised, the air should be introduced at low level, preferably behind radiators, and allowed to find its way out at ceiling level.

Three methods of mechanical ventilation are employed, the particular method to be used depending on circumstances.

(*a*) *Mechanical extract* only. Inlets are provided at a low level, preferably behind radiators, the air being extracted by means of a fan at high level. This method is commonly employed in halls seating under 1,000 people, and similar buildings.

(*b*) *Mechanical inlet* only. Inlets are provided at a low level or about 6 feet above floor level. A common distribution is one-third of the total area of gratings at low level and two-thirds at 6 feet up. Outlets are provided through the walls at high level or through the ceiling. This method is employed when fresh air from outside is not otherwise obtainable.

(*c*) *Both mechanical inlet and extract.* The extracts should be as (*a*) and the inlets as (*b*). This method is used for large halls and particularly difficult problems, such as basements and enclosed rooms. The amount of air extracted mechanically is usually 75 per cent. of the air introduced by the inlet fan.

Where circumstances will allow, it is often found an advantage to arrange the distributing ducts, inlets, and outlets in such a way that the direction of air currents in the room can be reversed for different atmospheric conditions and temperatures.

Air-Filtering, Washing and Conditioning.

It is always desirable and frequently essential that the air should be as free from dust and impurities as possible. This dust can be removed by filters and air washers. There are several types of air filters obtainable, but they can be divided into two main groups:

(*a*) *Dry filters.*—These consist of a series of perforated plates arranged so that the perforations are staggered, the plates being covered with a sheet of wadding on the up-stream side for the better collection of the dust. The filtering effect is dependent on the number of plates. To clean, the filter unit is detached from its frame and shaken.

(*b*) *Viscous fluid filters.*—In these, the air passes over a series of plates or troughs coated with a viscous fluid to which the dust adheres. To clean, the filter unit is washed in strong soda water, dried, and then dipped in the viscous fluid and allowed to drain before replacing.

Air washers commonly used consist of a series of fine sprays through which water is pumped, the air mixing thoroughly with this finely divided spray, afterwards passing over a series of eliminator plates which collect the surplus water together with the dust.

The use of dry or viscous filters removes the dust without altering the temperature or humidity of the air, while the air washer varies both the temperature and humidity, thus making it unsuitable for certain purposes although it is frequently advantageous, especially when applied to the ventilation of halls and theatres.

The endeavour to produce a pleasant and healthy atmosphere inside at all times of the year whatever the outside conditions may be, has received a good deal of attention of late years, and in this connection a term *Effective Temperature* has been brought into use. The 'Effective Temperature' is influenced by three factors: (1) Dry bulb temperature; (2) Wet bulb temperature; and (3) Velocity of the air. The most suitable 'Effective Temperature' may vary, according to circumstances, from 60° to 70° F. If, however, it were 90° F. outside, it would be very unwise and actually uncomfortable for people to go into a temperature of, say, 60° F. or even 70° F. In this case, the reduction of the dry bulb temperature by about 10° F., together with a perceptible air movement, would probably give rise to a marked degree of comfort.

Air Cooling.

Air can be cooled by an air washer, but the relative humidity will always rise. In a washer using recirculated water, the heat content will not be substantially changed although a certain cooling effect can be obtained with a consequent rise in relative humidity. The wet bulb temperature is unaffected by this method, so that the cooling is limited by the initial wet bulb temperature and will in practice always be a few degrees above this. Actually it will be found that the dry bulb temperature can be reduced to, say, 75 per cent. of the difference between the wet and dry bulb temperatures.

When artesian well water or some other water having a temperature of 52° F. is obtainable, the air can be cooled to about 57° F. with a consequent reduction in the vapour contents of the air should the initial dew-point be above 57° F., which is most probable in hot weather. The air will then be saturated.

When air is required with a lower temperature than can be obtained by the air washer, refrigeration will have to be employed. Cooling air is far more expensive than heating, owing to the condensation of the moisture in the air. For general ventilation, the air may be cooled from 10° F. to 15° F., but for industrial purposes, when it is required to have complete control over the relative humidity and temperature, it is usual to cool the air to 35° F. The amount of brine to be circulated should be calculated on a drop of 3° F. The specific heat of brine may be taken as 0·82° F., and in calculating the heat required for the reduction in temperature the condensation of the vapour must not be forgotten. The heat in British thermal units required, divided by 12,000, will give the size of plant required in tons of refrigeration per day of 24 hours.

The heat exchange in cooling or heating air over a given range of temperature is not materially different, provided the dew-point is not reached. Should, however, the initial humidity be such that the dew-point is reached in the cooling, another very important factor is introduced, a portion

of the aqueous vapour being condensed and giving up its latent heat in the process. The British thermal units involved in the heat exchange may be ascertained by the following, which is accurate enough for all practical purposes and shows the various points involved :

$$U = \overbrace{L_c (W_1 - W_2)}^{\text{SECTION A.}} + \overbrace{S_v(T_1 - T_2) W_1}^{\text{SECTION B.}} + \overbrace{S_a (T_1 - T_2)}^{\text{SECTION C.}},$$

where,

T_1 = Initial dry bulb temperature;
T_2 = final ,, ,, ,,
W_1 = weight of aqueous vapour per lb. of air at initial temperature T_1;
W_2 = ,, ,, ,, ,, final ,, T_2;
L_c = mean latent heat of vaporisation or condensation at initial dew-point, and T_2;
S_v = specific heat of vapour;
S_a = specific heat of air;
U = B.Th.U.'s to cool 1 lb. of air from T_1 to T_2;

Basic figures to be taken for this and other equations :

S_v = specific heat of aqueous vapour = 0·44;
S_a = ,, ,, air = 0·2415 *;
L_c = latent heat of vaporisation varies from 1073·4 B.Th.U.'s per lb. ar 32° F. to 1052·3 B.Th.U.'s per lb. at 70° F.

Section A is the heat required to be extracted to condense the water vapour.

Section B is the heat required to be extracted to reduce the temperature of the water-vapour contents of the air and is so small that it can usually be neglected.

Section C is the heat required to be extracted to cool the air from T_1 to T_2.

Velocities of Air Flow for Mechanical Ventilation and Flow of Air Through Ducts.

Where silence is essential, the velocities of flow of air in ducts must be kept low, the ideal velocity being from 10 to 15 feet per second, and on no account exceeding 20 feet per second. For factory work speeds up to 30 feet per second are allowable, but this velocity should not be exceeded. In general, the outlets to the rooms (if at low level) should be of such an area that the air issues at a velocity of from 3 to 5 feet per second, as such velocities do not give rise to a sensation of draught. In certain circumstances, however, such as during hot weather, it may be found desirable to direct a flow of air across a room at a velocity of 10 to 15 feet per second, this movement giving a feeling of coolness. (See 'Effective Temperature' p. 2618.) The following table gives the permissible velocities for different parts of the ventilation system. The lower figure refers to ventilation where silence is desired, and the higher gives the maximum permissible for factories, etc.

Fresh air intake	500 to 800 ft. per min.	Main ducts	800 to 1,200 ft. per min.
Heating battery	800 ,, 1,500 ,, ,,	Branch ducts	600 ,, 900 ,, ,,
Filter	150 ,, 350 ,, ,,	Registers	180 ,, 300 ,, ,,
Washer	300 ,, 500 ,, ,,		

The flow of air through a building resembles an electrical circuit, the inlets, outlets, and ducts acting as the resistance, and absorbing some of the aeromotive force. *To ensure an even flow in various ducts requires considerable care*; the resistance must be low, as the initial force in ventilation is small, usually equal to about 0·25 ins. of water, or 17 pneumatic units. In natural ventilation it is much less than this.

In order to obtain an even distribution of air throughout the system, the design of the ducts is very important. Changes in direction and section should be as gradual as possible, particular care being taken to avoid sudden changes. The finish of the ducts should always be of the best, as roughness and irregularities increase immensely the resistance to the flow of air. The ducts are usually made of galvanised iron or in builders' work. When made by the builders the ducts should be lined with glazed bricks, or, if this is too expensive, finished with Keene's cement or some similar hard, smooth finish. It is always advisable to whitewash the latter, as the presence of dust is then easily noticed and more likely to be removed. The importance of the foregoing will be appreciated when it is realised that in mechanical ventilation where silence is essential the fan should not work against a resistance exceeding 1in. W.G. and in other general work it is advisable not to exceed 2in. W.G., except for dust extraction and similar industrial processes.

* 0·238 was at one time the accepted figure for specific heat of air and will be found in many textbooks, but 0·2415, or sometimes 0·2400, is now accepted as the more correct value.

RESISTANCE OF FLOW OF AIR PER 100 LINEAL FEET OF CIRCULAR GALVANIZED IRON TRUNKING.

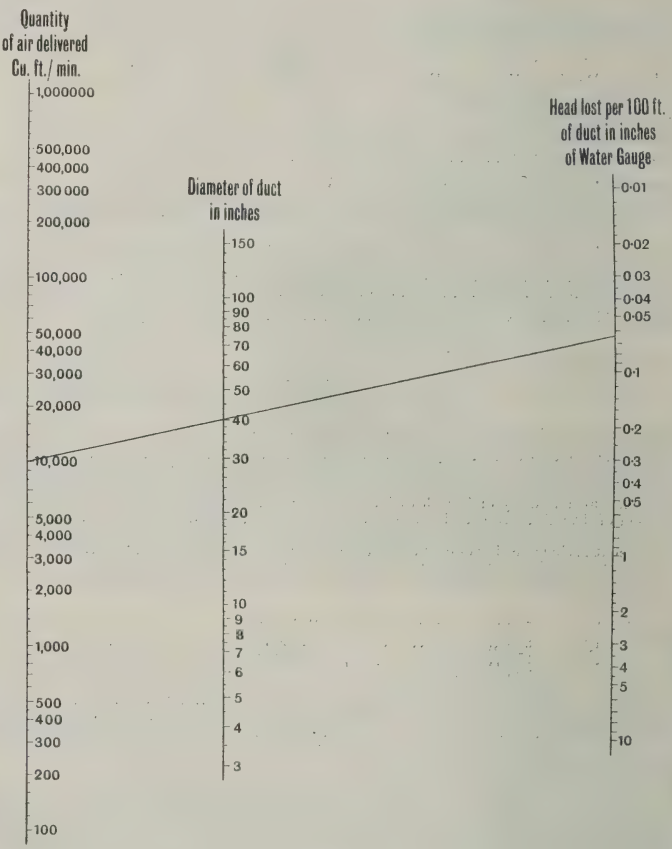


FIG. 1.

EXAMPLE: To find the loss of head in a 40-in. diam. duct carrying 10,000 cu. ft. of air per minute, lay a straight edge as shown connecting 10,000 cu. ft./min. and 40-in. diam., reading the loss of head 0·064 ins., W.G. per 100 ft., on the right.

There is yet a need for research work in connection with the flow of air in ducts, but the following will be found a useful formula for the flow of air through galvanised iron pipes :

$$C = 39d^{2\cdot5} \Big/ \left(\frac{L}{H}\right)^{\frac{1}{2}},$$

where,

C = cubic feet of air per minute ; L = length of pipe in feet ;
d = diameter of pipe in inches ; H = drop of pressure in inches of water.

The values derived from this formula are shown more conveniently in fig. 1.

For resistance in ducts of other materials than galvanised iron, multiply the result obtained from the diagram or by the above formula by the following factors :

Smooth concrete . . . 1·5 Brickwork . . . . . 2·0

When arriving at the total resistance in a duct, allowance must be made for bends and loss due to entry and egress at registers. The loss of a sharp elbow varies from 30 to 60 diameters, a smooth bend 5 diameters and a usual register 80 diameters.

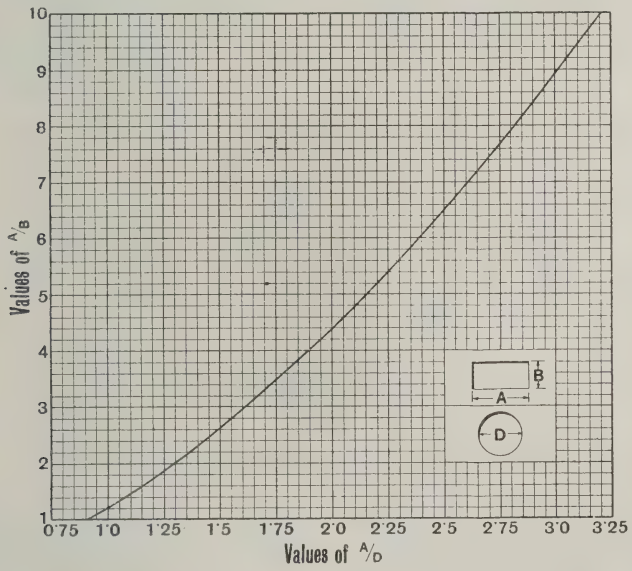


FIG. 2.

Diagram for obtaining equivalent Diameter of Round Duct carrying the same amount of air for the same loss as a Rectangular Duct.

To obtain the total head against which the fan has to work, the velocity head must be added to the resistance in the duct ; this is obtained from the following formula :

$$h = \frac{6D \cdot v^2}{gk},$$

where,

v = velocity in ft. per sec. ; h = velocity head in W.G. ;
g = 32·2 ; D = density of air at temperature required
k = density of water = 62·4 ; = 0·0749 for dry air at 70° F.

All the above is based on circular pipes. In the case of rectangular pipes, find the equivalent

circular pipe to give equal discharge for equal resistance from the following formula, and then use fig. 1 to find the resistance to flow :

$$D = \sqrt[5]{3{\cdot}25 \frac{A^3B^3}{A+B}}$$

where

A = long side of rectangular pipe ; B = short side of rectangular pipe ; D = dia. of equivalent round pipe.

This equivalent diameter can conveniently be found by means of fig. 2.

Movement of Air by Fans.

The choosing of a suitable fan is a most important factor in any scheme of mechanical ventilation.

There are two distinct types of fans used for this work : the propeller and centrifugal, one similar to a ship's propeller, and the other more like a paddle wheel.

A propeller is a very efficient and useful fan if used in the proper way, that is, either with free inlet and outlet, as, for instance, when fixed in a window or wall without any trunking attached, or with short trunking of ample size, causing very little resistance head.

In the average building a fan must run at a low speed to avoid noise. At this speed the efficiency of a propeller fan falls off very considerably against resistance, as will be seen from

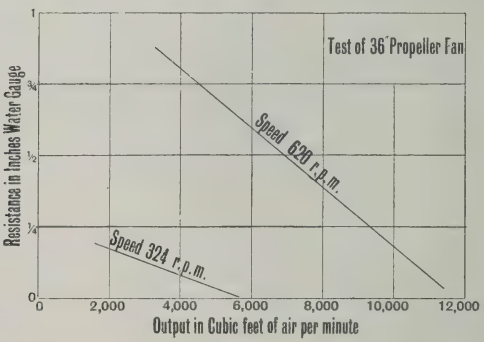


FIG. 3.

Diagram showing how the volume of air delivered by a propeller fan falls off with increased resistance.

fig. 3 (plotted from tests by the author). For factory and similar work where noise and extra power do not matter, the fan may be run at a much higher speed, in which case it can be used at a higher water gauge, see fig. 3, but even then the economical limit is soon reached. It must be remembered also that as the quantity of air decreases, due to resistance, the power increases.

The centrifugal fan can be used up to any reasonable pressure, but for general ventilation, where absence of noise and economy of power is desired, it is good practice not to exceed about 1 in. total head (resistance head + velocity head), the filters, heaters, ducts, etc., being designed accordingly.

When the resistance is increased, the amount of air falls—so also does the power taken. At a lesser resistance the volume of air increases—so also does the power.

Should a centrifugal fan and motor be designed for a given resistance, and then the resistance turns out to be less than expected, there is a great danger of the motor being overloaded, unless a specially large one has been installed. To meet this condition, a special fan has recently been put on the market, and the author's experience of this fan so far has been very satisfactory.

The Fan Standardisation Committee of the Institution af Heating and Ventilating Engineers, after devoting some five or six years to the study of the subject, has issued a very comprehensive report on the testing of fans, terms to be used, and method of cataloguing, etc.* to which the reader is referred.

* ' Report of the Fan Standardisation Committee, 1927 ' (price 2/6). Office of the Institution, 12 Russell Square, London, W.C. 1.

Movement of Air other than by Fans.

As the air in a flue such as a chimney or outlet duct from a warmed room is at a higher temperature than the outside atmosphere, its density is less, and the tendency is for it to rise. The resulting velocity of flow varies within wide limits, but is theoretically given by :

$$V = 480 \sqrt{\frac{h\,(T - t)}{t + 460}}$$

where,

V = velocity of air in feet per minute ; T = temperature of heated air °F ;
h = height of flue in feet ; t = outside temperature °F.

The actual velocity is considerably lower in practice, and the result from the above formula should be multiplied by 0·25 to 0·6, according to size, shape, and materials.

In addition to the movement of air due to difference of temperature, the wind has a considerable aspirating effect on a properly designed ventilator.

The following data are taken from some tables in the ' Robertson Ventilation Data Book ' :*

Temperature Difference, Fahrenheit.	Height of Ventilator above intake.	Wind Velocity in Miles per hour. 2	6	10
		Velocity of air exhausted, in feet per minute.		
0°	10′	61·5	185	309
	20′	61·5	185	309
	30′	61·5	185	309
	40′	61·5	185	309
	50′	61·5	185	309
	100′	61·5	185	309
5°	10′	139	262	385
	20′	170	294	416
	30′	195	318	441
	40′	215	337	460
	50′	233	355	479
	100′	300	422	545
10°	10′	170	294	416
	20′	215	337	460
	30′	250	372	496
	40′	279	401	525
	50′	304	426	549
	100′	402	526	650
20°	10′	215	337	460
	20′	279	401	525
	30′	326	449	572
	40′	367	491	614
	50′	402	526	650
	100′	545	670	792
30°	10′	250	372	496
	20′	326	449	572
	30′	386	509	633
	40′	437	560	684
	50′	481	604	728
	100′	655	779	902

Heating units such as gas jets, electric heating, steam or hot water coils, are sometimes used for the purpose of causing an up-draught. These should never be used if any other method is available, as they are very inefficient and costly in up-keep, costing, in the case of steam coils for example, on an average six times that of moving air by fans. (*J. Roger Preston.*)

* H. H. Robertson Co., Pittsburg, Pa., U.S.A.

Blowers and Exhausters.

STURTEVANT BLOWERS AND EXHAUSTERS.

(*Sturtevant Engineering Co. Ltd., London, E.C.* 4.)

The design of a fan depends upon the pressure against which it has to discharge. A blower designed for low pressure is not suitable for high, and *vice versâ*. Figs. 4 to 10 show various types.

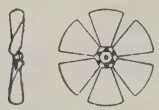


FIG. 4.

AIR PROPELLER.

For moving large volumes of air from one part to another where the resistance is low. Used for ventilating and exhausting.

MULTIVANE CENTRIFUGAL FAN.

For moving large volumes of air against 1 to 4 ins. water gauge. Fan wheel has shallow blades with corrugations to prevent lateral slip. Used for ventilating, drying apparatus, mechanical draught for boilers, etc.

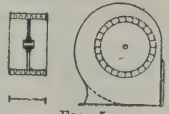


FIG. 5.

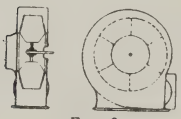


FIG. 6.

STEEL PLATE CASED CENTRIFUGAL FAN.

For air pressures up to 8 or 10 ins. water gauge. The square section steel plate casing is used, but the wheel has deeper blades and side plates (shrouds). Used for dust exhausting, conveying, etc.

CAST IRON CASING FAN.

For obtaining higher pressures up to 15 ins. water gauge, the casing must be more carefully designed and preferably made in cast iron. The wheel is narrower and deeper than that for lower pressures. Used for blowing forges, furnaces, etc.

FIG. 7.

FIG. 8.

HIGH PRESSURE FANS.

For blowing cupolas, etc., requiring air pressures from 12 to 30 ins. water gauge. The scroll must be most carefully designed and the wheel have a greater number of blades. For sizes and capacities of this fan see page 2663.

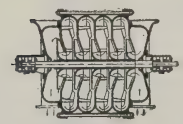


FIG. 9.

MULTISTAGE FANS.

For blowing and exhausting air at pressures up to 3 to 4 lbs. per square inch.

ROTARY BLOWERS.

These are used for many duties, for blowing and exhausting air or gases where pressures up to 5 lbs. per square inch are required.

FIG. 10.

POWER VARIATION OF A BLOWER AT CONSTANT SPEED.

Propeller Fans.—Power rises slightly with pressure; volume decreases rapidly as pressure rises.

Centrifugal Fans.—Power increases as pressure falls, and volume increases.

Rotary Blowers.—Power increases directly with pressure, volume remaining about constant.

The 'Wing' Fan.

(F. W Potter & Soar, Ltd., Phipp St., Great Eastern St., London, E.C. 2.)

The general arrangement of the blades of the Wing fan or air propeller is shown by fig. 11. It is claimed that the Wing fan, for a given amount of power, will move a greater volume of air than any other pattern, owing to the shape, angle, and curvature of the blades, and the method of securing them to the central boss. It has been found that an angle of 27½° to 30° is the best all-round angle for combined output and efficiency.

A distinctive and useful feature of the fan is the adjustability of the angle of the blades, the latter being readily set to any angle. If desired, the direction of the air current may be reversed, this reversal of the air current being effected without impairing the efficiency of the fan.

The blades of the Wing fan are usually made of steel, but for certain purposes they are made of copper, aluminium, or Muntz metal.

The fan wheel is mounted on a spindle running in bearings lined with best quality Babbitt metal, and the framing is constructed of cast iron or wrought iron according to circumstances.

The Wing fan is manufactured in all sizes, arranged with pulley for belt drive, or direct coupled to electric motor.

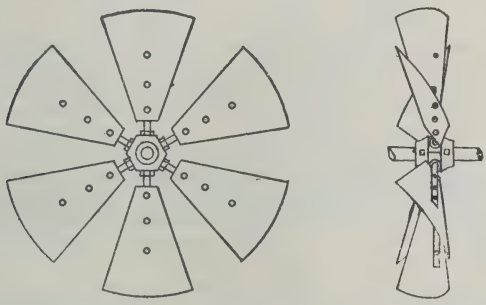


Fig. 11.

Economy of Power.—Where conditions permit of the choice between a propeller type and a centrifugal type of fan, the following rules are useful :—

(*a*) For open ventilation use propeller fans.

(*b*) Make main air duct same diameter as fan, or larger, avoiding acute or sharp angle bends.

(*c*) When small ducts and sharp bends are unavoidable use centrifugal fans.

(*d*) Use large fans at as low a speed as will overcome whatever resistance is in the way of free passage of air.

Example.—A 10 ft. Wing fan at 90 revs. per min. delivers 71,000 cu. ft. of air per min., absorbing 1·2 h.p.

A 5 ft Wing fan at 720 revs. per min. delivers 71,000 cu. ft. of air per min., absorbing 19·2 h.p. Volume for volume, therefore, the smaller fan uses 16 times the power absorbed by the larger fan.

(*e*) Use two fans of the same diameter at low speed instead of one at double the speed.

Example.—A 48 in. Wing fan at 600 revs. per min. delivers 30,200 cu. ft. of air per min., absorbing 3·6 h.p.

Two 48 in. Wing fans at 300 revs. per min. deliver 30,200 cu. ft. of air per min., absorbing ·9 h.p. together.

Volume for volume, therefore, one fan uses four times the power absorbed by two fans.

APPROXIMATE CAPACITIES OF 'WING' FANS (SMALLER SIZES).

Diam. of Fan Blades Ins.	Silent Speed.			For ordinary Ventilation.			For Factories and Engine Rooms.			Fast Speeds.		
	Revs. per Min.	Net H.P.	Cubic Ft. per Min.	Revs. per Min.	Net H.P.	Cubic Ft. per Min.	Revs. per Min.	Net H.P.	Cubic Ft. per Min.	Revs. per Min.	Net H.P.	Cubic Ft. per Min.
14	680	·012	850	1,090	·05	1,360	1,540	·13	1,930	2,000	·29	2,500
18	530	·02	1,410	850	·08	2,250	1,200	·22	3,170	1,600	·51	4,250
24	398	·034	2,510	640	·14	4,050	900	·40	5,680	1,200	·92	7,550
30	318	·052	3,900	510	·21	6,260	720	·63	8,850	960	1·50	11,800
36	265	·074	5,620	425	·30	9,000	600	·87	12,700	800	2·05	17,000
42	228	·10	7,680	364	·43	12,250	515	1·20	17,400	685	2·75	23,100
48	200	·14	10,100	318	·55	16,100	450	1·55	22,600	600	3·60	30,200
54	177	·17	12,700	283	·68	20,300	400	1·95	28,700	535	4·70	38,300
60	145	·19	14,300	255	·85	25,000	360	2·4	35,400	480	6·00	47,200

MOTORS FOR DRIVING VENTILATING FANS.

Too much cannot be said regarding the location of the motor. On account of the nature of the work it is frequently necessary to place the machine in an out-of-the-way place, although this does not necessarily mean that it must be in an inaccessible location. If it is so placed that it is difficult to care for, it will probably be neglected, with the result that the service will be impaired and the maintenance charge will be high. The placing of a motor in an inconvenient place because nothing else will go there, is false economy generally, and results in dissatisfaction. The question of motor efficiency in regard to fan drives is an important one and involves two factors—the efficiency of the motor when operating under normal speed, and the efficiency at other than normal speeds. Although the latter is, to some extent, influenced by inherent characteristics of the motor, the largest factor is the method of control, especially with direct-current motors. If, on account of its lower cost, a motor is selected, where adjustable speed is required, that will, when operating at normal speed, drive the fan at the maximum required speed, and it is necessary to operate for considerable periods at reduced speed, low efficiency will result.

The cost of the power lost in the armature external resistance will usually pay a handsome dividend on the additional capital required to install a motor and controller that, at normal speed, will drive the fan at the lowest speed required and beyond this point adjust the speed by field control.

The horse-power required to drive a centrifugal fan can be estimated from the following formula:

$$HP = \frac{5 \cdot 2 \times Q \times WG}{33{,}000 \times E}$$

where,

HP = horse-power of motor; Q = cubic feet of air per minute; WG = pressure of air in inches by water gauge; E = efficiency of fan, which is approximately 0·5 for a steel plate fan, 0·65 for Sirocco, and 0·45 for cone type.

The following mathematical relations are theoretically true for all centrifugal fans: Volume of air delivered varies as the speed; pressure of air delivered varies as the square of the speed; horse-power required varies as the cube of the speed. Hence a fan, which, when operated at 100 r.p.m. delivers 8,000 cub. ft. of air at ½ oz. pressure, requires 2 h.p., will, when operated at 200 r.p.m., deliver 16,000 cub. ft. of air at 2-oz. pressure and will require 16 h.p. It is therefore necessary, in selecting a motor for driving a fan, to figure the horse-power requirements at the maximum speed of the fan.

Reduction of the area of the inlet or discharge of a centrifugal fan will reduce the power required, whereas increasing the area of either will increase the power. Disc fans produce a relatively low pressure, and it is customary to figure their output on a basis of velocity. The relations between volume, speed, and horse-power for centrifugal fans hold good for disc fans. However, if the outlet of a disc fan is restricted, the pressure and the power required increase, and with a totally closed outlet the power needed is almost double that with an unrestricted outlet. (*'Power.'*)

Ventilators.

The Boyle 'Air-Pump' Ventilator.

In the latest patented forms of Boyle's patent 'air-pump' ventilators the wind, striking against and being deflected by curved vanes causes a suction effect which draws out the foul air and effects perfect ventilation. One form of the 'air-pump' ventilator, known as the *ridge* ventilator, is shown by fig. 12. The wind blows through the openings, and in passing over the curved plates causes a strong upward draught.

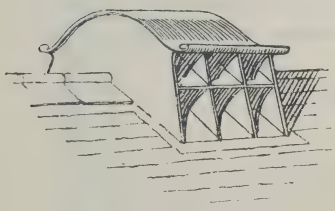


Fig. 12.

No. 175.

Fig. 13.

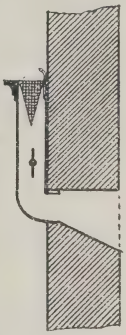


Fig. 14.

In the 'air-pump' ventilator (fig. 13) the curved vanes are set vertically.

For the necessary admission of fresh air to take the place of that extracted by the ventilator, Boyle's Air-Inlet Bracket, shown by fig. 14, may be adopted. The top of the inlet should be about 5 ft. 9 ins. above the level of the floor, and the outside opening should be covered with an iron grating.

The Plenum System.

This system (*The Sturtevant Engineering Co. Ltd.*) is well adapted for the economical warming and ventilation of workshops, large halls, theatres, and other large buildings. The

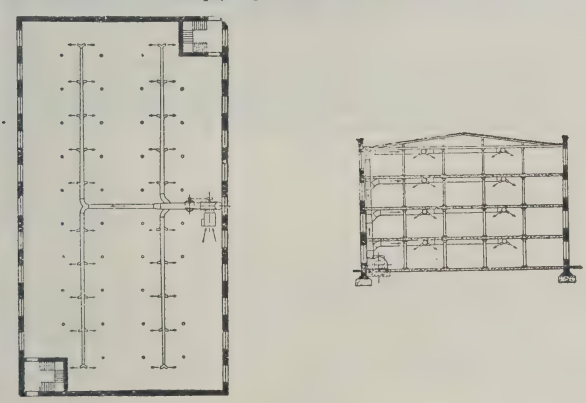


Fig. 15.

apparatus consists of a fan which draws in fresh air, delivering it through a heater where it is warmed to the required temperature, and then evenly distributed through a series of ducts to

every part of the building. The fan is usually of the steel-plate type driven by belt from line shafting, steam engine, or electric motor; the heater is built up with a nest of pipes for use with live or exhaust steam, hot water, furnace gases, etc. Ducts for distributing the air are constructed of galvanised sheet steel, masonry, wood, or other materials, and may be built in the walls or floors, or suspended from the roof or walls.

The system ensures even distribution of warmth throughout the whole building with proper ventilation. In the summer the heater can be eliminated and fresh air distributed throughout the buildings. Workshops can be quickly warmed up in the morning before work is commenced, by recirculating the air. The fan and heater can be installed at one central

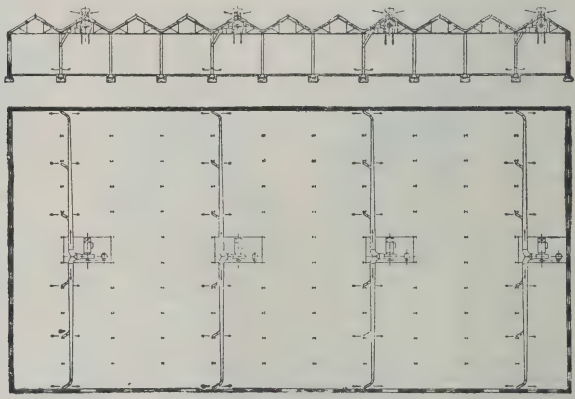


FIG. 16.

point either inside the building or outside, and they are under complete control of one man who can regulate the temperature of the air supply as required.

Two types of buildings equipped with this system are shown by figs. 15 and 16. Fig. 15 illustrates a building of several storeys, in which the apparatus is placed in the basement, one air main being carried up through the building with overhead horizontal distributing pipes on each floor.

Fig. 16 shows a large workshop of considerable height, in which the apparatus is divided into four units in order to keep down the size of the air piping. The sets of apparatus are placed out of the way in the roof, the air being distributed by overhead pipes and discharged at floor level through downtake pipes arranged alongside the column. An alternative arrangement would be to install one set of apparatus and connect it with each of the pipe mains.

SECTION XXXVIII

PART III

LIGHTING

(Revised by W. J. Jones, M.Sc., A.M.I.E.E.)

Terms in Illumination

International Candle.—This is the unit of light intensity standardised in Great Britain, the United States of America, and France (*Bougie decimale*), and is one-tenth the intensity of the standard 10 candle Harcourt Pentane lamp burning under specific conditions of barometric pressure and humidity.

The relationship between the International Candle and other units is shown in the following table:

Unit.	Is Equivalent to:		
	International Candle.	Hefner Candle.	Carcel Unit.
One International Candle .	1·0	1·1	0·104
One Hefner Candle . .	0·9	1·0	0·09375
One Carcel Unit . .	9·6	10·66	1·0

Mean Horizontal Candle Power.—As the intensity of a light source generally varies in all directions, this term is employed to denote the mean candle power obtained by averaging the candle-power readings measured in all directions on a plane passing through the centre of the light source and perpendicular to its axis.

Mean Spherical Candle Power (M.S.C.P.).—This is the mean of the candle powers measured in all directions.

Lumen.—The unit denoting the quantity of light, and is the amount of light flux contained in a unit solid angle of a sphere which has a uniform light source of one candle situated at the centre. A light source of one mean spherical candle emits 4π (12·57) lumens (see fig. 1).

Foot Candle.—The unit of illumination. It corresponds to the light density on a surface. If normal to its surface, every part of an area of 1 square foot receives light of an intensity of one candle; it is said to be illuminated to a density of 1 foot candle.

If a uniform point source of light of 1 candle intensity in all directions is situated at the centre of a sphere of unit radius, every square foot of the spherical surface will be illuminated to a density of 1 foot candle. Alternatively, it may be said that if one lumen illuminates an area of 1 square foot, the average light density will be 1 foot candle.

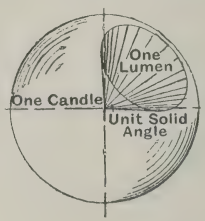


FIG. 1.

Lux or Metre Candle.—The Continental unit of light density is analogous to our foot candle unit. When an area of 1 square metre is illuminated by one lumen, the resultant illumination is 1 lux or metre candle. It follows that 1 foot candle equals 10·76 lux.

Brightness.—The brightness of a surface is the quotient of the luminous flux intensity in a given direction divided by the area of the surface projected on a plane perpendicular to that direction. For electric lamps and fittings this quantity is expressed in candles per square inch, while for large surfaces it is stated in candles per square foot.

Illumination.

Illumination, or the density of light received by an object, is measured in terms of foot candles, and can be shown to be directly proportional to the candle power and inversely proportional to the square of the distance from the light source, *i.e.* $I = \frac{C.P.}{D^2}$.

If a surface is oblique to the rays, $I = \frac{C.P.}{D^2} \cos \theta$, where θ is the angle subtended between the incident light and normal to the surface.

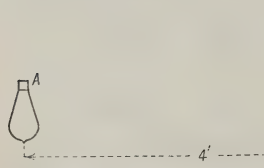


FIG. 2.

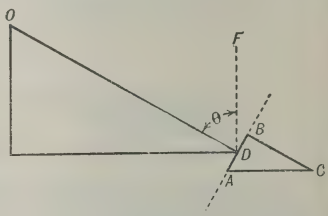


FIG. 3.

Example.—In fig. 2, if A is a lamp (electric or otherwise) giving 16 C.P. in a horizontal direction, the illumination at the point B, 4 ft. away, would be $\frac{16}{4^2} = 1$ foot candle, since the intensity of light varies inversely as the square of the distance. To get the normal illumination at any given point, the candle power in that particular direction must be divided by the square of the distance

to the point illuminated. As previously explained, if the surface illuminated is not at right angles to the direction of the light, the value of the illumination obtained, as above, must be multiplied by a reduction factor, taking into account the angle at which the rays strike. A beam of light coming in the direction of OD (fig. 3) falls upon a plane AB, illuminating it with a density of 1 foot candle, then the illumination on the plane AC, which intercepts the same amount of light as AB, would be less as the light is spread over a larger surface, the reduction being in the ratio of AB to AC, which is the cosine of the angle ODF. Thus the effective illumination on any plane at a given point will be:

$$\frac{\text{C.P}}{\text{distance}^2} \times \cos\theta$$

where θ is the angle between the direction of the ray and a perpendicular to the plane considered.

Brightness.

The brightness of an object under a given illumination depends on the amount of light reflected back by its surface, and this depends on the colour and the nature of its finish. The following table gives the approximate percentage of incident light reflected by various surfaces:

Approximate Reflecting Power of Surfaces.

	Per Cent.		Per Cent.
White, rough	82	Blue paper	25
Cream	80	Wood, dirty	20
White, smooth	70	Brown paper	13
White, printed matter (grey)	60	Brown, dark	4
Wood, clean	50	Black cloth	1·5
Yellow paper (light)	40	Black velvet	0·5
Yellow, dark	30		

Photometry.

As its name implies, photometry deals with the measurement of light. By comparing the unknown light source with a known standard and using the fundamental stated above—that is, that the intensity is inversely proportional to the square of the distance—photometric readings are reduced to the measurement of lengths. In a simple photometer the standard lamp and the light source to be measured are mounted on a horizontal bench with a movable screen placed between them (see fig. 4). The screen is then moved between the lamps until both its sides appear

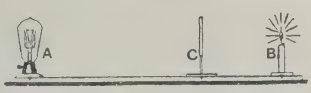


Fig. 4.

to have the same brightness. The distance AB and BC are then measured and the candle power calculated from the fact that:

$$\frac{\text{CP.A}}{\text{AC}^2} = \frac{\text{CP.B}}{\text{BC}^2} \qquad \therefore \text{CP.A} = \text{CP.B} \times \frac{\text{AC}^2}{\text{BC}^2}$$

While making such readings care must be taken that the screen is shielded from any light from other sources.

The Lummer-Brodhun photometer is a more accurate instrument in which prisms are used, so that on looking through a single eyepiece parts of both sides of the screen are seen superimposed; by this means very accurate comparisons of contrast can be made. With this instrument it is not the brightness of the surfaces that are balanced, but their contrasts, as seen in the two halves of the field. Fig. 5 shows a diagram of the Lummer-Brodhun photometer head, and also the appearance of the contrast field when viewed through the eyepiece.

The great difficulty in making candle-power measurements is the difference in the colour of various light sources, and for accurate work special means have to be adopted. One method employs colour filters, but variations in the light absorption with various lights of different complex colour composition renders this method rather unsatisfactory. Another method is to employ a flicker photometer such as the Simmance-Abady Flicker Photometer. This instrument is based on the fact that if a surface is illuminated alternately by two light sources it appears to flicker, due

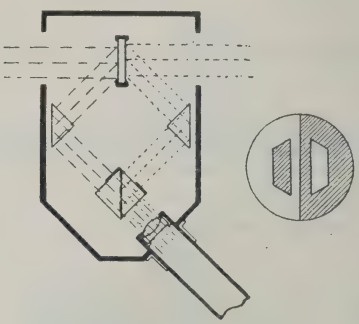


FIG. 5.

both to variations in light intensity and colour. By varying the speed of the alternations it is found that the eye is more sensitive to variations of intensities than to changes of colour; so by using a critical speed of exposure the surface assumes a neutral tint only affected by flicker due to variations in intensity, and so, under these conditions, the illuminations on the screen can be balanced in the usual way.

In the actual instrument these alterations are obtained by rotating a specially shaped disc between the light sources. The rotor is provided with surfaces that slope alternately to either side, so that at any instant during the rotation the portion in front of the eyepiece is only illuminated by one lamp.

INTEGRATING PHOTOMETER.

The sphere photometer is another form of instrument specially designed for measuring the mean spherical candle power of a light source. The apparatus consists of a large sphere with a whitened mat interior surface; the light source being placed in the centre. It can be shown that every part of the interior surface will receive the same amount of light by repeated reflections—in other words, the light is automatically averaged out; by placing a small window of diffusing glass in the side of the sphere this average intensity can be measured by the usual photometer method.

As mentioned, the effect depends on reflections in the interior, and so the window has to be carefully shielded from direct light from the source of light, also the readings must be multiplied by a constant depending on the nature of the internal surface.

See also British Standard Specification for Portable Photometers, No. 230, 1925.

Polar Curves.

Before suitable reflectors can be designed for a given source of light, or in fact before any real idea can be formed as to the characteristics of such a source, it is necessary to know how the light intensity varies in all directions. The polar curve is a very convenient way of showing these variations, and an example is given in fig. 6. The distance scaled along any line from the centre is proportional to the candle power in that particular direction. When handling light sources, the illuminating engineer needs information as to the nature of the light distribution complete with reflector. This data is recorded in a similar type of graph, and fig. 6 also shows how the light distribution has been completely changed by fitting a reflector; curve A shows the light distribution of a particular lamp alone, while curve B gives its distribution after a reflector has been installed. It will be seen that by means of the data from such polar curves, light can be properly controlled both as regards the design of fittings and installation for specific purposes.

Often it is necessary to compute the total lumens output from a polar curve, or more exactly the mean spherical candle power from which the lumens can be easily calculated. If the light source is symmetrical in its horizontal distribution and not too irregular in its vertical variations, this can be readily performed by taking readings at what are known as the Russel Angles; the mean spherical candle power being the average of such readings. Taking the lower vertical ordinate as our zero, readings should be taken at the following angles:

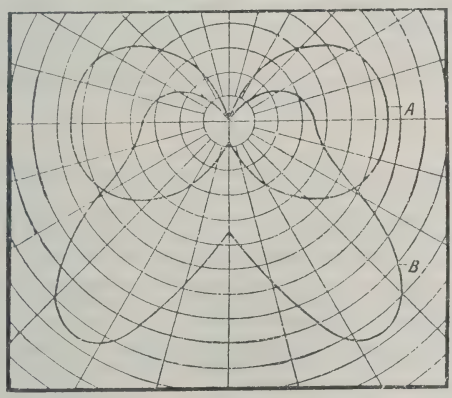


FIG. 6.

Degrees.				
25·8	60·0	84·3	107·5	134·4
55·6	72·5	95·7	120·0	154·2

As explained, the M.S.C.P. is the arithmetic mean of the ten readings; while if the lumen output is required this result has simply to be multiplied by 12·57.

Requirements of Good Illumination.

Illumination must be acceptable to the eye. We should be able to see objects clearly and with a minimum of fatigue. To accomplish this result certain conditions must be fulfilled.

(1) There must be sufficient illumination. Since objects are seen by means of the light which they reflect, more light must be thrown on dark objects than on light ones.

(2) Intensely bright lights in the field of vision should be avoided. The iris closes somewhat in order to afford a protection from such lights, and the amount of light received from illuminated objects is thereby so reduced that they cannot be seen clearly. This accounts for the well-known dazzling effects of searchlights and of many headlights. The intrinsic brilliancy or candle-power per square inch of luminous area should therefore be kept as low as possible; it should not ordinarily be higher than 4 to 6, if the source of light is in the field of vision. Clear bulbs should certainly never be used where the filaments are continuously in view. When reflectors are used it is generally desirable to employ some form of obscured lamp.

(3) Flickering lights should be avoided. Poorly regulated circuits, such as those having varying power loads, cause disagreeable flickering. Metal filament lamps have an inherently better regulation than carbon filament lamps.

(4) The light should have a continuous spectrum, *i.e.* one containing every colour, in order that the relative colour values of objects illuminated may be the same as when seen by daylight. For matching delicate tints, as in silk and paper mills, the light should be as white as possible. while for general illuminating purposes a 'warm' light is best; cold glaring lights too rich in green

and blue are to be avoided. Where colour matching is important special colour matching units should be employed, but for many industrial purposes absolutely correct daylight is not required, and the Daylight lamps can be used.

The size of the units is a question of design. In deciding upon the proper sizes of lamps to use, one must consider the spacing of present outlets, if building is already wired, and the location of pillars, beams, etc. In general the fewest units consistent with good distribution will be the most economical, as the total cost of outlets, fixtures, and lamps, as well as the maintenance charges, will be reduced. In many cases it is preferable to entirely rewire the premises, in order to obtain correct illumination.

Reports of Departmental (Home Office) Committee on Lighting in Factories and Workshops.

(1915–1922.)

Third Report of the Committee, H.M. Stationery Office, Adastral House, Kingsway, W.C. 2.

The minimum intensity of illumination desirable for the sake of general convenience and safety in parts of the workshops where work is normally carried on, and in yards, passages, and open spaces along which people are ordinarily liable to pass, is as follows :—

2. 'Over the "working areas" of workrooms the illumination measured on a horizontal plane at floor level shall be not less than 0·25 foot-candle, without prejudice to the illumination required for the work itself.'

3. 'In all parts of iron foundries in which work is carried on or over which any person is ordinarily liable to pass, the illumination measured on a horizontal plane at floor level shall be not less than 0·4 foot-candle.'

4. 'In all parts of factories and workshops (not included under recommendation 2) over which persons employed are liable to pass, the illumination measured on a horizontal plane at floor level shall be not less than 0·1 foot-candle.'

5. 'In all open places in which persons are employed during the period between one hour after sunset and one hour before sunrise, and in any dangerous parts of the regular road or way over a yard or other space forming the approach to any place of work, the illumination on a horizontal plane at ground level shall be not less than 0·25 foot-candle.'

(*a*) 'Every light source (except one of low brightness *) within a distance of 100 feet from any person employed shall be so shaded that no part of the filament, mantle, or flame is distinguishable through the shade, unless it be so placed that the angle between the line from the eye to an unshaded part of a source and a horizontal plane is not less than 20 deg., or in the case of any person employed at a distance of 6 ft. or less from the source, not less than 30 deg.'

(*b*) 'Adequate means shall be taken, either by suitable placing or screening of the light sources, or by some other effective method, to prevent direct reflection of the light from a smooth or polished surface into the eyes of the worker.'

(*c*) 'Adequate means shall be taken to prevent the formation of shadows which interfere with the safety or efficiency of any person employed.'

(*d*) 'No light sources which flicker or undergo abrupt changes of candle-power in such manner as to interfere with the safety or efficiency of any person employed shall be used for the illumination of a factory or workshop.'

---

* By 'low brightness' is meant an intrinsic brilliance of not more than five candles per square inch. 'Examples of sources of low brightness are a batswing burner, a paraffin flame, etc., and sources of about the same brightness as these are not regarded as coming within the scope of this recommendation.

'It should be remembered that the covering of a brilliant source by a very small shade (such as a silica cup covering a high-pressure gas mantle, or a small opal globe surrounding a high-power electric source) may give this shade or globe a brightness three or four times as great as the five candles per square inch, so that it may cause considerable glare and should consequently be treated as a source of light in itself.'

Two standards 'for fine work' and 'for very fine work' are recommended. For processes grouped in Schedule A the illumination shall not fall below three foot-candles, and for processes grouped in Schedule B shall not fall below five foot-candles.

The schedules give the class of work, and cover a large number of trades.

Illumination Calculations.

During recent years, the 'point-to-point' system referred to in the early part of this section has been abandoned in favour of the 'lumen' method of pre-determining the lighting for an installation. This is essentially due to the fact that, owing to the accumulated experience of illuminating engineers, it has been found possible to take into consideration the contributory effects due to reflection from the ceiling and the walls. The following gives the method of procedure that is adopted generally at the present time :—

Steps in the Design of a Lighting System.

1. *Intensity of Illumination.*

The foot-candle illumination according to present standards for different classes of operations are given on pp. 2533–2534. These values are the minimum to be adhered to if fully satisfactory lighting is to be assured. In many circumstances, particularly where dark materials are being used, or extremely fine work carried out, higher values of illumination are desirable.

2. *Type of Lighting Unit.*

The selection of the lighting unit depends, not only on the requirements of the work, but in some measure on the construction of the room and the colour of the ceiling and walls. For instance, it would be undesirable to install totally indirect lighting equipment where the ceilings are very dark. The choice of a particular fitting would depend as to which of the factors of glare, reflected glare, efficiency, etc., are the more important. For instance, in an office the criteria would rank as follows in importance :—

1. Freedom from direct glare.
2. Freedom from reflected glare.
3. Illumination on the horizontal plane.
4. Softness of shadows.
5. Favourable appearance of room.
6. Maintenance.
7. Illumination on the vertical plane.

In a machine shop the relative importance may be taken as :—

1. Freedom from direct glare.
2. Freedom from reflected glare.
3. Illumination on the horizontal plane.
4. Illumination on the vertical plane.
5. Softness of shadows.
6. Maintenance.
7. Favourable appearance of room.

Various representative types of fittings are rated according to these headings in Table II. on pp. 2535–2536. Four ratings are employed, Excellent, Good, Fair, and Poor. If a unit is rated 'Poor' for any characteristic, an installation of such units will very probably prove unsatisfactory if that particular characteristic is of prime importance for the installation under consideration. On the other hand it will be understood that a unit may not necessarily be disqualified because it is rated as 'Fair,' or even 'Poor,' in certain features, so long as its ratings are 'Excellent' or 'Good' in the essential characteristics for that specific installation.

3. *Location of Unit.*

Table III. shows the desirable spacing of units, according to the permissible mounting height. When these spacings are used, reasonably uniform illumination will result. Consult also the diagrams at bottom of Tables III. (*a*) and III. (*b*) pp. 2537–2538.

4. *Size of Lamp.*

After the position of the units has been settled on the plan according to the space permissible the size of the lamp can be determined by the consideration of the following factors :—

1. *The Room Index.*—In a large room with many points the system would have a higher coefficient of utilisation than in a small room. This is due to the fact that a number of units help to contribute directly the amount of light that reaches the table or bench without loss due to reflection from the walls, etc. The room index should be obtained from p. 2539, taking into consideration the proposed mounting height of a fitting.

2. *Depreciation Factor.*—Due to the settlement of dust and dirt on fittings and the deterioration in reflectivity of the walls and ceiling, the illumination will depreciate in the course of time. It is usual to allow a factor of 1·3 for reasonably clean locations and 1·5 where cleaning is infrequent or in dirty locations.

3. *Coefficient of Utilisation.*—On pp. 2540–2543 coefficiencies of utilisation are given for a particular fitting. Having chosen the fitting and obtained the value of the room index from Table IV., p. 2539, the corresponding coefficient of utilisation, according to the nature of the ceiling and walls can be readily obtained. This figure represents the percentage of the lumen generated by the lamp that reach the plane of work.

$$\left.\begin{matrix}\text{Lamp Lumens required}\\ \text{per sq. ft.}\end{matrix}\right\} = \frac{\text{Foot-candles} \times \text{depreciation factor}}{\text{Coefficient of utilisation}};$$

$$\text{Area in sq. ft. per unit} = \frac{\text{Total floor area in sq. ft.}}{\text{Number of units}};$$

$$\left.\begin{matrix}\text{Lamp Lumens required}\\ \text{per unit}\end{matrix}\right\} = (\text{Area in sq. ft. per unit}) \times (\text{Lamp lumens required per sq. ft.}).$$

Having thus determined the lamp lumens required per point, the wattage of the lamp to be used can be obtained by reference to Table below.

LUMENS RATINGS AND DIMENSIONS OF GAS-FILLED LAMPS.

(1930.)

Volts.	Rated. Watts.	Approx. Rated. Lumens.	Maximum Overall Dimensions. Length.	Dia. of Bulb.	Dia. of Neck.	Light Centre Length with Standard Cap.	Tolerance on Light Centre Length.	Standard Caps.
			mm.	mm.	mm.	mm.		
100 to 130	60	711*	122	65	45	85	± 3	B.C
	75	938*	130	70	51	90	± 3	,,
	100	1,300*	145	80	53	100	± 3	,,
	150	2,070	170	92	45	120	± 4	B.C. or E.S.
	200	2,880	215	103	48	150	± 5	E.S.
	300	4,650	250	113	55	180	± 6	G.E.S
	500	8,450	280	133	60	205	± 6	,,
	1,000	18,700	340	173	70	250	± 8	,,
	1,500	29,400	340	173	70	250	± 8	,,
200 to 260	60	576*	122	65	45	85	± 3	B.C
	75	780*	130	70	51	90	± 3	,,
	100	1,135*	145	80	53	100	± 3	,,
	150	1,875	170	92	45	120	± 4	B.C. or E.S.
	200	2,600	215	103	48	150	± 5	E.S.
	300	4,140	250	113	55	180	± 6	G.E.S
	500	7,500	280	133	60	205	± 6	,,
	1,000	16,900	340	173	70	250	± 8	,,
	1,500	27,000	340	173	70	250	± 8	,,

* Value applies to both Pearl and Clear Lamps.

When the lamp lumens required fall midway between two sizes, then it is usual to choose the larger rather than the smaller, unless it is certain that the less illumination from a smaller lamp will suffice.

ILLUMINATION DESIGN FOR A DEPARTMENTAL STORE MAIN FLOOR.

The following is an example in the procedure of designing illumination for a departmental main store:—

Store is 180 ft. long and 150 ft. wide with a 16-ft. ceiling. Columns divide the area into 15 ft. × 15 ft. bays. Ceiling, walls, and pillars are painted white. Supply voltage, 100. Height of counter, 3 ft.

Summary of Steps in the Design.	Reference.	Required Data.
1. Foot-candles recommended	Table I.	15
2. Type of fitting used	Table II.—p. 2535	Unit No. 16
3. Spacing and mounting height		
Possible mounting height above work	Depends upon ceiling height	12 ft.
Maximum spacing permissible	Table III.(*a*)—p. 2537	18 ft.
Actual spacing to suit interior	Symmetrical layout with respect to bays	15 ft.
Actual mounting height above work	—	10 ft.
Height above floor	—	13 ft.
4. Size of Lamp		
Room index	Table IV.—p. 2539	5·0
Coefficient of utilisation	Table V.—p. 2540	0·52
Depreciation Factor	According to dust and dirt conditions	1·3
Lamp lumens required per point	Formula—p. 2532	8437
Lamp size	Table—p. 2532	500-watt gasfilled

TABLE I.—RECOMMENDED FOOT-CANDLE ILLUMINATION.
SHOP INTERIORS.

Description.	Foot-Candles.	Description.	Foot-Candles.	Description.	Foot-Candles.
Large Departmental Store	15	Small Shops Types of (*cont.*)—		Small Shops Types of (*cont.*)—	
Shops of Medium Size	12	China	8	Jewellery	12
Small Shops Types of—		Confectionery	10	Leather Goods	8
Art	10	Draper	12	Milliners	10
Bakers	8	Electrical Supplies	12	Music	8
Boots	12	Florist	8*	Opticians	10
Chemist	10	Grocery	10	Stationer	8
		Hairdresser	10	Tailor	15
		Hardware	8	Toys	10

TABLE I (*cont.*)—COMMERCIAL AND PUBLIC BUILDINGS.

Description.	Foot-Candles.	Description.	Foot-Candles.	Description.	Foot-Candles.
Art Studio	15*	Hotel—		Restaurant	8
Bank	12	Lounge	6	School—	
Church—		Dining Room	6	Class Rooms, Library, and Office	8
Church	5	Writing Room	8	Corridors and Stairways	3
Church Hall	5–8	Bedrooms	5	Drawing and Art	15*
Club Room	5–8	Corridors	3	Laboratories	10
Dentist—Surgery	12	Billiards(General)	3	Lecture Theatre	5
Drawing Office	25	(on Table)	15	Stairways and Corridors	3
Garage Repair Department	12	Library—		Telephone Exchange	5
Gymnasium	10	Reading Rooms	10	Waiting Room	5
Hospital—		Book Room	5		
Corridors	3	Market	10		
Wards and Private Rooms	3	Office Building—			
Operating Table	100*	General Office	10		
		Private Offices	8		
		Typing and Bookkeeping Rooms	10		

* Installation where Daylight Lamps may be useful.
In all cases glare must be avoided.

TABLE I. (*cont.*)—INDUSTRIAL INTERIORS.

Description.	Foot-Candles.
Bookbinding	10
Cloth Products—	
Light Goods	10
Dark Goods	20
Electric Coil & Armature Winding, Mica Working, Insulating Processes	12
Engraving and Die Making	20–50
Foundry—	
Charging Floor, Cleaning, Pouring, and Shaking Out	5
Moulding and Core Making	8–12
Flour Grading	20*
Glass Works—	
Grinding, Cutting Glass to Size, Silvering	10
Glass Cutting (cut glass), Inspecting fine	10–50
Glove Manufacturing—	
Light Goods: Sorting, Stitching, Trimming, and Inspecting	12*
Dark Goods: Sorting, Stitching, Trimming, and Inspecting	25*
Laundries and Dry Cleaning	10
Leather Manufacturing—	
Cutting, Fleshing, and Stuffing	8
Finishing and Scarfing	12
Leather Working—	
Pressing & Winding:	
Light	10
Dark	12
Machine Shop—	
Rough Bench and Machine Work	8
Medium Bench & Machine Work	10
Fine Bench and Machine Work	10–20
Paint Manufacturing	10*
Paint Shop—	
Dipping, Spraying, Firing	8
Paint Shop (*cont.*)—	
Rubbing, Ordinary Hand-Painting, and Finishing	12*
Fine Hand-Painting and Finishing	15*
Extra Fine Hand-Painting and Finishing	20–50*
Paper Box Manufacturing—	
Light	8
Dark	10
Paper Manufacturing—	
Calendering	8
Finishing, Cutting, and Trimming	10
Power House—	
Boilers, Coal and Ash Handling, Storage Battery Rooms	5
Auxiliary Equipment, Oil Switches and Transformers	8
Switch Boards, Engines, Generators, Blowers, Compressors	8
Printing Industry—	
Matrixing & Casting, Presses	10
Proof Reading, Lithographing, Electrotyping	12*
Linotype, Monotype, Typesetting	20–50
Sheet Metal Works—	
Ordinary Bench Work	10
Punches, Presses, Shears, Stamps, Welders, Spinning	12
Tin Plate Inspection	15
Shoe Manufacturing	
Hand Turning, Bench and Machine Work	10
Cutting, Lasting, and Welting (light)	12
Stitching	20–50
Soap Manufacturing	6–8
Steel and Iron Mills—	
Soaking Pits and Reheating Furnaces	3
Charging and Casting Floors	5
Heavy Rolling, Shearing rough by gauge, Pickling and Cleaning	6
Plate Inspection	15
Automatic Machines, Rod, Light and Cold Rolling, Wire Drawing	10
Sugar Grading	15*
Sweet Making	10
Textile Mills—	
Cotton	5–8
Silk—	
Winding, Throwing, Dyeing	12
Quilling, Warping, Weaving, and Finishing:	
Light Goods	8
Dark Goods	10
Woollen—	
Carding, Picking, Washing, and Combing	4
Twisting and Dyeing	6
Weaving:	
Light Goods	8
Dark Goods	12
Knitting Machines	10
Welding	10
Wood Working—	
Sizing, Planing, Rough Sanding, Glueing, Veneering	10
Fine and Bench and Machine Working, Fine Sanding and Finishing	12

* Installation where Daylight Lamps may be useful.

In all cases glare must be avoided.

TABLE II.—GUIDE TO THE SELECTION OF REFLECTOR EQUIPMENT.

	LIGHTING UNIT		EFFICIENCY BASED UPON ILLUMINATION ON HORIZONTAL	EFFICIENCY BASED UPON ILLUMINATION ON VERTICAL	FAVOURABLE APPEARANCE OF LIGHTED ROOM	DIRECT GLARE	REFLECTED GLARE	SHADOWS	MAINTENANCE
DIRECT LIGHTING VITREOUS ENAMEL REFLECTORS.									
1	DISPERSIVE Clear Lamp		A+	B+	C+	C	D	C+	A+
2	DISPERSIVE Bowl-Sprayed Lamp		A−	B	B	B+	B	B+	A−
3	DIFFUSER REFLECTOR		B+	B	A−	A−	B+	A−	B+
4	DEEP BOWL Clear Lamp		B+	B−	C	C+	D	C	A
5	DEEP BOWL Bowl-Sprayed Lamp		B	C+	C	B	C+	C+	B+
6	DISTRIBUTING REFLECTOR		B	B	C−	C−	D	C−	B+
7	ENAMELLED IRON SHADE Clear Lamp		B−	C	C	D	D	C−	B
DIRECT LIGHTING OPEN GLASS REFLECTORS									
8	LIGHT DENSITY OPAL Clear Lamp		B+	B	B+	C+	D	B−	B
9	LIGHT DENSITY OPAL Bowl-Sprayed Lamp		B	B−	A−	B−	B−	B+	B−
10	DENSE OPAL Clear Lamp		A+	B+	B+	B	D	C+	A−
11	DENSE OPAL Bowl-Sprayed Lamp		B+	B−	A−	B+	B−	B	B
12	MIRRORED GLASS Clear Lamp		A	B	C	C+	D	C	A−
13	MIRRORED GLASS Bowl-Sprayed Lamp		B	C+	C	B−	C	C+	B−
14	PRISMATIC INDUSTRIAL Clear Lamp		A+	A−	B+	C+	D	C+	B−

TABLE II. (*contd.*)—GUIDE TO THE SELECTION OF REFLECTOR EQUIPMENT.

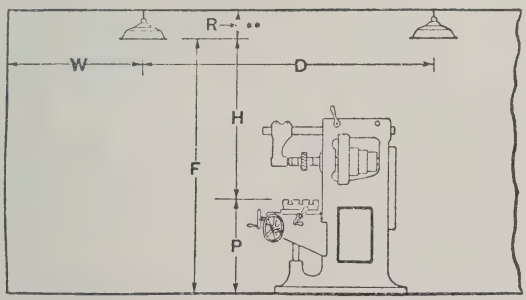
	LIGHTING UNIT		EFFICIENCY BASED UPON ILLUMINATION ON HORIZONTAL	EFFICIENCY BASED UPON ILLUMINATION ON VERTICAL	FAVOURABLE APPEARANCE OF LIGHTED ROOM	DIRECT GLARE	REFLECTED GLARE	SHADOWS	MAINTENANCE
DIRECT LIGHTING ENCLOSING AND SEMI-ENCLOSING UNITS									
15	DIFFUSING GLOBE Light Opal		B−	B−	A	B−	B	B+	B+
16	ONE PIECE OPAL Flattened Reflecting Top		B	B	A	B	B+	A−	A−
17	PRISMATIC ENCLOSING		B+	B	A	B	B−	B+	B+
18	DEEP DIFFUSING		B−	A	A	A	B+	A−	A−
19	ONE PIECE GLASS		B	A	A	B+	B+	A−	A−
20	TWO-PIECE GLASS		C+	B−	A	B+	B+	A−	B−
21	ONE-PIECE GLASS Enamelled Ref. and Bowl		B	B	A	B+	B+	A−	A−
SEMI-INDIRECT AND INDIRECT LIGHTING UNITS									
22	LIGHT OPAL		B−	C+	A	B+	B+	A−	C
23	DENSE OPAL for Light Opal and Bowl-Sprayed Lamp)		C+	C	A	A+	A−	A+	C
24	ALABASTER BOWL		C+	C	A	A+	A−	A+	C
25	PRISMATIC ENCLOSED		B+	C+	A	A−	A−	A−	B
26	CLEAR TOP ENCLOSED Enamelled Glass		B−	C+	A	B+	B+	A	B
27	MIRRORED INDIRECT		C+	C	B+*	A+	A	A+	C
28	ENAMELLED METAL INDIRECT		C	C	B+	A+	A	A+	C

* For luminous bowl type, Rate A.

TABLE III. (*a*).—SPACING—MOUNTING HEIGHT.

Direct Lighting Units, including Semi-Enclosing and Totally Enclosing Units, No. 1 to No. 21.

Mounting Height of Unit.			Distance between Points and Sidewalls.	
Above Plane of Work. (H)	Above Floor.* (F)	Distance between Points. (D)	In Usual Locations where Aisles & Storage are Next to Wall. (W)	In Offices or where Work Benches are Next to Wall. (W)
4	6½	6	3	2
5	7½	7½	3½	2½
6	8½	9	4½	3
7	9½	10½	5	3½
8	10½	12	6	4
9	11½	13½	6½	4½
10	12½	15	7½	5
11	13½	16½	8	5½
12	14½	18	9	6
13	15½	19½	9½	6½
14	16½	21	10½	7
15	17½	22½	11	7½
16	18½	24	12	8
18	20½	27	13½	9
20	22½	30	15	10
22	24½	33	16½	11
24	26½	36	18	12
27	29½	40½	20	13½
30	32½	45	22½	15
35	37½	52½	26	17½
40	42½	60	30	20



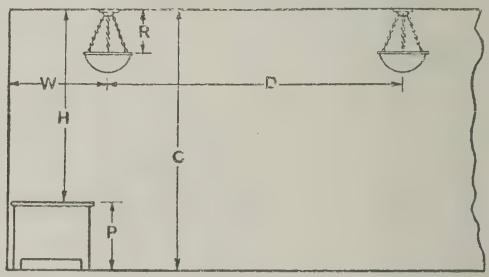
* Plane of work (P) assumed to be 2½ ft. above floor. When the plane of work is higher or lower than 2½ ft. above floor, neglect column (F) and work from column (H).

** Minimum allowance for (R) usually 1 ft.

TABLE III. (*b*) SPACING—MOUNTING HEIGHT.

Semi and Totally Indirect Lighting Units, No. 22 to No. 28.

Ceiling Height.		Spacing Distance Between Points.	Distance Between Points and Sidewalls.		Suspension Distance Ceiling to Top of Reflector.†
Above Plane of Work.	Above Floor.*		In Usual Locations where Aisles and Storage are Next to Wall	In Offices or where Work Benches are Next to Wall	
(H)	(C)	(D)	(W)	(W)	(R)
5	7½	7½	3½	2½	1½
6	8½	9	4½	3	1½
7	9½	10½	5	3½	1½
8	10½	12	6	4	2
9	11½	13½	6½	4½	2¼
10	12½	15	7½	5	2½
11	13½	16½	8	5½	2¾
12	14½	18	9	6	3
13	15⅓	19½	9½	6½	3¼
14	16½	21	10½	7	3½
15	17½	22½	11	7½	3¾
16	18½	24	12	8	4
18	20½	27	13½	9	4½
21	23½	31½	15½	10½	5½
24	26½	36	18	12	6
27	29½	40½	20	13½	6¾
30	32½	45	22½	15	7½
35	37½	52½	26	17½	8½
40	42½	60	30	20	10



* Plane of work (P) assumed to be 2½ feet above floor. When the plane of work is higher or lower than 2½ feet above floor, neglect column (C) and work from column (H).

† Suspension distances (R) in Table are based on best distribution of light and efficiency of utilisation for standard units. In some installations other considerations may require a different suspension distance.

Table IV.—Room Index.

Classify the room according to the proportions and the mounting height above the plane of work. Use upper column headings for direct-lighting units; for semi-indirect and totally indirect units choose column at bottom of page. Wherever the circumstances are such that the room index falls between two given figures, interpolate or use the smaller number.

Room Width.	Room Length.	Direct Lighting Units—Height Above Plane of Work. Feet												
Feet.	Feet.	4	5	6	7	8	9	10	12	14	16	20	24	30	40
8	10	1.0	0.8	0.8	0.6	0.6	0.6	0.6	..	..	..	..	..	..	..
	12	1.25	0.8	0.8	0.6	0.6	0.6	0.6	..	..	..	..	..	..	..
	16	1.25	1.0	0.8	0.6	0.6	0.6	0.6	..	..	..	..	..	..	..
	24	1.5	1.25	1.0	0.8	0.6	0.6	0.6	0.6	0.6	..	..	..	..	..
	35	1.5	1.25	1.25	1.0	0.8	0.8	0.6	0.6	0.6	0.6	..	..	..	..
	50	2.0	1.5	1.25	1.0	1.0	0.8	0.8	0.6	0.6	0.6	..	..	..	..
10	10	1.25	1.0	0.8	0.6	0.6	0.6	0.6	..	..	..	..	..	..	..
	14	1.5	1.0	1.0	0.8	0.6	0.6	0.6	0.6	..	..	..	..	..	..
	20	1.5	1.25	1.0	0.8	0.8	0.6	0.6	0.6	0.6	..	..	..	..	..
	30	2.0	1.5	1.25	1.0	0.8	0.8	0.6	0.6	0.6	0.6	..	..	..	..
	40	2.0	1.5	1.5	1.25	1.0	1.0	0.8	0.6	0.6	0.6	0.6	..	..	..
	70	2.0	1.5	1.5	1.25	1.25	1.0	1.0	0.8	0.8	0.6	0.6	..	..	..
12	12	1.5	1.25	1.0	0.8	0.8	0.6	0.6	0.6	..	..	..	..	..	..
	18	1.5	1.25	1.25	1.0	0.8	0.8	0.6	0.6	0.6	..	..	..	..	..
	24	2.0	1.5	1.25	1.0	1.0	0.8	0.8	0.6	0.6	0.6	..	..	..	..
	35	2.0	1.5	1.5	1.25	1.0	1.0	0.8	0.6	0.6	0.6	0.6	..	..	..
	50	2.0	2.0	1.5	1.5	1.25	1.0	1.0	0.8	0.6	0.6	0.6	0.6	..	..
	90	2.0	2.0	1.5	1.5	1.25	1.25	1.25	1.0	0.8	0.8	0.6	0.6	..	..
14	14	1.5	1.5	1.25	1.0	0.8	0.8	0.6	0.6	0.6	..	..	..	..	..
	24	2.0	1.5	1.5	1.25	1.0	1.0	0.8	0.6	0.6	0.6	..	..	..	..
	40	2.5	2.0	1.5	1.5	1.25	1.0	1.0	0.8	0.6	0.6	0.6	0.6	..	..
	70	2.5	2.0	2.0	1.5	1.5	1.25	1.25	1.0	0.8	0.8	0.6	0.6	..	..
	100	2.5	2.0	2.0	1.5	1.5	1.5	1.5	1.25	0.8	0.8	0.6	0.6	..	..
16	16	2.0	1.5	1.25	1.25	1.0	0.8	0.8	0.6	0.6	0.6	..	..	..	..
	30	2.5	2.0	1.5	1.5	1.25	1.0	1.0	0.8	0.6	0.6	0.6	..	..	..
	50	2.5	2.0	2.0	1.5	1.5	1.25	1.25	1.0	0.8	0.6	0.6	0.6	0.6	..
	80	2.5	2.0	2.0	2.0	2.0	1.5	1.5	1.25	1.0	0.8	0.6	0.6	0.6	..
	120	2.5	2.0	2.0	2.0	2.0	1.5	1.5	1.5	1.0	1.0	0.8	0.8	0.6	..
18	18	2.0	2.0	1.5	1.25	1.0	1.0	0.8	0.8	0.6	0.6	..	..	..	..
	35	3.0	2.5	2.0	1.5	1.5	1.25	1.0	1.0	0.8	0.6	0.6	0.6	..	..
	50	3.0	2.5	2.0	2.0	1.5	1.5	1.25	1.0	0.8	0.8	0.6	0.6	0.6	..
	80	3.0	2.5	2.0	2.0	2.0	1.5	1.5	1.25	1.0	0.8	0.8	0.6	0.6	..
	120	3.0	2.5	2.0	2.0	2.0	1.5	1.5	1.5	1.25	1.0	0.8	0.8	0.6	..
20	20	2.5	2.0	1.5	1.5	1.25	1.0	1.0	0.8	0.6	0.6	0.6	..	..	..
	40	3.0	2.5	2.0	2.0	1.5	1.5	1.25	1.0	0.8	0.8	0.6	0.6	..	..
	60	3.0	2.5	2.5	2.0	2.0	1.5	1.5	1.25	1.0	0.8	0.6	0.6	0.6	0.6
	100	3.0	2.5	2.5	2.0	2.0	2.0	1.5	1.5	1.25	1.0	0.8	0.6	0.6	0.6
	140	3.0	2.5	2.5	2.0	2.0	2.0	1.5	1.5	1.25	1.25	1.0	0.8	0.6	0.6
24	24	3.0	2.5	2.0	1.5	1.5	1.25	1.25	1.0	0.8	0.8	0.6	0.6	..	..
	50	3.0	3.0	2.5	2.0	2.0	1.5	1.5	1.25	1.0	1.0	0.8	0.6	0.6	..
	70	3.0	3.0	2.5	2.5	2.0	2.0	1.5	1.5	1.25	1.0	0.8	0.6	0.6	0.6
	100	3.0	3.0	2.5	2.5	2.0	2.0	2.0	1.5	1.5	1.25	1.0	0.8	0.6	0.6
	160	3.0	3.0	2.5	2.5	2.0	2.0	2.0	1.5	1.5	1.5	1.25	1.0	0.8	0.6
30	30	4.0	3.0	2.5	2.0	2.0	1.5	1.5	1.25	1.0	1.0	0.8	0.6	0.6	..
	60	4.0	3.0	3.0	2.5	2.5	2.0	2.0	1.5	1.25	1.25	1.0	0.8	0.6	0.6
	100	4.0	3.0	3.0	3.0	2.5	2.5	2.0	2.0	1.5	1.5	1.0	0.8	0.6	0.6
	140	4.0	3.0	3.0	3.0	2.5	2.5	2.0	2.0	2.0	1.5	1.25	1.0	0.8	0.6
35	35	4.0	3.0	3.0	2.5	2.0	2.0	1.5	1.5	1.25	1.0	0.8	0.6	0.6	..
	70	5.0	4.0	3.0	3.0	3.0	2.5	2.0	2.0	1.5	1.5	1.0	0.8	0.6	0.6
	100	5.0	4.0	3.0	3.0	3.0	2.5	2.5	2.0	2.0	1.5	1.25	1.0	0.8	0.6
	140	5.0	4.0	3.0	3.0	3.0	2.5	2.5	2.0	2.0	1.5	1.5	1.25	0.8	0.6
40	40	5.0	4.0	3.0	3.0	2.5	2.0	2.0	1.5	1.5	1.25	1.0	0.8	0.6	0.6
	80	5.0	4.0	3.0	3.0	3.0	2.5	2.5	2.0	2.0	1.5	1.25	1.0	0.8	0.6
	140	5.0	4.0	3.0	3.0	3.0	2.5	2.5	2.5	2.0	2.0	1.5	1.25	1.0	0.8
50	50	5.0	5.0	4.0	4.0	3.0	2.5	2.5	2.0	2.0	1.5	1.25	1.0	0.8	0.6
	100	5.0	5.0	4.0	4.0	3.0	3.0	2.5	2.5	2.0	2.0	1.5	1.25	1.0	0.8
	200	5.0	5.0	4.0	4.0	3.0	3.0	3.0	2.5	2.5	2.5	2.0	1.5	1.25	1.0
60	60	5.0	5.0	5.0	4.0	4.0	3.0	3.0	2.5	2.0	2.0	1.5	1.25	1.0	0.8
	100	5.0	5.0	5.0	5.0	4.0	4.0	3.0	3.0	2.5	2.0	1.5	1.5	1.25	1.0
	200	5.0	5.0	5.0	5.0	4.0	4.0	3.0	3.0	3.0	2.5	2.0	2.0	1.5	1.0
70	70	5.0	5.0	5.0	5.0	4.0	4.0	3.0	3.0	2.5	2.0	1.5	1.5	1.25	0.8
	140	5.0	5.0	5.0	5.0	5.0	4.0	4.0	3.0	3.0	3.0	2.0	2.0	1.5	1.0
	200	5.0	5.0	5.0	5.0	5.0	4.0	4.0	3.0	3.0	3.0	2.5	2.0	1.5	1.25
80	80	5.0	5.0	5.0	5.0	5.0	5.0	4.0	4.0	3.0	2.5	2.0	1.5	1.25	1.0
	140	5.0	5.0	5.0	5.0	5.0	5.0	4.0	4.0	3.0	3.0	3.0	2.5	2.0	1.25
100	100	5.0	5.0	5.0	5.0	5.0	5.0	5.0	4.0	4.0	3.0	2.5	2.0	1.5	1.25
	200	5.0	5.0	5.0	5.0	5.0	5.0	5.0	4.0	4.0	3.0	3.0	2.5	2.0	1.5
120	120	5.0	5.0	5.0	5.0	5.0	5.0	5.0	5.0	4.0	4.0	3.0	2.5	2.0	1.5
	200	5.0	5.0	5.0	5.0	5.0	5.0	5.0	5.0	5.0	4.0	3.0	3.0	2.5	1.5
		6	7½	9	10½	12	13½	15	18	21	24	30	36	45	60

Feet

Semi and Indirect Lighting Units—Ceiling Height Above Plane of Work.

TABLE V.—COEFFICIENTS OF UTILIZATION.
Find 'Room Index' from Table IV., p. 2539
DIRECT LIGHTING VITREOUS ENAMEL REFLECTORS.

	COLOR REFLECTION FACTOR	CEILING	VERY LIGHT (70%)			FAIRLY LIGHT (50%)			FAIRLY DARK (30%)	
		WALLS	FAIRLY LIGHT (50%)	FAIRLY DARK (30%)	VERY DARK (10%)	FAIRLY LIGHT (50%)	FAIRLY DARK (30%)	VERY DARK (10%)	FAIRLY DARK (30%)	VERY DARK (10%)
	REFLECTOR TYPE	ROOM INDEX	COEFFICIENTS OF UTILIZATION							
1	DISPERSIVE (R.L.M) Clear Lamp	0.6	.36	.32	.29	.36	.32	.29	.31	.29
		0.8	.44	.41	.39	.44	.41	.38	.40	.38
		1.0	.48	.45	.43	.48	.45	.43	.44	.43
		1.25	.52	.49	.47	.51	.49	.47	.48	.47
		1.5	.55	.52	.49	.53	.51	.49	.50	.49
		2.0	.59	.56	.53	.58	.56	.53	.55	.53
		2.5	.63	.60	.58	.61	.59	.58	.58	.57
		3.0	.64	.62	.59	.63	.61	.59	.60	.58
		4.0	.66	.64	.63	.65	.63	.62	.62	.61
		5.0	.68	.66	.64	.66	.64	.63	.63	.62
2	DISPERSIVE (R.L.M) Bowl Sprayed Lamp	0.6	.33	.29	.26	.32	.29	.26	.28	.26
		0.8	.40	.37	.35	.39	.36	.34	.36	.34
		1.0	.43	.41	.39	.43	.41	.39	.40	.39
		1.25	.47	.44	.42	.46	.44	.42	.43	.42
		1.5	.49	.47	.44	.48	.46	.44	.45	.44
		2.0	.53	.51	.48	.52	.50	.48	.49	.48
		2.5	.56	.54	.52	.55	.53	.52	.52	.52
		3.0	.58	.56	.53	.57	.54	.53	.54	.53
		4.0	.60	.58	.56	.58	.56	.56	.56	.55
		5.0	.61	.59	.57	.59	.58	.57	.57	.56
3	REFLECTOR DIFFUSER	0.6	.29	.25	.21	.28	.24	.21	.23	.21
		0.8	.36	.32	.29	.35	.31	.28	.31	.28
		1.0	.39	.36	.33	.38	.35	.33	.34	.32
		1.25	.42	.39	.36	.41	.38	.36	.37	.35
		1.5	.45	.42	.39	.43	.40	.38	.39	.38
		2.0	.49	.46	.43	.48	.45	.43	.44	.42
		2.5	.53	.50	.47	.51	.49	.47	.47	.46
		3.0	.54	.52	.49	.52	.50	.49	.49	.47
		4.0	.57	.55	.53	.55	.53	.51	.51	.50
		5.0	.58	.56	.54	.56	.54	.53	.52	.51
4	DEEP BOWL Clear Lamp	0.6	.31	.26	.23	.30	.26	.23	.25	.23
		0.8	.38	.34	.31	.37	.34	.31	.33	.31
		1.0	.41	.38	.35	.41	.38	.35	.37	.35
		1.25	.44	.41	.38	.44	.41	.38	.40	.38
		1.5	.47	.44	.41	.46	.43	.41	.43	.41
		2.0	.51	.48	.45	.50	.47	.45	.47	.45
		2.5	.54	.51	.49	.53	.51	.49	.51	.49
		3.0	.56	.54	.51	.55	.53	.51	.53	.51
		4.0	.58	.56	.54	.57	.55	.54	.55	.53
		5.0	.60	.58	.56	.58	.57	.55	.56	.55
5	DEEP BOWL Bowl Sprayed Lamp	0.6	.29	.26	.23	.29	.26	.23	.25	.23
		0.8	.35	.33	.31	.35	.32	.30	.32	.30
		1.0	.38	.36	.34	.38	.36	.34	.35	.34
		1.25	.41	.39	.37	.41	.39	.37	.38	.37
		1.5	.44	.41	.39	.43	.41	.39	.40	.39
		2.0	.47	.45	.43	.46	.44	.43	.43	.43
		2.5	.50	.48	.46	.49	.47	.46	.46	.46
		3.0	.51	.49	.47	.50	.48	.47	.47	.47
		4.0	.53	.51	.50	.52	.50	.49	.49	.49
		5.0	.54	.52	.51	.53	.51	.50	.50	.49
6	DISTRIBUTING REFLECTOR	0.6	.34	.29	.25	.33	.29	.25	.28	.25
		0.8	.42	.38	.35	.41	.37	.34	.37	.34
		1.0	.45	.42	.39	.45	.42	.39	.41	.39
		1.25	.49	.45	.43	.48	.45	.43	.44	.43
		1.5	.52	.48	.46	.51	.48	.45	.47	.45
		2.0	.56	.53	.51	.56	.53	.51	.52	.51
		2.5	.61	.58	.55	.59	.57	.55	.56	.55
		3.0	.62	.60	.57	.61	.59	.57	.58	.56
		4.0	.63	.63	.61	.64	.61	.60	.61	.59
		5.0	.67	.64	.62	.65	.63	.61	.62	.61
7	ENAMELLED IRON SHADE Clear Lamp	0.6	.26	.20	.16	.26	.19	.16	.19	.16
		0.8	.32	.26	.22	.32	.26	.22	.26	.22
		1.0	.36	.30	.26	.36	.30	.26	.30	.26
		1.25	.41	.35	.30	.39	.34	.30	.33	.30
		1.5	.44	.38	.33	.42	.37	.33	.36	.33
		2.0	.50	.44	.38	.48	.43	.38	.42	.38
		2.5	.54	.48	.42	.52	.47	.42	.46	.42
		3.0	.57	.51	.46	.55	.50	.45	.49	.45
		4.0	.62	.56	.51	.60	.54	.50	.53	.50
		5.0	.65	.60	.54	.62	.56	.53	.55	.53

TABLE V.—COEFFICIENTS OF UTILIZATION.
Find 'Room Index' from Table IV., p. 2539.
DIRECT LIGHTING OPEN GLASS REFLECTORS.

	COLOR REFLECTION FACTOR	CEILING	VERY LIGHT (70%)			FAIRLY LIGHT (50%)			FAIRLY DARK (30%)	
		WALLS	FAIRLY LIGHT (50%)	FAIRLY DARK (30%)	VERY DARK (10%)	FAIRLY LIGHT (50%)	FAIRLY DARK (30%)	VERY DARK (10%)	FAIRLY DARK (30%)	VERY DARK (10%)
	REFLECTOR TYPE	ROOM INDEX	COEFFICIENTS OF UTILIZATION							
8	LIGHT DENSITY OPAL Clear Lamp	0.6	.26	.21	.17	.24	.19	.16	.18	.15
		0.8	.32	.27	.23	.30	.25	.22	.24	.21
		1.0	.36	.31	.27	.34	.29	.26	.27	.24
		1.25	.40	.35	.31	.37	.32	.29	.30	.27
		1.5	.44	.38	.34	.40	.35	.32	.32	.30
		2.0	.49	.44	.39	.45	.40	.37	.37	.34
		2.5	.53	.48	.44	.49	.44	.41	.40	.38
		3.0	.56	.51	.47	.51	.46	.43	.42	.40
		4.0	.60	.55	.51	.55	.50	.47	.46	.44
		5.0	.62	.58	.54	.57	.53	.50	.48	.46
9	LIGHT DENSITY OPAL Bowl-Sprayed Lamp	0.6	.22	.17	.14	.20	.16	.13	.14	.12
		0.8	.27	.22	.19	.25	.21	.18	.19	.16
		1.0	.31	.26	.23	.28	.24	.21	.22	.19
		1.25	.35	.30	.26	.31	.27	.24	.24	.22
		1.5	.38	.33	.29	.34	.30	.26	.27	.24
		2.0	.43	.38	.34	.38	.34	.31	.30	.28
		2.5	.47	.42	.38	.42	.38	.34	.34	.31
		3.0	.49	.44	.40	.44	.40	.37	.36	.33
		4.0	.53	.49	.45	.48	.44	.41	.39	.37
		5.0	.56	.52	.48	.50	.46	.43	.40	.39
10	DENSE OPAL Clear Lamp	0.6	.32	.27	.23	.31	.26	.22	.25	.22
		0.8	.40	.35	.31	.38	.34	.31	.33	.30
		1.0	.44	.39	.36	.42	.38	.35	.37	.35
		1.25	.47	.43	.40	.46	.42	.39	.40	.38
		1.5	.51	.47	.43	.49	.45	.42	.43	.41
		2.0	.56	.52	.48	.54	.50	.47	.48	.46
		2.5	.60	.56	.53	.57	.54	.52	.52	.50
		3.0	.63	.59	.55	.60	.56	.54	.54	.52
		4.0	.66	.63	.60	.63	.60	.58	.57	.55
		5.0	.67	.65	.62	.65	.61	.59	.59	.57
11	DENSE OPAL Bowl-Sprayed Lamp	0.6	.29	.24	.20	.28	.23	.20	.22	.20
		0.8	.35	.31	.28	.34	.30	.27	.29	.27
		1.0	.39	.35	.32	.38	.34	.32	.33	.31
		1.25	.43	.39	.36	.41	.38	.35	.36	.34
		1.5	.46	.42	.38	.44	.40	.37	.38	.36
		2.0	.51	.47	.44	.48	.45	.42	.43	.41
		2.5	.55	.51	.48	.52	.49	.46	.47	.45
		3.0	.57	.54	.50	.54	.51	.48	.48	.46
		4.0	.60	.57	.54	.57	.54	.52	.51	.50
		5.0	.62	.59	.56	.58	.56	.54	.53	.52
12	MIRRORED GLASS Clear Lamp	0.6	.32	.27	.24	.31	.27	.24	.27	.24
		0.8	.39	.35	.32	.39	.35	.32	.35	.32
		1.0	.43	.39	.37	.42	.39	.37	.39	.37
		1.25	.46	.43	.40	.46	.43	.40	.42	.40
		1.5	.49	.46	.43	.48	.45	.43	.45	.43
		2.0	.53	.50	.48	.52	.50	.48	.49	.48
		2.5	.57	.54	.52	.56	.54	.52	.53	.52
		3.0	.58	.56	.54	.57	.55	.54	.54	.53
		4.0	.61	.59	.57	.60	.58	.56	.57	.56
		5.0	.63	.61	.58	.61	.59	.58	.58	.57
13	MIRRORED GLASS Bowl-Sprayed Lamp	0.6	.26	.22	.19	.25	.22	.19	.21	.19
		0.8	.32	.29	.26	.31	.28	.26	.28	.26
		1.0	.35	.32	.30	.34	.32	.30	.31	.30
		1.25	.38	.35	.33	.37	.35	.33	.34	.33
		1.5	.40	.37	.35	.39	.37	.35	.36	.35
		2.0	.43	.41	.39	.42	.40	.39	.40	.39
		2.5	.46	.44	.42	.45	.43	.42	.43	.42
		3.0	.47	.46	.44	.46	.45	.44	.44	.43
		4.0	.49	.48	.46	.48	.47	.46	.46	.45
		5.0	.50	.49	.47	.49	.48	.47	.47	.46
14	PRISMATIC INDUSTRIAL Clear Lamp	0.6	.33	.26	.21	.31	.25	.21	.24	.20
		0.8	.41	.35	.30	.39	.33	.29	.32	.29
		1.0	.45	.40	.35	.43	.39	.34	.37	.33
		1.25	.50	.44	.39	.47	.42	.38	.40	.37
		1.5	.52	.48	.43	.50	.45	.42	.43	.40
		2.0	.58	.54	.49	.56	.51	.47	.49	.46
		2.5	.63	.59	.54	.60	.56	.53	.54	.51
		3.0	.66	.62	.58	.63	.59	.56	.56	.54
		4.0	.71	.67	.63	.67	.63	.61	.60	.58
		5.0	.73	.69	.66	.69	.65	.63	.62	.60

TABLE V.—COEFFICIENTS OF UTILIZATION.
Find 'Room Index' from Table IV., p. 2539.
DIRECT LIGHTING ENCLOSING AND SEMI-ENCLOSING CHARTS.

	COLOR REFLECTION FACTOR	CEILING	VERY LIGHT (70%)			FAIRLY LIGHT (50%)			FAIRLY DARK (30%)	
		WALLS	FAIRLY LIGHT (50%)	FAIRLY DARK (30%)	VERY DARK (10%)	FAIRLY LIGHT (50%)	FAIRLY DARK (30%)	VERY DARK (10%)	FAIRLY DARK (30%)	VERY DARK (10%)
	REFLECTOR TYPE	ROOM INDEX	COEFFICIENTS OF UTILIZATION							
15	DIFFUSING GLOBE Light Opal	0.6	.18	.13	.10	.16	.12	.10	.10	.09
		0.8	.22	.17	.14	.20	.16	.13	.14	.12
		1.0	.26	.21	.18	.23	.19	.16	.17	.14
		1.25	.29	.24	.21	.26	.22	.19	.19	.16
		1.5	.32	.27	.23	.29	.24	.21	.22	.19
		2.0	.37	.32	.28	.32	.28	.25	.25	.22
		2.5	.40	.35	.31	.35	.31	.28	.28	.25
		3.0	.43	.38	.34	.38	.33	.30	.30	.27
		4.0	.47	.42	.38	.41	.37	.34	.33	.31
		5.0	.49	.45	.41	.43	.39	.36	.34	.33
16	ONE-PIECE OPAL Flattened Reflecting Top	0.6	.22	.18	.15	.20	.16	.14	.15	.13
		0.8	.27	.23	.20	.25	.21	.19	.20	.18
		1.0	.30	.26	.24	.28	.25	.22	.23	.21
		1.25	.34	.30	.27	.31	.27	.25	.25	.23
		1.5	.37	.33	.29	.34	.30	.27	.28	.25
		2.0	.41	.37	.33	.37	.34	.31	.31	.29
		2.5	.44	.40	.37	.40	.37	.34	.34	.32
		3.0	.47	.43	.39	.42	.39	.36	.36	.34
		4.0	.50	.46	.43	.46	.41	.40	.38	.37
		5.0	.52	.49	.45	.47	.43	.41	.40	.38
17	PRISMATIC ENCLOSING	0.6	.28	.22	.18	.26	.21	.17	.19	.16
		0.8	.35	.29	.25	.33	.28	.24	.26	.23
		1.0	.38	.33	.29	.36	.32	.28	.30	.27
		1.25	.43	.37	.33	.40	.35	.31	.33	.30
		1.5	.46	.41	.36	.43	.38	.34	.35	.33
		2.0	.51	.46	.42	.47	.43	.40	.40	.38
		2.5	.55	.51	.46	.51	.47	.44	.44	.42
		3.0	.58	.54	.50	.54	.50	.47	.46	.44
		4.0	.62	.58	.55	.57	.54	.51	.50	.48
		5.0	.65	.61	.57	.60	.56	.53	.52	.50
18	DEEP DIFFUSING	0.6	.18	.13	.10	.16	.12	.10	.11	.09
		0.8	.22	.18	.15	.21	.16	.14	.15	.13
		1.0	.25	.21	.18	.23	.19	.17	.18	.15
		1.25	.29	.24	.21	.26	.22	.19	.20	.17
		1.5	.32	.27	.23	.28	.24	.21	.22	.20
		2.0	.36	.31	.27	.32	.28	.25	.25	.23
		2.5	.39	.34	.31	.35	.31	.28	.28	.25
		3.0	.41	.37	.33	.37	.33	.30	.30	.28
		4.0	.45	.41	.37	.40	.36	.34	.33	.31
		5.0	.47	.43	.40	.42	.38	.36	.34	.33
19	ONE PIECE GLASS	0.6	.22	.17	.14	.21	.16	.14	.15	.14
		0.8	.27	.23	.20	.26	.22	.19	.20	.18
		1.0	.30	.26	.23	.29	.25	.22	.23	.21
		1.25	.34	.30	.26	.32	.28	.25	.26	.24
		1.5	.37	.33	.29	.34	.30	.27	.28	.26
		2.0	.41	.37	.33	.38	.34	.31	.32	.30
		2.5	.44	.40	.36	.41	.38	.35	.35	.33
		3.0	.47	.43	.39	.43	.40	.37	.37	.35
		4.0	.51	.47	.43	.47	.44	.41	.41	.39
		5.0	.53	.49	.45	.48	.45	.42	.42	.40
20	TWO PIECE GLASS	0.6	.20	.15	.13	.18	.14	.12	.13	.11
		0.8	.25	.20	.17	.23	.19	.16	.18	.15
		1.0	.28	.24	.21	.26	.22	.19	.20	.17
		1.25	.32	.27	.24	.28	.25	.22	.22	.20
		1.5	.34	.30	.26	.31	.27	.24	.25	.22
		2.0	.39	.34	.31	.35	.31	.28	.28	.25
		2.5	.42	.38	.34	.38	.34	.31	.31	.29
		3.0	.44	.40	.37	.40	.36	.33	.33	.30
		4.0	.48	.44	.41	.43	.39	.37	.35	.33
		5.0	.50	.46	.43	.45	.41	.39	.37	.35
21	ONE-PIECE GLASS Enamelled Reflector and Bowl	0.6	.22	.17	.14	.21	.16	.14	.15	.14
		0.8	.27	.23	.20	.26	.22	.19	.20	.18
		1.0	.30	.26	.23	.29	.25	.22	.23	.21
		1.25	.34	.30	.26	.32	.28	.25	.26	.24
		1.5	.37	.33	.29	.34	.30	.27	.28	.26
		2.0	.41	.37	.33	.38	.34	.31	.32	.30
		2.5	.44	.40	.36	.41	.38	.35	.35	.33
		3.0	.47	.43	.39	.43	.40	.37	.37	.35
		4.0	.51	.47	.43	.47	.44	.41	.41	.39
		5.0	.53	.49	.45	.48	.45	.42	.42	.40

TABLE V.—COEFFICIENTS OF UTILIZATION.
Find 'Room Index' from Table IV., p. 2539
SEMI-INDIRECT AND INDIRECT LIGHTING UNITS.

	COLOR REFLECTION FACTOR	CEILING	VERY LIGHT (70%)			FAIRLY LIGHT (50%)			FAIRLY DARK (30%)	
	WALLS		FAIRLY LIGHT (50%)	FAIRLY DARK (30%)	VERY DARK (10%)	FAIRLY LIGHT (50%)	FAIRLY DARK (30%)	VERY DARK (10%)	FAIRLY DARK (30%)	VERY DARK (10%)
	REFLECTOR TYPE	ROOM INDEX	COEFFICIENTS OF UTILIZATION							
22	LIGHT OPAL	0.6	.18	.14	.11	.15	.12	.09	.09	.07
		0.8	.22	.18	15	.19	.15	.12	.12	.10
		1.0	.26	.22	.18	.22	.18	.15	.14	.12
		1.25	.30	.25	.22	.25	.21	.18	.16	.14
		1.5	.33	.28	.24	.27	.23	.20	.18	.16
		2.0	.38	.33	.29	.31	.27	.24	.21	.19
		2.5	41	.36	.32	.34	.30	.27	.24	.22
		3.0	.44	.39	.35	.36	.32	.29	.25	.23
		4.0	49	.44	40	.40	.36	.33	.28	.26
		5.0	.51	.46	42	.42	.38	.35	.29	.28
23	DENSE OPAL (or Light Opal and Bowl-Sprayed Lamp)	0.6	.16	.13	.11	.12	.10	.08	.07	.06
		0.8	.20	.17	.15	.16	.13	.11	.09	.08
		1.0	.23	.20	.17	.18	.15	.13	.10	.09
		1.25	.27	.23	.20	.21	.18	.16	.12	.11
		1.5	.29	.26	.22	.23	.19	.17	.13	.12
		2.0	.33	.29	.26	.26	.22	.20	.15	.14
		2.5	.36	.32	.29	.28	.25	.23	.17	.16
		3.0	.39	.35	.32	.29	.27	.25	.18	.17
		4.0	.43	.39	.36	.32	.30	.28	.20	.19
		5.0	.45	.41	.38	.34	.32	.30	.22	.20
24	ALABASTER BOWL	0.6	.15	.12	.10	.11	.10	.08	.07	.06
		0.8	.18	.15	.13	.14	.12	.10	.09	.08
		1.0	.21	.18	.16	.16	.14	.12	.10	.09
		1.25	.24	.21	.18	.19	.16	.15	.11	.10
		1.5	.26	.23	.20	.20	.17	.16	.12	.11
		2.0	.29	.27	.24	.23	.20	.19	.14	.13
		2.5	.32	.29	.27	.24	.22	.22	.16	.15
		3.0	.34	.31	.29	.26	.24	.24	.17	.16
		4.0	.38	.35	.32	.28	.27	.27	.19	.18
		5.0	.39	.36	.35	.29	.28	.29	.21	.19
25	PRISMATIC ENCLOSED	0.6	.19	.15	.12	.16	.13	.10	.10	.09
		0.8	.24	.20	.17	.20	.17	.14	.14	.12
		1.0	.27	.23	.20	.23	.19	.17	.16	.14
		1.25	.31	.27	.23	.26	.22	.20	.18	.16
		1.5	.34	.29	.25	.28	.24	.22	.20	.18
		2.0	.38	.34	.30	.32	.28	.25	.22	.20
		2.5	.41	.37	.34	.34	.31	.28	.25	.23
		3.0	.44	.40	.36	.36	.33	.30	.26	.24
		4.0	.48	.44	.41	.40	.36	.34	.29	.27
		5.0	.50	.46	.43	.42	.38	.36	.30	.29
26	CLEAR TOP ENCLOSED Enamelled Glass	0.6	.16	.12	.10	.13	.10	.08	.07	.06
		0.8	.20	.16	.13	.17	.13	.11	.10	.09
		1.0	.23	.19	.16	.19	.15	.13	.12	.10
		1.25	.26	.22	.19	.21	.18	.15	.14	.12
		1.5	.29	.25	.21	.24	.20	.17	.15	.13
		2.0	.32	.28	.25	.26	.23	.20	.18	.16
		2.5	.36	.31	.28	.29	.26	.23	.20	.18
		3.0	.38	.34	.31	.31	.28	.25	.22	.20
		4.0	.42	.38	.35	.34	.31	.29	.24	.22
		5.0	.44	.40	.37	.36	.33	.31	.25	.24
27	MIRRORED INDIRECT	0.6	.15	.12	.10	.11	.09	.07	.05	.04
		0.8	.18	.15	.13	.13	.11	.09	.07	.06
		1.0	.22	.19	.16	.15	.13	.11	.08	.07
		1.25	.25	.22	.19	.18	.15	.13	.09	.08
		1.5	.27	.24	.21	.20	.17	.15	.10	.09
		2.0	.30	.27	.25	.22	.19	.17	.11	.10
		2.5	.34	.31	.28	.24	.22	.20	.13	.12
		3.0	.36	.33	.30	.26	.24	.22	.14	.13
		4.0	.40	.37	.34	.28	.26	.24	.15	.14
		5.0	.42	.39	.37	.30	.28	.26	.17	.15
28	ENAMELLED METAL INDIRECT	0.6	.14	.11	.10	.10	.08	.07	.04	.04
		0.8	.17	.14	.13	.13	.10	.09	.06	.05
		1.0	.20	.17	.15	.14	.12	.10	.07	.06
		1.25	.23	.20	.17	.17	.14	.13	.08	.07
		1.5	.25	.22	.19	.18	.15	.14	.09	.08
		2.0	.28	.25	.23	.21	.18	.16	.10	.10
		2.5	.31	.28	.26	.22	.20	.18	.12	.11
		3.0	.33	.30	.28	.24	.22	.20	.13	.12
		4.0	.37	.34	.32	.26	.24	.22	.14	.13
		5.0	.39	.36	.34	.28	.26	.24	.15	.14

Types of Reflectors.

In the following figs. are shown some typical reflectors (made by *Benjamin Electric, Ltd.*, Tottenham, London, N. 17), with their characteristic curves.

Biflector.

FIG. 7.

R.L.M.

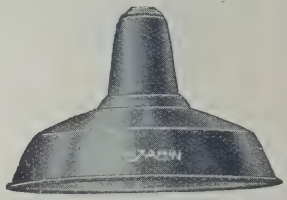


FIG. 8.

Characteristic Curves.

Biflector.

FIG. 9.

R.L.M.

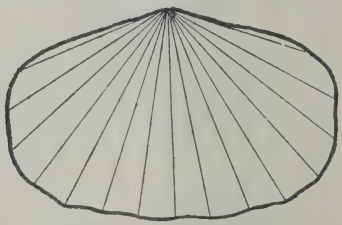


FIG. 10.

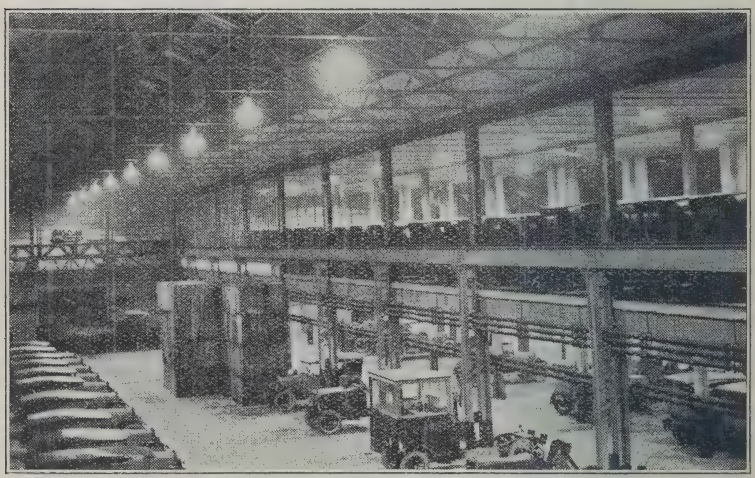


FIG. 11.—General Lighting in a large Motor Works (Benjamin R.L.M. Dispersive Reflectors).

It should be borne in mind that polar distribution curves represent the intensities corresponding with different angles to the vertical or horizontal axis of an illuminant. The area enclosed by such curves must not be associated with the total quantity of light emitted by the source.

Fig. 11 shows the excellent effect of using Benjamin R.L.M. Dispersive Reflectors in a large works.

See also British Standard Specification for Industrial Reflector Fittings for Electric Lighting, No. 232, 1926.

Gas Lighting in Factories.

This may be either of high pressure (60 to 100 inches water gauge) or low pressure (4 to 6 inches water gauge).

The amount of gas that must be consumed to provide a given number of foot-candles of illumination over the area of a workshop may be computed as follows:—

High pressure $\frac{A \times I}{200}$ = c. ft. per hour; Low pressure $\frac{A \times I}{100}$ = c. ft. per hour,

where A is the area to be lighted in square feet and I is the average illumination required in foot-candles.

Thus assume a workshop having a floor area of (say) 2,000 sq. ft., and an average illumination of 4 ft.-candles is required. According to the foregoing formula, the lighting units as a whole would have to consume,

High pressure $\frac{2{,}000 \times 4}{200} = 40$ c. ft. per hr. Low pressure $\frac{2{,}000 \times 4}{100} = 80$ c. ft. per hr.

Generally it will not pay to put in high-pressure plant unless the total candle-power exceeds 1,000 candles.

It remains to subdivide the installation into separate lighting units, so as to ensure as nearly as possible a uniform illumination in order that all parts of the workshop are equally available for practical use.

For instance, to develop the light from 80 cubic feet of low-pressure gas, any one of the following arrangements may be used.

No. of Lamps. . .	20	16	10	8
Cub. ft. per hour, each	4	5	8	10
Total of all Lamps .	80	80	80	80

Even larger lamps may be used, units ranging from 25 to 4,000 candle-power being available. The particular arrangement that may be used will depend partly upon the structural features of the building and the kind of work being done. Sometimes a uniform illumination must be aimed for, or in other cases it may be desirable to have certain areas extra-illuminated, the intervening areas being somewhat subdued.

Generally it is not advisable to rely upon the reflecting power of walls, because in the first place the use of reflecting shades prevents much of the light from the lamps reaching the walls, and secondly, although the whitewashed surfaces have a reflect on coefficient of 80 per cent. when new, this rapidly deteriorates to less than half this value, due to dust created by moving machinery.

The greater the candle-power of the lighting unit used, the higher it should be placed; and in order that the illumination from the separate lamps shall overlap they should be disposed so that the distance between their centres is 1½ to 2 times their height above the level at which work is being done.

The following points may be noted:—

If gas lamps have to be arranged amongst running shafting or belting, remote control may be used to ensure safety in lighting and extinguishing.

If there is excessive vibration the premature fracture of mantles can be prevented by using anti-vibrators.

Gas lamps may be arranged if required for use at various heights.

If the air round the lamps is extraordinarily dusty, the use of Keith-Blackman's 'group' system will ensure their perfect working.

For large work, lamps 9 feet to 10 feet above floor level will be found convenient This would need moderate size lamps, consuming 10 to 15 cubic feet per hour and spaced 9 to 12 feet between centres.

For smaller work, lamps consuming 5 cubic feet per hour, set at 7 to 8 feet above floor level and 6 to 9 feet between centres, will be found suitable.

Artificial Daylight.

On many occasions it is essential that artificial lighting shall approximate in quality to that obtained during the daytime. Particularly is this important in artists' studios and picture galleries, since the light obtained from an incandescent lamp has a greater proportion of red radiation than daylight. This has the effect of the blue and violet colours losing their life and the yellow and red becoming accentuated. Again in dye works and in some shops it is necessary to match colours, and in order to facilitate operation under such circumstances after daylight several expedients have been devised.

A gas-filled lamp is now available in which the filament operates in a selectively tinted blue daylight bulb, which has the effect of filtering the causes of red radiation. It is essential, however, to bear in mind that when daylight lamps of this order are being used that a lamp of sufficient wattage be employed. It is a good rule to bear in mind that, if under ordinary circumstances a 60-watt lamp is required, then in order to produce approximately daylight effects a 100-watt daylight lamp will be required.

A further use for these gas-filled lamps is their employment in the grading of flour or the grading of leather. It sometimes happens that even the colour which is obtained in this way as not sufficiently accurate, in which case special glass filters and reflecting materials have been designed. In such few cases in which they must be employed it is still more important to have an ample size of electric lamp. (See 'Standard Types of Gas-filled Lamps,' p. 2532.)

Street Lighting.

See British Standard Specification for Street Lighting, No 307—1927.

Terms used in Illumination and Photometry.

See British Standard Glossary, No. 233—1925.

Lighting Fittings.

Neck and Flange dimensions of illumination glassware and carriers, see British Standard Specification 364.

Translucent glassware illumination fittings for interior lighting, see British Standard Specification 324.

Industrial electrical fittings for electric lighting, see British Standard Specification 232.

Portable Photometers.

See British Standard Specification for Portable Photometers, No. 230.

Electric Lamps.

See British Standard Specification for Tungsten filament electric lamps, No. 161.

SECTION XXXVIII

PART IV

FIRE EXTINCTION.

STANDARDS OF FIRE-RESISTANCE.

STANDARD TABLE FOR FIRE-RESISTING SINGLE DOORS, WITH OR WITHOUT FRAMES.

Classification.	Sub-Class.	Duration of Test at Least.	Minimum Temperature.	Thickness of Material.	Minimum Superficial Area under Test.	Minimum Time for Application of Water under Pressure.
		Min.	Deg. F.		Sq. Ft.	Min.
Temporary protective class	A	45	1500	2 in. and under	20	2
	B	60	1500	Optional	20	2
Partial protective class	A	90	1800	2½ in. and under	20	2
	B	120	1800	Optional	20	2
Full protective class	A	150	1800	½ in. and under	25	2
	B	240	1800	Optional	25	5

(*International Fire-prevention Congress.*)

Fire-Resisting Substances.

Exhaustive tests seem to show that sulphate of aluminium is one of the best fire-resisting substances for wood, as it checks combustion by forming an infusible and non-conducting coating, while such substances as ammonium sulphate or ammonium phosphate when heated check combustion by emitting an incombustible gas.

The following is the formula for what is known as the United States 'Government whitewash mixture,' which also acts as a fire-retarding coating over interior wooden surfaces:—Slake a half bushel of quicklime with boiling water, keeping it covered during process; strain and add one peck of salt dissolved in warm water; put 3 lbs. ground rice in water and boil to a thin paste; ½ lb. of powdered Spanish whiting; 1 lb. of clean glue dissolved in hot water. Mix well and let it stand for several days. Keep the mixture in a kettle or receptacle, and apply as hot as possible with a whitewash or paint brush.

(See also p. 333.)

Fire Streams.

TABLE SHOWING PRESSURE REQUIRED AT NOZZLE AND AT PUMP, WITH QUANTITY AND PRESSURE NECESSARY TO THROW GOOD EFFECTIVE STREAMS VARIOUS DISTANCES THROUGH DIFFERENT SIZE NOZZLES, USING 100 FEET OF ORDINARY 2½-INCH RUBBER-LINED HOSE AND SMOOTH NOZZLES.

Size of Nozzle, ¾ inch.

Pressure at nozzle, in lbs. per sq. in.	40	50	60	70	80	90	100
Pressure at pump, „ „ „	46	57	68	80	91	102	114
Imperial gallons per minute . .	86	96	105	114	122	129	136
Distance thrown horizontal, in ft.	44	50	54	58	62	65	68
Distance thrown vertical, in ft. .	60	67	72	76	79	81	83

Size of Nozzle, ⅞ inch.

Pressure at nozzle, in lbs. per sq. in.	40	50	60	70	80	90	100
Pressure at pump, „ „ „	50	63	75	88	101	113	126
Imperial gallons per minute . .	118	132	144	156	167	177	186
Distance thrown horizontal, in ft.	49	55	61	66	70	74	76
Distance thrown vertical, in ft. .	62	71	77	81	85	88	90

Size of Nozzle, 1 inch.

Pressure at nozzle, in lbs. per sq. in.	40	50	60	70	80	90	100
Pressure at pump, „ „ „	58	72	87	101	115	130	144
Imperial gallons per minute . .	154	173	189	204	218	232	245
Distance thrown horizontal, in ft.	55	61	67	72	76	80	83
Distance thrown vertical, in ft. .	64	73	79	85	89	92	96

Size of Nozzle, 1⅛ inches.

Pressure at nozzle, in lbs. per sq. in.	40	50	60	70	80	90	100
Pressure at pump, „ „ „	69	86	103	120	138	155	172
Imperial gallons per minute . .	197	221	241	260	279	295	312
Distance thrown horizontal, in ft.	59	66	72	77	81	85	89
Distance thrown vertical, in ft. .	65	75	83	88	92	96	99

Size of Nozzle, 1¼ inches.

Pressure at nozzle, in lbs. per sq. in.	40	50	60	70	80	90	100
Pressure at pump, „ „ „	84	106	127	148	169	190	211
Imperial gallons per minute . .	246	275	301	325	348	368	388
Distance thrown horizontal, in ft.	63	70	76	81	85	90	93
Distance thrown vertical, in ft. .	67	77	85	91	95	99	101

Size of Nozzle, 1⅜ inches.

Pressure at nozzle, in lbs. per sq. in.	40	50	60	70	80	90	100
Pressure at pump, „ „ „	107	134	160	187	214	240	268
Imperial gallons per minute . .	301	337	369	398	426	452	476
Distance thrown horizontal, in ft.	66	73	79	84	88	92	96
Distance thrown vertical, in ft. .	69	79	87	92	97	100	103

N.B.—The above pressures are based on the supposition that the hose is coupled direct to the delivery of the pump and while the stream is flowing; if, however, the hose is coupled to a hydrant which is supplied direct from the pump, then the corresponding fire pump pressure must be greater than the hydrant pressure by an amount equal to friction loss and difference of head between hydrant and pump.

The pressures given in the table are *indicated* pressures, not effective pressures; effective pressures would be slightly greater.

The distances given are for *effective* fire *streams* adapted for fire purposes, and are not for mere isolated drops. (*Freeman.*)

Fire Streams and Electric Circuits.

It has always been believed to be dangerous to those handling a hose line to allow the stream to come in contact with an electric circuit. Prof. F. C. Caldwell has found by experiment that there is no danger of shocks when the distance between the nozzle and circuit is more than 25 ft., irrespective of the voltage. (*'Power.'*)

Fire Extinguishers.

Sawdust possesses some special merits as a fire extinguisher when dealing with small outbreaks of liquid combustibles, such as lacquer and petrol, which are usually difficult to extinguish by ordinary means. Sand is generally looked upon as the best substance to use in such cases when it can be applied promptly, but tests have shown sawdust to be superior.

The British Fire-Prevention Committee carried out a series of tests, the results of which tended to prove that sawdust intimately mixed with bicarbonate of soda and applied in bulk has certain advantages over sand and similar materials as an extinguishing medium for comparatively small fires occasioned by the ignition of the vapour of petroleum spirit (petrol) and other inflammable liquids. Ordinary sawdust, as obtained from saw-mills, is the most suitable for this purpose. It must be free from shavings and chips of wood, but it does not require to be either specially dried or to contain added moisture. The addition of bicarbonate of soda, although not essential, is advantageous. The Committee found an effective proportion of bicarbonate of soda to sawdust to be 10 lbs. to 1 bushel (or, say, 12 lbs.) of sawdust. The Committee wished to emphasise the fact that so far their investigations had been limited to small quantities of inflammable liquids, not exceeding 2 gallons, and the areas over which they were spread did not exceed 6 ft. square. These conditions are met with in motor garages and hangars, for which, in the opinion of the Committee, the proposed mixture is applicable and economical. The Committee added that their tests showed that the application of this or any similar mixture in small quantities from tubes or canisters does not give satisfactory results.

Some extensive tests carried out by the Standard Oil Company have proved that the foam method of fighting oil fires is ineffective unless the foam is applied gently at the surface of the burning oil. A 55,000-barrel tank burned right out although enough foam was dropped into it to form a blanket 6 ft. thick. When, however, the foam was applied through asbestos pipes reaching to the surface, experimental fires in the same tank were quickly put out.

British Standard Specification for Materials and Constructional Strength of Chemical Fire Extinguishers.*

(*Abstract.*) (No. 138—1922.)

Definition.

1. The term 'Chemical Fire Extinguisher' under this Specification denotes a vessel or machine manufactured as a Fire Appliance in which the pressure necessary to project the extinguishing fluid is generated automatically either by chemical action or by the release of compressed gas in a capsule within or affixed to the vessel itself.

Size.

2. This Specification applies only to Extinguishers having a fluid capacity of three Imperial gallons and under.

Shape.

3. The shape of the vessel may be cylindrical or conical.

Method of Actuating.

4. The machine may be actuated in any of the following ways :—

(*a*) By breaking a hermetically sealed bottle containing acid and thereby allowing this acid to come into contact with an alkaline solution.

(*b*) By turning over the vessel and thereby spilling acid from a bottle into the alkaline solution.

(*c*) By opening a sealed capsule containing high-pressure gas within or affixed to the vessel.

These operations may be performed by striking a knob on the top of the machine, by a blow on the side, or by other means.

* By permission of the British Engineering Standards Association.

SECTION XXXIX

(Contributed and revised by Emeritus Prof. Henry Louis, M.A., D.Sc., A.R.S.M., M.Inst.C.E., F.I.C., F.G.S., etc., Past-President Iron & Steel Institute, Past-President Inst. Mining Engrs.)

PART I

MINING (pp. 2555-2649)

PART II

MINERAL VALUATION (pp. 2651-2656)

SECTION XXXIX

PART I

MINING

(Contributed and revised by Emeritus Professor Henry Louis, M.A., D.Sc., A.R.S.M., M.Inst.C.E., F.I.C., F.G.S., etc.)

LEGAL—CLASSIFICATION OF MINERAL DEPOSITS—COAL—METALLIFEROUS MINERALS—BORING—DRILLING AND BLASTING—DRIFTS—SHAFTS—COAL CUTTERS—SHAFT PILLARS—MINING IN VEINS AND MASSES—HAULAGE—WINDING—WIRE ROPES—ELECTRIC CABLES—PUMPS AND PUMPING—VENTILATION—SAFETY LAMPS—RESCUE APPARATUS—DRESSING OF ORES—GOLD MILLING—ALLUVIAL MINING.

Legal.

DEFINITIONS OF MINERALS AND MINES.

By the term 'mineral,' as used in legal documents, is meant a constituent of the earth's crust which possesses in its severed condition an intrinsic value by virtue of its composition, that is to say, 'any substance that can be got from within the surface of the earth which possesses a value in use apart from its mere possession of bulk and weight' (Lord Fletcher Moulton, Court of Appeal, Nov. 23, 1908). The meaning of the word 'mineral' may be affected by the context; according to Lord Halsbury, in 'a grant of *mines* and *minerals*,' it 'is a question of fact what these words meant in the vernacular of the mining world, the commercial world and landowners at the time when they were used in the instrument' (Lord Provost of Glasgow *v.* Fairie, 13 App. Cas. 1888).

In this country a *mine* is defined as a working for mineral carried on by artificial light without intentionally disturbing the surface, whilst a *quarry* is an excavation for the extraction of mineral open to the day; the distinction is important from a legal point of view. It must be noted that a quarry less than 20 feet deep is by law classed as a factory. In many other countries, notably in the United States of America, the distinction between a 'mine' and a 'quarry' depends not upon the method of working but upon the mineral wrought, any working for e.g. building stone, whether underground or open, being classed as a quarry. In legal instruments 'mine' sometimes has the wider meaning of the space containing the minerals, or even from which the minerals have been extracted.

Ownership of Minerals.

In this country the mineral occurring within any area is the property of the owner of the freehold of the surface, with some few apparent exceptions, the most notable of which are the Forest of Dean and the Isle of Man, where the minerals are Crown property because the Crown owns the surface. Minerals containing precious metals—where these metals form the essential value of the mineral—are the property of the Crown wherever found (except in Scotland). Minerals under copyhold land were mostly the property of the Lord of the Manor, who could not, however, as a rule, deal with them save with the consent of the copyholder. Since by the Law of Property Act, 1922 (12 & 13 Geo. V. ch 16, s. 128), copyhold has been abolished, the minerals are as a rule enfranchised and become the property of the tenant, although the latter may, if he so desire, transfer his right to the lord. This law came into force on January 1, 1926 (Law of Property Act (Postponement) Act, 1924).

Generally speaking, in this country the minerals are leased to those exploiting them. 'A mining lease is really a sale payable by instalments' (Henn Collins, Master of the Rolls, *In re* Aldam's Settled Estates, Chanc. 2, 1902). (See Presidential Address by H. Louis, *Trans. Inst. Min. Eng.*, vol. lxxiv (1928), p. 277.) Most often the so-called rental is based on the tonnage exploited, the rate per ton or *royalty* being fixed. For coal this may range from 2*d.* to 2*s.*, but is usually between 4*d.* and 7*d.*, the average for the United Kingdom having been pre-war probably just about 5½*d.* per ton (Royalties Commission, 1889); it is now (1928) 5·74*d.* including wayleaves

per ton of saleable coal raised. Sometimes the royalty is a fraction of the value of the coal, this ranging from $\frac{1}{2}$ to $\frac{1}{50}$; sometimes the rental is a proportion of the profit made; sometimes it is a price per inch thickness of coal seam per acre wrought out. In addition to royalty, wayleaves have often to be paid for; average wayleave in United Kingdom is about $\frac{1}{4}d.$ per ton (Royalties Commission, 1889).

Royalties on stratified ironstone vary from $2\frac{1}{2}d.$ to 1*s.* 6*d.* per ton, averaging about 6*d.*

In the hæmatite mines of Cumberland and Lancashire the royalties vary normally from 1*s.* to 2*s.*, averaging about 1*s.* 9*d.* In the Forest of Dean royalties on iron ore vary from 3*d.* to 5*d.* per ton.

Royalties in Cornwall on tin and copper ores range from $\frac{1}{10}$ to $\frac{1}{20}$ of the value of the mineral raised.

Royalties on lead and zinc ores range from $\frac{1}{4}$ to $\frac{1}{6}$ of the value of the ore raised.

Royalties are often on a sliding scale, the ratio increasing with the market price of the mineral or of its metallic contents.

List of Principal Legislative Enactments affecting Mines and Minerals.

Land Tax Redemption Act, 1798, and subsequent Acts; 'The Mining Code,' Railway Clauses Consolidation Act, 1845, Ss. 74–85; Waterworks Clauses Consolidation Act, 1847, Ss. 18–27; Public Health (Support of Sewers) Act, 1883—see, however, New Moss Colliery *v.* Manchester Corporation, 1908, and Howley Park Coal and Canal Co. *v.* L. & N.W. Railway, 1912; Copyhold Act, S. 48, 1852; Coal Mines Act, 1866; Metalliferous Mines Acts, 1872, 1875; Explosives Act, 1875; Rating Act, 1874; Stratified Ironstone Mines (Gunpowder) Act, 1881; Conveyancing Act, S. 2, 1881; Slate Mines (Gunpowder) Act, 1882; Coal Mines Regulation Act, 1887; Quarries (Fencing) Act, 1887; Quarries Act, 1894; Coal Mines (Check Weigher) Act, 1894; Workmen's Compensation Act, 1897; Coal Mines (Weighing of Minerals) Act, 1905; Workmen's Compensation Act, 1906; Notice of Accidents Act, 1906; Coal Mines Regulation (Eight Hours) Act, 1908; Small Holdings and Allotments Acts, Schedule I., 1908; Mine Accidents (Rescue and Aid) Act, 1910; Coal Mines Act, 1911; Coal Mines (Minimum Wage) Act, 1912; Coal Mines Act, 1914; Feudal Casualties (Scotland) Act, S. 7, 1914; Price of Coal (Limitation) Act, 1915; Coal Mines Control Agreement (Confirmation) Act, 1917; Coal Mines Regulation (Amendment) Act, 1917; Petroleum (Production) Act, 1918; Coal Mines Act, 1919; Coal Industry Commission Act, 1919; Coal Mines (Emergency) Act, 1920; Mining Industry Act, 1920; Finance (1909–10) Act, 1910, Ss. 1–5, 20–24, 25–27, 33, 34; Finance Act, 1912, S. 11; Finance (No. 2) Act, 1915, S. 43; Finance Act, 1916, S. 46; Finance Act, 1917, S. 21; Income Tax Act, 1918, Schedule III.; Finance Act, 1919, S. 33; Finance Act, 1920, Ss. 49, 7; Mines (Working Facilities and Support) Act, 1923. See Report of Mining Cases decided by the Railway and Canal Commission Court under the Mines (Working Facilities and Support) Act, 1923, by P. G. Bamber. Coal Mines Act, 1926 (Amendment of 1908 Eight Hours' Act for 5 years); Mining Industry Act, 1926; Rating and Valuation Act, 1925; Rating and Valuation (Apportionment) Act, 1928; Coal Mines Act, 1930.

Principal Clauses in Colliery Lease.

Description of contracting parties (lessor and lessee) and of property demised.

Powers granted to lessee:—
- To break surface, make shafts, adits, etc.
- To win and work coal, bring to bank, make merchantable, dispose of.
- To work by instroke or outstroke or both.
- To make and dispose of coke and other products.
- To have necessary surface accommodation.
- To have necessary wayleaves.
- To construct roads, tramways, railways, etc.
- To erect buildings, machinery, plant, etc.
- To work quarries of lime, clay, etc. (sometimes for own use only).
- To remove coals, dead and live stock, plant, etc., after expiration of lease.
- To relinquish at stated times before termination of lease.
- To renew lease for a further period.
- To generally do all things necessary for getting coal, etc.

Reservations to lessor:—
- Of mines and minerals not demised.
- Of certain rights and wayleaves.
- Of use of ways, railways, etc.

'Habendum':—
- Lessee to have and hold property for a fixed term of years.

'Redendum':—
- Annual certain rent or deadrent, usually merging into royalty.
- Royalty per ton or otherwise.
- Rent for surface occupied.
- Rent for wayleaves, waterleaves, or other facilities.

Provisoes:—
- No royalty payable in respect of colliery consumption.
- Power to make up short workings of previous years.
- Power to carry over excess workings to subsequent years.
- For distress in case of default in payments.
- For re-entry if rents, etc., remain unpaid for certain period.

Covenants on part of lessee :—
For due payment of rents and royalties.
For due payment of taxes.
For compensating for damage done to surface or to neighbours.
For erecting (or against erecting) buildings, houses, coke-ovens, etc.
For keeping proper accounts accessible to lessor, and rendering the same.
For keeping proper mine plans, and permitting inspection of mine.
For leaving proper barriers.
For working in approved and miner-like manner.
Against sub-letting without authority.
For giving up possession in good condition at determination of lease.
For keeping mine, plant, buildings, etc., in good repair.
For properly fencing or securing shafts, adit mouths, etc.
For giving lessor power to purchase plant, etc., at a valuation on determination of lease.
Covenants on part of lessor :—
For proper title.
For quiet enjoyment.
Arbitration clause.

Leases of ironstone mines as also of metalliferous mines follow practically the same lines; as regards the latter, it is usually set forth that the royalty shall be a certain part of the value of the minerals, cleaned, dressed, smelted, or otherwise rendered merchantable, as the case may demand, at the expense of the lessee. There are often clauses specifying that a certain amount of development work, driving, sinking, etc., shall be performed.

A tacknote or license to prospect is often granted ; this differs from a lease, inasmuch as it transfers no interest in the property. Its usual form is a license for one year to explore for all or for certain specified minerals within a definite area, usually for a nominal consideration, the licensee making compensation for all damages, and paying a royalty on all minerals gotten and sold. The license is generally renewable at the option of the licensee for a term of years, usually three, and the licensee is entitled to a lease upon conditions specified ; this specification often takes the form of a schedule embodying the conditions of a draft lease.

Classification of Mineral Deposits.

A mineral deposit may be defined as a portion of the earth's crust containing mineral matter of intrinsic value under favourable conditions.

The mineral substances that form the subject of mining operations may be classified as under :—

Group I., Fuels ; II., Ores ; III., Salts ; IV., Gems ; V., Rocks.

Group I. contains the different varieties of coal, asphalts, bitumen, and other solid hydrocarbons, mineral oils, etc., while sulphur may conveniently be included.

Group II. contains the various heavy metals either in the native state or in combination.

Group III. contains the haloids, sulphates, carbonates, silicates, phosphates, borates, etc., of the alkaline earths and alkalies.

Group IV. contains those minerals that are prized as gems on account of their hardness, brilliancy, and transparency, as well as the semi-gems, such as opal, turquoise, etc., which though opaque are prized as precious stones on account of the beauty of their colouring and their rarity.

Group V. comprises the various rock masses, such as granite, slate, limestone, etc., the exploitation of which is more properly considered under quarrying.

With respect to their mode of occurrence, mineral deposits may be classified morphologically as follows :—

Tabular = Beds, veins.
Irregular = Masses, lenses, pipes, etc.

A more complete classification, based on genetic as well as morphological characters, is as follows :—

Class I.—Syphythic; Syngenetic, contemporaneous or stratified
- Group *a*, clastic.
- „ *b*, formed by chemical agencies.
- „ *c*, formed by organic agencies.
- „ *d*, disseminated through beds.

Class II.—Epactic; Epigenetic, subsequent or non-stratified.
- Sub-Class A—Veins.
  - Group *a*, fissure.
  - „ *b*, bedded.
  - „ *c*, contact.
  - „ *d*, gash.
- Sub-Class B—Masses.
  - Group *a*, stockworks.
  - „ *b*, disseminated in igneous rocks.
  - „ *c*, connected with igneous rocks.
  - „ *d*, connected with soluble rocks.

Class I.—Group *a* includes auriferous alluvials, tin-bearing alluvials, ironsands, etc. Group *b* contains deposits due to chemical precipitation, to evaporation or to metasomatic changes. Practically all the stratified ironstones belong to this group. Of group *c* the most important

representative is coal. Group *d* includes the Knottensandstein of Mechernich, the Kupferschiefer of Mansfeld, the Banket beds of Johannesburg, and similar deposits.

Class II.—Sub-Class A. Group *a* includes most of the important vein systems of the world, such as those of Cornwall, etc.

Group *b* includes veins parallel to the stratification.

Group *c* embraces certain veins following the contact of dissimilar rocks.

Group *d* contains veins carrying galena and zinc blende that characterise certain limestone districts, e.g., that of Missouri.

Sub-Class B. Group *a* consists of masses traversed by minute veinlets carrying ore, such as the tin deposits of Saxony.

Group *b* comprises such deposits as the Kimberley diamond-bearing pipes, the copper deposits of Lake Superior, etc.

Group *c* are very largely lenticular in form; the pyrite deposits of the Huelva district are a typical example.

Group *d* may be represented by the red hæmatite deposits of Cumberland and Lancashire.

Branches of Mining.

I. Prospecting, the most important part of which is 'boring.'

II. Exploiting, including the various operations of opening up mineral deposits by shafts or adits, extracting and transporting the mineral, securing, pumping and ventilating the workings, and the proper care of the men employed.

III. Dressing or preparing the mineral extracted for the market.

COAL.

Classification.

Coal is generally classified under the three main heads of 'lignite' or 'brown coal,' 'bituminous coal,' and 'anthracite.' Bituminous coal is further classified according to the purposes for which it is best adapted, such terms as gas coals, house coals, furnace coals, smithy coals, steam coals, etc., being usually employed. A classification that is widely adopted is based upon Gruner's work, and is as follows :—

Description of Coal.	Composition of Ash-free Coal.			Percentage of Fixed Residue.	Nature of Residue.	Calorific Power.
	Carbon.	Hydrogen.	Oxygen.			Cal.
Lignite . .	70–75	5·6	15·25	50	Non-coking .	6,000
Dry coals with long flame (furnace coals)	75–80	4·5–5·5	15–19·5	55–66	Pulverulent or very slightly coking	8,000–8,500
Fat coals with long flame (gas coals)	80–85	5·0–5·8	10·0–14·2	60–68	Completely agglomerated, porous	8,500–8,800
Fat coals proper (smithy coals)	84–89	5·0–5·5	5·5–11·0	68–74	Completely agglomerated, more or less porous	8,800–9,300
Fat coals with short flame (coking coals)	89–91	4·5–5·5	5·5–6·5	74–82	Coked, compact	9,300–9,600
Anthracite .	90–93	4·0–4·5	3·0–5·5	82–90	Pulverulent .	9,200–9,500

Seyler's classification is based upon the percentage of hydrogen in the ash-free coal and the ratio of carbon to hydrogen :—

Genus.	H per cent.	C : H.
1. Anthracite	< 4	> 22
2. Carbonaceous	4–4·5	20·3–23·35
3. Semi-bituminous	4·5–5	16·6–20·4
4. Bituminous	5–5·8	14·6–18·2
5. Per-bituminous	> 5·8	12·9–14·7

(*Strahan & Pollard.*)

CLASSIFICATION OF THE INTERNATIONAL GEOLOGICAL CONGRESS, CANADA.

Class.	Name.	Fuel Ratio.	Composition.			Percentage of Volatile Combustible Matter.	Calorific Power B.Th.U.
			C.	H.	O & N.		
A_1	Anthracite	12 or more	93–95	2–4	3–5	3–5	14,500–15,000
A_2	Semi-Anthracite	7–12	90–93	4 4·5	3–5·5	7–12	15,000–15,500
B_1	High Carbon Bituminous	4–7	80–90	4·5–5	5·5–12	12–15	15,200–16,000
B_2	Bituminous	1·2–7	75·90	4·5–5·5	6–15	12–26	14,000–16,000
B_3	Low Carbon Bituminous	—	70–80	4·5–6	18–20	up to 35	12,000–14,000
C	Cannel	—	—	—	—	30–40	12,000–16,000
D_1	Lignitic	—	60–75	6–6·5	20–30	—	10,000–13,000
D_2	Lignite	—	45–65	6–6·8	30–45	—	7,000–11,000

Class B_2 comprises most of the coking coals; in Class D_1 moisture is 6–20 per cent., in D_2 over 20 per cent.

CLASSIFICATION OF THE IMPERIAL MINERAL RESOURCES BUREAU.

A, Brown coal or lignite; B_1, gas coal and cannel; B_2, coking coal: B_3, house coal; B_4 manufacturing or iron smelting coal; B_5, steam coal; C, anthracite.

Other classifications are also used; one much liked in America is based upon the 'fuel ratio,' that is, the figure obtained by dividing the percentage of fixed carbonaceous residue by the percentage of volatile matter.

CONSTITUTION OF COAL.

The four constituents of coal, as identified by Dr. Marie Stopes, are:—

(1) Fusain. (2) Durain. (3) Clarain. (4) Vitrain.

Some authorities (*e.g.* Stach, *Glückauf*, 1927, p. 759, Kellett, *Trans. Inst. Min. Eng.*, vol. lxxv., 1928, p. 400) hold that Clairain is merely a mixture of Vitrain and Durain. In the U.S.A. these three classes are generally spoken of as Anthraxylon, Attritus, and Fusain. Fusain forms, usually, under 10 per cent. of a coal seam and is the highest in ash, this ash being always high in lime; it also usually carries more salt than the rest of the seam. Vitrain is very low in ash, generally under 0·5 per cent. Durain ash contains mainly silica and alumina. Fusain is non-coking, but its calorific power is high and it is valuable for dust-firing.

CALORIFIC POWER.

The calorific power may be calculated from the ultimate analysis by the 'Deutscher Verband' formula:—

$$\text{Cal. power in calories} = 81\,C + 290\left(H - \frac{O+N}{8}\right) + 25\,S - 6\,W,$$

where C, H, O, N, S, and W are the percentages, respectively, of Carbon, Hydrogen, Oxygen, Nitrogen, Sulphur, and Moisture.

Gontal's formula is applied where only the proximate analysis (percentages of fixed carbon, volatile matter and ash) is known.

Let C be percentage of fixed carbon, V of volatile matter, A of ash; then

$$\text{Cal. power in calories} = 82\,C + a\,V,$$

where a is a coefficient depending upon the percentage of volatile matter in the ash-free coal, *i.e.* upon $\frac{100\ V}{100-A}$ as follows:—

$\frac{100\ V}{100-A}$	5	10	15	20	25	30	35	40
a	145	130	117	109	103	98	94	80

The ash of coals varies usually between 3 and 15 per cent.

1 lb. of ordinary coal requires theoretically 130 cub. ft. of air for complete combustion.

COKE.

Ordinary bituminous coal, containing 25 to 30 per cent. of volatile matter, coked in a modern regenerative bye-product coke-oven plant, yields approximately:—

Coke	70–75 per cent.
Tar	3·5–4·5 „
Ammonia as ammonic sulphate	1–1.25 „
Crude benzol	1 „

Surplus gas, 3,000 to 5,000 cub. ft. per ton of coal.

The composition of this gas is about as follows:—Hydrogen, 48 to 55 per cent.; methane, 25 to 34 per cent.; carbonic oxide, 6 to 8 per cent.; carbonic acid, 2 to 2·5 per cent.; hydrocarbons, 2 to 4 per cent.; nitrogen, etc., 5 to 12 per cent. The calorific power is about 480 to 600 B.Th.U. gross, or 425 to 550 B.Th.U. net.

Weight of Coal.

The specific gravity of coal varies from 1·1 to 1·6, most bituminous coals approximating to 1·3. The following table shows the weight of coal contained in one acre of superficial area per inch thickness of seam.

Table of the Weight of Coal under Different Circumstances.

Specific Gravity.	Weight in the Natural Bed, per Acre, per Inch Thick, in Tons.	Weight of a Cubic Foot in the Broken State, in Lbs.	
		Large Coal.	Small Coal.
1·10	111·411	42·62	37·12
1·15	116·475	44·56	38·81
1·20	121·540	46·50	40·50
1·25	126·604	48·43	42·18
1·30	131·668	50·37	43·87
1·35	136·732	52·31	45·56
1·40	141·796	54·25	47·25
1·45	146·860	56·18	48·93
1·50	151·925	58·12	50·62

Specific gravity × 101·283 = weight in tons per inch thick per acre.

To find the contents of coal of any given specific gravity per acre multiply the figure in the table by the thickness of the seam in inches; the thickness measured along a vertical line should always be taken, as the correct result will thus be obtained, whatever may be the inclination of the coal seam.

The weight of coal in its broken state, that is, as it comes to the surface, in tubs or otherwise, will depend on its largeness; the large has here, however, been computed to weigh in proportion to the solid coal as 62 to 100, and the small as 54 to 100.

The calculation now most commonly adopted in the North of England to ascertain the weight of coal in the solid state is to reckon 19 cwts. per cubic yard = 78·815 lbs. per cubic foot. On this calculation, a seam 1 foot in thickness would contain 1,532 tons 13 cwts. 1 qr. 10 lbs. per acre, and a seam 1 yard thick would contain 4,598 tons per acre.

A summary way of reckoning the quantity of available coal in a given area of a seam is to take an acre of coal 1 inch thick to produce 100 tons. This leaves a sufficient margin for faults and losses in barriers and in working.

Sizing of Coal.

In the coal trade coal is often sold by size, but there is no uniformity in practice even in the same field. Above 4 ins. to 6 ins. is usually sold as Best Coal, House, or Best House; a very general scheme of designations is:—

Cobbles	between 5 ins. to 6 ins.	and about	$3\frac{1}{2}$ ins. to 3 ins.
Treble nuts	„ $3\frac{1}{2}$ „ 3 „	„	$1\frac{1}{2}$ „
Double nuts	„ $1\frac{1}{2}$ ins.	„	1 in. to $\frac{3}{4}$ in.
Single nuts	„ 1 in. to $\frac{3}{4}$ in.	„	$\frac{5}{8}$ in.

Duff, slack or gum below $\frac{5}{8}$ in.

Some collieries also make beans, say about $\frac{5}{8}$ in. to $\frac{3}{4}$ in. and peas, say about $\frac{3}{4}$ in. to $\frac{7}{16}$ in. The dimensions may signify plate screens with round or with square holes, bar screens or wire gauze; in the two latter they may be measured in the clear or centre to centre. All these different methods are in use.

With reasonably closely sized coal of whatever dimensions it is found that as ordinarily heaped the coal occupies 55 per cent. of the total volume of the heap.

METALLIFEROUS MINERALS.

By *metalliferous* minerals or *metallic* minerals are meant those minerals that are technically utilisable for the extraction of the ordinary commercial heavy metals, metals of the alkalies and alkaline earths being excluded; this is almost identical with the meaning of *ores*, or with Group II. of above classification. In United States of America and Canadian statistics, a wider meaning is given to *non-metallic* minerals, which includes all minerals not actually used for the extraction of metals, *e.g.* ochre, although this is an oxide of iron, because it is used as a pigment and not as a source of iron.

COMPOSITION OF MORE IMPORTANT ORES.

Name.	Formula.	Percentage of Metal.
Magnetite	Fe_3O_4	Fe 72·4
Hæmatite	Fe_2O_3	„ 70
Limonite	Fe_2O_3,H_2O	„ 60
Brown hæmatite	Hydrated ferric oxide, with 10-15 per cent. H_2O	„ 45-55
Siderite	$FeCO_3$	„ 48·3
Ilmenite	$(TiFe)_2O_3$	„ 30-40
Pyrites	FeS_2	„ 46·7
Pyrrhotite	Fe_7S_8	„ 60·5
Pyrolusite	MnO_2	Mn 63·3
Manganite	$Mn_2O_3H_2O$	„ 62·5
Wad	Impure hydrated manganese oxide	„ 25-50
Chalcopyrite	$Cu_2SFe_2S_3$	Cu 34·6
Chalcocite	Cu_2S	„ 79·8
Malachite	$2(CuO)CO_2,H_2O$	„ 56
Cinnabar	HgS	Hg 86·2
Galena	PbS	Pb 86·6
Cerussite	$PbCO_3$	„ 77·5
Anglesite	$PbSO_4$	„ 68·3
Blende	ZnS	Zn 67
Calamine	$ZnCO_3$	„ 52
Stibnite	Sb_2S_3	Sb 71·8
Mispickel	$FeAs_2S_2$	As 46
Cassiterite	SnO_2	Sn 78·7
Wolfram	$(FeMn)OWO_3$	W 70
Scheelite	$CaOWO_3$	„ 74·8

IRONSTONE.

Ironstone is the term applied to stratified or symphytic deposits workable for iron, iron ore to epactic deposits, veins or masses. Bessemer ore or hæmatite ore is generally understood to mean iron ore with less than 0·035 per cent. of phosphorus when the ore contains about 50 per cent. of iron. Such ores are usually quoted on a basis of 50 per cent. of iron, with a definite increase or deduction in price for each unit of iron above or below that figure, and a penalty for every unit of silica above a given percentage, often 8 per cent.

The yield of ironstone is often taken roughly at 2,000 tons per foot thick per acre.

HAEMATITES.

These weigh 3–3·5 tons per cubic metre in place, 2¼ tons per cubic metre in mine cars, 2 tons per cubic metre stacked.

TABLE OF BRITISH MESOZOIC IRONSTONES.

Formations.			*Name of Ironstone.*
CRETACEOUS—Neocomian	Wealden	Lower Greensand	Seend, Wiltshire.
		Weald Clay	
		Hastings Sand	Sussex and Kent.
			Westbury, Wiltshire.
			Claxby, Lincolnshire.
JURASSIC	Upper Oolite		
	Middle Oolite	Corallian	Dover, Kent.
		Oxford Clay	Westbury, Wiltshire.
	Lower Oolite	Great Oolite	
		Estuarine Beds	
		Northampton Sands	Northamptonshire.
			Cleveland Dogger or Top Seam.
	Upper Lias	Upper Lias Clay	
		Toarcian	Isle of Raasay.
	Middle Lias	Middle Lias Limestone	Cleveland Main Seam.
			Cleveland Pecten Seam.
			Cleveland Avicula Seam.
		Marlstone	Blenheim, Oxfordshire.
			Melton Mowbray, Leicestershire.
			Caythorpe, Lincolnshire.
	Lower Lias		Frodingham, Lincolnshire.

COMPOSITION OF BRITISH MESOZOIC IRONSTONES, DRIED AT 100° C.

	Iron.	Silica.	Lime.	Sulphur.	Phosphorus	Loss on Ignition
	per cent.	per cent.	per cent.	per cent.	per cent.	per cent.
Cleveland Main Seam	30	12	5	0·3	0·5	26
Northamptonshire	38	17	3	0·15	0·7	32
Lincolnshire	25	9	21	0·2	0·35	36
Oxfordshire	28	13	11	0·15	0·3	35
Raasay	25	10	17	0·5	1·0	20

COPPER, TIN, LEAD AND ZINC ORES.

Copper, tin, lead, and zinc ores are usually sold by assay. The value is calculated on the current market price of the metal multiplied by the percentage of metal present less the percentage estimated to be lost in the operation of smelting; from the value thus found there is then deducted the 'returning charge,' being the estimated cost of smelting the ore, including the smelter's profit, etc. The following were the generally accepted (pre-war) methods :—

COPPER ORES.—Weight paid for is gross weight less moisture and less 1 per cent. draftage. Wet assay less 1 per cent. for ores assaying 3–4 per cent. (deduction varied proportionately for richer or poorer ores); for value per unit (1 per cent.) take 0·01 of price of best selected copper, less returning charge of 2*s.* 3*d.* to 2*s.* 6*d.*

Example.—Cargo of 1,000 tons ore assaying 4 per cent. copper, ¼ per cent. moisture. Net weight $= 1{,}000 - \left(\frac{1{\cdot}25}{100} \times 1{,}000\right) = 987{\cdot}5$ tons.

4 per cent. − 1 per cent. = 3 per cent. copper to be paid for.

Best selected quoted that day 82*l.* 10*s.* per ton = 16*s.* 6*d.* per unit, less 2*s.* 3*d.* returning charge = 14*s.* 3*d.* 3 units at 14*s.* 3*d.* = 42*s.* 9*d.* per ton of ore.

Value of cargo 987·5 tons at 42*s.* 9*d.* = 2,110*l.* 15*s.* 7½*d.*

LEAD ORE.—Weight paid for is net dry weight. Percentage paid for is dry assay less 4 units, on basis of soft foreign lead price in London, subtracting 40*s.* per ton for returning charge. Silver above 3 oz. per ton paid for at current rate for fine silver. Price is understood to be delivered smelter's works.

Example.—150 tons net of galena assaying 78 per cent. of lead and 5 oz. of silver per ton quotations being : lead, 20*l.* per ton; silver, 2*s.* 3*d.* per oz. Lead, 150 tons at (78 − 4) per cent. = 111 tons at 20*l.* = 2,220*l.*, less 150 tons at 40*s.* = 1,920*l.* Silver, 150 × (5 − 3) oz. = 300 oz. at 2*s.* 3*d.* = 33*l.* 15*s.* Total value of ore, 1,953*l.* 15*s.*

ZINC ORE.—Weight paid for is net dry weight. From percentage of zinc, as determined by wet assay, subtract 9 units, multiply by $\frac{18}{20}$ of price of spelter, and subtract returning charge of 3*l.*; difference is price per ton delivered smelter's works.

Example.—If ore assays 50 per cent., London quotation for spelter being 18*l.*, the ore is worth $£\left(\left[\frac{50-9}{100} \cdot \frac{18}{20} \cdot 18\right] - 3\right) = 4l.\ 0s.\ 3d.$ per ton.

The general gangue of most ores is quartz; solid quartz averages about 165 lbs. per cub. ft. (13½ cub. ft. to the ton), broken about 100 lbs. (22½ cub. ft. to the ton), but varies considerably, the weight of veinstuff containing metalliferous minerals can be calculated from the specific gravity of the minerals.

TABLE OF SPECIFIC GRAVITIES AND WEIGHTS PER CUBIC FOOT OF MINERALS.

Mineral.	Sp. gr.	Cwts. per cub. ft.	Mineral.	Sp. gr.	Cwts. per cub. ft.
Copper pyrites	4·1–4·3	2·34	Hæmatite	4·5–5·3	2 73
Copper glance	5·5–5·8	3·15	Brown hæmatite	3·4–4·4	2·17
Malachite	3·7–4·0	2·20	Spathic ore	3·4–3·9	2·04
Cinnabar	8·9–9·0	5·00	Pyrolusite	4·8–4·9	2·67
Galena	7·3–7·7	4·18	Cassiterite	6·4–7·1	3·76
Cerussite	6·4–6·5	3·60	Wolfram	7·1–7·6	4·12
Anglesite	6·1–6·4	3·50	Scheelite	5·9–6·1	3·34
Zinc blende	3·9–4·2	2·23	Quartz	2·5–2·8	1·50
Calamine	4·0–4·5	2·40	Calcspar	2·5–2·8	1·50
Stibnite	4·5–4·6	2·50	Dolomite	2·8–2·9	1·56
Pyrites	4·8–5·2	2·78	Fluorspar	3·0–3·2	1·73
Mispickel	6·0–6·4	3·45	Felspar	2·5–2·8	1·50
Magnetite	4·9–5·2	2·80	Barytes	4·3–4·7	2.51
Ilmenite	4·5–5·0	2·64			

A rib of **mineral 1 in. thick** = 3 cub. ft. per square fathom.

Calculation of Mineral Mixtures.

The percentages of two minerals present in a mixture can easily be calculated if the sp. gr. of the mixture and of the ingredients is known. A convenient method is that proposed by R. T. Hancock.

Take a weight of x grms. of the mixture containing two ingredients of sp. gr., a and b respectively, where $\frac{1}{x} = \frac{1}{a} - \frac{1}{b}$ or $x = \frac{ab}{b-a}$; then the volume of this weight of a alone will differ from that of the same weight of b alone by 1 c.c., and for every 1 per cent. of a (the lighter substance) present in the mixture, the volume will be greater by 0·01 c.c. than the volume of b alone and for every 1 per cent. of b present, less by 0·01 c.c. than the volume of b alone. It is usually convenient to take $10x$ grms., when the differences will be 0·1 c.c.

Example.—To determine percentage of cassiterite (sp. gr. 7) and 'black sand' (sp. gr. 4·5) in a mixture of these. Since $\frac{1}{4{\cdot}5} - \frac{1}{7} = \frac{1}{12{\cdot}6}$, take 126 grms. Volume of mixture, measured by pouring the mixture into a burette partly filled with water and noting the difference in the burette readings, = 20 c.c. Volume of 126 grms. of black sand would be $\frac{126}{4{\cdot}5} = 28$ c.c. Hence percentage of cassiterite = 10 (28 − 20) = 80 per cent.

BORING.

Boreholes are put down by one or other of the following methods :—

I. Percussive: (*a*) Using solid rigid rods. (*b*) Using ropes. (*c*) Using hollow rods (water-flush method). (*d*) Using a hydraulic ram.

II. Rotary: A. Core drills. (*a*) Using diamonds. (*b*) Using steel cutting teeth. (*c*) Using chilled shot. B. Hydraulic rotary oil drill.

Generally speaking, percussive methods are used when the primary object is to get a hole down rapidly, where information as to the nature of the ground passed through is of less importance, whilst rotary core drills, by the use of which a more or less continuous core is obtained, are employed when an accurate knowledge of the strata passed through is the first consideration. Sometimes both methods are combined ; thus in some of the boreholes put down in Kent, boring through the secondary rocks was performed by percussion, whilst rotary core-boring was substituted after the coal measures had been reached. Percussion boring by the ordinary 'English' method can be used to depths of about 500 ft., costing about 10*s.* to 20*s.* per foot, and advancing 2 ft. to 6 ft. per twenty-four hours. Beyond this depth a free-falling cutter must be used, and in this way depths of 3,000 ft. or more may be reached.

Rope boring with round ropes on the American system is used almost exclusively in boring for oil, salt, etc. The plant for a hole 2,000 ft. deep would weigh about 80 tons complete and cost about 2,000*l.* ; holes range from 6 ins. to 15 ins. in diameter. In Pennsylvania and Ohio such boring has been done for as little as 2*s.* per foot, and contracts are usually let at prices ranging from 4*s.* to 8*s.* The rate of boring is from 20 ft. per day upwards. In Middlesbrough, 8-in. holes have been put down to depths of about 1,000 ft. at a cost of about 8*s.* per foot with a speed of 60 ft. per day. Boring with flat ropes by the Mather and Platt system is used for very large boreholes, as for instance for waterworks ; holes up to 45 ins. in diameter have been bored by this method.

The water-flush method, of which there are many varieties (Fauck, Raky, etc.), is used in hard rocks for boreholes of medium size (rarely under 8 ins. in diameter). Very fast boring can be done in this way ; e.g. in the Ruhr district a borehole was put down to a depth of nearly 2,000 ft. at an average speed of 157 ft. per twenty-four hours ; 30 to 50 ft. is quite a usual rate down to depths of about 2,000 ft., and over 20 ft. a day has been bored down to a depth of over 3,000 ft.

The hydraulic ram method appears to offer certain important advantages, especially for boreholes of moderately large diameter, but has hardly passed the experimental stage.

Core Boring.

Diamond boring is in use to-day for hard rocks ; the main drawback is the high price of the 'diamonds' (bort, used in the softer rocks, costs 2*l.* to 3*l.** per carat, carbon costs 6*l.* to 12*l.* per carat), which tells severely against boreholes larger than 6 ins. in diameter. The cost of diamond boring in this country is about 7*s.* 6*d.* per foot down to 500 ft. and 12*s.* to 18*s.* down to 1,500 ft. for a 5-in. borehole. A 5-in. borehole put down in Scotland to 1,800 ft. cost only 6*s.* 4*d.* per foot ; the rate of advance was about 6 ft. per day. In British Columbia a borehole 2,200 ft. deep cost 8*s.* 6*d.* per foot, and was bored at the rate of 10 ft. per day. In Victoria the Government does much boring to aid prospectors ; the bores are mostly shallow, but in

* All costs throughout this article refer to prices before 1913.

one case a depth of 1,110 ft. was reached in about 120 days; the average cost of boring in coal measures in 1912 was 9*s.* 3*d.* and in the harder rocks 14*s.* per foot. In Nova Scotia 2 in. diamond boreholes put down by the Government in moderately hard ground to depths of 200 to 400 ft. averaged about 9*s.* per foot in 1913.

Steel cutting teeth can only be used in tolerably soft ground. The cutter should make 15–25 r.p.m. A 10-in. borehole in Kent was put down by this method by the Calyx Company to a depth of 1,400 ft., the speed at times reaching 10 ft. per hour.

The chilled shot method is very generally applicable, and is used for relatively large bore holes in hard ground. The cutter should run at 100–150 r.p.m.; about No. 5 shot is generally used. It is usually somewhat slower but cheaper than diamond boring. The consumption of shot with a cutter 4½ ins. in diameter ranges from 0·15 lb. per foot bored in shale to 1·2 lbs. in hard sandstone, but may be much higher in broken ground, where shot escapes into the fissures. In Nova Scotia 6 in. Government boreholes by this method averaged 10*s.* 6*d.* per foot in 1913.

The hydraulic rotary drill is the name given to a form now much used in boring oil-wells; it cuts out the whole of the material, leaving no core, the debris being brought to the surface by a water-flush. Over 100 ft. per day has been bored by this method.

Boreholes, especially those bored by rotary methods, are apt to deviate from the vertical, and a number of ingenious appliances have been devised for determining the amount and direction of the deviation. This is specially important when boreholes are put down in connection with the freezing method of shaft sinking (see p. 2579).

Location of Bore-holes for Valuation of Alluvial.*

When ground is systematically tested by pits or boreholes at regular intervals, three conventional views are possible:—

1. That there should be so many pits or holes per acre, or
2. That the distance between pits or holes should be so much, or
3. That no point on the ground should lie beyond a certain distance from a pit or hole.

The first two are those most generally held, but the last seems the most logical, and to attain it with the least amount of work the holes should lie at the corners of a series of equilateral triangles, the length of whose sides, or the distance of the pits apart, is equal to the maximum distance under condition (3) multiplied by $\sqrt{3}$. A line is run in any desired direction and staked at intervals equal to the distance between pits. From these stakes lines are turned off at angles of 60° or 120°, and staked at similar intervals starting from the turning-off point. The stakes are numbered to indicate their position on their own line and the number of the turning-off point, so that the pits sunk at them can be readily identified.

The advantages of this method over the more usual square lay-out are considerable. There is 6 per cent. less work involved in the lay-out of the holes, and 23 per cent. less in the sinking and sampling. The computation of values can also be more accurately done. Interpolation of values can be done on the plan, much as in contouring from spot levels, and a reliable assay plan prepared. (*R. T. Hancock.*)

The following is a list of some of the more important boreholes in the world:—

	£ *s.* *d.*
Lake Well, nr. Fairmont, Marian Co., W. Virginia . 7,579 ft.	
Goff Well, nr. Clarksburg, Harrison Co., W. Virginia. . 7,386 „	
Czuchow, Rybnik, Upper Silesia . 7,350 „	2 3 4 per ft.
Gardena No.1, Dominguez, Cal. (rotary boring). . 7,319 „	
Virginia U.S.A.(1918)7,310 „	
R. A. Geary, Macdonald, Penn., U.S.A. 7,248 „	
Deepest producing oil-well in U.S.A. . 8,700 „	

	£ *s.* *d.*
Paruschowitz, Silesia 6,572 ft.	0 11 6 per ft.
Johannesburg, Transvaal . . 6,340 „	
Dentingen, Lorraine 5,925 „	
Bezuidenville, Transvaal . . 5,800 „	2 2 0 „
Schladebach, Prussia . . . 5,737 „	1 1 2 „
Dornkloof, Transvaal . . . 5,582 „	
Wheeling, West Virginia . . . 4,500 „	
Balfour Main, Fifeshire . . . 4,060 „	

Geophysics for the Location of Minerals.

Magnetic methods; electric methods (self-potential method, equi-potential method, parallel wire method, single electrode method), electro-magnetic methods; gravitational methods (torsion balance); seismic methods; radio-active methods; geothermal methods. A Chair of Geophysics has been established at the Imperial College of Science, South Kensington.

* See 'The Valuation of Alluvial Deposits,' by W. R. Rumbold. *Trans. Inst. Min. Met.*, vol. xxxvii, 1928, p. 437.

Mechanical Drilling.

Drills either rotary or percussive. Rotary drills mostly of the twist-drill type confined to the softer rocks; Steavenson electric drill averages 40 ins. per minute in ironstone, uses 6 h.p. Siemens and others make small electric drills, taking 1 to 2 h.p., boring 6 to 10 ins. per minute in coal. For boring in hard rock the Brandt machine has been used in the Simplon tunnel and in some German collieries; hydraulically driven under head of 1,500 lbs. per sq. in.; bores 15 ft. per hour in gneiss. The Kellow drill used in North Wales is similar; has drilled 5 ft. per minute in slates.

Percussive drills are almost exclusively worked by compressed air. The drills are of two kinds: piston drills, in which the drill bit is rigidly connected with the piston rod, and hammer drills, in which the piston rod strikes on the end of the drill bit, the latter being lighter, smaller, and running at much higher speeds.

Drills may be classified by their valve action :—

1. Valve moved by external tappets.
2. „ „ internal „
3. „ „ air with auxiliary valve moved by tappet.
4. Valve moved by air.
5. Valveless, the piston itself acting as a valve.

Most modern drills belong to Class 3, but several types belonging to Classes 2 and 4 are also in use; very many hammer drills belong to Class 4.

Piston drills are further classified by their piston diameter, ranging from 2 to 4 ins.; 2½-in. drills are largely used for stoping, 2¾ to 3½ ins. for drifting, 3 to 3½ ins. for sinking. In 1902 70 per cent. of the drills in use on the Rand were 3½ drills. Such drills use 60 to 90 cu. ft. of free air per minute at pressures between 60 and 90 lbs. per sq. in. A 3-in. drill evolves about 1¾ h.p., for which about 5 h.p. must be applied to the drill piston and 10 to 25 h.p. to the steam end of a direct-driven air compressor.

At the Rose Deep Mine, Transvaal, a trial run of 31 drills for 6 hours gave air consumption for 3½-in. drills 81 cu. ft. compressed air, free air compressed to 70 lbs. pressure per minute; each drill required the consumption of 43 lbs. coal per hour to supply this air, 3·4 lbs. coal developing 1 h.p. hour. The average power of the compressor engine was 12·7 i.h.p. per drill; average drilling speed 4·5 ft. drill hole per drill per hour.

Relative air consumption of drills :—2½-in. drill, 0·445; 3½-in. drill, 1·000; $3\frac{5}{16}$-in. drill, 1·069; 3¾-in. drill, 1·123.

Average speed of boring, including setting up, is 10 to 25 ft. per 10-hour shift. Costs of boring in hard rock in Sweden are about 8*d.* to 10*d.* per foot bored, of which 40 per cent. is for wages and 30 per cent. for power; average boring speed is 5 ft. 6 ins. to 8 ft. per hour for downward holes and 4 ft. to 6 ft. 6 ins. for upward holes.

Hammer drills use 25 to 50 cubic feet of free air per minute; speed of boring about the same as piston drills, but holes are usually smaller. Hammer drills with a relatively rapid turning action of the bit and using bits with a chisel edge, but specially twisted steel, are used very successfully in coal.

A complete 6-piston-drill plant costs about 500*l.*

Two-stage compressors, delivering air at 100 lbs. gauge pressure, weigh on the average 27 lbs. and cost on the average 12*s.* in the U.S.A. per cubic foot piston displacement.

Sharpening Steel Bits.

A good blacksmith and striker can sharpen 30 to 40 bits per hour.

By the use of power drill-sharpeners, many of the shortcomings attendant upon the hand-sharpening process are got rid of, and drills can be kept true to gauge, with the result that where these machines are used it is possible to accomplish more drilling, whilst the cost of sharpening is considerably reduced.

According to experiment, the fastest cutting bit is the Carr (double chisel bit with central hole), the Z, + and 6 edged star in this order; cutting speed $\propto$ (drill diameter) $^{-2}$; cutting speed $\propto$ air pressure; pressure of 85 lbs. per square inch most generally satisfactory.

(*Bull. Amer. Inst. Min. Eng.*, August 1917, p. 1079.)

The Leyner compressed air operated drill sharpener, which can be handled by one man, is capable of sharpening up to 100 drill bits per hour, the number depending upon the size of the bits and steels to be handled.

In the case of small quarry plants or contract work, where drilling is done in the open, dry steam at a pressure of from 80 to 100 lbs. can be used, but will not give such good results as compressed air. Where a number of drills are used, a compressed air plant should preferably be installed, and is indispensable in underground drilling. With a properly laid pipe line of suitable size, there is very little loss in conveying compressed air from a central plant over very large distances.

Detachable drill bits are giving good results at several large mines, saving in cost of transport of drills. (*Trans. Amer. Inst. Min. Met. Eng.*, 1930 Year-book, p. 139.)

Holman Pedestal Grinder.

With these grinders (fig. 1) it is possible to keep the shanks of drill steels straight and true in addition to trueing up the edges of drill bits and repointing picks when necessary. The machines can be used as general workshop grinders for all kinds of tool grinding.

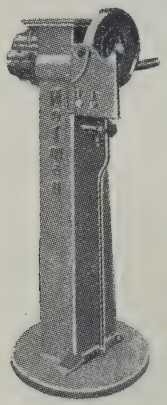


Fig. 1.

The machine does not require any foundation, but will stand wherever required, and is easily moved from place to place.

The machine provides a very economical means for cheap grinding, as it only uses power while the foot control is depressed and when released stops immediately. Full power and speed are attained immediately it is started.

The Holman Twingrip Sharpener.

There are two sizes of the new Twingrip, the smaller being suitable for bits up to 2¾ ins. and the larger for bits up to 5½ ins. in diameter.

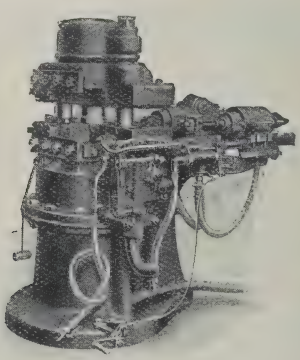


Fig. 2.

All operations except punching are controlled by one handle. A short movement of this towards the rear brings the jaw down to grip the steel in the gripping dies and a further short

[*Continued on p.* 2568.

TABLE I.—CUBIC FEET OF FREE AIR REQUIRED TO RUN ONE DRILL.

Gauge Pressure.	Size and Cylinder Diameter of Drill.												
	A35	A32 A86	B	C	D	D	D	E	F	F	G	H	H9
	2 ins.	2¼ ins.	2⅜ ins.	2¾ ins.	3 ins.	3⅛ ins.	3 7/16 ins.	3¼ ins.	3½ ins.	3⅝ ins.	4¼ ins.	5 ins.	5½ ins.
60	50	60	68	82	90	95	97	100	108	113	130	150	164
70	56	68	77	93	102	108	110	113	124	129	147	170	181
80	63	76	86	104	114	120	123	127	131	143	164	190	207
90	70	84	95	115	126	133	136	141	152	159	182	210	230
100	77	92	104	126	138	146	149	154	166	174	199	240	252

TABLE II.—MULTIPLIERS TO DETERMINE CAPACITY OF COMPRESSOR REQUIRED TO OPERATE FROM 1 TO 70 ROCK DRILLS AT ALTITUDES COMPARED WITH SEA LEVEL.

Altitude Above Sea Level.	Number of Drills.																		
	1	2	3	4	5	6	7	8	9	10	12	15	20	25	30	40	50	60	70
	Multipliers.																		
0	1·0	1·8	2·7	3·4	4·1	4·8	5·4	6·0	6·5	7·1	8·1	9·5	11·7	13·7	15·8	21·4	25·5	29·4	33·2
1,000	1·03	1·85	2·78	3·5	4·22	4·94	5·56	6·18	6·69	7·3	8·34	9·78	12·05	14·1	16·3	22·0	26·26	30·3	34·2
2,000	1·07	1·92	2·89	3·64	4·39	5·14	5·78	6·42	6·95	7·60	8·67	10·17	12·52	14·66	16·9	22·9	27·28	31·46	35·52
3,000	1·10	1·98	2·97	3·74	4·51	5·28	5·94	6·6	7·15	7·81	8·91	10·45	12·87	15·07	17·38	23·54	28·05	32·34	36·52
4,000	1·14	2·05	3·08	3·88	4·67	5·47	6·15	6·84	7·41	8·09	9·23	10·83	13·34	15·62	18·01	24·4	29·07	33·52	37·8
5,000	1·17	2·10	3·16	3·98	4·8	5·62	6·32	7·02	7·61	8·31	9·48	11·12	13·69	16·03	18·49	25·04	29·84	34·4	38·84
6,000	1·20	2·16	3·24	4·08	4·9	5·76	6·48	7·2	7·8	8·52	9·72	11·4	14·04	16·44	18·96	25·68	30·6	35·4	39·84
7,000	1·23	2·21	3·32	4·18	5·04	5·9	6·64	7·38	7·99	8·73	9·96	11·68	14·39	16·85	19·43	26·32	31·36	36·16	40·84
8,000	1·26	2·27	3·40	4·28	5·17	6·05	6·8	7·56	8·19	8·95	10·21	11·97	14·74	17·26	19·9	26·96	32·13	37·04	41·83
9,000	1·29	2·32	3·48	4·39	5·29	6·19	6·96	7·74	8·38	9·16	10·45	12·26	15·09	17·67	20·38	27·6	32·9	37·92	42·83
10,000	1·32	2·38	3·56	4·49	5·41	6·34	7·13	7·92	8·58	9·37	10·69	12·54	15·44	18·08	20·86	28·25	33·66	38·8	43·82
12,000	1·37	2·47	3·7	4·66	5·62	6·57	7·4	8·22	8·9	9·73	11·1	13·02	16·03	18·77	21·64	29·32	34·94	40·28	45·48
15,000	1·43	2·57	3·86	4·86	5·86	6·86	7·72	8·58	9·3	10·15	11·58	13·58	16·73	19·59	22·59	30·6	36·46	42·04	47·47

EXAMPLE.—Required the amount of free air necessary to operate thirty 5-in. 'H' drills at 9,000 ft. altitude, using to operate these drills air at a gauge pressure of 80 pounds per sq. in.

From Table I we find, when operating the drills at 80 pounds gauge pressure at sea level, that one 5-inch 'H' drill requires 190 cubic feet of free air per minute.

From Table II we also find that the factor for 30 drills at 9,000 feet altitude is 20·38; multiplying 190 cubic feet by 20·38 gives 3,872 cubic feet air per minute, which is the displacement of a compressor for the above outfit under average conditions, to which must be added pipe line losses, such as friction and leakage.

movement in the same direction sets the dollying machine in operation to upset the end of the steel. Swaging and sizing of the bits are performed by moving the handle backwards and forwards, causing the top jaw to strike heavy blows whilst the steel is in the formers or the gauging blocks. To punch the hole in hollow steel after making the bit, the steel is first gripped as for upsetting and the punching machine then brought into operation by depressing the foot pedal. The punch is automatically fed forward by air pressure and when the foot pedal is released it is hammered out, the machine ceasing to operate when the pin is clear of the steel.

In the Twingrip there are two operating pistons—one above and one below the jaws. The advantages of this design are :—

A very great gripping force is obtained ; the pressure on the top piston acting directly on the upper jaw is not transmitted through the rod ; the upper jaw and piston being in one piece, the former is adequately guided and the piston rod thus relieved of bending strains ; correct alignment of the jaws is ensured.

Notes on Rock Drills.

In mining practice, drills are required to be of rapid boring capacity, and at the same time of light weight, so that the work may be both rapidly advanced and the whole outfit moved in awkward places with a minimum of trouble. In driving main headings, the piston type of drill, mounted on columns, is generally used. Where space is limited, it is not uncommon to mount two drills on arms attached to one column, two runners and a helper being sufficient to handle the machines so arranged ; in large tunnels the drills, from 1 to 4 in number, and of heavy pattern, are frequently mounted on drill carriages, thus enabling the equipment to be rapidly moved away from the face when the round of holes is ready for firing, without the necessity of dismantling. In stopes, raises, or small headings, the telescope-feed hammer drill, which is largely used, is a one-man drill, and is a rapid boring machine. Where the rock is of such a nature as to cause a considerable amount of dust, a condition which is to be met with in almost every metalliferous mine, drills having an automatic water feed are used. In the case of the heavy drills, the water is fed through the piston and drill bit under pressure ; of this type is the Leyner-Ingersoll. Telescope-feed stope drills can also be arranged for water feed through the heads and drill bits, which have a central hole running from the shank to the cutting face for this purpose.

For hammer drill work, drill bits are furnished with single chisel, double chisel, 6 point and 'Z' cutters, the steels themselves being sometimes hollow and sometimes round, hexagonal or spiral, the section depending upon the class of rock to be bored.

Compressed Air Drilling Plants.

The Hardy-Simplex Air Hammer Rock Drill (*Patent*).

(*The Hardy Patent Pick Co., Ltd., Sheffield.*)

In this drill, fig 3, the valve action is as follows:—When the air pressure is suddenly reduced at one end of the valve, the piston uncovers the main exhaust ports, and the full pressure acting on the opposite end causes the valve to travel to the other side of the valve box and thereby change the direction of the air pressure and reverse the piston or hammer. The action of the valve does not depend on compression set up by the piston in order to accomplish its reversal. It is not dependent on a tight piston and cylinder. It moves only $\frac{1}{14}$ in. in order to cut off the main air supply, but it has a further considerable travel before readmitting air. By the arrangement, all valve losses are practically eliminated, and the delay in readmitting air in quantity for the reversal of the piston diminishes internal resistance, so that a very powerful blow is given with a very small air consumption.

The valve will preserve its air tightness practically indefinitely, whilst at the same time it is perfectly balanced, and so light as to have great rapidity of action. The cylinder and valve box are forged together in one piece of special steel, bored, case-hardened to a glass hardness, and then accurately ground. The valve and valve bush are both manufactured from special alloy steels, both parts being hardened to suit the demands made on them. The rotation is effected by means of a rifle bar and a ratchet wheel situated at the rear of the machine in combination with a number of straight grooves cut in the piston or hammer working through a steel guide nut, situated in the chuck or drill holder. All parts of the rotation device are of hardened steel and are readily renewable.

The drills are retained by means of a cap which is provided with a strong round thread screwing over the front of the drilling machine and embracing the collar on the drill shank. The cap is slotted its full length, thereby enabling the machine to be at once disengaged from the drill steel should this for any reason become necessary.

The drills are made in three sizes, viz., the B5, B6, and B7, weighing 23 lbs., 27 lbs., and 40 lbs. respectively. The B5 pattern is for all overhead work in any class of rock ; it is also used for coal boring on long wall faces and in headings, etc. The B6 pattern will bore easily

holes up to 10 ft. in depth where necessary, in rock of any hardness, the speed of boring varying from 6 to 8 ins. per min. in hard granite to 30 ins. per min. in soft sandstone, and up to 48 ins. per min. in hard coal. Air at 60 to 80 lbs. per sq. in. The B7 pattern has been designed specially

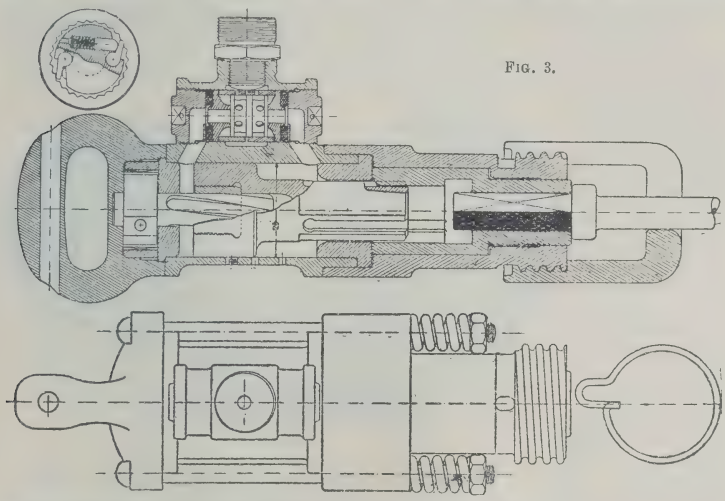


FIG. 3.

for 'down holes' in very hard ground in mine shafts and in quarries. It will easily drill holes up to 25 ft. in depth, starting at 3 ins. in diameter. Its normal boring speed in hard granite is 8 to 10 ins. per min. with air at 80 lbs. per sq. in.

THE HARDY AIR COMPRESSOR.

(Patent.)

(*The Hardy Patent Pick Co., Ltd., Sheffield.*)

This compressor is of the vertical single-acting type, single stage. The idea has been to provide a first-class small compressor of a simple type for driving rock drills, pneumatic hammers, etc., for use in mines, quarries, and engineering shops, and for such purposes up to 80 or 90 lbs. pressure per sq. in.

The lubrication of the whole machine is purely automatic and on the 'splash' principle.

The special feature of this compressor is the valve system. The inlet valve is situated in the piston itself, and consists of a simple ring controlling an annular inlet port in the piston head. This valve opens instantly on the down stroke, and remains open practically until the precise moment when the piston is reversed, admitting air both on its inside and outside peripheries and thereby ensuring a very high volumetric efficiency. The great port opening combined with the absence of spring load, gives an unbreakable inlet valve with a maximum port area and a minimum of complication.

The discharge valve is of the same ring type, made like the inlet valve of a special metal but double-ported and spring-loaded for quick closing to prevent reflux of compressed air after discharge. The discharge valve also uses both inside and outside peripheries, giving an ample port area to prevent undue excess pressure at the moment of discharge. The springs are seated partly in the discharge valve guard and partly in grooves in the discharge valve. They are very light and simple, and may be relied upon to give satisfactory service.

The water-jacketing of the cylinder walls is very thoroughly carried out and results in cool working and moderate discharge temperatures, thereby ensuring a high over-all efficiency of compression. The cylinders are of high-grade iron, crank and connecting rod of forged steel, of full dimensions; large inspecting doors and oil reservoirs are provided. In action the compressor is almost silent.

All machines are now fitted with automatic shut-off valve, closing the suction as soon as the desired pressure is attained; this reopens only when the pressure falls.

The compressors are made in three sizes, giving capacities of 50, 75, and 140 cu. ft., respectively.

The Holman Rock Drills.

The latest Holman drills are known as the 'Streamline' range and consist of various sizes of handrils (hand-held drills) drifters, sinkers and an air feed stoper. The SL 14, an intermediate size drifter, illustrated herewith, may be taken as representing the series.

The almost unbroken exterior of the machine is at once noticeable, the fact accounting for the name 'Streamline.' This feature, apart from other considerations, is of great assistance to the operator, as the absence of projections facilitates the changing of steels and enables holes to be drilled close to the roof, floor or walls of a tunnel. This symmetry is attained by placing the valve at the back end of the cylinder, concentric with the bore.

The valve, which is operated by the difference of pressure on either side as the piston travels backwards and forwards, is 'hat-shaped.' In diameter it is almost as large as the cylinder bore, and as the admission of air is controlled by the circumference, its travel is remarkably small--about 15/1,000 ins. This makes it almost impossible to break a valve in actual work, as the travel is insufficient for it to acquire any great velocity and the impact is consequently light. As the ports are closed by the faces of the valve, wear on the inner and outer circumferences does not cause air leakage.

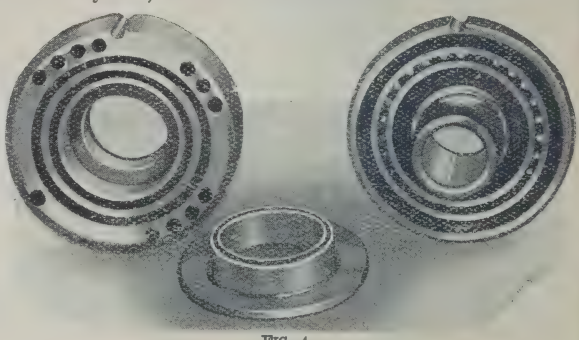


Fig. 4.

On the working stroke, the air passes almost directly on to the piston, the ports being extremely short. This results in a high speed and economy in air consumption. For the return stroke the air has to pass through ports the length of the cylinder, giving the piston time to deliver a full uncushioned blow. The piston of this particular model strikes over 2,000 blows per min. when working at 100 lbs. per sq. in. air pressure.

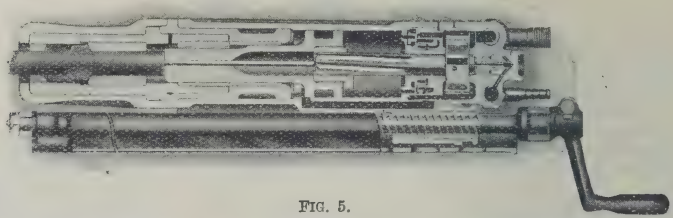


Fig. 5.

Rotation of the drill-steel is effected by the twist bar with spiral flutes working in a twist nut in the rear of the piston. On the forward stroke the twist bar is free to rotate but on the return it is held by the ratchet-gear, and the piston is given a rotary motion. There are straight flutes on the forward end of the piston engaging in corresponding grooves in the chuck guide nut, which transmits the motion of the drill-steel through the chuck, chuck key and bushing, and the lugs formed on the shank of the drill-steel itself.

Water is forced, under pressure, through the water tube, passing through the twist bar and piston and entering a short distance into the hollow drill-steel, right to the bottom of the hole being drilled.

On the cylinder are two machined slides for securing the drill to the cradle. The latter is equipped with a feed screw by which the machine is fed forward as the depth of the hole increases. The cradle is secured by means of a cone clamp to a column for tunnelling work or to a bar or tripod for quarry or open cut working. With the Holman Sliding Cone Cradle, a depth of 56 ins. may be drilled without changing the drill-steel—providing that the rock is such that the wear on the drill bit does not necessitate more frequent changes. This type of cradle also facilitates the 'setting-up' of a drill when starting a new hole.

Metallic Tubing.

Flexible Metallic Tubing (*The United Flexible Metallic Tubing Co. Ltd.*, 112 Queen Victoria Street, London) is specially suitable for mining work. Its durability renders it economic, and it diminishes 'breakdowns,' as the tubing cannot kink and is not easily crushed ; it is also unaffected by changes of atmosphere, or by vermin, white ants, etc.

Wire Clamping Tool for Compressed Air Hose.

Fig. 6 shows a tool in use in a number of the mines for winding wire around the hose to hold the nipple in place, thereby doing away with the usual clamps and bolts.

The dimensions given are approximate for a tool used in clamping a 1-in. four-ply hose. On one end of a piece of ¾-in. pipe 6 ins. long a standard pipe thread is cut, and on the other end a 1½-in. bolt thread about 3 ins. long. The end with the bolt thread is inserted in the hose about 4 ins., and the hose is ready for the wire clamping.

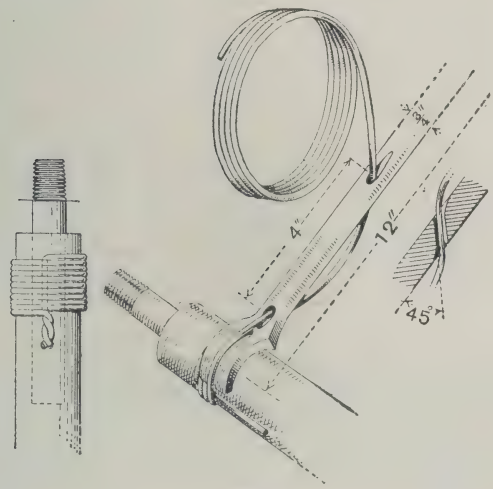


Fig. 6.

A No. 14 Stubs gauge soft steel wire is used. The wire is cut in 9-ft. lengths for winding a 1-in. hose, shorter lengths being used for smaller hose. The wire is first wound up on a piece of 3-in. pipe with a slot cut in it so that the wire can be hooked in it and held in place while the bundle is being wound. When a hose is to be clamped, a bundle of wire is taken and started on the tool as shown in the figure. The wire is fed through the tool until about 4 ins. extends beyond the point of the tool ; this is then bent at a right angle, and is laid on the hose lengthwise. The wire is then wound around the hose layer after layer, and when the final turn is made the two ends are given two twists and the job is done. Any number of lengths of hose may be connected in this manner. The two holes through the handle are drilled at an angle of 45 deg., which gives the proper tension to the wire when winding.

Air Receivers.

An air receiver is not intended so much for storage as to equalise pressure, and to allow of an even flow of air through the pipe, there being a certain amount of pulsation from the air compressor discharge itself. In long pipe lines a smaller receiver should be placed in the pipe line near the point of application of the compressed air, furnished with suitable drain cocks, for the purpose of drawing off the superfluous moisture which might collect, and the consequent

prevention of freezing in the line. The following table gives sizes and details of receivers which are usually considered standard practice :—

TABLE III.—AIR RECEIVERS AND PRESSURE TANKS FOR STANDARD WORKING PRESSURES. Tested to 165 lbs. Water Pressure.

Diam. in Ins.	Length in Ft.	Contents Cub. Ft. (about).	Thickness of Shell. Ins.	Thickness of Heads. Ins.	Weight Lbs. (about).	Diam. of Safety Valve. Ins.	Diam. of Inlet and Discharge Openings. Ins.	Compressor Capacity Receiver is Best Adapted for in Cub. Ft. Free Air per Minute.
18	6	10	3/16	5/16	350	1	2½	90
24	6	18	7/32	5/16	575	1¼	2½	120
30	6	29	¼	⅜	950	1½	3	150
36	6	42	¼	⅜	1,000	1½	3½	150 to 200
36	8	56	¼	⅜	1,350	1½	4	200 to 300
42	8	77	9/32	⅜	1,750	2	5	300 to 500
42	10	96	9/32	⅜	2,000	2	6	500 to 700
48	8	100	11/32	7/16	2,480	2½	6	500 to 800
48	12	150	11/32	7/16	3,000	2½	7	700 to 1,200
54	12	190	⅜	7/16	3,300	2½	8	1,200 to 2,100
60	14	275	13/32	½	5,500	2½	9	2,000 to 3,000
66	18	437	7/16	9/16	7,500	2½	10	3,000 and over
24	6	18	7/32	5/16	625	1½	4	These are only furnished horizontal style and are used as water traps in air lines.
36	6	42	¼	⅜	1,100	1½	6	
48	8	100	11/32	7/16	2,200	2½	6	

(For Air Compressors see p. 1190.)

BLASTING.

Explosives are divided into low explosives, with a slow rending action and fired by simple ignition, and high explosives, with a rapid shattering action, fired by means of a detonator. The rate of explosion of gunpowder, typical of the former, is 3 to 4 metres per second, and of blasting gelatine, typical of the latter, up to 7,000 metres per second.

For blasting in fiery coal-mines explosives known as safety explosives are employed, which have a lower temperature of detonation. No explosive can, however, be considered really safe in a fiery atmosphere. In this country only those explosives are allowed to be used that have passed the Government tests and which are scheduled in the Home Office Order in force for the time being. A shot-hole in coal should never be placed in the solid, i.e. in coal that has not previously been kirved.

If D be the depth of a properly placed shot-hole, the volume of rock blasted out averages with one free face 0·37 D^3 ; with two free faces volume = 0·84 D^3, and with three free faces 1·33 D^3. In drifting and shaft sinking the volume blasted out is approximately 0·75 D^3. If L be the line of least resistance, D=about 1·75 L.

The quantity of explosive required for charging a shot-hole may be calculated by the following formula, modified from Chalon :—

Let Q be the quantity of explosive in lbs., L the line of least resistance in feet, then

$$Q = \epsilon\kappa L^2,$$

where ϵ is a coefficient depending on the explosive and κ a coefficient depending on the character of the rock.

VALUES OF ϵ.

Blasting gelatine	0·14	No. 3 dynamite (33 per cent. nitroglycerine)	0·37
No. 1 gelatine dynamite	0·20	Compressed powder	0·41
Tonite	0·245	Ordinary blasting powder	0·61
Roburite, bellite and similar 'safety' explosives	0·25		

VALUES OF κ.

Very hard rocks, quartzite, hard igneous rocks, etc.	1·0
Hard igneous and metamorphic rocks, etc.	1·0 to 0·8
Schists, crystalline limestones, hard sandstones, etc.	0·8 „ 0·5
Coal measure shales, soft sandstones, etc.	0·5 „ 0·3
Chalk, soft shales, etc.	0·3 „ 0·15
Coal	0·2 „ 0·1
Gravel and other non-coherent rocks	0·1 „ 0·05

Low-tension electric fuses require a tension of about 1 volt; medium, about 4 volts; high, about 100 volts. Low tension fuses best in series, high tension in parallel.

Liquid air is used extensively in the ironstone mines of Lorraine; a cartridge containing sawdust, powdered cork, soot, or a similar substance impregnated with liquid air or liquid oxygen is employed. No detonator is required, but the shot must be fired within 15 minutes of charging. The addition of aluminium powder to the charge greatly increases the detonating action. The quantity of liquid air used is 0·3 litre per ton of ironstone, and the cost of this explosive per ton blasted is given as 0·303 per cent. as against 0·571 per cent. with cheddite and 0·597 per cent. with compressed black powder. Liquid oxygen is preferred to liquid air, because the latter is apt to produce fumes containing oxides of nitrogen which injuriously affect the miners, whereas with the former no noxious gases at all are evolved. It is also used in some of the Lincolnshire * ironstone quarries.

Under Explosives Order No. 1194 of 1921, the production in the mine of an explosive by impregnating carbonaceous material with liquid air or oxygen is now permitted under licence, thus modifying Section 50 of the Explosives Act, 1875. (See p. 2656.)

Detonators.

Detonators.—These are known by numbers, the charge of which varies somewhat. The following is about the average practice:—

Detonator numbers	1	2	3	4	5	6	7	8
Weight of mixture in grains	4½	6	8	10	12	15	23	30

The mixture consists of 80 per cent. fulminate of mercury, 20 per cent. chlorate of potash. Tetryl and Lead Azide mixture are also now used. (See *Proc. Roy. Soc. N.S.W.*, vol. lxiii, p. 35.)

Detonating Cord (Cordeau détonant) consists of a flat tube of lead, tin or aluminium usually 7 mm. wide, 3 mm. thick, filled with nitro-methylaniline or trinitrotoluol, containing usually about 10 grams of explosive per metre. An explosion is propagated through this at a velocity of about 7,000 metres per second. A piece of this cord is attached along a row of blasting cartridges and ensures their complete and instantaneous detonation, even in wet holes. It is stated that the efficiency of the explosive is increased by 20 per cent., and that experiments at Lievin have shown that an explosive mixture of firedamp is not ignited by the detonation of 3 metres of cord.

Ordinary fuse burns at the rate of 30 seconds per foot.

(See also EXPLOSIVES, p. 2797.)

ELIMINATION OF MISFIRING OF CENTRE SHOTS.

As a means towards the elimination of misfiring of centre shots in large-scale blasting in coal mines, the United States Bureau of Mines recommends that the following practices be used whenever shots are to be fired in wet holes: Use waterproofed electric detonators with enamelled leg wires. When making connections with enamelled leg wires, care must be taken to scrape the ends of the wires well, otherwise the enamel will prevent good electrical contact. Fire the shots from an ungrounded power circuit that has a capacity of at least 30 kw. Use extra care when tamping the holes in order not to damage the insulation of the leg wires. Arrange the connections between the detonators so that they are supported clear of the earth or any other conducting medium. (See *Technical Publication* No 257, *Amer. Inst. Min. Met. Eng.*, 1929.)

Drifts.

In collieries, drifts of ordinary size, say 35 to 50 square feet in sectional area in ordinary coal measures, range in price for hand labour in 1913 from 1*s.* to 2*s.* per linear yard for each square foot, according to nature of strata and width of drift, being less the wider the drift. The cost in shale is about two-thirds of the cost in ordinary sandstone. Mean rate of advance of such drifts, about 4 to 6 yards per week in sandstone, 6 to 8 yards per week in shale, 3 shifts per day, two miners in face with two to three labourers.

For inclined drifts rising at high angles the prices will be up to 50 per cent. higher; for drifts dipping at a high angle up to 100 per cent. In very hard sandstone the speed of driving will be about one-half, and the costs about double those given above, and in hard tough igneous rocks, such as basalts, the speed may fall to one-fourth, and the cost rise to four times these figures.

By the use of rock drills the speed may be greatly increased, especially in hard rocks, but the actual drivage costs are not much diminished, except in the harder rocks.

In making height where a top or bottom canch of stone has to be blasted, the price for a top canch of shale ranges from 1½*d.* to 2*d.* per inch for a width of 6 ft.; sandstone, 2*d.* to 3*d.* Bottom canches cost about 20 per cent. more than top canches. These prices include stowing the stone.

Driving in hard carboniferous limestone, drift 30 sq. ft. in cross-section, cost about 60*s.* per yard; three men in three shifts can drive about 4 to 5 yards per week by hand. In one case with

* *Journ. Iron and Steel Inst.*, vol. cxvii., 1928, Pt. I., p. 186.

hand-drilling the cost was 55*s.* per yard, and rate of advance 4 yards per week, and with machine drilling the cost fell to 25*s.* and the rate of advance rose to 10 yards.

In metalliferous mines.—*By hand.*—A drift 5 ft. by 7 ft. in the clear, in rock of medium hardness, can be driven by hand at a rate of 30 to 40 feet per month by employing two ten-hour shifts per twenty-four hours. In soft rock as much as 100 ft. per month can be penetrated. The cost in moderate rock with labour at 12*s.* per shift will average about 2*l.* per foot, including timber and strap rails.

By machine.—A 5-ft. by 7½-ft. drift was run in Colorado, U.S.A., at the rate of 236 feet per month. This speed is above the average and was due to a well-trained crew (three shifts per day) working on a bonus system. Rock was granite, gneiss, and schist, very hard to drill and breaking badly. Labour, 16*s.* per shift. Cost, 2*l.* 14*s.* per foot.

At Cripple Creek, wages 13*s.* 6*d.* per eight-hour shift, ground moderately hard, 896 ft. of 5 ft. by 7 ft. drifts were run by machines, at a cost of 1*l.* 4*s.* per foot; 1,229 ft. of 5 ft. by 7 ft. crosscuts were run at a cost of 1*l.* 9*s.* per foot.

At Park City, Utah, with miners' wages at 12*s.* per eight-hour shift, a timbered drift 7 ft. by 11 ft. was run in quartzite by means of 3½ inch air drill, 3 shifts per 24 hours, at rate of 4·75 ft. per day, at a cost of 2*l.* 4*s.* per foot of drift, or 7*s.* 1*d.* per ton of material broken. No hoisting.

At the same mine crosscuts 7 ft. by 7 ft. in the clear are run by air drills at a rate of 8 ft. per day at a cost of 2*l.* 15*s.* per foot or 15*s.* per ton of material broken.

The Chipeta Adit in Colorado, U.S.A., 7½ ft. by 7½ ft. in clear, was run in hard rock a distance of 1,712½ ft. in 152 days. The best monthly average was 342½. Two 3½ in. air drills were employed. The cost, including track and pipe, power, timber, labour and explosive, was 2*l.* 4*s.* per foot. The amount of powder (40 per cent.) consumed was 14½ lbs. per foot of tunnel. Wages 16*s.* per day.

In Western Australia the drifts vary in size from 5 ft. by 7½ ft. to 6 ft. by 7 ft., and the crosscuts from 5 ft. by 7 ft. to 8 ft. by 8 ft. In eleven of the principal mines the rate of driving and crosscutting varies from 25 ft. to 90 ft. per month, averaging 60 ft. Working costs range from 46*s.* to 103*s.* per foot, averaging 60*s.* The total costs of drives were from 3*l.* 6*s.* to 4*l.* 2*s.* per foot and crosscuts from 3*l.* 3*s.* to 3*l.* 14*s.* At the Braden Copper Mine, Chili, all development work in the nature of drifts, crosscuts and upraises with average section of 4·53 square metres are run at an average cost of 14*s.* per foot driven. This cost includes all labour (native Chilian), supplies, explosives, machinery, and tramming. No timber is used. The work is done under contract and rock is moderately hard and breaks well. Machine drills are used. The rate of progress is about 130 ft. per month or 13 ft. per month per man.

Timbering Drifts.

Props 3 ins. to 5 ins. diameter and crowntrees 5 × 2½ ins. are often used in colliery drifts; in main roads balks of much larger size may be used.

Resistance to crushing of pit props in tons per square inch of small end:

Scandinavian white wood, 1½; Larch, 1½; Scotch Fir, 1½; Oak, 1; Beech, 1⅜.

Prof. Hudson Beare gives the following:—

Scots Pine, 1·767; Spruce, 1·892; Larch, 1·871; Japanese Larch, 1·379; Finnish Red and White Wood, 1·784 tons per square inch of mean area of prop.

In main roads ferro-concrete props are coming into use; their crushing strength is about 1·3 tons per sq. in. Tests by W. A. Machin (*Trans. Nat. Assoc. Col. Man.*, xv., p. 155) gave 1·58 to 1·82 tons per sq. in. on Keedon props 6 ft. long, about 4½ ins. square. Steel props are being largely used often in connection with corrugated steel straps (see *Trans. Inst. Min. Eng.*, vol. lxxvii., 1928–9, p. 158).

Instead of balks, steel H girders are often used. The following are some of the usual sizes:

Depth by width, in ins.	6×6	6×5	6×4	5×4	4½×4
Weight in lbs. per linear foot	31	27	24	22	18
Tons that will be carried in middle by beam 1 ft. long, supported at both ends	160	130	120	105	60

Drystone arching for roadway 6 to 7 ft. high and 3 ft. to 4 ft. 6 ins. wide in Alston district costs about 12*s.* 6*d.* per linear yard for labour. The cost of labour for ordinary arching ranges from 7*s.* to 10*s.* per linear yard. Concrete arching varies from 6*s.* to 10*s.* per cubic yard of concrete put in. Arches of steel H girders supporting galvanised sheets have been used successfully in big drifts. Steel arching, both H and channel, is being extensively used in many collieries (see *Trans. Inst. Min. Eng.*, vol. lxv., 1922–3, p. 124, vol. lxxv., 1927–8, pp. 12 and 93; also *Brit. Eng. Stand. Assoc.*, No. 227, 1926).

A coating of magnesia cement on the timbers of mines is stated to be an economical and efficient assurance against fire, especially in the arid regions where timber becomes highly inflammable and is difficult to replace. It is elastic, adheres firmly to almost any surface, and is stable and durable.

A layer of cement (Gunite) put on by means of a 'cement gun' is sometimes used for covering timbers, also the sides and roofs of drifts, etc.

In drifts in very heavy ground the Schaefer system of lining may be used; this consists of blocks of reinforced concrete, set dry, but keyed into each other, with rows of wooden blocks to take up the squeeze; such linings may deform under heavy pressure, but will keep the drift open. (*Trans. Nat. Assoc. Coll. Managers*, vol. xxiv., 1926-7, p. 122.)

SHAFTS.

(See *' Vertical Shaft Sinking' by G. O. Forster Brown*, 1927.)

Modern shafts are generally either rectangular or circular in cross-section; the former are used in sinking upon narrow veins or seams, where the ground is not very heavy nor the amount of water excessive, or where good timber is cheap; under other conditions round shafts are preferred, but local custom also exercises much influence. Inclined shafts are practically always rectangular; elliptical shafts have been sunk in some places (Forest of Dean) but are now rarely used.

Rectangular Shafts.

Rectangular shafts for large mines may be up to 25 ft. long by 15 ft. wide in the clear. Exceptionally large shafts are used in Pennsylvania, e.g., No. 5 Wilkesbarre anthracite mines, 52 ft. by 12 ft., Exeter Red Ash, 46 ft. 2 ins. by 13 ft. 10 ins.

Round shafts, say from 8 ft. to 24 ft. diameter in the clear; a colliery shaft to wind 1,000 tons per day is usually 16 to 18 ft. diameter.

Rectangular shafts are mostly wood-lined; they are either cribbed or timbered with sets holding lagging in place. In Fifeshire shafts up to 30 ft. by 12 ft. are cribbed with ' barring ' of 9 ins. by 3 ins. up to 6 ins. by 12 ins. The timber for a shaft 23 ft. by 8 ft., including buntons, etc., would amount to about 125 cubic feet, and would cost, including nails, wedges, etc., about 13*l.* 10*s.* per fathom. The cost of sinking (the ground is usually easy) and putting in the timber is about 30*l.* per fathom. In the iron-ore mines of Cumberland and Lancashire shafts about 10 ft. by 5 ft. are cribbed with 10 ins. × 5 ins. plank, the ends being half checked; the total cost including sinking, timbering, timber, stores, etc., is about 30*l.* per fathom, the ground being carboniferous limestone that requires blasting. Sets are made of squared timber 4 ins. × 6 ins. for a small shaft, up to 9 ins. × 15 ins. for a very large one; sets are 3 ft. to 6 ft. apart, held together by hook bolts of 1¼-in. round iron with wing nuts.

Horned sets or bearer sets are put in about every 6 sets or so, depending on the nature of the ground; lagging is often 9 ins. × 3 ins. plank.

Round Shafts.

Round shafts are lined with brick or concrete in ordinary strata; special methods are adopted where shaft traverses water-bearing strata and the lining is required to keep back water. For shafts over 18 ft. in diameter ordinary brick may be used; for smaller shafts wedge bricks should be employed. Let f be the width of the inner face of the brick, b of the outer face, l the length of the brick and R the radius of the shaft. Then $b = \frac{(R + l)f}{R}$; if, as is often the case, $l = 2f$, this becomes $b = f + \frac{2f^2}{R}$. Instead of bricks, firebrick lumps 6 ins. to 8 ins. × 12 ins. to 8 ins. × 10 ins. to 15 ins. may be used. They should also be made wedge shaped in accordance with the above formula. A shaft 10 ft. in diameter lined one brick thick requires about 350 bricks laid in mortar, per foot of depth. The thickness of the lining depends on the nature of the ground and the diameter of the shaft.

COST OF SINKING.

The cost of sinking may be taken as made up of five main items: Supervision, labour, materials, power, and interest on capital expended.

Sinking in ordinary coal measures costs 1*l.* to 2*l.* per fathom for each foot in diameter of the shaft; to this must be added surface labour, fuel, etc., 30*l.* to 50*l.* per week. If more water than 100 gallons per minute is met with, these costs will increase rapidly. Walling, including the cost of the bricks, amounts to between 15*s.* and 30*s.* per cubic yard of brickwork. Walling a shaft 13 ft. in the clear, 15 ft. over all, cost about 15*l.* per fathom = 30*s.* per cubic yard; walling a shaft 22 ft. in the clear, 25 ft. over all, cost 24*l.* 5*s.* per fathom = 15*s.* 6*d.* per cubic yard. A walling curb costs about 10*l.* Sinking a shaft 12 ft. in the clear cost 24*l.* per fathom (sinkers and waiters-on, 13*l.* 10*s.*; general labour, 7*l.* 10*s.*; explosives and stores, 3*l.*). Sinking a shaft 13 ft. in the clear costs 23*l.* to 30*l.* for labour in ordinary strata, and 70*l.* to 100*l.* in basalt, per fathom.

Sinking two shafts at Hylton colliery in 1897–99 cost as follows per fathom:—

	Shaft 20 ft. in the clear.						Shaft 15 ft. in the clear.					
	£	s.	d.	£	s.	d.	£	s.	d.	£	s.	d.
Labour—Sinkers (incl. explosives)	21	0	6				17	0	0			
Wallers	7	7	2				5	11	0			
Miscellaneous	8	7	4				2	16	0			
				36	15	0				25	7	0
Materials—Bricks	9	12	0				5	2	0			
Cement, lime, sand	4	16	0				2	11	0			
Walling cribs	2	6	0				2	0	0			
Miscellaneous	0	15	0				0	10	0			
				17	9	0				10	3	0
Total per fathom				54	4	0				35	10	0

Sinking three 20 ft. shafts at Horden, each 210 fathoms deep, against a flow of 9,000 gallons of water per minute, cost 350,000*l.* (*J. J. Prest.*)

The cost of sinking and tubbing a section of 22-ft. shaft at Blackhall with feeders of water from 300 to 2,885 gallons per minute was as follows :—

	£	s.	d.	
Underground labour	70	10	10	per linear yard.
Surface „	26	13	7	
Materials, coal and stores	87	2	1	
	£184	6	6	

The cost of sinking a 22-ft. shaft through limestone and tubbing it, against feeders of water averaging 3,000 gallons p.m., may be averaged at about 275*l.* per yard of depth.
(*J. J. Prest*, Tr. Inst. M. E. xlvii. p. 611.)

Shafts may be lined with cement, either by setting sheet iron cylinders and ramming concrete round these, lifting the cylinders when the cement has set, or by walling with thin blocks or slabs (about 2 ft. 6 ins. × 3 ft. 6 ins. × 5 ins.), and ramming cement behind this facing ring. The cost of an 18-in. monolithic lining in a 22-ft. finished shaft was made up as follows :—

	s.	d.	
Mixing concrete	1	0	per cub. yd.
Cement	4	2	„
Lining shaft with concrete	2	6	„
Coal and stores	0	9	„
Steel cylinders and other plant	1	4	„
Total	9	9	per cub. yd.

There were 28 cubic yards to the fathom, hence the cost per fathom was 13*l.* 13*s.* It took seven hours to complete the concreting of one fathom.

The cost of walling a similar pit with brickwork was 24*l.* 3*s.* 3*d.*

Lining a 22 ft. shaft at Blackhall with a monolithic cement concrete (6 : 1) lining cost per linear yard :—

	£	s.	d.
Labour in loading sand and gravel	1	18	6
Enginemen and attending mixer	0	4	1
Cement	3	13	6
Interest and depreciation on shuttering, etc.	0	19	0
Coal, stores, etc.	0	10	6
Labour in shaft	3	1	7
	£10	7	2

Lining with concrete blocks 3 ft. 6 ins. × 2 ft. 6 ins. × 5 ins. backed solid by ramming in concrete, all also 6 : 1, cost practically the same. The average cost of sinking and walling with 18 in. concrete was 26*l.* 1*s.* 8*d.* inclusive. Suitable bricks would have cost 45*s.* per 1,000, and at this price brickwork in cement would have cost 15*l.* per running yard. (*Loc. cit.*)

In a 14 ft. pit the total costs of labour for sinking and concreting came to 15*l.* 15*s.* per fathom. The cost of the concrete was about 21*s.*, and the labour of lining the shaft with it about 1*s.* 6*d.*, or altogether 22*s.* 6*d.* per cubic yard. There were 10½ cubic yards per fathom, so that the cost of the concrete lining amounted to 11*l.* 16*s.* per fathom ; with four men at the surface and four men in the pit, the concrete was put in at the rate of a fathom in six hours. At Karwin, Silesia, an 18 ft. shaft was lined with concrete put in from above downwards at the rate of a fathom in fourteen hours at a cost of 62*l.* per fathom complete ; the same shaft lined in the upper part with brickwork had previously cost 132*l.* per fathom. Ferro-concrete has been used in place of simple concrete. At Zweckel, near Gladbeck, a shaft was lined with ferro-concrete slabs backed with concrete, into which the iron framework of the slabs was continued, for 23*l.* per fathom. At Rheinelbe a shaft 20 ft. in diameter was lined with Monier ferro-concrete 11 ins. thick at a cost of 2*l.* 5*s.* 6*d.* per cubic metre of lining, or 30*l.* 12*s.* per fathom of shaft : for a similar shaft at the Hansa Colliery lined 9½ ins. thick the prices were 2*l.* 3*s.* 6*d.* per cubic yard or 29*l.* 14*s.* per fathom.

Shaft Lining.

In heavily watered strata a lining has to be put in strong enough to resist the pressure of water. The thickness of such lining may be calculated as follows :—

Let R be the internal radius of the shaft, s the maximum admissible stress in the wall, p the pressure to be resisted, all to the same units, which are preferably inches and pounds, then :—

$$\text{Thickness of wall} = R\left(\sqrt{\frac{s}{s-2p}}-1\right)$$

(*Aldis*, N.E. Inst. M. E. xxxii. p. 201.)

The resistance to crushing of the various materials used in lining shafts is :—

Common brick	800 to 1,000	lbs.	per square inch.
Fire brick	1,500 to 2,000	,,	,, ,,
Sandstone	2,000 to 5,000	,,	,, ,,
Concrete 1 : 3 : 6 to 1 : 1 : 3.	1,500 to 5,000	,,	,, ,,
Average	3,000	,,	,, ,,
Cast iron	90,000 to 120,000	,,	,, ,,

Safe loads may be taken as $\frac{1}{10}$ of these : for best brick laid in 1 : 3 Portland cement up to 250 lbs.; for ferro-concrete up to 600 lbs. per sq. in.; for frozen sand fully saturated with water at −12° C. up to 340 lbs. per sq. in. (*Alby*, Bull. Soc. Ind. Min. 1895, p. 273.) L. Sauvestre has shown that frozen sand is highly plastic, and that its resistance to crushing increases by 28½ lbs. per sq. in. for each drop of 1° C. (*Annales des Mines*, Ser. xi., vol. ix., 1920, Pt. I, p. 5.)

The pressure to be resisted is usually taken merely as the head of water—namely, 0·434 h lbs. per sq. in., where h is the head expressed in feet. For quicksands Poetsch adopts a formula equivalent to $p = 1{\cdot}042\ h \tan^2\left(45° - \frac{\alpha}{2}\right)$, where α is the angle of repose of the sand ; Poetsch gives $\alpha = 35°$ for fine quicksand, and 20° for coarse sand. These figures are used in calculating the thickness of the ice-wall required in the freezing method of sinking.

Many approximations are used for ordinary shaft linings :—

J. J. Atkinson : Thickness in inches $T = \frac{5{\cdot}2\ R\ h}{s - 0{\cdot}434\ h}$, or for cast iron $T = \frac{5{\cdot}2\ R\ h}{9{,}000 - 0{\cdot}434\ h}$.

Chaudron: for cast iron, $T = 0{\cdot}8 + \frac{0{\cdot}434\ R\ h}{600}$ where $h < 165$ ft. and $T = \frac{130}{h} + \frac{0{\cdot}434\ R\ h}{600}$ where $h > 165$ ft. (here s is taken at only 7,200).

Riemer takes $s = 11{,}400$, and writes :—

$$T = 0{\cdot}8 + \frac{0{\cdot}434\ R\ h}{950} \text{ for } h < 165,$$

$$\text{and } T = \frac{130}{h} + \frac{0{\cdot}434\ R\ h}{950} \text{ for } h > 165.$$

Cast-iron tubbing is always stiffened by ribs and flanges, and these should be taken into account (see *Morrow*, Trans. Inst. M.E. xxxiv. p. 100).

Coffering.

Coffering may be used for moderate pressures and shallow depths ; costs range from 20*l.* 10*s.* per fathom (16-foot shaft) to 38*l.* per fathom (18-foot shaft), being considerably less than cost of cast-iron tubbing.

Tubbing.

Tubbing is put in either by English or German method ; in the former, segments are 2 ft. to 3 ft. deep, flanges are external, not machined, and the joints are made with wood sheeting and wedges ; in the latter, segments are 5 ft. to 6 ft. deep, flanges are internal, are machined, and the joints are made with lead sheeting and bolts. In both forms the tubbing is carried upon wedging curbs, followed usually by a foundation ring. In some few recent cases the wedging curb has been dispensed with, and the tubbing set direct upon a good bed.

Cost of Tubbing.

Cost of English tubbing varies with the price of pig iron, etc., from 5*l.* to 7*l.* 10*s.* per ton ; putting in the tubbing costs 2*l.* to 2*l.* 10*s.* per ton of tubbing put in, or 5*l.* to 7*l.* per fathom of shaft. The following are the detailed costs of tubbing a 15 ft. shaft with 12 segments to the ring, 2 ft. deep, ¾ in. thick, each ring weighing 2 tons 10 cwt. :—

	£	*s.*	*d.*
Three rings, 7 tons 10 cwt. at 7*l.* 10*s.*	56	5	0
Sheeting, 213 ft. at 8*s.* per 100	0	17	0
Wedges, 3,500 at 10*s.* per 1,000	1	15	0
Labour, putting in and wedging	6	0	0
Total	64	17	0
Cost per fathom, about	64	0	0

To the above must be added a due proportion of the wedging curb. The castings for the curb of a 15-ft. shaft cost 25*l.* to 35*l.*, sheeting and wedges 5*l.*, putting in and wedging 10*l.* to 15*l.* Dressing the curb seat may cost from 10*l.* to 40*l.* according to circumstances. About ten men are usually employed in the shaft in wedging tubbing, and they can wedge two to three rings per shift.

The cost of 'German' tubbing usually comes out a little less than English tubbing per fathom. The seams and joints are all machined, this costing 3*l.* 10*s.* to 4*l.* per ring; three shifts of four men can put in and bolt up four rings (20 to 24 ft.) per 24 hours.

The cost of 'German' tubbing in an 18-ft. shaft, the tubbing being 1 in. to 1½ in. thick, 12 segments to the ring 5 ft. deep, was as follows in this country :—

	£	*s.*	*d.*	
Cast-iron tubbing, 13 tons 4 cwt. at 7*l.* 10*s.* .	99	0	0	per fathom
Lead for sheeting and washers, 6 cwt. . .	6	0	0	
Bolts and nuts	3	5	0	
Labour	7	10	0	
Cost per fathom	115	15	0	

In the Minister Achenbach Colliery a shaft 17 ft. in diameter was tubbed with 18 rings of tubbing 8 ft. 2 ins. high, 1 in. to 1¼ in. thick, each ring composed of 10 segments. The weights of material employed were :—

Tubbing . . .	192 tons 10 cwt.	Lead washers . . .	1 ton 2 cwt.
Lead strips . .	2 „ 8 „	Bolts, nuts and washers .	6 „ 3 „

Where the ground is weak and apt to run, suspended tubbing can be used to much advantage. This method was used at Astley Green (*Trans. Inst. M. E.* xxxix. p. 543), also at Hamsterley (*Trans. Inst. M. E.* xxxviii. p. 320); the latter shaft was 10 ft. in diameter and 35 ft. deep. The cost was :—

Materials, 627*l.* 2*s.* 10*d.*, labour, 141*l.* 0*s.* 3*d.*, equal to 83*l.* 16*s.* per fathom. The method is to be recommended for safety, speed and economy.

The above, which may be called ordinary methods of sinking, can only be used where a moderate influx of water has to be contended with; the greatest quantity ever dealt with by these means has been in the sinkings at Blackhall Rocks, where the Horden Colliery Company has sunk two shafts through the Magnesian Limestone in which 14,300 gallons per minute were being pumped from a depth of about 90 yards. Very much less than this is generally sufficient to cause special methods to be resorted to, 5,000 gallons per minute being generally considered prohibitive. The increase of cost due to a heavy flow of water in deep sinkings is well shown by a table of estimated costs of sinking in wet ground in Westphalia :—

Costs of Sinking Shafts 16 to 17 Ft. in Diameter in Westphalia per Fathom.

To Depth of	Where the Average Flow of Water is	
	880 Gallons per Minute.	1,760 Gallons per Minute.
Fathoms.	£	£
27	458	548
55	566	750
82	686	960
110	804	1,170
137	914	1,316
164	1,006	1,554
219	1,188	1,920
273	1,316	2,286
328	1,554	2,652

(*L. Hoffman*, 'Glückauf,' xxxvii. p. 775.)

Cost of Shaft Sinking in Gold Mines.

A shaft 4 ft. by 8 ft. in clear can be sunk in California, where wages are 14*s.* per shift, by hand drilling, at a cost of 4*l.* 10*s.* per foot. This includes labour, timbering, explosives and supplies. The hoisting done by a horse whim and no water to pump. The rate of progress is about one foot per day.

In Nevada a shaft 4 ft. by 8 ft. in clear was sunk 100 ft. in 31½ days by means of hand drilling. The cost per foot was 6*l.* 10*s.* Labour 16*s.* per shift.

Large Shafts.—In Western Australia shafts vary from 4 ft. by 10 ft. to 5½ ft. by 13½ ft. On eleven of the mines in the district the cost per foot of shaft is from 11*l.* to 16*l.*, and from 2*s.* 9*d.* to 5*s.* 6*d.* per cub. foot of material excavated. The average monthly advance was 34 ft. to 60 ft. with an average of 50 ft. Machine drills used on all above shafts.

In Utah, with wages 12*s.* per shift, a shaft 17 ft. by 65 ft. was sunk in ground of medium hardness at a cost of 12*l.* per foot sunk or 1*l.* 16*s.* per ton of material broken. Machine drills used. Pumping from 300 ft. Rate of sinking 0·9 ft. per day.

On the Rand a shaft (Brakpan, Ltd.) 43 ft. by 9 ft. in the clear was sunk at a rate of 204 ft. per month. The working cost was 14*l.* 10*s.* per foot and the total cost 22*l.* 14*s.* per foot. Ground soft. Hand drilling employed.

The following are some of the Rand shafts and the cost per ton of material handled :—

Shaft.	Size.	Speed per Month.	Cost per Ton.	Method Employed.
			s. *d.*	
Rand Collieries . .	34 ft. by 9 ft.	120 ft.	10 9	Machine
Kleinfontein . .	34 ft. by 9 ft.	107 ft.	11 8	,,
Brakpan . . .	43 ft. by 9 ft.	124 ft.	8 11	Hand.
City Reef . . .	46 ft. by 9 ft.	125 ft. 6 ins.	9 2	,,
Hercules . . .	47 ft. by 9 ft.	119 ft.	8 5	,,
Wolhunter . . .	46 ft. by 9 ft.	95 ft.	10 6	,,

The Inspiration Copper Company, Miami, Arizona, sunk and completed a shaft 12 ft. 4 in. by 16 ft. 8 in., 1,400 ft. deep in 10 months.

Some exceptionally fast sinking has been done on the Rand ; thus, in one of the shafts on the Crown Mines 318 ft. were sunk in August, 1925, and in one on the Randfontein Estates 386 ft. were sunk in June 1926.

Ferro-concrete Tubbing.

(The François Cementation Co., Ltd., Bently, Doncaster.)

Ferro-concrete tubbing of monolithic construction is rapidly supplanting cast-iron tubbing as shaft lining. As compared with cast-iron tubbing the cost is, roughly, the same for the first 75 yds. in depth, but thereafter it becomes relatively very much cheaper. Other advantages are that it does not corrode, does not require wedging, repairing, or renewing, but increases in strength with age ; with great inertia it has great flexibility ; it is not appreciably affected by temperature variations, and it is easily adaptable to change of size while sinking. The cost of ferro-concrete tubbing, to resist a static pressure of 250 lbs. to the sq. in., with a safety factor of 2 to 3·5 for an 18-ft. diameter shaft, is calculated at 65*l.* per lineal yard.

Winzing and Raising.

At Park City, Utah, winzes 11 ft. by 7·5 ft. in quartzite are sunk at a cost of 6*l.* per foot sunk, or 1*l.* 10*s.* per ton of material broken. The advance is about 1 ft. per day. In same mine raises, 17·5 ft. by 5·5 ft., are put up at a cost of 1*l.* 14*s.* per foot or 4*s.* per ton of material broken. The advance is at the rate of 3·3 ft. per day.

In Western Australia winzes are from 4 to 5 ft. in width and 6 to 8 ft. long. They are 50 ft. to 200 ft. deep, and the average total cost per foot in eleven of the mines is 4*l.* 15*s.*

The rises in the same district extend from 25 ft. to 120 ft. in height, and are 4 ft. to 5 ft. wide and 6 ft. to 8 ft. long. The total cost on the average is 3*l.* 18*s.* 7*d.* per foot.

Special Methods of Sinking.

In hard, very wet ground :—

Kind-Chaudron : maximum diameter of shaft, 15 ft. ; greatest depth yet attained by it, 1,335 ft. ; Nos. 1 and 2 pits Marsden Colliery (the first sunk in this country by this method) cost respectively 362*l.* 8*s.* and 293*l.* per fathom ; in the North of France shafts 8 to 12 ft. diameter to depth of 100 fathoms cost 80*l.* to 240*l.* per fathom, and the rate of sinking varied from 2 to 6 fathoms per month. In Germany it is estimated that to sink 55 fathoms by this process at a depth of about 300 fathoms would cost about 1,400*l.* per fathom for a shaft 14 ft. 6 ins. in diameter. (See *Bull. Ind. Min.*, 1909, ser. 4, vol. ii, p. 342.

Poetsch Freezing Process.—Applicable both to fissured firm ground and to running ground. The brine used is a 25 per cent. solution of magnesium chloride cooled down to —20°C., its freezing point being about —35°C. The following table shows the pressures at which ammonia gas liquefies at different temperatures :—

Temperatures.		Pressures in Lbs. per Sq. In.
—20°C.	— 4°F.	27
—10°C.	+14°F.	41
0°C.	+32°F.	62
+10°C.	+50°F.	89
+20°C.	+68°F.	125

It is estimated that to produce 100,000 negative calories per hour at —15° C., 50-60 h.p. are required at the freezing plant, and that 0·25 lb. of ammonia and 2 gallons of brine are required for each yard in length of the freezing circuit.

Saturated sand (165 parts water to 1,000 parts sand) frozen at 32° F. has a crushing strength of 242–341 lbs. per sq. in., and at — 13° F. of 2,845 lbs. per sq. in.

Shafts have been sunk to 130 fathoms by this method, the cost for shafts 13 ft. to 16 ft. in diameter ranging from about 150*l.* to 250*l.* per fathom. The cost of sinking two pits, 12 ft. and 16 ft. 6 ins. respectively in diameter, at Anzin, in 1895, to a depth of 50 fathoms, was as follows :—

	£ s. d.	£ s. d.
Cost of freezing plant	. .	73 2 0
Cost of freezing operations . .	. .	48 6 0
Cost of sinking and tubbing : labour .	32 16 0	
Cost of tubbing . . .	66 10 0	
		99 6 0
Total cost per fathom		220 14 0

The cost of sinking a pair of 20 ft. shafts by freezing under ordinary conditions should be :—

	£
To a depth of 50 fathoms	17,500
„ „ 100 „	28,000
„ „ 200 „	50,000

(*W. W. Wilson*, Tr. Inst. M.E. xli. 353.)

Freezing an 18-ft. shaft at Monkwearmouth cost 230*l.* per fathom to a depth of 60 fathoms.

A deep sinking was carried out by this process at Beeringen in 1920 ; a depth of 1,640 ft. of clay, quicksand, and water-bearing chalk was first sunk through in this way, and subsequently the process was applied to the more difficult task of passing through a bed of running sand 46 ft. thick, at a depth of 2,000 ft. under a pressure of 926 lbs. to the sq. in.

(*Annales des Mines*, Ser. xi., vol. ix., 1920, Pt. I. p. 5.)

At Seaham, Co. Durham, there were three freezing units, one of 300,000 negative calories and two of 150,000 negative calories each per hour (300 and 150 h.p. respectively). There were five double-acting brine pumps 10 in. diam., 10-in. stroke ; the brine was calcium chloride solution of sp. gr. 1·25. Two shafts have been sunk 21 ft. diam. in the clear to a depth of about 490 ft. (See 'The Sinking of Londonderry Colliery, Seaham Harbour, Co. Durham, by the Freezing Process,' by J. L. Henrard and J. T. Whetton, *Trans. Inst. Min. Eng.*, vol. lxxv., 1928, p. 358).

In the sinking of No. 19 pit at Courrières in 1909, which was sunk by the freezing method, difficulties were encountered in the upper part of the shaft owing to running water, and these were overcome by the injection of cement to a depth of 57 ft. ; over 1,000 tons of cement were used. (*Annales des Mines*, Ser. xi., vol. ix., 1920, Pt. I. p. 38.)

In sinking through strata containing brine, *e.g.* in certain potash mines, the temperature must be reduced below —35° C. Liquid carbonic acid must be used instead of ammonia in the freezing plant and calcium chloride brine. (*Glückauf*, vol. 63, 1927, p. 293.)

Sheet piling can be used for relatively shallow sinking, near the surface. At Bowburn Colliery Co., Durham, 6 fathoms of wet sand were sunk through by wooden piles in 1906 at a cost of 26*l.* per fathom for timber and 34*l.* 10*s.* for labour. Interlocking steel piling (p. 504) of joists of various forms has also been used.

The dropshaft method is employed in various forms, sometimes by weighting a cutting shoe and curb, on which an iron or brickwork cylinder is built up, or more usually by building a brick or concrete pressure block against which work hydraulic rams that force down tubbing of cast iron, or in exceptionally difficult cases of cast steel. The former method (forcing down a shoe by dead-weight) was found much superior to forcing a shoe down by jacks in sinking a shaft 14 ft. by 8 ft. through 70 ft. of loose strata at Des Moines, Iowa (*Trans. Amer. Inst. Min. Eng.*, vol. lv., p. 232). The following table shows probable costs for shafts of normal size in ordinary running ground (Hoffman) :—

For depth.	Cost per fathom sunk.	For depth.	Cost per fathom sunk.
	£		£
From 15 fathoms to 27 fathoms .	320	From 82 fathoms to 110 fathoms .	1,280
„ 27 „ to 55 „ .	732	„ 110 „ to 137 „ .	1,554
„ 55 „ to 82 „	1,006	„ 137 „ to 165 „ .	1,830

A sinking with a timber drop shaft lining is described in the February, 1922, *Bulletin of the Canadian Institute of Mining and Metallurgy.*

The *Honigmann* system has been used in a few cases down to depths of 70 fathoms on the Continent; the shaft is bored out by a dredge-borer, the walls being kept up by internal hydrostatic pressure, and the shaft is lined with tubbing composed of rings of channel iron riveted together and subsequently lined with concrete.

Cement Injections.—The Sonnenschein cementation process for sinking through quicksands consists in boring a number of holes about 5 m. deep, relatively close together, through the concrete plug in the shaft bottom. When a sufficient number of holes have been bored, water is forced down a bore-hole adjoining the one to be treated, thus forcing the quicksand up the latter, and leaving a conical cavity in the quicksand below the plug. Cement slurry is then forced down the borehole to be treated until the cavity previously formed is filled with cement, and the operation is repeated until a continuous body of cement, about 5 m. deep, is formed below the plug. The cement is then excavated until only enough is left to form a fresh plug, and the entire operation is repeated until the quicksand has been traversed.

The François Cementation Process.

(The François Cementation Co., Ltd., Bentley, Doncaster.)

The origin of the use of cement grout in mine shafts is not certain. Probably the first fully authenticated use was in the injection behind shaft lining at the Rheinpreussen shaft, near Homberg, in 1864, at a pressure of 210 lbs. to the square inch. Monsieur H. Portier at Courrières in 1900 successfully injected and sealed a considerable feeder of water behind an old shaft lining of wood. Cement injection was successfully employed at a shaft sinking in Germany by R. A. Weide, at Pohlhauer, Saxony, in 1901. Early examples of its use in France were by Monsieur Reumaux at Lens; and by Monsieur Deffieux at the Mine de Marles in 1908. In 1911 Mr. Albert J. François, who had previously spent some years on the Continent in experimenting with various systems of cement injection behind shaft lining and through various classes of strata, introduced his process in England, where its possibilities were at once recognised by colliery engineers, and where Mr. François was given every opportunity to improve his patents and develop his process. Over the previous crude systems his methods were as distinctive and advanced as their success was marked, and his process proved a valuable aid to the development and maintenance of coal mines in wet areas.

The process has since been introduced into France, Belgium, Germany, and South Africa, where it is extensively used. In all, forty shafts have been sunk by its means. The preliminary silicatisation of the ground, which is an essential part of the process, was first tried at Hatfield Main, in 1911, with complete success. In porous or finely fissured strata, or alluvial deposits, or generally where the injection of cement slurry proves difficult, the preliminary preparation of the ground by the injection of solutions of silicate of soda and sulphate of alumina is used; the shaft is sunk by ordinary methods until the presence of water is indicated by means of test holes. A number of holes are then bored in the shaft bottom, pipes being cemented in them, their number, position, and inclination being dependent upon the size of the shaft and the character of the strata. Through these pipes boreholes are sunk by means of a diamond drill. On the tapping of feeders by one of the holes the boring is immediately stopped and the pipe in the hole connected by means of a length of high pressure flexible hose to a column of pipes in the shaft, leading in turn to a cement pump and mixing tank on the surface. The holes are arranged in series for cementation and silicatisation, and into them cement pulp and mixtures of chemicals are pumped, the strength and proportion of the mixtures of chemicals, the percentage of cement, and the pressure employed being such as experience has shown to be best suited to the class of strata and pressure of water to be treated. The holes are cleaned out and reinjected or further deepened as necessary, the process being repeated with the various holes, boring and injecting of the different holes taking place simultaneously. The ground is treated in this manner until the desired length of ground has been rendered sufficiently watertight to allow mining and walling operations to proceed. The ground is excavated in the ordinary manner and reinforced concrete walling built up, any water not sealed off by the cementation of the boreholes being shut off by injections between the completed lining and the ground. On the walling being completed a second length of ground is excavated and walled, this procedure being continued till the water-bearing strata have been passed through. The lining employed through water-bearing or bad ground is ferro-concrete tubbing of monolithic construction. Cementation in ordinary ground will cost about 50*l.* per yard for a 20-ft. shaft (see *Transactions of Mining Engineers*, vol. liii., 1916, p. 22).

The process has been used in sinking a shaft 16 ft. in diameter at the Holditch Iron Mine, Chesterton, Staffs. The shaft is sunk through the water-bearing Keele strata to a depth of 632 ft., the upper bed giving off 30,000 gallons of water per hour. The total time occupied was 69 weeks, and the costs were 40*l.* 4*s.* 4*d.* for labour and 36*l.* 3*s.* 9*d.* for material per yard of finished shaft (1916–17), or together 76*l.* 8*s.* 1*d.*; it is estimated that sinking with underhung tubbing would have cost 91*l.* 4*s.* 8*d.* The shaft was lined with reinforced concrete. Cement for injection of boreholes 350 tons 13 cwt., for the concrete walling 998 tons 6 cwt., and for the injection of the walling 50 tons 10 cwt. (*A. Hassam and T. T. Mawson*, Trans. Inst. M.E. lviii. p. 16.)

In porous sandstones *The François Cementation Co.* first injects separately solutions of silicate of soda and sulphate of alumina, and follows this up with cement. This process was used successfully at Hatfield Main, Thorne, etc. Two shafts were sunk successfully at Ollerton by this process.

For a more complete description of the process see paper by H. Standish Ball, *Proceedings of South Wales Institute of Engineers*, vol. xxxvi. No. 2, p. 111.

Mr. Eustace Mitton estimates the cost of sinking through the Bunter sandstone by cementation, including the installation of ferro-concrete tubbing to a depth of 200 yards for an 18-ft. shaft, at £350 per lineal yard. (*Midland Counties Institute of Engineers*, October 15, 1921.)

At two shafts recently completed at Harworth Main, where there was an indicated quantity of water of 3,000 gallons per minute, the shafts were rapidly and successfully sunk by the François process, with a total residual water in both shafts of 6 gallons per minute (see 'Harworth Sinkings,' *Colliery Guardian*, Nov. 17, 1922 ; see also p. 2578) ; while at two shafts recently completed at Ollerton Colliery, where the estimated amount of water was even greater than this, the shafts were sunk by the François process with an average of residual water of only two gallons per minute in each shaft. (See paper read by Mr. H. Eustace Mitton before the Institution of Mining Engineers (*Transactions*, vol. lxx.), November 4, 1925.)

The François cementation process was successfully employed in preference to sheet piling at Haugh Colliery, Kilsyth, Scotland, for sinking from the surface to some 6 fathoms in depth through running ground.

See also L. Souvestrie, *Annales des Mines*, 12th series ; *Colliery Guardian*, Nov. 10, 1922 ; an article in the *Iron and Coal Trades Review*, Sept. 8, 1922 ; *Kropf, Industriebau*, Oct. 15, 1922 ; 'Cementation des Grès Vosgiens,' by F. Arguillère, *Rev. Ind. Min.*, Nov. 1, 1927, p. 451. Papers by A. A. Barnes and J. R. Fox, *Inst. of Water Engineers*, June 1927.

The time required to cement a 20-yd. length of shaft in various rocks is as follows :

Bunter Sandstone . . .	16 days	Limestone . . .	18–19 days
Pennant Sandstone . . .	16 ,,	Chalk . . .	12 ,,

and the quantity of cement required per yard of 20–22 ft. shaft has been found to be :—

Pennant Sandstone . . .	15 cwt.–3 tons	Bunter Sandstone . .	5 tons–6 tons
Magnesian Limestone . .	2 tons–4 tons	Chalk	up to 4 tons.
Triassic Sandstone . . .	3 tons–6 tons		

Where silicatisation is employed, the consumption of silicate of soda is 1½ to 5 tons per linear yd., and that of sulphate of alumina about half of this.

(*Vertical Shaft Sinking*, by E. O. Forster Brown, 1927, pp. 147, 148.)

Under special conditions when very rapid setting cement is required, 'ciment fondu' has been used in place of ordinary Portland cement, and in intensely cold ground a mixture of magnesia and magnesic chloride has been used.

LIST OF DEEP SHAFTS IN THE WORLD (1912).

Mine and Shaft.	Locality.	Depth, Ft.	Mineral Wrought.
St. John Del Rey	Brazil	7,300	Gold
No. 15 Shaft, Crown Mines Ltd. .	Central Rand	6,450	,,
Village Deep Main	,,	6,263	,,
Ooregaum	Kolar, India	6,250	,,
Champion Reef	,,	6,094	,,
Tamarack, No. 3	L. Superior	5,309	Copper
Calumet and Hecla, Red Jacket .	,,	4,920	,,
Victoria Reef Quartz	Bendigo	4,614	Gold
Tamarack, No. 4	L. Superior	4,450	Copper
New Chum Railway	Bendigo	4,318	Gold
Flenu, Ste. Henriette	Belgium	3,809	Coal
Viviers, Gilly	,,	3,750	,,
Przibram, Maria	Bohemia	3,694	Silver, Lead
Przibram, Adalbert	,,	3,665	,,
Lazarus New Chum	Bendigo	3,682	Gold
Kimberley Main Rock Shaft . .	Kimberley	3,601	Diamonds
Kimberley Prospect Shaft . .	,,	3,520	,,
Kennedy	California	3,555	Gold
Westfalen I	Ahlen, Westphalia	3,566	Coal
Sachsen I	Heesen, ,,	3,560	,,
Zwickau, Morgenstern	Saxony	3,550	,,
Long Tunnel Walhalla (incline shaft)	Walhalla	3,450	Gold
Oelsnitz, Frisches Glück . . .	Saxony	3,055	Coal
Dolcoath, William Shaft . . .	Cornwall	3,000	Tin
Kaiser Wilhelm II.	Clausthal	2,960	Silver, Lead
Ashton Moss	Lancashire	2,880	Coal
Cooks Kitchen	Cornwall	2,580	Tin
De Beers Main Rock Shaft . .	Kimberley	2,466	Diamonds
Rosebridge	Lancashire	2,446	Coal

The Barnsley Bed was cut in the Thorne sinkings at 2,765 ft. on August 22, 1924.

The deepest working in the world at present is in the Village Deep at 7,638 ft., 2,044 ft. below sea level.

In 1924 the total depth of all shafts in mines in Great Britain coming under the C.M.R. Act was 845,000 yds, and the ratio of shaft yardage to average tons of daily output was 1 : 1·183 ; in the U.S.A. the ratio of output appears to be at least 15 times as great.

(*Vertical Shaft Sinking*, by E. O. Forster Brown, 1927, p. 372.)

Coal Cutters.

In coal-getting by longwall, props with lids are set along the working face at definite distances apart, there being 1, 2, or 3 rows, as conditions may require. As far as possible, these props are drawn as the face advances, the control of the roof being a matter of prime importance. In the U.S.A. some collieries have adopted adjustable chocks, which can be advanced by screw or hydraulic jacks. Steel props, generally in combination with corrugated steel straps or lids are used extensively in place of wood, and have been attended with a marked diminution of accidents from falls of roof, etc., at the face. Both rigid steel props and composite steel props capable of yielding to roof pressure have been used according to the character of the roof. Steel props have greatly facilitated roof control, the importance of which is generally recognised. (See *Trans. Inst. Min. Eng.*, vol. lxxvi., 1928–9, p. 236; vol. lxxvii., 1928–9, p. 158.) Steel props are generally used in combination with Intensive Mining, in which all operations at the face are carried out systematically in proper succession.

In hand kirving, the average height of a kirving is 13½ ins. in front and 1⅜ in. at back, by a depth of 3 ft., giving a volume of 1·9 cub. ft. of coal kirved out per linear foot of coal face (equal to 16 per cent. on a 4 ft. seam). In a 6-hour shift a hewer in ordinary coal will kirve a bord 5 yds. wide to a depth of 1 yd. or about 7½ sq. ft. per hour of shift. A man will kirve about 9 sq. ft. per hour's actual hewing time after the jud has been nicked in a bord; in a wall he can kirve about three-quarters as much as in a bord, and in longwall work he can kirve one and a-half times as much as in average narrow work. Allowing for his other work a hewer in Northumberland and Durham averages about 38 sq. ft. of kirving per shift.

Machine cutting gives up to about 65 per cent. more output per man employed; on the average about 10 per cent. more round coal is made by machine than by hand and a greater output of coal can be obtained from a given seam.

Coal-cutting machines for longwall work are chain-cutters, disc-cutters and bar-cutters, the former being applicable in seams where the coal is fairly hard, the cleat not too well marked, and the roof tolerably good. The disc machine makes a kirving 3½ ins. to 5½ ins. deep, the bar machine a kirving tapering from 7½ ins. in front to about 3½ ins. at the back. Machines may be air-driven, using compressed air at 35 to 65 lbs. per sq. in. at the machine, or electrically driven (direct, alternating, or tri-phase) at voltages of 250 to 450. The power consumption varies between wide limits, between 12 and 30 h.p. Compressed air coal-cutters cost 250*l.* to 300*l.*, electrically-driven coal-cutters 350*l.* to 450*l.*; a complete plant for a pit using 10 machines would cost about 10,000*l.* with the former and 11,000*l.* with the latter form. The area undercut per hour varies from 45 to 140 sq. ft., 90 sq. ft. being about an average, per hour of running time. An average shift's work may be taken as about 500 sq. ft. Air consumption 1,000 to 3,000 cub. ft. free air per sq. yd. undercut.

According to Mr. Mavor (*Trans. Inst. Min. Eng.*, vol. l., p. 641) a good compressor will compress 6 cub. ft. free air per minute to 60 lbs. per sq. in. per b.h.p. Tests showed coal cutters to use 143 to 1,146 cub. ft. free air per minute, or 620 to 3,642 cub. ft. per sq. yd., at mean pressure of 23 lbs. per sq. in. at machine and 68 lbs. at compressor.

COST.

Mr. Mavor gives average costs as follows:—

	Per shift. s. d.	Per ton. d.
Interest 5 per cent. on 450*l.*	1 9	0·21
Depreciation 12½ per cent.	4 4	0·52
Labour	33 0	4·00
Air power	23 0	2·77
Picks, oil, etc.	8 4	1·00
Repairs	12 10	1·50
Total	83 3	10·00

Gate-end loaders are coming into use. Face conveyors are largely used.*

In narrow work percussive coal-cutters are mostly used, also chain-heading machines and arc wall machines. Percussive cutters supported on a fixed column are largely in favour; they work with compressed air at 50 to 70 lbs. per sq. in., the air consumption being 100 to 300 cu. ft. of compressed air per sq. yd. undercut; different makes run at 300 to 600 blows per minute. A machine has undercut 30 sq. ft. per hour, but their usual performance is 80 to 100 sq. ft. per shift. Percussive cutters of the Ingersoll type on wheels are also used; these machines weigh 5 to 7½ cwts.; they use air at 70 to 90 lbs. pressure per sq in., with an average consumption of 100 cu. ft. of free air per min., and strike about 150 blows per min.; they will undercut 15 to 50 sq. ft. per hour of running time, or 50 to 200 sq. ft. per shift.

Chain heading machines may be driven electrically or by compressed air, using 10 to 20 h.p.; these machines can undercut about 50 sq. ft. per hour, but it is rare that 200 sq. ft. can be kirved per shift. See 'The Driving of Narrow Places,' by S. Walton-Brown (*Trans. Inst., M.E.*, vol. lxxiii, 1927, p. 506).

* See "Underground conveying and Loading of Coal by Mechanical Means," *So. & W. Yorks Coalowners Assocn. and Midland Inst. Min. Eng.*, 1929.

Arcwall machines will cut 10 to 15 places per shift in roads 4 to 5 ft. wide, the usual depth of undercut being about 6 ft.; these machines weigh about 3½ tons, and their motors develop about 40 h.p.

The Stanley header cuts roads circular in cross-section through coal, using 15 to 40 i.h.p. A road 5½ ft. in diameter can be driven through coal at the rate of 10 ft. per 8-hour shift, or six times as fast as handwork, the cost being sometimes rather less, sometimes rather more than hand labour, namely 2*s*. 6*d*. to 4*s*. per foot of advance, everything included, as against 2*s*.8*d*. by hand.

The Jeffrey Entry driver cuts, breaks down, and loads the coal into tubs.

In the U.S.A. Longwall machines with jibs 40 ft. long cut and load 350 tons per shift. A machine with a 50 ft. jib is worked by five men, and in a 6 ft. seam with a strong sandstone roof cuts and loads 200 tons per shift or 400 tons per day; a machine with a 200 ft. jib, to cut and load 1,000 tons per shift. is being built.

In connection with coal-cutters, in thin seams face conveyors are much used for bringing the coal to the gateways. There are four main types:—(1) troughs containing scraper-chains, either endless or reciprocating, (2) endless belts, (3) shallow trucks or trays hauled by ropes, (4) shaker conveyors. The Blackett conveyor is an example of the first type; the chain is 6-inch pitch and rests in a trough 12 ins. wide at the bottom, 18 ins. at the top, 9 ins. deep, made in 6 ft. lengths; it will carry up to 30 tons per hour along faces 100 yds. in length. Most of the other forms have lower capacities. and are suited to smaller outputs or shorter faces. In one case where hand-cutting and filling cost 2*s*. 4*d*. per ton (face costs only), machine cutting and hand filling cost about 1*s*. 8*d*. per ton (1*s*. 4*d*. face costs, plus 4*d*. for power, repairs, stores, depreciation), and machine cutting and conveyor filling 1*s*. 3*d*. (9*d*. and 6*d*. respectively). In most cases a considerable saving is shown, especially in stonework, but the amount varies considerably according to circumstances.

See Report of an Investigation of the Underground Conveying and Loading of Coal by Mechanical Means. *Midland Inst. Min. Eng.* (R. Olive), 1929.

On the Continent shaking trough conveyors are much used, and are also coming into use in this country. (See *Glückauf*, vol. 61, 1925, pp. 89–149, and vol. 63, 1927, p. 1449.)

Face loaders are coming into use in the U.S.A. Amongst the leading types are:

O'Toole loader—undercuts the coal so that it falls on to a belt conveyor.
Jeffrey loader—somewhat similar in principle; it has been introduced into Great Britain.
Wilson chain loader—has gathering arms combined with a conveyor.
Joy loader—is a combined digger and conveyor.
Goodman machine—somewhat similar in principle.
Myers Whalley—mechanical shovel loading into tubs.

Jeffrey Co. and others make scrapers combined with conveyors.

Wages and general costs may be taken as 50 per cent. higher in 1930 than all those here given.

The 'Hardiax' Coal Cutter.
(The Hardy Patent Pick Co., Ltd., Sheffield.)

This cutter consists essentially of a long stroke air-drill, which is fixed in position and held up to its work by means of a solid drawn steel tube provided with powerful jack-screw and

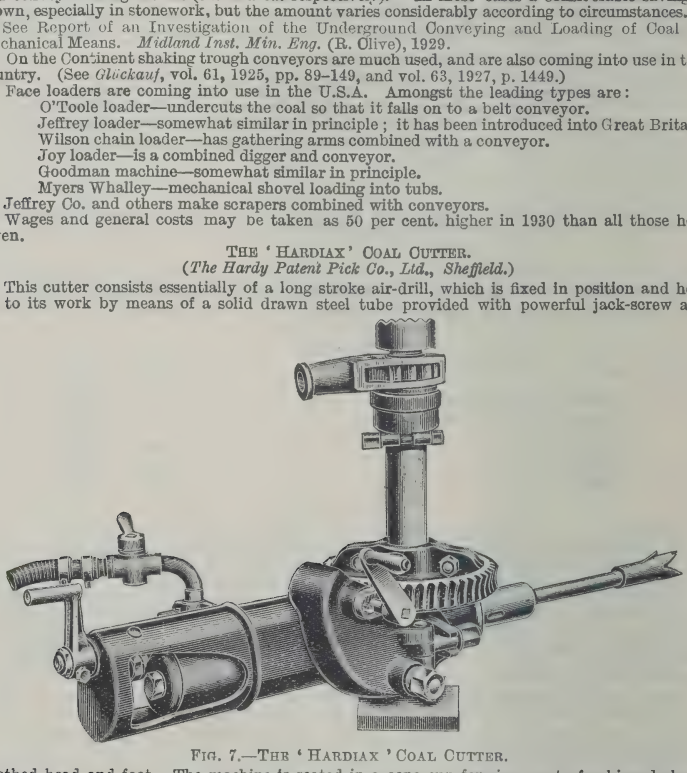


Fig. 7.—The 'Hardiax' Coal Cutter.

toothed head and foot. The machine is seated in a cone cup forming part of a hinged clamp attached loosely to the column. This clamp carries a worm, the teeth of which mesh with those

of a worm-wheel bolted rigidly on the column. By turning the worm-handle the coal cutter is caused to move around the column, whilst the forward movement into the coal is obtained by the feed screw controlled by a handle at the back of the machine. Figs. 7 and 8 give side views of the complete apparatus fixed for work at floor level. The machine may be fixed to cut at any height between floor and roof. The friction between the hinged clamp carrying the machine and the column itself is regulated to a nicety, relieving the worm of the shock of the recoil without in any way impeding the free movement of the machine right and left across the coal face.

When air is admitted to the machine the rapid and powerful blows quickly cut away the coal or other material, whilst the machine is swung right and left by the worm gear and fed forward as is necessary. An arc-shaped and continuous channel or undercut is thus speedily made, increasing in depth and width as each successive length of cutter-bar is inserted in the piston of the machine.

As soon as the desired depth is attained at the centre of the working place the operator commences to play in gradually decreasing sweeps upon the right- and left-hand sides of the arc-shaped cut, using longer cutter-bars where necessary until the undercut or channel is of uniform depth at every point, and square at the sides and corners. When pillars are taken out, or in working on a longwall face, the successive cuts break into one another, giving a continuous undercut of perfectly even depth the whole length of the coal face, whatever this may be. The cut may be any width up to, say, 20 ft. from one setting of the column. The depth of the cut is generally from 4 ft. to 6 ft. and the height 3¼ ins. to 4¼ ins. The diameter of the cutters is varied to suit different requirements.

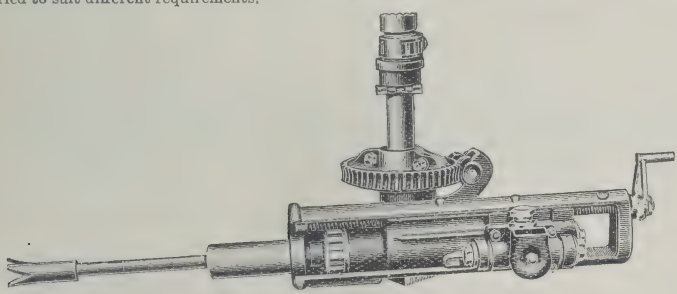


FIG. 8.—THE 'HARDIAX' COAL CUTTER.

The special constructional features of the machine are: (1) Crucible Cast Steel Cylinder. (2) Crucible Cast Steel Valve Box. (3) Crucible Cast Steel Cradle. (4) Crucible Cast Steel Gears. (5) Valve and Main Piston Hardened and Ground.

The machine is provided with a new patented piston valve action, giving great cutting power with low air consumption.

The air-pressure should, preferably, be from 40 to 60 lbs. per sq. in. or more, at which pressure the air consumption is approximately 120 cub. ft. free air per minute.

Shaft Pillars.

The 'draw' in ordinary Coal Measure strata, practically horizontal, is generally between 5° and 15° to the vertical. In Staffordshire 'draws' between 8° 45′ and 11° 10′ were observed in strata dipping 1 in 24 (W. Hay, *Trans. Inst. Min. Eng.* xxxvi. p. 47). The Final Report of the Coal Conservation Committee (Cd. 9084), 1918, p. 64, implies that the maximum angle of 'draw' to be expected is 26° 35′. In secondary strata the 'draw' is generally between 15° and 30°.

There are reasons for assuming that the angle of draw in practically horizontal strata approximates to $45° - \frac{\text{angle of repose}}{2}$.

In Westphalia (*Zeitsch. f. B. H. u. S. W.*, 1897) the draw in the Coal Measure has been found to be 15° on both rise and dip sides when the dip of the seam is $< 15°$; for dips exceeding 15°, the draw is 15° on the rise side and increases uniformly up to 35° on the dip side, reaching this value with seams dipping 35°. In the overlying clay deposits the draw is uniformly about 20°. The depth of subsidence is $kT \cos a$, where T is the thickness of the seam and a its inclination, k is a coefficient, which is 0·8 when the goaf is not stowed at all; with a well-stowed goaf, k is 0·4 when a is 0° to 10°, 0·3 when a is 10° to 35°, and 0·25 when a is 35°.

When a structure on the surface has to be protected from damage, a portion of the seam of coal or other mineral, usually spoken of as a pillar, has to be left unworked, and its position and dimensions must be calculated according to the draw. The simplest case arises when a single bed of mineral, lying horizontally, is worked out, and if the problem is to support a single point on the

surface, obviously the pillar will be circular in plan, having a radius equal to D tan b, where D is the depth of the seam below the point to be protected and b the angle of draw. In fig. 9, PS= D, the depth of the seam, and SM or SN is the radius of the protecting pillar. It follows that if a number of seams one below the other are being similarly worked out, the pillars left in the lower

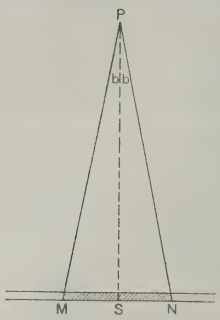


FIG. 9.

seams will be progressively larger in proportion to the depth below the surface at which the seams occur. This method may be fairly well applied as long as the dip of the seam is less than 3°.

When the seam is inclined the position becomes more complicated, and is not thoroughly understood yet. The following may be taken as a general rule: Let b_1 be the angle of draw on the

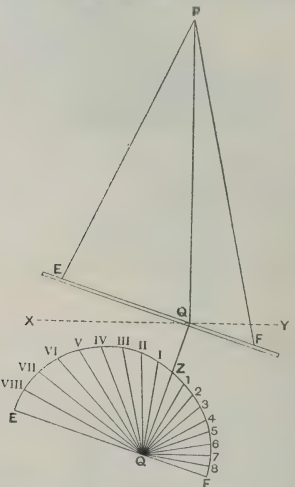


FIG. 10.

lower side of the excavation, *i.e.*, on the rise side of the pillar, and let b_u be the angle of draw on the upper side of the excavation or the dip side of the pillar, b being the normal draw when the strata are horizontal,

then

$$b_l = b + ckf(i) \text{ and } b_u = b - kf(i),$$

where i is the inclination of the seam of mineral measured in the usual way to the horizontal, c and k are coefficients, and $f(i)$ is some function of the inclination. This formula is purely empirical and should not be applied where i exceeds 30°; furthermore, $kf(i)$ can never exceed b.

Apparently $f(i)$ may be taken as sin i, k (in the coal measures) has approximately the value of 18, and c between 1·8 and 2.

Based on this formula, it is possible to calculate the dimensions of the pillar to be left in an inclined seam below a point on the surface which is to be supported, the depth of the seam below the point being, as before, D. It is easiest to calculate the dimensions of the pillar in the plane of the seam, and these can readily be projected on a horizontal plane if required.

The length (in the plane of the seam) of the pillar in the direction of full rise is given by the expression D sin b_l/cos $(b_l - i)$ and in the direction of full dip by D sin b_u/cos $(b_u + i)$, whilst the width at right angles to the dip will be D tan b on either side of the centre. Intermediate points must now be calculated as shown in fig. 10, where P is the point to be supported, Q being a point in the seam vertically beneath it. QZ is drawn in the line of strike, QE to the full rise, and QF to the full dip, the lengths of these lines being determined as above. Divide the quadrant ZQE into equal parts by radial lines set out at angles of 10° to each other, such as the lines Q I, Q II . . . Q VIII. In an inclined plane dipping at an angle of $i°$, a vertical plane making an angle of $x°$ with the horizontal will cut the inclined plane in a line having a dip of $y°$, such that tan y= sin x tan i. Thus the tangent of the inclination of the line Q I = sin 10° tan i, of the line Q II= sin 20° tan i, etc. By substituting the values of these angles of dip for i in the formula given above, it is possible to calculate the angles on the rise and dip sides respectively, and in this way the radii of the pillar in the plane of the seam can be determined, as shown in fig. 10, where D is assumed to be 500 ft., i 18°, and b 10°. The shape of the pillar thus obtained is an inconvenient one, and in practice it is best replaced by a rectangle with truncated corners just sufficiently large to include the curved area obtained by the above calculation. For approximations a circle drawn on the line of full dip may be substituted, or spirals increasing regularly from the width of the pillar on the strike to the length of the pillar on the full rise and decreasing in the other direction from the width of the pillar on the strike to the length of the pillar on the full dip.

(See ' Protection Against Damage by Subsidence,' by H. Louis, *Trans. Inst. Min. Surv.*, 1928.)

Another method, being practically that of Mr. T. A. O'Donahue,* is as follows :—If D be the depth of the seam and T its thickness, in a horizontal seam a pillar of radius R should be set out such that R = (0·19 + 0·01 T)D, all dimensions being in feet. If the seam is inclined at an angle $d°$ as above, the distance from the vertical through the point to be protected to the edge of the pillar on the rise side should be $R + \frac{2D}{3}\left(\frac{\tan d}{1 + \tan^2 d}\right)$, and on the dip side $R - \frac{D}{3}\left(\frac{\tan d}{1 + \tan^2 d}\right)$. A circular pillar should be set out with radius $R + \frac{D}{6}\left(\frac{\tan d}{1 + \tan^2 d}\right)$, the centre of the circle being at a distance $\frac{D}{2}\left(\frac{\tan d}{1 + \tan^2 d}\right)$ to the rise side of the point S. If a rectangular pillar is preferred, the square circumscribing this circle may be set out. Various other formulas have been proposed, but the two above given appear to be satisfactory.

In practice it is generally an area and not merely a point that has to be protected, the area being usually a circle, a square, or a rectangle. As a general rule this can be done in various ways. If the area is a circle or a square the radii of the pillar can easily be calculated. In other cases a number of suitable points on the periphery of the area (*e.g.*, the corners of a rectangle) should be selected and the protecting pillar for each be determined as above; the irregular figure thus obtained should be enclosed in a figure of some simple form (*e.g.*, a rectangle with truncated corners), re-entering angles being avoided. In cases like this, where the solution of the problem involves so many uncertainties, an approximate result is all that can be obtained, and a liberal margin should be allowed for safety.

Bord and Pillar working.—Pillars may be square but are mostly rectangular, sides varying from 30 to 80 yds.; 30 yds. by 40 yds. to 40 yds. by 60 yds. are usual sizes. Some engineers give areas proportional to depth; for seams 100 to 200 yds. from surface, pillars about 30 by 40 yds.; 200 yds. to 300 yds., 35 by 50 yds.; 300 to 400 yds., 40 by 60 yds.

The Coal Conservation Committee of the Ministry of Reconstruction recommends the following proportions :—

Depth from Surface.	Area of Pillar.
< 100 yds.	300 sq. yds.
100–200 ,,	500 ,,
200–300 ,,	800 ,,
300–400 ,,	1100 ,,
400–500 ,,	1400 ,,
500–600 ,,	1700 ,,

* See 'Subsidence caused by Coal-Mining,' by T. A. O'Donahue, *Trans. Inst. Min. Eng.*, vol. lxxviii, 1929, p. 91.

In *panel working* a rib 30 to 60 yds. wide is left between the districts, the latter being usually 20 to 50 acres in area. In American mines the width of rib is often determined by the expression T (5 + 0·01 D), where T is the thickness of the seam and D its depth below surface.

Welsh Stall working.—Single stalls are usually about 12 yds. wide with 12-yd. ribs, double stalls up to 18 yds. wide with 24-yd. ribs between them. In America the following method is sometimes used :—

Width of pillar $W = \frac{T\ D}{100}$, and width of stall $= \frac{D}{W}$, where T is the thickness of seam and D its depth, both in feet.

Subsidences.

(See ' Surface Subsidence caused by Mining,' by H. Louis, *Soc. Ing. Civ. France*, 1928. Also Second and Final Report of the Royal Commission on Mining Subsidence. Cmd. 2899. 1927. ' Mining Subsidence,' by Hy. Briggs, 1929.)

According to O'Donahue, the working out of a coal seam 4 ft. thick will cause a subsidence of the surface of 3 ft. if the seam be not more than 200 yds. deep, and from 12 to 18 ins. if it be 800 yds. deep. It was shown that at Monkwearmouth, working two seams 6 ft. and 4 ft. thick respectively at depths of 1,725 ft. and 1,600 ft., caused surface subsidences of 6 ft. to 6 ft. 9 ins.

At Dunkirk collieries working two seams 5 ft. and 6 ft. thick, 50 yds. apart, at a depth of 445 yds., caused a subsidence of 6·8 ft. in five years and 8·92 ft. in ten years, after which it ceased.

In Staffordshire working a 5 ft. seam at a depth of 1,626 ft. caused surface subsidences varying from 1·34 to 1·47 ft.: these reached their maximum in about 29 months.

In the Doncaster district the surface subsidence may be expected to be about 60 to 70 per cent. of the thickness of the mineral extracted. (Report on Mining and Drainage in an area around the County Borough of Doncaster. Ministry of Agriculture and Fisheries, 1928.)

Final subsidence at the surface is equal to the thickness of the seam actually worked out (*i.e.* the thickness of the seam after allowing for any stowage) less the expansion due to the breaking up of the superincumbent measures. If t is the thickness of the seam and d the depth of the seam, both in feet, and k the coefficient of expansion of the strata, the surface subsidence equals $t - kd$; in ordinary coal measures with seams of moderate thickness and at moderate depths k appears to be of the order of 0·2 per cent., so that if D is the depth of the superincumbent strata in hundreds of feet, then the subsidence equals $t - 0{\cdot}2$ D feet. The coefficient appears to increase with increasing thickness of the seam, which gives the strata more room to break, and it decreases as depth increases, probably because the weight of the overlying strata re-compresses to some extent the strata expanded by fracture, and this effect appears to be sufficiently well marked to prevent $t - kd$ ever coming down to zero, though it is not impossible that it might do so in the case of narrow seams at very great depth.

There are too few data available to give numerical expressions for the above general statements.

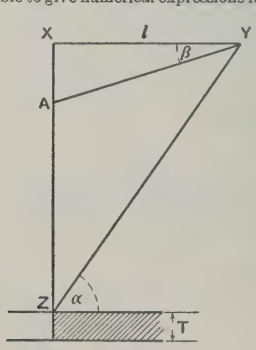


FIG. 11.

Besides the vertical subsidence over the worked-out area, a zone of draw exists over the solid coal, the breadth of which = depth of seam × tan angle of draw as given above. Let XY, fig. 11, represent the zone of draw.

Let S = XA be full vertical subsidence; l = XY be length of draw (width of zone of draw); H = depth of seam; T = thickness of seam; $\alpha°$ = angle of draw measured to the horizontal, said to be about 70° in coal measures; $\beta°$ = angle of inclination of drawn surface.

There is a horizontal displacement from Y to X, and this is a maximum in the middle of the zone of draw, becoming zero at X and Y.

Tan $\beta = \frac{S.}{l}$ l = H cot α. For seams 4ft. to 10 ft. thick, S = (0·5 to 1·0) T, when unstowed, (0·25 to 0·5) T, when hand-stowed. Maximum horizontal displacement varies from 0·05 to 0·3 T and = $\frac{Sl}{4H} = \frac{S \cot \alpha}{4}$.

(*A. H. Goldreich, 'Die Bodenbewegungen im Kohlenrevier und deren Einfluss auf die Tagesoberfläche.'* 1926.)

In the Zwickau (Saxony) district seams dipping 15°, aggregating 28 ft., worked at depths from 2,000 ft. to 3,000 ft., cause subsidence of 70 per cent. of thickness worked by goafing, 40 per cent. by hand-stowage, 10 to 5 per cent., averaging 7 per cent., by hydraulic stowage. The angle of draw by the two former methods is 80°, flatter by last-named method. In mining under densely built-over areas hydraulic stowage gives better protection against damage than leaving safety pillars. (*Goldreich, op. cit.*)

Working Costs in Getting Coal.

Working brokens is usually about 25 per cent. cheaper than working in the whole coal.

(But see 'The Driving of Narrow Places,' by S. Walter-Brown, *Trans. Inst. M.E.*, vol. lxxiii. 1923, p. 506.)

Typical examples of basis yard prices in whole coal to show relative costs:

	2 yds. wide.	3 yds. wide.
Winning headways	1*s.* 6*d.*	1*s.* 2*d.* per yard.
Holing walls	1*s.* 5*d.*	1*s.* 2*d.* ,,
Bords	1*s.* 1*d.*	0*s.* 11*d.* ,,

In brokens.—Jenkins, 3 yds. wide, 8*d.*; half-pillar walls, 3 yds. wide, 10*d.*; skirtings, 3 yds. wide, 6*d.* per yard.

Allowances for wet places ½*d.* to 1*d.* per ton; for band up to 2½ ins. thick, ½*d.* per ton, with ½*d.* additional for every further inch in thickness of band. For taking down ramble 3 ins. to 6 ins. thick, 1*d.* additional per ton, with 1*d.* additional for every further 3 ins.

Longwall.—Hewing price is usually about halfway between whole and broken tonnage price. An average hewing price may be taken to be 1*s.* 6*d.* per ton for a 3-ft. seam, the price rising or falling 1*d.* per ton for every 1 in. decrease or increase in thickness of seam.

All these prices are subject to local percentage increases.

PERCENTAGES OF ITEMS IN PRICE OF COAL AT PIT'S MOUTH.

Year	1913.	1901.	1866–96.	1700–50.
	Per cent.	Per cent.	Per cent.	Per cent.
Royalty	5·35	7·96	8·5	12·5
Wages	62·55	67·37	66·0	37·5
Materials	16·45	16·54	17·0	25·0
Administration	7·00			
Interest and Profit	8·65	8·13	8·5	25·0

Recent costs are given as follows:—

	Average for 1913.*	Average for Year ending March 31, 1920.*	Average for 1922.	Average for 1923.†	Average for 1924.†	Average for 1925 (10 months.)†
	s. d.	*s. d.*	*s. d.*	*s.*	*s.*	*s.*
Labour	6 4	19 7¼	12 1	11·90	13·24	13·05
Timber and stores	1 0	3 10	2 4	1·97	1·99	1·79
Other costs	0 11	1 6		2·72	2·86	3·03
Administration, etc.	—	0 1	3 2½	—	—	—
Control and contingencies	—	0 2		—	—	—
Royalties	0 5½	0 6½	0 7	0·42	0·44	0·45
Capital adjustments	—	0 4	—	—	—	—
Owners' profits	1 5	1 2	0 11½	0·53	0·54	−0·08 (loss)
Pithead value	10 1½	27 3¼	19 2	17·54	19·07	18·24

Hydraulic Stowage.—The best built packwall will compress down to 70 per cent. of its original height, whilst hydraulically stowed material will not compress to less than 90 per cent. Material used in Germany varies from sand to washery residues about 1½ in. cube. Cast-iron pipes 6 to 8 ins. diameter are used, often lined with hard cast-iron or porcelain. A cast-iron tube 7½ ins. in diameter ¾ in. thick will transport about 1,300,000 cub. yds. of stowage material before it is worn out. The average cost of tubes is 3*d.* per cub. yd. stowed. Average rate of stowage 25 to 30 cub. yds. per hour; some mines stow over 4,000 cub. yds. per day. The total cost of hydraulic stowage varies from 4*d.* to 1*s.* 6*d.* per ton of coals got—average appears to be under 1*s.*

In a trial run in Scotland the working costs amounted to 9*d.* per cub. yd.

(Professor R. W. Dron, *Institute of Mining Engineers*, November 1923.)

According to P. Francke (*Glückauf*, 1926, p. 1156) hand stowage produces a settlement of 40 per cent. of the thickness of the seam, including 10 per cent. settlement before stowing, and hydraulic stowage a settlement of 5 to 15 per cent., averaging 10 per cent.

A modified form intermediate between hydraulic and hand stowage is described in *Ann. Mines Belg.* vol. 28, 1927, p. 501. Pneumatic and mechanical stowage are also being used.

Colliery Consumption.

According to the Coal Conservation Committee of the Ministry of Reconstruction, 1918, colliery consumption in Britain varies from 5 to 9·43 per cent., averaging 6·8 per cent., of which 6·2 per cent. is produced by the direct combustion of coal under boilers, and the balance from outside sources. In 1923 colliery consumption amounted to 6·11 per cent. and miners' coal to 2·34 per cent. of the output, and in 1924 to 6·21 per cent. and 2·46 per cent. respectively.

The capital cost of a modern colliery may be taken at 20*s.* to 30*s.* per ton of coal got per annum.

Mining in Veins and Masses.

Methods employed in veins are:

Underhand stoping.
Overhand stoping.
Combined underhand and overhand stoping.
Rill stoping.
Shrinkage stoping or magazine mining.

Irregular deposits, wide veins, thick and steeply inclined seams may be worked by the following methods:—

Lost pillars.
Vertical slices.
Horizontal slices taken transversely.
Horizontal slices taken longitudinally.
Top slicing.
Caving.
Magazine mining.
Square set timbering.

Of these, overhand stoping is the most general, underhand stoping being only used in quite special cases, as when there is only a small quantity of payground to be taken below a given level. Overhand stoping can be used for veins of medium width, say up to 10 ft. on the average. Rill stoping is mainly employed in Western Australia. Magazine mining suits veins of any width, provided that they are not too flat, that the walls are fairly good, and that the whole of the veinstuff has to be extracted. Slicing and caving are used in wide veins and irregular masses when the value of the mineral is not great and where cheap methods of working are essential; it is largely used in the hæmatite mines of Cumberland and Lancashire. Square sets can only be used where the mineral is valuable and where large timber is abundant and cheap.

* *Third Report of Fuel Economy Committee. Brit. Assoc.*, 1920.
† Royal Commission on Coal Industry (1925).

The following classification has been drawn up by the Committee on Mining Methods of the *Amer. Inst. Min. Met. Engs.*, vol. lxxii. 1925.

- **Open cut with or without stripping**
  - Benches
  - Glory holes and milling
  - Placer mining
    - Hand sluicing
    - Dredging
    - Hydraulicing
- **Underground methods**
  - Breast stoping (tabular flat-dipping deposits or beds)
    - Continuous horizon
    - Casual pillars
    - Room-and-pillar
  - Underhand stoping (veins and large masses)
    - Glory hole and milling
    - Open stope
      - With pillars
      - With stalls
      - With square set timbering
  - Overhand stoping (steep-dipping veins or masses)
    - Open stope
      - With pillars
      - With stalls
      - With square set timbering
    - Filled stope
      - Horizontal cut and fill (flat-back)
      - Inclined cut and fill (rill)
      - Square set timbering
      - Piercing or stripping
    - Shrinkage stope
    - Sub-level stopes
  - Top slicing (wide veins or masses)
    - Continuous horizon
    - Panel slicing
    - Block slicing
    - Inclined slicing
  - Caving (large masses)
    - Sub-level caving
    - Block caving
      - Undercut from main level
      - Undercut from sub-levels
      - Back caving into chutes
      - With branch raises
- **Combined methods (large masses)**
  - Alternate shrinkage stope, and pillar
  - Pillars mined by: top slicing, sub-level caving, or block caving
  - Square set timbering: pillars mined by slicing

See 'The Working of Unstratified Mineral Deposits' by G. J. Young, 1927.

The costs of stoping vary within very wide limits, and may be illustrated by typical examples.

In Western Australia, with the contract system in force, a miner breaks 7·31 tons of ore per shift. Formerly a miner on day's pay broke only 4 tons per shift.

In eleven of the West Australian mines treating from 9,000 to 22,000 tons of ore per month with veins varying from 4–20 ft. to 6–60 ft. in width the stoping costs per ton of ore ranges from 6*s.* 4*d.* to 9*s.*, averaging 7*s.* This includes breaking ore, filling stopes, trucking, hoisting, etc. On the Rand it costs from 3*s.* 6*d.* to 5*s.* 6*d.* to stope a ton of milling ore in stopes 40 to 60 inches in height. To these figures must be added cost of compressed air, rock drill maintenance, steel sharpening, and administration underground. The first three factors amount roughly to about 1*s.* per ton, while the last would not exceed 1*d.* In some Rand mines one man can break 12 tons per shift per machine, and uses one case (50 lbs.) of blasting gelatine to break 3·5 fathoms of ground. At other mines only 7 tons are broken per man and one case of explosives is used to break 2·5 fathoms.

In Nevada at the Silver Peak Mines each miner breaks 8·6 tons of rock per shift. The total stoping cost is 5*s.* per ton. Miners here receive 16*s.* per shift.

At the Braden Copper Mines native miners break 12 tons of ore per man per day with rock drills in moderately hard rock. The total cost per ton of ore broken, including superintendence and general charges, is given as 1*s.* 9*d.*

In Utah in quartzite with a stoping width of 10 ft., square set system, small air drills, the stoping cost per ton varies from 6*s.* to 8*s.*

At Douglas Island, Alaska, the ore is a mineralised igneous rock of fair hardness, the workable portions of which vary upwards from 30 ft. in width. Wages 12*s*. Machine used 3¾ inches piston diameter. The following figures are taken over a year's working :—

COST OF BREAKING GROUND AT DOUGLAS ISLAND, ALASKA.

General Details.	Ready Bullion.	700 Ft.	Mexican.	Treadwell.
Average number of drills used	10·12	11·586	14·917	39·13
Tons ore broken	151,460	238,696	232,875	830,551
Tons waste broken	2,756	—	10,509	1,670
Tons broken, total	154,216	238,696	243,384	832,221
Total footage drilled	241,679	269,990	362,519	803,542
Tons ground broken per machine per shift :				
Stoping	47·36	42·998	41·26	—
Developing	8·65	11·323	6·58	—
Cutting out	14·30	7·334	10·34	—
Average	23·06	30·41	24·19	28·47
Footage drilled per machine per shift :				
Stoping	29·39	33·587	33·50	—
Developing	39·14	37·78	39·71	—
Cutting out	40·11	33·954	34·79	—
Average	36·12	34·37	35·7	29·41
Tons broken per foot of hole drilled :				
Stoping	1·61	1·28	1·23	—
Developing	0·22	0·19	0·165	—
Cutting out	0·31	0·33	0·30	—
Average	0·64	0·88	0·669	0·96
Costs of breaking ground per shift per machine :	£ *s*. *d*.	£ *s*. *d*.	£ *s*. *d*.	£ *s*. *d*.
Labour	1 14 0	1 8 0	1 10 0	1 9 0
Explosives	9 8	11 4	12 2	11 4
Supplies and power	1 0 0	18 2	1 0 0	1 0 0
Total	3 3 8	2 17 6	3 2 2	3 0 4
Cost per foot of hole drilled	1 10	1 8	1 9	2 0
Cost per ton of ground broken	2 10	11	2 8	2 1
Total cost of mining, including development	4 0	4 9	4 10	4 6
Percentage of total mining cost	2 9	8	2 2	1 11

In the Granite Gold Mine, B.C., on a vein of hard quartz from 8 inches to 5 feet wide dipping 35° to 50°, the stoping costs per ton are as follows :—

	s.	*d*.
Stoping by hand	14	0
„ by ¾-in. piston drills	11	6
„ by 2½-in. „ „	8	4
„ by 2-in. hammer drills	6	4

Mining costs at the Elko Prince Mine, Nevada, in 1917, stoping a 2 ft. 6 in. vertical vein of hard quartz by magazine mining, using hammer drills with 1⅛ in. cruciform steel, were as follows :—

	Per Ton of Ore $
Timbering, including ore shoots	0·5588
Blacksmith	0·0995
Stoping	0·8083
Tramming	0·2822
Hoisting	0·1956
Compressed air	0·1288
Explosives, detonators and fuse	0·4452
Repairs	0·0486
Drill steel	0·0570
Supervision, taxes, assaying, etc.	0·4987
Cost per ton	$3·1227 = 13*s*.

(*Bull. Amer. Inst. Min. Eng.*, Aug. 1918.)

Costs of stoping in 1903 in two mines on Cripple Creek, Col., in narrow veins: machinemen, $4; miners, $3 for 8 hour shift; coal, $4·60 per ton; timber, $20 per mil.

	I.	II.	
Machinemen and miners	$0·34	$0·369	
Shovellers and trammers	0·41	0·401	
Timbermen	0·23	0·215	
Total labour underground	$0·98		$0·985
Compressed air, rock drills, sharpening	0·15		0·179
Tramming, trams, etc.	0·03		0·034
Explosives	0·13		0·139
Timber	0·28		0·310
Hoisting and surface transport	0·23		0·254
Miscellaneous stores	0·04		0·291
Supervision and general expenses	0·23		
	$2·07		$2·192
	= 8s. 7½d.		= 9s. 1½d.

For detailed costs of mining operations in various West Australian mines, see pages 2594–2597.

Drilling.

Ordinarily two men can drill double-handed 25 to 33 ft. of 1-inch holes per day in limestone, 4 to 8 ft. in granite. With double hammers 10 ft. of 2-inch holes can be drilled in medium rock.

On the Rand generally one white man and five Kaffirs run two air-drills; in some cases one white man and seven Kaffirs run three drills. Eight 6-ft. holes is considered a shift's work for two machines; these holes break on an average two fathoms of ground. The cost is as follows:—

	£	s.	d.
One white man	1	5	0
Five Kaffirs		15	0
Powder for two machines		16	0
Maintenance		4	0
Total	3	0	0 per fathom.

In Chili, with native labour, contracts are let on the basis of the number of holes drilled. The rock is moderately hard.

By hand.—1,008 man-days, by single hand work, drilled 13,855 ft. at a cost of 4·1*d.* per foot. Average amount drilled per man per day 13·8 ft.

By machine.—The actual cost of drilling for one month was as follows:—164 man-days (9 hours) with two ¾-in. air drills, drilled 5,082 ft. at a cost of 1·01*d.* per foot. Average amount drilled per man-day, 31 ft.

It is well known that drilling in siliceous rocks produces a dangerous dust, recognised as the origin of Miners' Phthisis. The amount of dust suspended in the air is tested by the Kotzé konimeter; the atmosphere is considered safe if there are less than 300 dust particles per c.c. (Final Report of Miners' Phthisis Prevention Committee, Pretoria, S. Africa.)

The Safety in Mines Research Board has devised a filter bag connected by a flexible tube with the hole which is being drilled; this catches the particles of dust produced in drilling. (Paper No. 23, 1926.)

See "Silicosis and Coal Mining," by J. S. Haldane, *Inst. Min. Eng.*, Jan. 1931.

HAULAGE.

Rails.

Mine rails weigh 10 to 40 lbs. per yard. The following table shows the least weight that should be used for the given loads (weight of 1 loaded tub) and low speeds:—

Loads, in cwts.	7	10	15	20	25	30
Weight of rails per yard, lbs.	10	12	15	18	20	22
Gauge of track, in inches	15–18	15–18	16–20	18–22	20–26	20–30

For speeds above 5 miles per hour heavier rails are to be preferred.

Weight of 1 mile of single track in tons = 1·571 × weight of rail in lbs. per yard.

Sleepers 2 ft. 6 ins. to 3 ft. 6 ins., usually 3 ft. apart.

Sleepers 8 ins. to 12 ins. longer than gauge of track, 3–4 ins. deep, 6 ins. wide. A pair of fishplates and 4 bolts for 18 lb. rails, weigh 4–8 lbs., usually 5 lbs.; 100 dogspikes for same weigh about 14 lbs. Weight of fishplates, bolts and spikes is approximately 1 cwt. per ton of rails.

[*Continued on p.* 2598.

COST OF MINING OPERATIONS IN VARIOUS WEST AUSTRALIAN MINES.

General Details.	Shaft Sinking.			
	Oroya Brownhill.	Sons of Gwalia.	Lake View.	Great Fingall.
Size of machine used . . . ins.	3¼	3¼	3⅝	—
Time taken rigging and pulling down machine	1 hour (3 men)	5 hours	4 hours	—
Time boring	8 hours	24 hours	16 hours (2 machines and 4 men)	—
Number of holes bored in face . . .	9	31	26	—
Average depth of holes . . . ins.	60	72	78	84
Feet bored in 8 hours	45	62	42·9 (1 machine)	—
Time taken to fire out end . . .	7·5 hours	6·5 hours	5 hours	—
Depth of cut . . . . . ft. ins.	3	5	6	5·2
Cost of compressed air per hour to run drill	1*s.* 11·5*d.*	2*s.* 1*d.*	2*s.* 6*d.*	—
Total cost of air to bore out end . .	15*s.* 5*d.*	2*l.* 10*s.*	4*l.*	—
Amount of explosives used in firing out cut (3 holes) . . . . . lbs.	—	—	—	—
Amount of nitro used in firing out cut (3 holes) . . . . . lbs.	—	—	—	—
Cost of explosives used in firing out cut .	—	—	—	—
Amount of explosives used in firing out remaining holes . . . . . lbs.	—	—	—	—
Amount of nitro used in firing out remaining holes . . . . . lbs.	—	—	—	—
Cost of explosives used in firing out remaining holes	—	—	—	—
Length of fuse used in firing out end . ft.	90	216	168	—
Number of detonators used in firing out end	15	36	30	—
Cost of detonators used in firing out end .	9*d.*	1*s.* 3½*d.*	1*s.* 1·2*d.*	—
Total cost of explosives, fuse, and detonators used to fire out end	1*l.* 12*s.* 10*d.*	3*l.* 19*s.* 1*d.*	4*l.* 3*s.* 4*d.*	—
Cubic contents of ground broken . .	180 cu. ft.	840 cu. ft. 5′ × 21′ × 8′	630 cu. ft. 15′ × 7′ × 6′	706 cu. ft. 17′ × 8′ × 5·2′
Amount of explosives used per foot lbs.	13·3 Gel.	17½ Gel.	12½ B.G.	24·2 Gel.
Amount of nitro used per foot . . lbs.	8·64	11·37	12·5	15·73
Amount of explosives used per ton . lbs.	2·44 Gel.	1¼ Gel.	1·42 B.G.	2·12 Gel.
Amount of nitro used per ton . . lbs.	1·58	0·81	1·42	1·37
Amount of explosives used in firing out end . . . . . lbs.	40 Gel.	87½ Gel.	75 B.G.	125 Gel.
Amount of nitro used in firing out end lbs.	26	56·87	75	81·75
Cost of fuse used in firing out end . .	2*s.* 1*d.*	4*s.* 10½*d.*	3*s.* 6*d.*	—
Cost of explosives used to fire out end .	1*l.* 10*s.*	3*l.* 12*s.* 11*d.*	3*l.* 18*s.* 9*d.*	—
Tons broken	16·36	70	52·5	58·8
Amount of explosives per hole . . lbs.	4·45 Gel.	2·82 Gel.	2·88 B.G.	—
Amount of nitro per hole . . . lbs.	2·89	1·83	2·88	—
Tons broken per hole bored . . .	1·81	2·26	2·02	—
Method of boring end	Stope cut	Centre cut	Centre cut	Centre cut

COST OF MINING OPERATIONS IN VARIOUS WEST AUSTRALIAN MINES—*continued.*

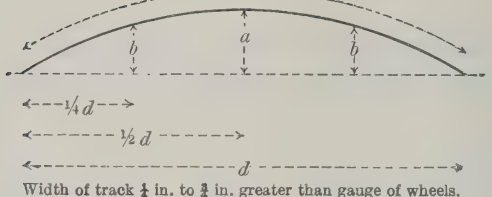
Driving.				Cross Cutting.			
Oroya Brownhill.	Sons of Gwalia.	Lake View.	Great Fingall.	Oroya Brownhill.	Sons of Gwalia.	Lake View.	Great Fingall.
2½	3½	3⅝	—	2½	3½	3⅝	—
1 hour (1 man)	2 hours	2 hours	—	1 hour (1 man)	2 hours	2 hours	—
20 hours	20 hours	24 hours	—	17 hours	21 hours	24 hours	—
15	15	15	—	15	15	14	
66	72	72	—	72	72	72	
32·5	36	30	—	42	34·3	28	
1·5 hour	6 hours	4 hours	—	1·5 hour	7 hours	4 hours	—
5	5′ 6″	5	4·15	5′ 6″	5′ 6″	5′ 6″	4·72
1*s.* 6*d.*	2*s.* 1*d.*	2*s.* 6*d.*	—	1*s.* 6*d.*	2*s.* 1*d.*	2*s.* 6*d.*	—
1*l.* 10*s.*	2*l.* 1*s.* 8*d.*	3*l.*	—	1*l.* 5*s.* 6*d.*	2*l.* 3*s.* 9*d.*	3*l.*	—
20 Gel.	22½ B.G.	13¾ B.G.	—	20 Gel.	17½ B.G.	—	—
13	22·5	13·75	—	13	17·5	—	—
15*s.*	1*l.* 5*s.* 1½*d.*	14*s.* 5*d.*	—	15*s.*	19*s.* 6½*d.*	—	—
30 Gel.	45 Gel.	50 Gel.	—	30 Gel.	40 Gel.	—	—
19·5	29·2	32·5	—	19·5	26	—	—
1*l.* 2*s.* 6*d.*	1*l.* 17*s.* 6*d.*	1*l.* 17*s.* 6*d.*	—	1*l.* 2*s.* 6*d.*	1*l.* 13*s.* 4*d.*	—	—
96	96	120	—	96	100	108	—
16	18	20	—	16	20	20	—
9*d.*	7½*d.*	8¾*d.*	—	9*d.*	9*d.*	8¾*d.*	
2*l.* 0*s.* 4*d.*	3*l.* 5*s.* 5*d.*	2*l.* 15*s.* 1¾*d.*	—	2*l.* 0*s.* 4*d.*	2*l.* 15*s.* 11*d.*	2*l.* 5*s.* 1¾*d.*	—
210 cu. ft.	221·375	210 cu. ft.	174 cu. ft.	231 cu. ft.	211·75	231 cu. ft.	198 cu. ft.
7′×6′×5′	7′×5′9″×5′6″	7′×6 ×5′	7′×6′×4·15′	7′×6′×5′6″	5′6″×7′×5′3″	7′×6 ×5′6′	7′×6′×4·72
10 Gel.	4·09 B.G. 8·18 Gel.	2¾ B.G. 10 Gel.	10·84 G.	9·1 Gel.	3 B.G. 7·27 Gel.	10·23 Gel.	13·6 Gel.
6·5	9·40	9·25	7·04	5·91	7·73	6·64	8·84
2·62 G.	1·11 B.G. 2·23 Gel.	0·82 B.G. 3 Gel.	3·8 Gel.	2·38 Gel.	0·99 B.G. 2·26 Gel.	2·87 Gel.	—
1·70	2·55	2·77	2·47	1·54	2·45	1·86	—
50 Gel.	22·5 B.G. 45 Gel.	13¾ B.G. 50 Gel.	55 Gel.	50 Gel.	17½ B.G. 40 Gel.	56·25 Gel.	64 Gel.
32·5	51·7	46·25	35·75	32·5	43·5	36·56	41·6
2*s.* 1*d.*	2*s.* 2*d.*	2*s.* 6*d.*	—	2*s.* 1*d.*	2*s.* 3½*d.*	2*s.* 3*d.*	—
1*l.* 17*s.* 6*d.*	3*l.* 2*s.* 7½*d.*	2*l.* 11*s.* 11*d.*	—	1*l.* 17*s.* 6*d.*	2*l.* 12*s.* 10½*d.*	2*l.* 2*s.* 2*d.*	—
19·09	20·11	17·5	—	21	17·64	19·25	—
3·34 Gel.	1·43 B.G. 3 Gel.	0·91 B.G. 3·34 Gel.	—	3·34 Gel.	1·16 B.G. 2·67 Gel.	4·01 Gel.	—
2·17	3·38	3·08	—	2·17	2·89	2·60	—
1·26	1·34	1·16	—	1·4	1·16	1·37	—
Centre cut	Centre cut	Centre cut	—	Centre cut	Centre cut	Centre cut	—

COST OF MINING OPERATIONS IN VARIOUS WEST AUSTRALIAN MINES—*continued.*

General Details.	Rising.			
	Oroya Brownhill.	Sons of Gwalia.	Lake View.	Great Fingall.
Size of machine used . . . . ins.	2½	3¼	3⅝	—
Time taken rigging and pulling down machine	2 hours (2 men)	2 hours	4 hours	—
Time boring	20·8 hours	17 hours	26 hours	—
Number of holes bored in face . . .	13	11	17	10
Average depth of holes . . . ins.	60	59	72	54
Feet bored in 8 hours	25	25′ 4″	31·15	—
Time taken to fire out end . . .	3 hours	4 hours	6 hours	—
Depth of cut . . . . ft. ins.	4′ 6″	4′ 6″	5	4·28
Cost of compressed air per hour to run drill	1*s.* 6*d.*	2*s.* 1*d.*	2*s.* 6*d.*	—
Total cost of air to bore out end . .	1*l.* 11*s.* 2*d.*	1*l.* 15*s.* 5*d.*	3*l.* 5*s.*	—
Amount of explosives used in firing out cut (3 holes) . . . . . lbs	20 Gel.	18¾ B.G.	20 B.G.	—
Amount of nitro used in firing out cut (3 holes) . . . . . lbs.	13	18¾	20	—
Cost of explosives used in firing out cut .	15*s.*	1*l.* 1*s.*	1*l.*	—
Amount of explosives used in firing out remaining holes . . . . lbs.	30 Gel.	23¾ Gel.	50 Gel.	—
Amount of nitro used in firing out remaining holes . . . . . lbs.	19·5	15·4	32·5	—
Cost of explosives used in firing out remaining holes	1*l.* 2*s.* 6*d.*	19*s.* 9½*d.*	1*l.* 17*s.* 6*d.*	—
Length of fuse used in firing out end . ft.	96	90	120	—
Number of detonators used in firing out end	16	18	20	—
Cost of detonators used in firing out end .	9*d.*	7½*d.*	8¾*d.*	—
Total cost of explosives, fuse, and detonators used to fire out end	2*l.* 0*s.* 4*d.*	2*l.* 3*s.* 5*d.*	3*l.* 0*s.* 8¾*d.*	—
Cubic contents of ground broken . .	180 cu. ft.	189 cu. ft. 4′6″ × 6′ × 7′	175 cu. ft. 7′ × 5′ × 5′	116 cu. ft. 4′6′ × 6′ × 4·28′
Amount of explosives used per foot lbs.	11·1 Gel.	4·15 B.G. 5·3 Gel.	4 B.G. 10 Gel.	9·3 Gel.
Amount of nitro used per foot . . lbs.	7·21	7·57	10·5	6·04
Amount of explosives used per ton . lbs.	3·05 Gel.	1·12 B.G. 1·5 Gel.	1·37 B.G. 3·42 Gel.	—
Amount of nitro used per ton . . lbs.	1·98	2·09	3·59	—
Amount of explosives used in firing out end . . . . . lbs.	50 Gel.	18·75 B.G. 23·75 Gel.	20 B.G. 50 Gel.	40 Gel.
Amount of nitro used in firing out end lbs.	32·5	34·15	52·5	26
Cost of fuse used in firing out end . .	2*s.* 1*d.*	2*s.*	2*s.* 6*d.*	—
Cost of explosives used to fire out end .	1*l.* 17*s.* 6*d.*	2*l.* 0*s.* 9½*d.*	2*l.* 17*s.* 6*d.*	—
Tons broken	16·36	15·75	14·58	—
Amount of explosives per hole . . lbs.	3·84 Gel.	1·7 B.G. 2·16 Gel.	1·37 B.G. 3·43 Gel.	—
Amount of nitro per hole . . lbs.	2·49	3·1	3·59	—
Tons broken per hole bored . . .	1·25	1·42	0·84	—
Method of boring end	Centre cut	Centre cut	Centre cut	—

COST OF MINING OPERATIONS IN VARIOUS WEST AUSTRALIAN MINES—*continued.*

Winzing.				Stoping.			
Oroya Brownhill.	Sons of Gwalia.	Lake View.	Great Fingall.	Oroya Brownhill.	Sons of Gwalia.	Lake View.	Great Fingall.
					(Average for 3 faces)	(Stope 14′×12′)	
2½	3¼	3⅝	—	2½	2½	2½, 3¼, 3⅝	—
1 hour	3 hours	6 hours	—	2 hours	6 hours (3 times)	4 hours	—
(2 men) 12 hours	29 hours	26 hours	—	8 hours	56 hours	28 hours	—
9	14	17	16	5	24	16	8
60	62	72	60	72	62	96	60·72
30	24	31·15	—	30	17′ 8″	36′ 6″	—
7½ Winch	4 hours	48 hours	—	4 hours	6 hours (3 faces)	24 hours	—
15 Windlass							
3	4′ 9″	5	5·10	6	4′ 6″	9	—
1*s.* 6*d.*	2*s.* 1*d.*	2*s.* 6*d.*	—	1*s.* 1·37*d.*	1*s.* 3*d.*	1*s.* 10*d.*	—
18*s.*	3*l.* 2*s.* 6*d.*	3*l.* 5*s.*	—	1*l.* 13*s.* 3*d.*	3*l.* 10*s.* (3 faces)	2*l.* 11*s.* 5½*d.*	—
15 Gel.	17½ B.G.	20 B.G.	—	—	—	—	—
9·75	11·37	20	—	—	—	—	—
11*s.* 3*d.*	14*s.* 7*d.*	1*l.*	—	—	—	—	—
15 Gel.	30 Gel.	50 Gel.	—	—	—	—	—
9·75	19·5	32·5	—	—	—	—	—
11*s.* 3*d.*	1*l.* 5*s.*	1*l.* 17*s.* 6*d.*	—	—	—	—	—
72	85	120	—	72	140	120	—
12	17	20	—	12	28	20	—
7*d.*	7½*d.*	8¾*d.*	—	—	1*s.*	8¾*d.*	—
1*l.* 4*s.* 8*d.*	2*l.* 2*s.* 1½*d.*	3*l.* 0*s.* 8¾*d.*	—	2*l.* 5*s.* 6*d.*	3*l.* 6*s.* 8*d.* (3 faces)	2*l.* 0*s.* 8¾*d.*	—
120 cu. ft.	182·87 cu. ft. 4′9″×5′6″×7′	175 cu. ft. 5′×7′×5′	275 cu. ft. 9′×6′×5·10′	720 cu. ft.	1262·25 cu. ft. 13′6″×11′×8′6″	1,512 cu. ft. 14′×12′×9′	—
10 Gel.	10 Gel.	4 B.G. 10 Gel.	11·9 Gel.	—	—	—	—
6·5	6·5	10·5	7·73	—	—	—	—
2·75 Gel.	3·12 Gel.	1·37 B.G. 3·42 Gel.	—	0·9 Gel.	0·714 Gel.	0·39 Gel.	—
1·77	2·02	3·59	—	0·58	0·464	0·25	—
30 Gel.	47·5 Gel.	20 B.G. 50 Gel.	61 Gel.	58 Gel.	75 Gel.	50 Gel.	—
19·5	30·87	52·5	39·65	37·70	48·7	32·5	—
1*s.* 7*d.*	1*s.* 11*d.*	2*s.* 6*d.*	—	1*s.* 7*d.*	3*s.* 2*d.*	2*s.* 6*d.*	—
1*l.* 2*s.* 6*d.*	1*l.* 19*s.* 7*d.*	2*l.* 17*s.* 6*d.*	—	2*l.* 3*s.* 4*d.*	3*l.* 2*s.* 6*d.* (3 faces)	1*l.* 17*s.* 6*d.*	—
10·9	15 24	14·58	—	65	—	126	—
3·34 Gel.	3·4 Gel.	1·37 B.G. 3·43 Gel.	—	4·8 Gel.	3·125 Gel.	3·12 Gel.	—
2·17	2·21	3·59	—	3·12	2·03	2·02	—
1·21	1·08	0·84	—	5·4	4·37	7·87	—
Stope cut	Centre cut	Centre cut	—	—	—	—	—



To bend rails to any curve, let l be length of rail, r radius of curve, both in feet, a the middle ordinate in the centre of the curve, b the quarter ordinate, midway between a and the end of the rail (both a and b in inches):—

Then

$$a = 1{\cdot}56\,\frac{l^2}{r}, \qquad b = 1{\cdot}17\,\frac{l^2}{r}.$$

Width of track ¼ in. to ⅜ in. greater than gauge of wheels.

Additional width round curves:—

Gauge of track	15	18	24	30 ins.
Additional width	¼	½	⅝	¾ in.

In going round a curve with locomotive traction or on self-acting inclines, the outer rail must be elevated; with rope haulage the inner rail may have to be elevated.

Let

e = super-elevation in feet; s = gauge of track in feet; v = velocity in feet per second; r = radius of curve in feet;

then,

$$\text{for outer rail, } e = \frac{v^2\,s}{32{\cdot}2\,r}.$$

For inner rail, the elevation depends upon whether the centrifugal force due to the velocity of running is greater or less than the inward pull on the drawbar; it will be greater at high speeds, say, exceeding 20 miles per hour, and less at low speeds; under ordinary conditions at 15 miles per hour the elevation of the inner rail should be about 2 ins.

Colliery Tubs.

Colliery tubs carry from 7 to 40 cwts. of coals.

In North country average dimensions of tub body are about 3 ft. 9 ins. × 3 ft. × 2 ft.; wheels 11 ins. diameter, axles 1¾ in. diameter.

Such a tub if made of wood will carry about 7½ cwts. of coals, if of iron about 9 cwts.; wooden tubs weigh about 55 per cent. of the load, iron tubs about 50 per cent. Tubs cost approximately 16*s.* to 17*s.* per cwt., and their upkeep is about 10*s.* per annum. Upkeep is often reckoned at 1*d.* per ton of coal. In South Wales much larger tubs, holding up to 2 tons, are used; the general tendency in other parts of the country is towards increasing the size of tubs.

Using ordinary tubs carrying about 10 cwt. of coals, it is usually reckoned that a colliery requires for its equipment about one tub per ton of coals raised per day.

In U.S.A. colliery tubs carrying 1 to 4 tons are in use; average 3-ton tub, 9 ft. 6 ins. long, 4 ft. 3 ins. wide, 1 ft. 3 ins. high.

Coefficient of friction of tubs on well-kept mine road (level), 1 per cent. to 3 per cent., average say 1·5 per cent.; with tubs badly greased on roads in bad repair may be two to three times as much. Coefficient of sliding friction (wheel spragged) is from 12 to 20 per cent.; if m be coefficient of rolling friction, m' of sliding friction, the coefficient of friction with:—

1 wheel spragged $= \frac{3m + m'}{4}$; 2 wheels spragged $= \frac{m + m'}{2}$; 3 wheels spragged $= \frac{m + 3m'}{4}$.

Additional friction in going round curve of radius r, where the gauge of the track $= s$ (both in feet) is $\frac{s}{2r}\,m'$.

Tractive effort with 18-in. wheels on good level track, 40-in. gauge, at 5 to 6 m.p.h. = 24·3 lbs. per ton with plain bearings, 12·8 lbs. with roller bearings. (*Trans. Amer. Inst. Min. Eng.*, vol. lv. p. 29.)

A wooden tub with plain bearings at speed of 1·75 m.p.h. showed coefficients of friction:—

$\frac{1}{3\cdot0}$ with poor, $\frac{1}{6\cdot5}$ with average, $\frac{1}{6\cdot8}$ with good lubrication.

Coefficient on starting from rest with average lubrication $\frac{1}{5\cdot0}$; tests with plain bearings at collieries gave $\frac{1}{6\cdot0}$ $\frac{1}{3\cdot3}$, and $\frac{1}{5\cdot5}$ respectively. A steel tub with plain bearings gave $\frac{1}{5\cdot8}$ with Climax bearings $\frac{1}{6\cdot5}$, with Skefko ball bearings $\frac{1}{13\cdot5}$; tub with Rowbotham wheels $\frac{1}{6\cdot5}$.

A 9-ft. radius curve trebles, a 12-ft. radius curve doubles, the coefficient of friction with fixed wheels; loose wheels give ⅔ of increase of fixed wheels.

Starting effort to get up to a speed of 1·75 m.p.h. = 2 to 3 times the effort required to maintain that speed with plain bearings, 1½ to 2 times with Rowbotham wheels, and 1 to 1½ times with Skefko ball bearings. (J. Wilson, *Trans. Inst. Min. Eng.*, lxiii., p. 294.)

With 'wheel' greasers the consumption of grease averages about ¼ lb. per tub per diem for ordinary collieries.

On the average a putter can put 10 tons of coal over a distance of 200 yards (level) in an 8-hour shift.

A horse does its best work when travelling at a speed in feet per second $= \frac{1}{4}h$, where h is the height of the horse in hands, and exerting a pull in lbs. of $\frac{60\,c^2}{h}$, where c is the circumference of the horse's chest in feet (*Baron*). Average pull is about 1¼ cwt.

Maximum gradient for hand putting, 17 per cent.

Maximum gradient for horse haulage, 12 per cent.

Gradient at which resistance of full tub downhill equals that of empty tub uphill:—Let W be the weight of a loaded tub, w that of an empty one, θ the inclination of the road, μ the coefficient of friction:—$\tan\theta = \frac{\mu\,(W - w)}{W + w}$ (usually $\tan\theta = 0{\cdot}015$ about).

Work done in hauling a tub of weight W up or down an incline of length $l = l\,(\mu\,W\cos\theta \pm W\sin\theta)$

Self-acting Incline.

Self-acting Incline.—Let the sets consist of n tubs each of weight w lbs., and capable of carrying W lbs. of mineral. Let the slope of the incline, assumed uniform, be $\alpha°$. Let l be the length of the rope (= length of incline) in feet, c its weight in lbs. per foot.

Let the rope be supported on rollers, the weight of which per foot of rope (= weight of one roller divided by the distance in feet between the centres of successive rollers) is r lbs. Let D be the diameter of the rollers, d of the spindle. Let μ be the coefficient of friction of the tub on rails (usually say about 0·015) and μ_1 the coefficient of friction of lubricated bearings (say 0·05 to 0·08). Let F be the frictional resistance of the drum at the head of the incline, including that of the rope coiled upon it—a figure that is always relatively small and may often be neglected. Then at any distance x feet from the starting point the full set is just able to move down the incline when:—

$$\tan\alpha = \frac{\mu n\,(W + 2w) + \mu_1\,l(c + r)\,\frac{d}{D} + F}{nW - c\,(l - 2x)}$$

By putting $x = 0$ the above expression gives the angle at which the set will just start from the top of the incline.

The time occupied in descending a distance x will be:—

$$t \text{ seconds} = \sqrt{\frac{2x\,[n\,(W + 2w) + (c + r)\,l + M]}{g\left[\sin\alpha\,(nw - cl) - \cos\alpha\left(l\,(c + r)\,\frac{d}{D}\,\mu_1 + n\,(W + 2w)\,\mu\right) - F\right]}}$$

and the velocity in feet per second v attained after travelling a distance of x feet will be:—

$$v = \sqrt{\frac{2gx\left[\sin\alpha\,(nw - cl) - \cos\alpha\left(l\,(c + r)\,\frac{d}{D}\,\mu_1 + n\,(W + 2w)\,\mu\right) - F\right]}{n\,(W + 2w) + (c + r)\,l + M}},$$

where M is the weight of the drum and its fittings and g the acceleration due to gravity.

All the above expressions are approximations, but are accurate enough for practical purposes.

When the slope is not uniform, the calculations become more complicated. If H be the total vertical fall of the incline and L its horizontal length, it is usually sufficient, with the low gradients prevailing in collieries, to use the above formulas, putting $\alpha = \tan^{-1}\frac{H}{L}$. As the denominator of the expression given above for $\tan\alpha$ increases with x, it is obvious that the incline should, whenever possible, flatten going downwards, or be concave, though not so much so as to cause the rope to rise too high above the rails. The exact shape of the curve is not very important; it may with advantage be a parabola, which may be set out by making the height h at any point distant y ft. horizontally from the foot of the incline $= \frac{y^2 H}{L^2}$. The rope will then follow the curve whenever the relations are such that $H = \frac{cl^2}{4T}$, where T is the tension of the rope.

Main-and-Tail Rope Haulage.

Main-and-Tail Rope Haulage.—Loaded tubs in sets up to 40 are hauled outbye by motors (steam-engines, compressed-air engines or electro-motors) at speeds up to 20 miles per hour, whilst the empty tubs are hauled inbye by means of the tail rope. The size of the engine depends upon the pull to be exercised when one set (nearly always the full set) is in its least favourable position. Let T be the tension in the rope in such position, made up of the weight of the set plus rope × sine of angle of inclination of the roadway, the friction of the set and of both ropes on the rollers and the pull required to accelerate the moving parts. Let O be the diameter of the steam cylinder

in inches, one of which is alone assumed to be effective at the start, let K C be the stroke of the engine in feet (K is usually $\frac{1}{6}$ to $\frac{1}{4}$), S the steam pressure in lbs. per sq. in., s the ratio of the number of teeth in the spur wheel on the drum shaft to the number of teeth in the pinion on the crankshaft, D the diameter of the drum in feet, then

$$C = \sqrt[3]{\frac{4DT}{\pi SK s}}$$

For any portion of the trip except the start (i.e. with set already in motion), both cylinders are effective, and the value as found above must be divided by $\sqrt{2}$.

These engines are usually small and run at a piston speed of 500 to 800 ft. per min.

Endless Chain and Endless Rope.

Endless Chain and Endless Rope.—Endless chain is now rarely used except on steep gradients and for short distances; it is less reliable than endless rope on long roads, requires more power and is rather more expensive to work, but admits of rather higher speeds; it may work up to 6 miles per hour; endless rope rarely exceeds 4 miles.

The following calculations apply to both methods unless otherwise stated:—

If Q be the quantity of mineral in tons to be transported per hour, W the weight in lbs. carried by one tub, V the speed of haulage in miles per hour, f the distance apart of the tubs in feet, then

$$f = 2{\cdot}36 \frac{V \cdot W}{Q}.$$

Endless chain is always over-tub haulage, endless rope may be over-tub or under-tub. In over-tub haulage the tubs must be so near together that the rope does not touch the ground between them. Let T be the least tension in the hauling rope or chain (*i.e.* between the two tubs nearest the driving pulley on the empty side), let c be the weight of the rope or chain per foot, h the height of the tub, then the distance f_1 of the tubs apart is given by

$$f_1 = \sqrt{\frac{8hT}{c}}.$$

If f is greater than f_1 empty tubs must be run on the full side.

The pull to be exerted by the engine driving the haulage is

$$\mu\left[n\ (2w + W) + 2c \cdot l\right] \cos \alpha \pm W \sin \alpha,$$

using the same notation as above, n being the number of tubs on any one side of the rope at any time, so that $n = \frac{l}{f}$; the lower sign is used when the gradient is with the full tubs, the upper one when against it. If this expression has the $-$ sign, the arrangement is self-acting.

The return wheel of an endless rope haulage is often used as a tightening pulley, being capable of sliding and pulled by a weight W.

Let the radius of the pulley, fig. 12, be R and of its spindle r.

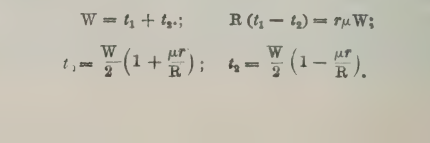


FIG. 12.

Let the coefficient of friction between the spindle and its bearing be μ.

Let the tension in the rope coming off the pulley be t_1, and in the rope leading to the pulley be t_2,

then,

$$W = t_1 + t_2; \qquad R(t_1 - t_2) = r\mu W;$$

whence

$$t_1 = \frac{W}{2}\left(1 + \frac{\mu r}{R}\right); \quad t_2 = \frac{W}{2}\left(1 - \frac{\mu r}{R}\right).$$

Horse-Power required for Haulage.

TABLE SHOWING H.P. REQUIRED FOR MAIN AND TAIL HAULAGE AT 10 MILES PER HOUR.

Actual Incline in ins. per yd. (i)	Virtual Incline* in ins. per yd.	Load in Tons. (W)										
		5	7·5	10	15	20	25	30	35	40	45	50
—2	0	0	0	0	0	0	0	0	0	0	0	0
—1	1	8·3	12·5	16·6	25	33·2	41·4	50	58	66·4	76	80·8
0	2	16·7	25	33·3	50	66·6	83	100	116	133	150	166
1	3	25	37·7	50	75	100	125	150	175	200	225	250
2	4	33·4	50	67·5	100	134	167	200	233	270	300	334
3	5	41·5	62	83·5	125	167	208	250	290	334	375	416
4	6	50	75	100	150	200	250	300	350	400	450	500
5	7	58·3	87	117	175	234	294	350	408	468	525	588
6	8	66·3	100	133	200	267	333	400	465	532	600	666
7	9	75	112	150	225	300	375	450	520	600	675	750
8	10	83·5	124	166	250	333	420	500	580	664	750	830
9	11	91	137	183	275	366	459	550	640	732	825	918
10	12	99	150	200	300	400	500	600	696	800	900	1000
11	13	108	162	217	325	433	542	650	755	868	975	1080
12	14	116	174	233	350	466	584	700	815	932	1050	1168

$$\text{H.P.} = \frac{5}{3}(i + 2)\,\text{W}.$$

TABLE SHOWING H.P. REQUIRED FOR ENDLESS ROPE HAULAGE ON ROAD 1,000 YARDS LONG

Actual Incline in ins. per yd. (i)	Virtual Incline* in ins. per yd.	Output in Lbs. per Minute. (l)																
		200	300	400	500	600	700	800	900	1000	1250	1500	1750	2000	2250	2500	2750	3000
	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
—1	1	·5	·75	1	1·2	1·5	1·7	2	2·3	2·5	3·1	3·7	4·4	5	5·6	6·2	6·9	7·5
0	2	1	1·5	2	2·5	3	3·5	4	4·6	5	6·3	7·6	8·9	10	11·4	12·7	14	15
1	3	1·5	2·3	3	3·8	4·6	5·3	6	6·8	7·6	9·5	11·4	13·3	15·2	17	19	21	23
2	4	2	3	4	5	6	7	8	9	10	12·6	15	17·6	20	23	26	28	30
3	5	2·5	3·8	5	6·3	7·6	8·9	10	11·4	12·7	15·8	19	22·3	25·4	28	32	35	38
4	6	3	4·5	6	7·6	9	10·6	12	13·7	15·2	19	22·8	26·6	30·4	34	38	42	45
5	7	3·5	5·3	7	8·8	10·5	12·4	14	16	17·7	22	26·5	31	35	40	44	49	52
6	8	4	6	8	10	12	14·2	16	18·2	20·4	25·3	30·4	35	41	45	51	55	60
7	9	4·5	6·8	9	11·4	13·7	16	18	20·5	22·8	28·5	34	40	45	51	57	63	68
8	10	5	7·6	10	12·6	15·2	17·7	20	22·5	25·3	31·7	38	44	51	57	63	69	76
9	11	5·5	8·3	11	14	16·7	19·5	22	25	27·8	34·7	41·7	49	55	62	69	76	83
10	12	6	9	12	15·2	18·2	21·3	24	27·3	30	38	45	53	61	68	76	83	90
11	13	6·5	9·8	13	16·4	19·8	23	26	29·6	33	41	49	57	66	74	82	90	98
12	14	7	10·6	14	17·7	21·3	24·8	28	31·8	35·4	44	53	62	71	80	88	97	106

(*Ernest Scott and Mountain.*)

$$\text{H.P.} = \cdot00253\,(i + 2)\,l.$$

* 'Virtual Incline' is the incline to which the 'Actual Incline' is equivalent to when a load is hauled up it, the difference between the two being due to friction.

The 'Virtual Incline' is taken as equivalent to 'Actual Incline $+\frac{1}{2}$.'

Cost of Haulage.

COST OF UNDERGROUND HAULAGE.

System of Haulage.	Average Gradient for Full Tubs.	Cost in Pence per Ton per Mile. Ropes or Chains.	Tubs.	Grease and Oil.	Coals.	Repairs to Engines and Boilers.	Maintenance of Way.	Labour.	Total.
Tail Rope	Rise 1 in 213	·276	·114	·186	·558	·098	·064	·583	1·879
Endless Chain	„ 1 „ 59	·083	·173	·155	·256	·072	·068	·572	1·379
Endless Rope	„ 1 „ 36	·252	·309	·138	·323	·196	·083	1·692	2·993

COST OF SECONDARY HAULAGE IN NORTH OF ENGLAND COLLIERIES.

	Average Distance.	Tons per Animal.	Cost per Ton in Pence. Wages.	Maintenance.	Interest & Deprecn.	Total.
Pony putting . .	180 yds.	27·0	1·584	0·500	0·068	2·152
Horse driving . .	423 „	30·0	0·531	0·615	0·087	1·233
Putting and driving.	603 „	14·1	2·115	1·115	0·155	3·385

(*Trans. Inst. Min. Eng.* xv. 1897, p. 136.)

Cost of underground locomotive haulage in Westphalia in 1912, including redemption of capital outlay :—

Petrol	2·51 pence per ton-mile.
Continuous current (with conductors) . . .	2·16 „ „
Alternating „ „ „ . . .	1·58 „ „
Continuous „ (with accumulators) . . .	2·65 „ „
Compressed air	2·12 „ „

So-called fireless locomotives using superheated water, and oil or coke-fired steam locomotives are also used.

Steam Mine Locomotives.

Mine locomotives (*steam*) usually have cylinders 5 ins. to 8 ins. diameter, stroke 1½ diameters, driving wheels 20 ins. to 30 ins. diameter, wheel base 3 ft. to 6 ft., weight complete 4 to 15 tons, drawbar pull ⅓ to ⅛ weight. Boiler pressure 120 to 150 lbs. Speed with full load in miles per hour ⅛ diameter of driving wheel in inches.

When D is piston diameter, l length of stroke, d diameter of driving wheel, all in inches, and P the gauge pressure of the boiler in lbs. per sq. in. then drawbar pull = approximately, $\frac{0{\cdot}85\ P\ l\ D^2}{d}$.

Compressed Air Locomotives.

Compressed air locomotives are worked at 600 to 1,000 lbs. per sq. in. gauge pressure, charged from underground receivers, in which air is compressed to 50 to 100 lbs. higher. Pressure is reduced to about 150 lbs. per sq. in. before the air enters the cylinders. The formula above given for drawbar pull may be used, P being the pressure of air as supplied to the cylinders, writing 0·95 instead of 0·85.

Electric Locomotives.

With trolley wires, electric locomotives may use either continuous or alternating current. Weight complete is usually 7 to 15 tons, and drawbar pull $\frac{1}{6}$ to $\frac{1}{8}$, usually $\frac{1}{7}$ of weight; coefficient of adhesion, 25 to 30 per cent. 75 per cent. to 80 per cent. of electric imput is converted into tractive effort, 1 per cent. to 2 per cent. of which will move the locomotive on the level, the remainder being available for hauling load. Speed on the level usually 6 to 8 miles per hour.

With accumulators, small locomotives are used in places for secondary haulage. The B.E.V. locomotive has a battery capacity of 193 amps. at 65 volts, and can pull or push 3-ton trucks on the level. Its use is on the increase in this country.

WINDING.

In sinking pits, the excavated material is usually hoisted in steel buckets or hoppits, usually unguided, and usually single. The load is thus wholly unbalanced, geared engines are often used, and the winding speed is slow. Such buckets have capacities of 25 to 50 cubic feet, say 30 to 60 cwts., the weight of the bucket being from $\frac{1}{2}$ to $\frac{3}{4}$ of its load. For ordinary colliery work cages carrying two to twelve tubs are used, always in pairs running on wooden, rail or rope guides; in collieries in U.S.A. large skips are sometimes used instead of cages.

Skips are largely used instead of cages in metalliferous mines; when the mineral has to be crushed the fact that the skip breaks the mineral up more than does the cage is no drawback. Self-dumping skips are used on large mines, whilst underground bins and loading stations at the shaft enable the skips to be filled very quickly. Skips usually hold from 20 to 100 cubic feet, say up to about 5 tons of rock; they are made of $\frac{1}{4}$ in. to $\frac{3}{8}$ in. steel plate.

In some Illinois collieries, where the coal is to be crushed afterwards, skips about 6 ft. × 6 ft. × 12 ft. 6 ins., holding about 12 tons of coal, are used; the speed of winding is up to 1,600 ft. per min.; the average power consumption is 0·485 kw. hour per ton of coal; up to 1,000 tons of coal has been hoisted per hour from a depth of 200 yards.

On the Continent the total load lifted is about: cage 30 per cent., tubs 20 per cent., coals 50 per cent. German practice is: Let W be the weight of coals, T weight of the tubs, and Q the total weight of coals, tubs, and cage (including chains, hook, etc.); then

$$\frac{W}{T} = \frac{1}{0{\cdot}38} \text{ to } \frac{1}{0{\cdot}78}, \text{ averaging } \frac{1}{0{\cdot}56} \text{ and } \frac{W}{Q} = \frac{0{\cdot}27}{1} \text{ to } \frac{0{\cdot}44}{1}, \text{ averaging } \frac{0{\cdot}36}{1}.$$

In skip hoisting the ratio of weight of skip to that of load of mineral is 0·6 : 1 to 0·9 to 1; in hoisting in cages, weight of cage and tubs : weight of coal : : 1·5 : 1 to 1·9 : 1 (*Zsch. f. B.H. u.S.W.*, 1915, p. 285).

In Great Britain the cage usually weighs 35 to 45 per cent. of the total load, or approximately as much as the useful load of coals lifted. It is carried by bridle chains, usually six in number. Their factor of safety should be at least 10, and the size of the chains may be calculated thus:

Let d be the diameter of the chain in eighths of an inch, and W the safe working load in tons then $d = \sqrt{10W}$.

Slack adds greatly to strain on rope; according to numerous experiments, if the pull required to lift a given cage with rope taut be taken as 1, it requires a pull equal 1·5 to 1·7 with 3 ins. of slack, to 2–2·25 with 6 ins., 2·25 to 2·5 with 9 ins., and up to a pull of 3 with 12 ins.

The velocity of winding varies from 15 to 75 ft. per second; a rule given by O'Donahue is: allow a speed of 20 ft. per second for shafts up to 200 yards in depth, and add 5 ft. per second for every additional 100 yards.

Wooden guides are usually 3 ins. × 4 ins. to 4 ins. × 6 ins.; rail guides 50 to 70 lbs. per yard. Rope guides made of single strand of 7 to 15 wires each $\frac{1}{2}$ to $\frac{1}{4}$ in. diameter, stretched by weight of $1\frac{1}{2}$ to 5 tons; weight is often calculated at 1 ton for every 250 yards depth of shaft. The friction on the guides is but small; it usually amounts to about 0·025 of the total load on the rope with wooden or rail guides and 0·02 of the load with wire rope guides.

Size of Winding Engine.

To calculate the size of the winding engine the conditions of the wind must be known or assumed. The diameter of the cylinders must be such that (*a*) one cylinder is capable of starting the unbalanced load and (*b*) that the two cylinders taking steam on the full stroke shall be capable of accelerating the total load to be set in motion.

If w be the weight of a cage and fittings with the empty tubs, W the weight of minerals lifted per wind, h the depth of the shaft in feet, c the weight of the rope in lbs. per foot, v the velocity in feet per second up to which the load is to be accelerated during a lift of f feet; let P be the steam pressure in lbs. per sq. in., d the diameter of the piston in ins., S the length of the stroke in feet, D the diameter of the drum in feet, M the weight of drum, pulleys, etc., and allowing 20 per cent. to cover the friction in the shaft, of the pulleys, drum, and engine, etc., then to fulfil condition (*a*)

$$d = \sqrt{\frac{D\,(W + ch)}{0{\cdot}471\ P.S.}}.$$

To fulfil condition (*b*)

$$d = \sqrt{\frac{4\,D\left[\left\{W + c(h - f)\right\} f + (W + 2w + 1{\cdot}5\,ch + 0{\cdot}5\,M)\,\frac{v^2}{2}\right]}{3\ .\ P.f.S.}} \qquad (b)$$

The larger value of d as found by (*a*) or (*b*) should be adopted.

This formula only applies to the ordinary form of winding engine with the drum carried direct on the crankshaft and the two cranks at right angles.

On the average the total load, namely, coal, tubs, and cage, equals 2·66W; assuming a factor of safety of 10 and a rope of 100 tons steel, $c = \frac{26 \cdot 6W}{16{,}000 - 10h}$.

The diameter of a rope in inches equals approximately $0 \cdot 45 \sqrt{c}$; taking the diameter of the drum as 120 times the diameter of the rope, $D = 4 \cdot 5 \sqrt{c}$.

On these assumptions, formula (*a*) becomes

$$d = 3 \cdot 856 \sqrt{\frac{\sqrt{c}\,(W + ch)}{P}}.$$

Assuming that $S = \frac{d}{6}$, and that the diameter of the drum is 120 times the diameter of the rope, this expression takes the following form:—

$$d = 4 \cdot 59 \sqrt[3]{\frac{(W + ch)\sqrt{c}}{P}}.$$

An empirical formula which gives reasonably good results is:

$$d = 3 \sqrt{\frac{WD}{P}}.$$

Usually $S = \frac{d}{6}$ and the length of the connecting rod is $2\frac{1}{2} - 3$ S. Average piston speed 500 ft. per min., varying from 300 to 800 ft.

If the load is partially or wholly balanced, the formula must be correspondingly modified.

Once the load has been accelerated to its full speed, the steam admission is reduced by means of valve gears with variable cut-offs, and during the last part of the wind work must be absorbed so as to retard the load and bring it to rest gradually; powerful brakes are employed or the engine may be reversed as well as the brakes applied.

The weight of the rope may be balanced by:—

1. Chain counterbalance, suitable for slow winding only, and adding enormously to the weight to be accelerated.

2. Balance rope beneath the cages; this adds to the weight to be accelerated, but the chief objection is the additional strain put upon the cappings of the winding ropes. Some engineers use a balance rope heavier than the winding rope, so as to overbalance the latter, to help the engine to accelerate the load at start and to aid in retardation at the finish of the wind.

3. The conical or scroll drum. The ascending load is attached to the end of the rope which is at the small end of the drum, and the descending load to the end at the large end. Using the same notation as before, but calling r the mean radius of the drum, r_s the smaller radius, r_d the larger, Q_a the resistance due to the ascending load and Q_d the force derived from the descending load and n the total number of revolutions of the drum during the wind:—

$$r = \frac{r_s + r_d}{2} = \frac{h}{2\pi n}$$

$$r_s = r\left(1 - \frac{hc}{Q_a + Q_d + hc}\right); \qquad r_d = r\left(1 + \frac{hc}{Q_a + Q_d + hc}\right);$$

$$\frac{r_s}{r_d} = \frac{Q_a + Q_d}{Q_a + Q_d + 2hc}$$

If δ be the diameter of the rope, the width of the drum will be approximately $1 \cdot 5\, n\, \delta$.

Cylindro-conical drums, giving partial counterbalance (where most needed), are often used.

Flat ropes wound on a reel are equivalent to a spiral drum; they will counterbalance in the three main positions when

$$r_d - r_s = nt \text{ and } r = \sqrt{\frac{Q_a + Q_d + hc}{4\pi c}\, t},$$

where t is the thickness of the flat rope.

The time of banking varies considerably; it may be averaged at about 10 to 15 seconds for one deck and 25 to 45 seconds for double decks. Up to 3,000 tons may be wound in 10 hours, but 2,000 tons is not often exceeded.

The consumption of coal in winding engines varies from 0·5 to 1·5 per cent. of the weight of coals wound. Steam consumption 50 to 100 lbs. per shaft h.p. hour.

The cost of a winding engine and boilers erected varies from about 3,500*l.* to about 10,000*l.*

The cost of winding, exclusive of interest and depreciation on the winding plant, varies from about 3*s.* to 10*s.* 6*d.* per 100 tons wound.

Electric Winding Plants.

Electric winding plants are coming increasingly into favour; according to W. Mountain (*Trans. Inst. M.E.* xxxi., p. 329) the first cost of electric winders is considerably in excess of steam winders, so that interest and depreciation of the plant form serious items in the working costs. Some of the large geared electric winders in use on the Rand raise a 3-ton load from a depth of

2,500 ft., taking 2,000 e.h.p. An electric winder at a Durham pit raises 63 cwts. of coal per wind from 1,350 ft., uses 3-phase induction motor, 700 h.p., 2,900 volts; motor runs at 230 r.p.m., geared 5·45 to 1; cylindro-conical drum 10–20 ft. diam., weighing 45¾ tons; overall efficiency 51 per cent. A winder in a South Wales pit uses 2,000 volts alternating, lifts 55 cwts. from 600 ft., uses 174 kW. per wind; drum cylindro-conical, 8 ft. to 13 ft.; overall efficiency, 48·25 per cent.

ELECTRIC WINDERS ON WITWATERSRAND.

Name.	Depth in ft.	Tons (2,000 lbs.) per hour.	Weight of Mineral.	Weight of Skip.	Maximum Rope Speed.	Drum Diameter.	Winding Rope Diameter.	Tail Rope Diameter.	H.P.	System.
			lbs.	lbs.	ft. per min.	ft	in.	in.		
City Deep . .	3,210	157·5	9,000	8,000	3,500	11	$1\frac{9}{16}$	$1\frac{9}{16}$	1,600	A.C.
Village Deep . .	4,200	152·0	10,600	7,000	3,500	11	$1\frac{9}{16}$	$1\frac{9}{16}$	1,600	A.C.
Brakpan . . .	3,800	180·0	10,000	6,000	3,500	11	$1\frac{1}{2}$	$1\frac{1}{2}$	1,450	Ward-Leonard
Village Deep, Turf Shaft	4,093	150·0	10,000	7,000	3,500	12	$1\frac{1}{2}$	—	1,600	A.C.

The last-named winder (Whiting hoist) shows an overall efficiency of 77 per cent.
(J. F. Perry, *Trans. Nat. Assoc. Col. Man.*, xv. p. 299.)

At the East Geduld Mines an electric winder hoists from a depth of 3,350 ft. 1,400 lbs. per wind, 230 (short) tons per hr.; drum cylindrical 14 ft. diameter, 5 ft. between flanges, maximum speed 2,580 ft. per min.; maximum peak capacity to be 4,570 h.p.

At Aberdare colliery, N.S.W., an electric winder raises 7½ tons net per wind, 490 tons per hr. from a depth of 1,260 ft.; it is rated at 4,500–11,250 h.p. Another shaft at this colliery has an electric winder making 133 trips per hour, with a net load of 4½ tons, the maximum winding speed being 2,555 ft. per min.

An electric winder is used at Orient No. 2 mine at West Frankfort, Ill. (Chicago, Wilmington, & Franklin Coal Co.). Coal skips normally weigh 10 tons, maximum weight, 11·5 tons; coal dumped at shaft bottom from mine cars by tipplers; depth of shaft, 202 yds.; normal output, 1500 tons per hour; maximum, 2,000 tons per hour; time of wind, 16 secs.; maximum winding speed, 4,000 ft. per minute; peak load during acceleration, 8,000 h.p.; two motors, each 2,000 h.p., one placed at each end of drum shaft; motor generator set, 40 tons; fly wheel, 12 ft. diameter, equipped with two 1,800 kw., 600 volt, 600 r.p.m., d.c. generators, driven by a 2,200 h.p. induction motor 220 volts, 3-phase, 60 cycles, 600 r.p.m. At the Nemacolin Mines, Pa., the seam is 250 ft. below and the tipping platform 90 ft. above ground level; the skip carries 16 tons and the plant can wind 1,500 tons per hour; the electric winder develops 1,400 h.p., the drum being cylindro-conical.

The largest electric winder in Gt. Britain is at the Carlton Main Colliery, Grimethorpe; it raises 300 tons per hour from a depth of 2,610 ft., and can develop 7,500 h.p. at 49·6 r.p.m.: it has a 23-ton flywheel Ilgner set.

For full information on electric winders see "Electric Winders" by H. H. Broughton, 1927.

Headgears.

Headgears.—Steel-framed headgears usually range from ⅝ to 1 ton per foot of height, and cost erected 12*l.* to 18*l.* per ton; headgears composed of angles and joists only, not lattice girders, weigh ⅝ to ⅞ ton per foot of height. Wooden headgears weigh about ¼ ton per foot of height and cost 10*l.* to 12*l.* per ton. The size of the main legs of timber headgears is given approximately by the expression $A = 0{\cdot}03\ H^4\sqrt{W}$, where W is the total working load on the winding ropes in tons, H the height in feet of the headgear from the ground to the pulley centres, and A the area of the cross-section of the main leg; the area of the backstay = 0·64 A.
(R. Boyle, *Scott. Inst. Min. Stud.* 1908.)

Ferro-concrete headgears have been erected in several places and have proved quite satisfactory.

Pulleys.

Pulleys.—Usually 10 to 20 ft. in diameter, mostly of the same diameter as the drum. Pillow blocks may be carried on strong springs to diminish shock at starting.

If W be the weight of a pulley in cwts. and *d* its diameter in ft., $W = 9{\cdot}2\ (d-8)$ approximately for large colliery headgear pulleys. $W = \frac{d^2}{4}$ is a useful approximation, but is too low for pulleys under 6 ft. in diameter.

The angle of lead from the drum to the head gear pulleys should not exceed 2° on either side of the centre line; the horizontal distance between the drum and pulley centres should not exceed 150 ft., otherwise the rope is apt to surge.

The first Koepe winder, supported directly over the shaft, thus dispensing with the usual headgear and pulleys, to be erected in this country, is in operation at Plenmeller Colliery, Northumberland. The Koepe driving pulley is 11 ft. 6 ins. in diameter, driven through single reduction gearing at 200 r.p.m. by a 240 h.p., d.c. motor, fitted with Ward-Leonard control. Current is supplied by a 200 kW., 350 volt generator direct coupled to a 3-phase 50 cycle 600 volt induction motor running at 730 r.p.m. The depth of the shaft is 815 ft., and single deck cages taking 3 tubs holding altogether 48 cwts. of coal are used. The whole of the hoisting machinery is contained in a two-storied steel-framed house with corrugated iron sides and roof mounted on a tall lattice-girder framework. An overall efficiency of 60 per cent. is attained.
(*Trans. Inst. Min. Eng.*, lv. p. 170.)

Another has been erected at Murton Colliery, Co. Durham. The motor here is 400 h.p. on a 2,000 volt, 40 period current; it draws 4 tons of coals per wind from a depth of 1431 ft. in 80 secs., The coefficient of friction between rope and driving pulley has been found to be 0·407. The overall efficiency is 61 per cent. (G. Raw, *Trans. Inst. M.E.*, vol. lxxiii., 1927, p. 380.)

On the Rand, hoisting with skips costs 8*d.* to 1*s.* per ton from depth of about 1,000 ft.

ROPES* FOR HAULING AND WINDING.

Weight in lbs. of rope of circumference C inches per yard :—

Of ordinary hauling or winding wire ropes with hemp cores about	$0{\cdot}55\ C^2$.
,, ,, wire ropes with wire cores . . . . . ,,	$0{\cdot}6\ C^2$.
,, flattened strand rope	$0{\cdot}57\ C^2$.
,, locked coil rope	$0{\cdot}75\ C^2$.
,, all-steel guide ropes	$0{\cdot}65\ C^2$.
,, hemp ropes	$0{\cdot}2\ C^2$.
,, coir ,, ...	$0{\cdot}12\ C^2$.
,, aloes ,, ...	$0{\cdot}1\ C^2$.

The number of wires of diameter d in a wire rope of diameter D is approximately $0{\cdot}75\ \frac{D^2}{d^2}$.

The breaking strength in tons of ropes of circumference C inches is approximately :

Ordinary hemp rope	$0{\cdot}3\ C^2$.	Aloes fibre rope .	$0{\cdot}4\ C^2$.	Mild steel . .	$1{\cdot}75\ C^2$.
Best ,,	$0{\cdot}6\ C^2$.	Charcoal iron .	$1{\cdot}5\ C^2$.		

For ordinary ropes (with hemp cores) of	80–90 Ton wire . . .	$3{\cdot}3\ C^2$.
,, ,, ,,	90–100 ,, . . .	$3{\cdot}7\ C^2$.
,, ,, ,,	100–110 ,, . . .	$4{\cdot}0\ C^2$.
,, ,, ,,	110–120 ,, . . .	$4{\cdot}4\ C^2$.
,, ,, ,,	>120 ,, . . .	$4{\cdot}6\ C^2$.

The weight of rope in lbs. per yard which shall have a breaking strength of W tons = kW, where k has values :

For 80–90 Ton wire .	0·167	For 100–110 Ton wire .	0·138	For >120 Ton wire .	0·120
,, 90–100 ,, .	0·148	,, 110–120 ,, .	0·125		

If S is the breaking strength of the wire in tons, the strength of the rope is approximately

$$0{\cdot}04\ C^2\ S.$$

If a rope n yards in length is required to carry a load of W tons with a factor of safety of f, using wire having a breaking strength of S tons per sq. in., weight of rope in lbs. per yd. is

$$\frac{2{,}240\ f\ W}{160\ S - f n} \quad \text{or} \quad \frac{2{,}240\ k\ f\ W}{2{,}240 - k\ f\ n}.$$

If $160\ S$ is $< f n$, a stronger wire must be selected.

The British Association of Wire Rope Manufacturers has adopted the following distinctive nomenclature for the steel wire ordinarily used in rope manufacture :—

Description.	*Breaking Strain.*
Best patent steel wire	80 to 90 tons per sq. in.
Special improved patent steel wire	90 to 100 ,, ,,
Best plough steel wire	100 to 110 ,, ,,
Special improved plough steel wire	110 to 120 ,, ,,
Extra special improved plough steel wire . . .	above 120 ,, ,,

Mild steel used for special purposes has breaking strength of 26 to 80 tons per sq. in.

For wire ropes of	6	strands of	7	wires	deduct	5%	off sum of strength of wires.				
,,	6	,,	12	,,	,,	10%	,,	,,	,,	,,	
,,	6	,,	30	,,	,,	15%	,,	,,	,,	,,	
For locked coil ropes deduct . . .						5%	,,	,,	,,	,,	

* See the B.E.S.A. specifications for winding and hauling ropes.

Wire rope should never be bent round a drum or pulley whose diameter is less than 100 D, or 1,000 d; 150 D, or 1,500 d, are preferable, and the respective coefficients 120 and 1,200 are quite frequently employed. In Belgium, diameters of pulleys greater than 750 d for flat ropes, or greater than 1,000 d for round ropes, are allowed. In the Pas de Calais the following proportions of drum diameter have been adopted: for Lang's lay, greater than 68 D, for flattened strand greater than 72 D, and for lock coil, greater than 80 D.

(J. F. Perry, '*Mining Engineer*,' vol. iv., No. 34, p. 24.)

The maximum bending stress in pounds per square inch produced by bending a rope round a pulley of diameter $\triangle$ is theoretically $E\frac{d}{\triangle}$, where E is the modulus of elasticity of the wire used; it ranges from 10,000,000 to 30,000,000 lbs. Owing to the strain not acting directly at right angles to the cross section of the whole rope, it is found in practice that the mean stress is more correctly represented by the expression $\frac{3}{8}E\frac{d}{\triangle}$. According to R. W. Chapman (*Proc. Austr. Inst. Min. Eng.*, vol. iii., No. 3, April 1908), if a is the angle of pitch of the wires in a strand and b of the strands in a rope, the bending stress is $E\frac{d}{\triangle}\cos^2 a \cos^2 b$.

If s be the mean bending stress as above, T the breaking strength in lbs. per sq. in., m the factor of safety (ratio of total breaking load to total working load), W the total load to be carried, and A the total cross-section of steel in the rope, $A = \frac{mW}{T - s}$.

For ordinary winding ropes $A = 0{\cdot}6\ D^2$ approximately.

Factor of safety for haulages, from 5 to 6, for winding ropes 8 to 10.

Coefficients of friction of wire rope :—

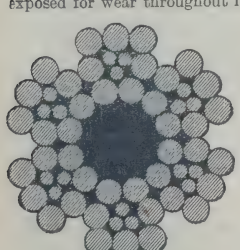
Dry rope on a grooved iron drum . .	0·120	Greased rope on a wood filled sheave	0·140
Wet " " " " . .	0·070	Dry rope on rubber or leather filling	0·495
Greased " " " . .	0·70	Wet " " " " .	0·400
Dry rope on a wood filled sheave . .	0·235	Greased " " " " .	0·205
Wet " " " " . .	0·170		

According to G. Raw (*Trans. Inst. Min. Eng.*, iv., p. 180) the coefficient of friction of a slightly oiled flattened strand rope upon an elm-filled sheave, fibres on end, is between 0·3 and 0·35.

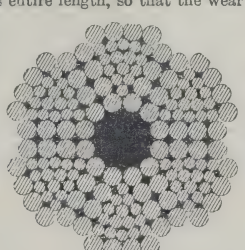
The resistance to bending of a rope under tension T lbs. round a pulley of radius r ft. $= 1{\cdot}26 + 0{\cdot}002276\frac{T}{r}$ (*Weisbach*). The maximum admissible stress in a medium wire rope may be taken as averaging 5,000 times the weight of 1 foot of the rope.

Flattened Strand Ropes.

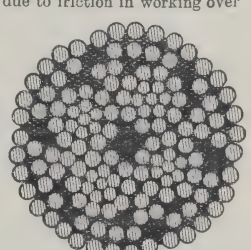
Flattened strand ropes are constructed of strands each of which has a flattened surface exposed for wear throughout its entire length, so that the wear due to friction in working over



No. 48 K.
Haulage Rope.



No. 49 K.
Winding Rope.



No. 163.
Flexible Crane Rope (Non-rotating).

pulleys is distributed over a larger number of wires than is the case with ordinary round stranded ropes. In the latter the wear is very severe on the crown wires of each strand, which are consequently, considerably reduced in sectional area before any appreciable wear has taken place in the adjoining wires. The smooth surface of flattened strand ropes materially reduces wear and tear on drums and pulleys; owing to the wear being so light on the ropes, it is not necessary to have so large a margin between the working load and the breaking strain as in ropes of ordinary make. Being more compact, they contain a larger metallic area than ordinary ropes, and therefore a flattened strand rope is of smaller size than an ordinary rope of equal breaking

strain, and is not so liable to be forced out of shape by the crushing stress it has to sustain. When required, flattened strand ropes are made non-spinning for winding and sinking purposes. They are very suitable as balance ropes, as, being absolutely dead, they can be used with or without pulleys in the sump.

Flattened strand haulage ropes splice more effectively than ordinary ropes, as the core is slightly smaller than the strands, and there is consequently less risk of the splice drawing. This construction is of special utility in ropes used for power transmission purposes, aerial ropeways, etc., where an efficient splice is essential. The greater surface of flattened strand ropes increases the holding power of grips or clips used in haulage work. For use on cranes and lifts, these smaller ropes possess special advantages, as the construction gives greater strength than the ordinary make without increase in size or loss of flexibility. Non-rotating ropes of special formation are in great demand for crane work, particularly for cranes used in the erection of large buildings, and on quays and shipping wharves, as any tendency of the rope to spin is entirely obviated.

Approximate Circum.	Approximate Diameter.		Approximate Weight. Per ftlm.	Per metre.	Actual Breaking Loads. Best Patent Steel, 80–90 Tons per Sq. In.	Special Improved Patent Steel, 90–100 Tons per Sq. In.	Best Plough Steel, 100–110 Tons per Sq. In.	Special Improved Plough Steel, 110–120 Tons per Sq. In.	Extra Special Improved Plough Steel, 115–125 Tons per Sq. In.
Ins.	Ins.	Mm.	Lbs.	Kilos.	Tons.	Tons.	Tons.	Tons.	Tons.
2	$\frac{8}{8}$	16	4·63	1·15	13·12	14·56	15·87	17·08	18·19
$2\frac{1}{4}$	$\frac{11}{16}$	18	5·82	1·44	16·63	18·52	20·14	21·75	22·90
$2\frac{3}{8}$	$\frac{12}{16}$	19	6·51	1·61	18·59	20·44	21·75	24·25	25·27
$2\frac{1}{2}$	$1\frac{3}{16}$	20	7·18	1·78	20·71	22·97	25·07	27·05	27·95
$2\frac{3}{4}$	$\frac{14}{16}$	22	8·69	2·15	25·12	27·79	30·46	32·82	34·40
3	$1\frac{5}{16}$	24	10·13	2·51	29·50	32·73	35·70	38·64	40·25
$3\frac{1}{8}$	1	25	11·1	2·75	31·99	35·73	38·90	42·04	43·48
$3\frac{1}{4}$	$1\frac{1}{16}$	26	12·1	3·00	34·81	38·72	42·08	45·80	47·60
$3\frac{1}{2}$	$1\frac{1}{8}$	28	14·0	3·47	40·74	45·11	49·14	53·67	51·30
$3\frac{3}{4}$	$1\frac{3}{16}$	30	16·0	3·97	46·62	51·84	56·42	61·00	63·35
4	$1\frac{1}{4}$	32	18·2	4·51	52·82	58·41	63·90	69·30	71·96
$4\frac{1}{4}$	$1\frac{5}{16}$	34	20·4	5·05	59·16	65·40	71·62	77·62	81·10
$4\frac{3}{8}$	$1\frac{3}{8}$	35	21·72	5·39	63·37	70·29	78·50	84·30	86·90
$4\frac{1}{2}$	$1\frac{7}{16}$	36	23·0	5·70	66·07	73·34	79·52	86·21	91·05
$4\frac{3}{4}$	$1\frac{1}{2}$	38	25·5	6·32	73·47	81·13	89·45	97·21	100·96
5	$1\frac{5}{8}$	40	28·2	6·99	81·40	90·12	98·99	107·68	111·65
$5\frac{1}{4}$	$1\frac{11}{16}$	42	30·9	7·66	90·15	99·61	109·91	119·51	123·84
$5\frac{1}{2}$	$1\frac{3}{4}$	44	33·8	8·38	98·76	109·22	120·85	130·81	136·60
$5\frac{3}{4}$	$1\frac{13}{16}$	46	36·9	9·15	107·23	118·57	130·93	142·02	146·09
6	$1\frac{7}{8}$	48	40·2	9·96	116·73	130·77	142·47	154·33	163·09
$6\frac{1}{4}$	2	50	43·5	10·80	127·22	141·78	154·48	166·90	174·56
$6\frac{1}{2}$	$2\frac{1}{16}$	52	47·7	11·80	136·05	153·90	167·63	181·70	192·97
$6\frac{3}{4}$	$2\frac{1}{8}$	54	51·3	12·70	149·32	165·64	179·95	196·04	204·23
7	$2\frac{1}{4}$	56	55·0	13·60	161·26	178·88	195·10	211·71	221·13

NOTE.—The figures given above are calculated on flattened strand ropes, having six strands with hemp main core, see No. 495, p. 2607. (*Wrights' Ropes Ltd., Birmingham.*)

Locked Coil Ropes.

Locked coil ropes are in many cases advantageous where heavy loads have to be raised from great depths. Strength for strength, they are of less diameter than ordinary ropes, and consequently admit of corresponding reduction in width of drum, thus minimising lateral friction on drums and pulleys. They are absolutely free from any tendency to rotate, and are therefore of special utility in sinking operations and for winding in shafts where there is not much clearance, as the possibility of collision between the cages is obviated. Frictional wear on guides is also very much reduced by the use of locked coil ropes in winding. They are of longer duration than any other form of rope, as owing to their smooth surface the external wear is evenly distributed over the periphery of the rope, and internal friction, which is so marked a feature of ordinary ropes, is reduced to a minimum. In the event of any wire being broken in use, it is held in position and does not, as in an ordinary rope, unravel and lay across the adjacent wires, causing other fractures in working over drums and pulleys. The tendency to internal corrosion is less in locked coil ropes than in any other type of rope, as during the process of manufacture the successive annular layers of wires of which the ropes are composed are coated with a special

WRIGHTS' LOCKED COIL ROPES

for Aerial Ropeways, Cableways, Colliery Winding, Guide Ropes, etc., "immensely strong, non-rotating, perfectly balanced and smooth running in operation."

WRIGHTS' TRICOR-KEYSTONE ROPES

for Transporters, Haulage, Winding, Dredger Ladder Lines, etc., "the rope of ropes for arduous conditions—strong, resists frictional wear and cannot be crushed."

WRIGHTS' "OXOVAL" NON-ROTATING FLATTENED STRAND ROPES

for winding and hoisting

"absolutely non-rotating, and 17½% stronger than any Round Strand of corresponding size and quality."

WRIGHTS' "FOROVAL" NON-ROTATING ROPES

For Cranes, Hoists, Grabs, etc., "foolproof and effective in cases where a non-twisting rope is desirable."

WRIGHTS' "MODISEAL" AND "TRUSEAL" ROPES

For Dragline Excavators, etc., "the finest of their type in the world—breaking records wherever installed."

WRIGHTS' "RUST-RESISTING" STEEL ROPES

"The latest contribution of Science to ropecraft."

Further particulars of these and Wrights' other special Ropes are available on request.

WRIGHTS' ROPES LIMITED

BIRMINGHAM, ENGLAND

LONDON OFFICE: Salisbury House, London Wall, E.C. 2

'Phone: Metropolitan 5611

See articles on Flattened Strand, Locked Coil and "Rust-Resisting" ropes on pages 2607, 2608, and 2610 respectively.

F. 2608

LATCH & BATCHELOR

LIMITED

AMALGAMATED WITH

WEBSTER & HORSFALL

ESTABLISHED 1730

WIRE DRAWERS

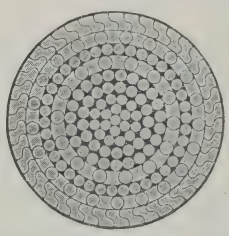
AND

WIRE ROPE MANUFACTURERS

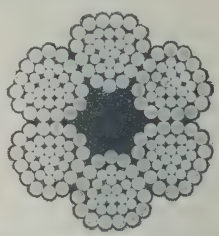
Inventors, Patentees and Originators of

PATENT, PLOUGH and EXTRA PLOUGH STEEL WIRES, LOCKED COIL, FLATTENED STRAND and FLATTENED CONCENTRIC STRAND ROPES, and other specialities in Wire Ropes for WINDING, OIL WELLS, AERIAL ROPE-WAYS and CABLEWAYS, DREDGING, Suspension Bridges, Cranes, Excavators, Lifts, Power Transmission, Steam Ploughing, Ferries, etc.

Manufacturers of Steel Wire in all Sections and with Tensile Strengths up to 200 tons per square inch. Originators of Anti-Corrosive Steel Wire and Steel Wire Ropes.



Locked Coil Ropes of every Type
Patented by us in 1884.



Flattened Strand Ropes of every Type
Patented by us in 1888.

Head Office and Works:

Hay Mills, BIRMINGHAM, England

Telegraphic Address:
"Latch, Hay Mills, England."

Telephone:
Birmingham, Victoria 0555 (3 lines)

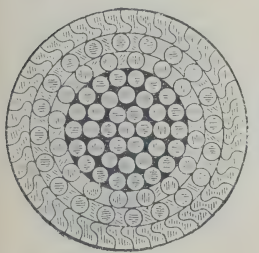
Cable Codes: A 1 and A B C (4th & 5th Editions). Marconi, Bentley's, Western Union, Hamilton's Rope Code, Private Code.

F .2609

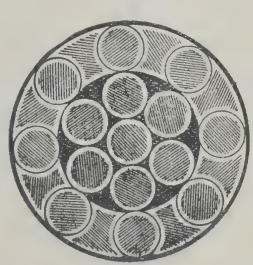
rust and acid-proof lubricant, which cannot be worn off during work owing to the protection afforded by the interlocking wires of the outer cover. In consequence of the wear on these ropes being so light, they may be worked with a lower margin of safety than is the case with ordinary ropes, as with the latter, in estimating factor of safety required, provision has to be made for the rapid wear which takes place.

As guide ropes, the advantages of the locked coil construction may be readily understood when it is borne in mind that, in the event of a rod breaking in the ordinary twisted guide rope, it would strand out from the rope, and be caught by the cage gland with disastrous results; in a locked coil rope the broken rod would be securely held in position.

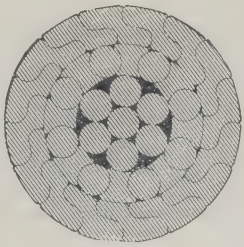
Locked coil ropes are eminently desirable as standing ropes for aerial ropeways, as their smooth surface affords very easy running for the wheels of the carrier; consequently less haulage power is required and wear and tear considerably reduced.



No. 63.
Winding Rope.



No. 64.
Guide Rope.



No. 66A.
Aerial Rope.

Locked Coil Steel Wire Ropes for Winding, Sinking, etc.

Circum.		Diameter.			Approximate Weight.		Approximate Breaking Strain.		Minimum Diameter of Drum and Pulley.	Guides, Rubber Ropes, Standing Ropes for Aerial Ropeways, etc. Approximate Breaking Strain.		
					Per fthm.	Per metre.	Special Improved Patent Steel.	Special Improved Plough Steel.		Special Mild Steel.	Best Patent Steel.	Special Improved Patent Steel.
Ins.	mm.	Ins.	Ins.	mm.	Lbs.	Kilos.	Tons.	Tons.	Feet.	Tons.	Tons.	Tons.
3¼	79	1	·994	25	15½	3·86	49	60	9½	23	30	46
3¼	82	1 1/16	1·03	26	16½	4·1	54	70	9½	25	34	50
3½	88	1⅛	1·11	28	18	4·5	65	80	10½	31	41	62
3¾	94	1 3/16	1·19	30	21	5·21	75	87	11¼	36	48	72
4	101	1¼	1·27	32	24	5·95	86	100	12	41	55	82
4¼	107	1 5/16	1·35	34	26½	6·45	95	111	12¾	45	60	90
4½	114	1 7/16	1·43	36	30	7·44	107	123	13½	51	68	102
4¾	120	1½	1·51	38	34	8·43	118	140	14¼	57	76	114
5	127	1⅝	1·59	40	38	9·43	132	154	15	63	84	126
5¼	133	1 11/16	1·67	42	41	10·25	144	170	15¾	68	91	136
5½	139	1¾	1·75	44	45	11·16	157	185	16½	74	99	148
5¾	146	1 13/16	1·83	46	49	12·23	173	203	17¼	82	109	164
6	152	1⅞	1·91	48	53	13·14	190	222	18	90	120	180
6¼	158	2	1·98	50	59	14·71	203	243	18¾			
6½	165	2 1/16	2·06	52	62	15·5	215	260	19½			
6¾	171	2 5/16	2·14	54	65	16·24	235	281	20¼			
7	177	2 3/16	2·22	56	74	18·44	253	303	21			

(Wrights' Ropes Ltd., Birmingham.)

Size and Weight in Lbs. per Yard, and Safe Working Load in Cwts., of Round Wire Ropes.

IRON ROPES.

Circumference in Inches.	Weight in Lbs. per Yard.	Working Load in Cwts.	Circumference in Inches.	Weight in Lbs. per Yard.	Working Load in Cwts.	Circumference in Inches.	Weight in Lbs. per Yard.	Working Load in Cwts.	Circumference in Inches.	Weight in Lbs. per Yard.	Working Load in Cwts.
Best Charcoal Iron.											
1	⅛	4	2⅛	2¼	20	3⅜	4½	36	4⅛	9	72
1¼	⅛	6	2¼	2½	22	3½	5	40	4¼	9½	76
1⅜	1	8	2⅜	3	24	3⅝	5½	44	4⅜	10	80
1½	1¼	10	2½	3¼	26	3¾	6	48	5	11	88
1⅝	1½	12	2⅝	3¾	28	3⅞	6½	52	5¼	11½	94
2	1¾	14	3	3¾	30	4	7	56	5½	12¼	100
2⅛	2	16	3¼	4	32	4¼	7½	60	6	15	120
2¼	2¼	18	3¼	4¼	34	4⅜	8	64	6¼	16½	132

STEEL ROPES.

Circumference in Inches.	Weight in Lbs. per Yard.	Working Load in Cwts.	Circumference in Inches.	Weight in Lbs. per Yard.	Working Load in Cwts.	Circumference in Inches.	Weight in Lbs. per Yard.	Working Load in Cwts.	Circumference in Inches.	Weight in Lbs. per Yard.	Working Load in Cwts.
Best Crucible Steel.											
1	⅛	6	2¼	2¼	30	3⅛	4	56	4¼	7⅓	100
1¼	⅛	8	2⅜	2½	32	3⅜	4½	60	4⅜	9	120
1⅜	1	10	2½	2¾	36	3½	5	64	4½	10	132
1½	1	12	2⅝	3	40	3⅝	5½	72	5	11	154
1⅝	1¼	16	2¾	3¼	44	3¾	6	76	5¼	12½	175
1⅞	1½	20	2⅞	3¼	48	3⅞	6½	80	5⅜	13⅓	189
2	1⅝	22	3	3⅜	52	4	7	94	6	15	210
2⅛	2	26									
Plough Steel.											
1	⅛	10	2⅛	2	36	3	3⅜	72	3⅞	6½	120
1¼	⅛	12	2¼	2¼	44	3⅛	4	76	4	7	132
1⅜	1	16	2⅜	2½	48	3½	4¼	80	4⅛	8	154
1½	1¼	22	2½	2⅝	52	3⅝	4½	88	4¼	9	175
1⅝	1½	26	2⅝	3	56	3¾	5½	94	5	11	189
2	1¾	32	2¾	3¼	64	3¾	6	100			

See also British Standard Specification No. 300, 1927, for 'Round Strand Steel Wire Ropes for Colliery Winding Purposes.'

'Rust-Resisting' Steel Ropes.

(Wrights' Ropes Ltd., Birmingham.)

These ropes are constructed from steel wires treated by a special rust-proofing process which gives them immunity from corrosive influences. The protective constituent is actually part of the structure of the steel itself and does not affect in the slightest degree its physical characteristics. The ropes have been installed in some of the leading collieries in the country and are stated to be working with complete success.

'Tru-Lay' Ropes.

'Tru-lay' ropes have the individual wires in each strand and the individual strands of the rope pre-formed into their proper shape before they are laid up, with the result that the rope has no tendency to open or to untwist, and that it can be cut without needing previous binding. Such ropes can be well secured in the 'Tru-loc' attachment, made of mild steel, which is cold-swaged on to the rope. ('Tru-lay Wire Ropes and Tru-loc Fittings,' by A. T. Adam, *Trans. Inst. Min. Eng.*, vol. lxxvi, 1928, p. 77.)

Flat Ropes.

W = width, in inches. t = thickness, in inches. $k = 35$ for charcoal iron, 55 for crucible, 70 for plough steel.

Safe working load = Wtk cwts.

Size and Weight in Lbs. per Yard, and Safe Working Load in Cwts., of Flat Wire Ropes.

Dimensions in Inches.	Weight in Lbs. per Yard.	Working Load in Cwts.	Dimensions in Inches.	Weight in Lbs. per Yard.	Working Load in Cwts.	Dimensions in Inches.	Weight in Lbs. per Yard.	Working Load in Cwts.	Dimensions in Inches.	Weight in Lbs. per Yard.	Working Load in Cwts.
Best Charcoal Iron.											
$2\frac{1}{2} \times \frac{1}{2}$	5	40	$3 \times \frac{5}{8}$	8	64	$4 \times \frac{11}{16}$	11	88	$4\frac{5}{8} \times \frac{3}{4}$	$15\frac{1}{2}$	124
$2\frac{1}{2} \times \frac{7}{16}$	6	48	$3\frac{1}{2} \times \frac{11}{16}$	9	72	$4\frac{1}{2} \times \frac{11}{16}$	$12\frac{1}{2}$	100	$4\frac{5}{8} \times \frac{13}{16}$	17	136
$2\frac{3}{8} \times \frac{7}{16}$	7	56	$3\frac{3}{4} \times \frac{11}{16}$	10	80	$4\frac{3}{8} \times \frac{3}{4}$	14	112	$4\frac{5}{8} \times \frac{7}{8}$	18	144
Best Crucible Steel.											
$2\frac{1}{2} \times \frac{1}{2}$	5	64	$2\frac{3}{4} \times \frac{5}{8}$	7	80	$3\frac{1}{4} \times \frac{5}{8}$	9	124	$3\frac{3}{4} \times \frac{11}{16}$	11	144
$2\frac{1}{2} \times \frac{7}{16}$	6	72	$3 \times \frac{5}{8}$	8	100	$3\frac{1}{2} \times \frac{5}{8}$	10	136	$4\frac{1}{4} \times \frac{3}{4}$	$15\frac{1}{2}$	180
Plough Steel.											
$2\frac{1}{2} \times \frac{1}{2}$	5	80	$2\frac{3}{4} \times \frac{5}{8}$	7	124	$3 \times \frac{5}{8}$	8	144	$3\frac{1}{2} \times \frac{5}{8}$	10	180
$2\frac{1}{2} \times \frac{7}{16}$	6	100									

Damping Dynamic Stresses in Mine Winding Ropes.

Investigation has shown that dynamic shocks have a marked influence on the life of steel winding ropes, and has led to the design of apparatus for the continuous control of the stresses in the ropes.

The elements of the shock-damping buffer are springs and pistons moving in hydraulic cylinders; the buffer (Koepe system) is preferably interposed at the lower end of the rope, as shown in fig. 14.

The buffer is shown in fig. 15. The main bar (*b*) passes through the roof of the cage (*a*), and is connected to the piston-rod (*c*) by a bolt (*d*). The buffer itself consists of the piston (*f*), the spring (*g*), and the cylinder (*e*), and is suspended by the two trunnions (*h*), the chains (*k*), and the headpiece (*l*). The dynamic effect of the brake depends on the throttle valves (*m*) (of which there are four), through which the fluid (glycerine) contained in the cylinder is forced as the cylinder is displaced. A feather (*i*) prevents the piston from turning. An expansion space (n) is provided above the main space (*e*) of the cylinder, and connected to it by a hole (*o*). This space (*n*), which is filled only partly with glycerine, serves to accommodate the expansion of the glycerine through heat and its varying displacement by the piston-rod.

From fig. 14 it will be seen that when the lower cage is stopped at the bottom of the pit the heaviest shocks come when the portion L1 of the winding rope is above the upper cage. Some of the shocks are taken up by the natural elasticity of the rope, but if L1 is only short it cannot reduce them very much. In the first pit in which the brake was applied the dimensions of the piston and of the spring were so calculated that the effect of the damping apparatus was equivalent to that of a wire rope 400 ft. long. As the length L1 was about 200 ft., the total elasticity was tripled by the arrangement. This holds for a new rope; old ropes have not the same elasticity as new ones, and with them the relative effect of the damping apparatus is therefore much greater. The first buffers were installed in 1924 at the Hannover pit (Germany), and have since been in continuous service. The springs were designed for a normal load of 21·5 tons

which compresses them by about 6 ins. They may be compressed for 3 ins. more with a load up to 32 tons. The total height of the spring is 24 ins., and its section is 1·4 in. by 2·6 ins. There are about 12 coils in each spring.

The practical value of the buffers can be seen from the records of a winding rope. At the Hannover pit 140 wires had broken in a rope without a buffer in two years, and the rope had to be

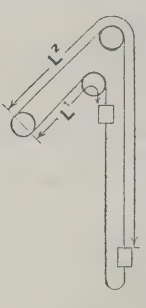


FIG. 14.

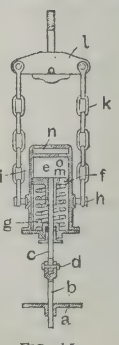


FIG. 15.

removed after this time, but after the buffers had been fitted, only 20 wires were broken in two years of use; moreover, the breaks in the wires of the rope without buffers were concentrated at certain points, but were regularly distributed in the rope with buffers. The use of this rope has been permitted up to four years, so that its life is at least doubled. Another advantage is that the renewal of the rope, which causes delays, is required less frequently.

(*Times Engineering Supplement*, *Oct.* 1928.)

STEEL WIRE ROPES FOR OIL-WELL DRILLING.

Wire ropes are largely employed in drilling oil-wells, particularly where the American system of drilling is in use, which is known as the Californian Cable Rig. This system uses a drilling line for manipulating the tools, a casing line for manipulating the casing, and a sand line for cleaning out the wells. The two former are generally 6 × 19 construction, and the latter 6 × 7.

The Russian or Galician system employs jointed rods in place of drilling lines, but wire ropes are used for cleaning out the wells and for manipulating the casing and the string of tools. The sand, pump, or bailing lines are usually 6 × 7 construction, and the other ropes, which run up to 1½ ins. diameter, are 6 × 19, 6 × 24, or 6 × 30.

Oil-well drilling, being exceptionally severe, calls for ropes of high quality.

Galvanized ropes are sometimes advisable on account of the corrosive action of the water in the wells.

It is a matter of general practice to have the drilling ropes left-hand lay.

Electrical Cables for Collieries.

See MINES AND QUARRIES FORM NO. 11, MINES DEPT. (*April* 1921).*

PUMPS AND PUMPING.

Cornish pumps from 4 ft. to 12 ft. stroke, 6 ins. to 30 ins. diameter, 12 to 16 strokes per min.; piston speeds vary from 60 to 120 ft. per min. A rule sometimes adopted is that speed in feet per second $= \sqrt{l}$, where l is the length of stroke in feet. Ram pumps work at speeds of 100 to 200 ft. per minute.

If n be number of complete strokes (i.e. up and down strokes) per min., and d the diameter of the ram or bucket, the theoretical discharge G in gallons per min. is given by $G = 0{\cdot}034d^2\ n.l$. The actual discharge is less than this by the slip, which ranges from 3 per cent. to 30 per cent.

* Printed and published by H.M. Stationery Office. Price 1*s.* net; 1*s.* 2*d.* post free.

in extreme cases. In practice it is usually between 10 per cent. and 15 per cent. It must be deducted from the theoretical discharge to get the actual discharge. Maximum velocity of water in rising main 200 to 300 ft. per min., rarely exceeds 100 ft. in small pumps; maximum velocity in windbore, 40 ft. per min.

Pressure at bottom of rising main = pressure due to static head + pressure required to overcome resistance of pipe.

If v be the velocity of discharge in feet per second, H the vertical head in feet, L the length in feet of the pipe, D its diameter in inches, and **P** the total pressure at bottom of rising main, then,

$$P = 0{\cdot}341\, D^2 \left(H + 0{\cdot}024 \left(\frac{L}{12D}\right)^4 \frac{v^2}{2g}\right).$$

The pressure in the steam cylinder must be capable of overcoming this pressure, allowing for the mechanical efficiency of the engine (say 0·66).

The content of a pipe in gallons of water = $0{\cdot}034\, D^2L$.

The thickness of a pipe can be calculated thus :—

Thickness in inches = 0·000125 P.D. + 0·5.

Thickness of pump-barrel = $0{\cdot}5 + \frac{P.D.}{2{,}000}$, where P is pressure in lbs. per sq. in.

Weight of pipe in lbs. = $2{\cdot}5\ L\ (D^2 - d^2)$, where D and d are external and internal diameters. The flange = weight of 1 ft. of pipe.

Horse-power in the water lifted = $\frac{g\ H}{3{,}300}$, where g is the actual delivery in gallons per min.

The '**duty**' of a Cornish pumping engine is the number of millions of pounds of water lifted one foot high by the consumption of 112 lbs. of coal (before 1856 of one bushel = 94 lbs., and in America of 100 lbs. of coal) under the boilers. To convert duty into lbs. of coal burnt per h.p. in the water, divide 222 by the duty. It is always theoretical discharge that is taken in calculating duty. Cornish engines have worked with as little as 1·5 lb. coal per h.p.; 2·5 to 3 lbs. is a usual figure.

For double-acting pumps the formula for discharge previously given becomes

$$G = 0{\cdot}068\ d^2 n.l.$$

If A be the area of the piston, n number of strokes per minute, S length of stroke, b the height of the water barometer, l the length of the windbore, a its area, h the vertical height of the suction lift, g the acceleration of gravity,

$$a > 0{\cdot}0055\ \frac{An^2\,Sl}{g(b-h)}$$ for single-acting pumps, and $> 0{\cdot}00275\ \frac{An^2\,Sl}{g(b-h)}$ for double-acting pumps,

or the piston will rise faster than the water can follow it, and hammering will result.

The air-vessel should have a volume = 4 times that of barrel for single-acting, and 1·6 times for double-acting pumps. (*Weisbach.*)

Centrifugal Pumps.

Let u be peripheral velocity of impeller in ft. per sec.

r radius of the impeller in feet.

n number of revolutions per second.

d depth of openings in impeller in feet.

i number of impellers in series.

h height of total lift in feet.

then

$u = 2\,n\,\pi\,r$. Velocity of radial water discharge from impeller = K u, where K is a coefficient between 0·1 and 0·3, averaging 0·2. Quantity of water discharged per second = $K\,u\,2\,\pi\,r\,d = 4\,K\,n\,\pi^2\,r^2\,d$ cub. ft. $h = c\,i\,\frac{u^2}{2g}$, where c is a coefficient that varies considerably, but is mostly between 0·75 and 1.

Mechanical efficiency (m) is between 0·5 and 0·8, usually about 0·7.

Hence h.p. required $= \frac{8\ K\ c\ \pi^4\ n^3\ r^4\ i\ d}{8{\cdot}8\ m\ g}$, or, taking average values, h.p. = about $\frac{2}{3}\ n^3\ r^4\ i\ d$.

Direct coupled motor-driven centrifugal pumps to pump 1,500 gallons p.m. to a height of 1,000 ft. have been found to have an overall efficiency of 71·6 per cent.; electrically driven ram pumps would have a similar efficiency of 80 per cent. *(Prest & Leggat.)*

(See also p. 1266).

VENTILATION.

Mine Gases and Ventilation.

Normal air contains 79 vols. nitrogen, 21 vols. oxygen, carbonic acid 0·06 to 0·11 per cent. With 17·0 vols. of oxygen a candle is extinguished, but breathing is still possible; with 15·2 vols. of oxygen breathing becomes more difficult; with 10 vols. of oxygen, very difficult; and with 5 or 6 vols. practically impossible. Prussian rule for volume of air in mines is 70 cubic feet per min. per man underground, reckoning each horse as four men; it must never amount to less than 53 cubic feet per ton of daily output.

Approximately in Great Britain 5·8 tons of air pass through coal mines per ton of coal raised.

The blood in a man's body is capable of absorbing about one pint (about 20 per cent. of its volume) of either oxygen or carbonic oxide when saturated. If it be half saturated with carbonic oxide, unconsciousness sets in, and with higher proportions death follows. In air containing 0·1 per cent. of carbonic oxide the blood would become half saturated in about an hour in the case of a man at rest, and in about half an hour in the case of a man taking exercise. Anything above 0·15 per cent. of carbonic oxide must be considered dangerous; 0·16 per cent. will cause slight distress to mice in an hour, 0·2 per cent. distress in 8 minutes, 0·3 per cent. distress in 4 minutes, 0·45 per cent. distress in 2 minutes and collapse in 4 minutes, 0·57 per cent. distress in 1 minute, collapse in 6½ minutes, and death in 12½ minutes. With canaries 0·09 per cent. causes distress in one hour, 0·15 per cent. distress in 3 minutes, fall from perch in 18 minutes, 0·2 per cent. distress in 1½ minutes, fall from perch in 5 minutes, and 0·3 per cent. fall from perch in 2½ minutes.

Fire-damp.—Fire-damp consists essentially (up to 95 per cent. or more) of marsh gas or methane, CH_4, the remainder consisting of nitrogen and carbonic acid; hydrogen and some of the higher hydrocarbons have also been found in some samples. Haller and Moureu have found helium with traces of neon (0·003–0·05 per cent.), and argon with traces of krypton and xenon (0·003–0·04 per cent.) in fire-damp. A mixture of air and marsh gas will explode if the mixture contains between 5 and 15 per cent. of marsh gas by volume. The maximum of explosive violence is attained with 9 to 10 per cent. of marsh gas. The igniting point is between 650° and 750° C., and there is always a perceptible 'lag' or delay in the ignition, the amount of which depends upon the temperature. According to Taffanel and Le Floch, mixtures containing only 3 per cent. of marsh gas will ignite at 680° C. and as much as 30 per cent. at 875°. In a mixture of 6·5 per cent. of marsh gas with air, the quantity of gas burnt per second varies from 0·00075 per cent. at 535° C. to 0·05 per cent. at 640° C. According to H. Le Chatelier the igniting point of acetylene is 450° C., hydrogen 550° C., methane and carbonic oxide 650° C. The velocity of propagation of ignition of methane ranges from 0·03 m. per sec. with 6 per cent. of methane to a maximum of 0·61 m. per sec. with 12 per cent., and falls again to 0·1 m. per sec. with 16 per cent. of methane in air. The velocity of the explosive wave in a mixture of methane with pure oxygen is 2,500 m. per sec. Marsh gas has been given off in quantities up to 4,500 cubic feet per ton of coal wrought, and has been shown to exist in coal under pressures up to 40 atmospheres. It is on record that in one mine as much as four million cubic feet of fire-damp has been given off in 24 hours.

Experiments on the ignition of mixtures of methane and air by electric sparks have been recorded by Dr. Thornton (*Trans. Inst. Min. Eng.*, xliv. p. 145, and xlvi. p. 112); the most inflammable mixture is one containing about 7 per cent. of methane. With sparks between copper poles the least continuous current capable of igniting a 9 per cent. mixture was found to be:—

50	100	240	480 volts.
4·0	0·9	0·175	0·05 amps.

With iron poles slightly greater currents are required. With alternating currents of low periodicity, the current required is far greater, ranging from 3 times at 50 volts to 12½ times at 500 volts. The higher the frequency, the larger the current required for ignition.

It has been shown that mixtures of methane and air can be ignited by the impact of siliceous rocks on each other. (Safety in Mines Research Board, Paper No. 46.)

Principles of Ventilation.

(See 'The Ventilation of Mines,' by Hy. Briggs, 1928.)

The difference of pressure producing ventilating currents in mines is measured by the water-gauge; a column of 1 in. of water corresponds to about 66 ft. of air at 60° F and 30 ins. of barometric pressure. The theoretical velocity of air due to such pressure is given by the expression

$v = 66\sqrt{h}$, where v is the velocity in feet per second and h the water-gauge in inches. If the friction in the airways of a mine is taken into account, according to Murgue—

$$h = K\frac{l\,p\,v^2}{a} \text{ or } h = K\frac{l\,p\,Q^2}{a^3}$$

where l is the length of the air passage, p its perimeter, a its area, all in feet, and Q the quantity of air in cubic feet per second. Values of coefficient K: for drifts, bricked and arched, K = 0·000,0012; in ordinary rock, untimbered, K = 0·000,0034; timbered in the usual way, dimensions taken inside timbers, K = 0·000,0057. If the quantities and velocities be given in thousands of feet per minute, these coefficients become 0·00033, 0·00094 and 0·00156 respectively; some engineers use Mr. Fairley's coefficient, which is 0·002. The critical velocity of air flowing in circular pipes $= \frac{18}{D}$ feet per min., where D is the diameter of the pipe in feet.

The work required to move a quantity Q of air under a water-gauge of h inches = 5·2 hQ foot-pounds.

Mr. J. T. Storrow has found that the quantity of air flowing through material under 0·25 in. in size at a slow rate varies directly with the pressure and not with the square root.

(*Trans. Inst. Min. Eng.*, lv. 313.)

The loss by scaling through pack walls, etc., would therefore, for the same difference in pressure, be greater than in an open airway.

Measurement of flow of air. — In a pipe circular in section a velocity equal to the mean flow is at 0·15 of radius from the wall of the pipe. In a mine drift the mean of two measurements halfway between roof and floor and one-sixth of the width of the airway from either side will approximate to the mean velocity of the air in the drift. The mean velocity is between 0·7 and 0·85 of the maximum velocity at the centre of the drift, averaging 0·78.

With the ordinary plain Pitot tube, if v be the velocity of airflow in feet per second and h the water-gauge in inches, $v = 66\sqrt{h}$; with the double Pitot tube (one branch facing the other pointing in the direction of the flow) $v = 56{\cdot}5\sqrt{h}$.

If the pressure of air on either side of a regulator or other diaphragm with an aperture of a square feet be determined by water-gauge, and the difference of gauge pressure be d inches, the quantity of air flowing is $40\,a\sqrt{d}$ cubic feet.

Natural Ventilation and Furnace Ventilation.

Natural Ventilation and Furnace Ventilation.—If there be two shafts, the mean temperature of the air in each of which is respectively T and t (in degrees Fahrenheit), the water-gauge due to this difference of temperature will be $\frac{H}{66}\left(\frac{T-t}{459+T}\right)$, where H is the depth of the pit in feet, assuming that the temperature of the atmosphere is 60° and the pressure 30 ins., under which conditions a cubic foot of air weighs 0·0765 lb.; under any barometric pressure of β inches and temperature $t°$ Fahrenheit, the weight of a cubic foot of air is $\frac{1{\cdot}326\,\beta}{459+t}$, by means of which formula the necessary correction can be applied. The number of heat units required to produce the above water gauge will be $\frac{1{\cdot}326\ Q\ \beta\ (T-t)}{459+t} \times 0{\cdot}2375 = \frac{0{\cdot}315\ Q\ \beta\ (T-t)}{459+t}$ heat units per second.

The grate area is often taken $= \frac{34}{\sqrt{H}} \times \frac{5{\cdot}2\,hQ}{550}$ sq. ft.

In practice, to pass 300,000 cubic feet of air per min. with a 2-in. water-gauge about one ton of coal is burnt per hour.

Fan Ventilation.

Centrifugal Fan.—If u be the peripheral velocity of the fan blades in feet per second, the theoretical water-gauge produced $= \frac{u^2}{68g} = \frac{u^2}{2180}$.

The manometrical efficiency of a fan ranges from about 0·45 for an open-running fan up to about 0·9 for a good modern enclosed fan.

The volumetric efficiency under given conditions is the volume of air actually passed by the fan divided by the volume of the fan × number of revolutions.

The mechanical efficiency is the horse-power required to drive the fan (or of the fan and fan-engine, as the case may be) divided by the horse-power in the quantity of air moved $= \frac{5{\cdot}2\,hQ}{550}$. It varies from about 45 to about 75 per cent. in different types of fans.

The equivalent orifice of a mine $o = 0{\cdot}4\,\frac{q}{\sqrt{h}}$, where q represents the quantity of air in

thousands of feet per minute, and o the area in sq. ft. The equivalent orifice of British mines is usually between 20 and 30 sq. ft.

If H be the initial water-gauge of a given fan (maximum water-gauge, produced on exhausting out of a closed vessel), the equivalent orifice of a fan, working on a mine with equivalent orifice o, should be $= o\sqrt{\frac{h}{H-h}}$. The inlet area of a fan should be about 20 per cent. greater than its equivalent orifice. The equivalent orifice of a fan should be about three times that of the mine on which it works. The velocity at which air enters the fan orifice should be between 1,000 and 1,500 ft. per min. If k be the manometric efficiency of any fan, according to Murgue—

$$u = \sqrt{\frac{g\,68\,h}{k}}.$$

To find the horse-power of a motor or engine required to drive a fan under average mine conditions, multiply the number of cubic feet of air by the water-gauge and divide this product by 4,500. Example: If a fan is delivering 100,000 cub. ft. of air at 2-in. water-gauge, what should be the horse-power of the motor or engine required to drive it?

$$\text{H.P.} = \frac{100{,}000 \times 2}{4{,}500} = 44{\cdot}4.$$

(*Machinery.*)

The Steart fan is an air-screw or propeller fan which is stated to give a very high efficiency and to be readily adaptable to the changing conditions of a colliery. See *Journ. Chem. Met. and Min. Soc. So. Afr.*, vol. xxiv., 1923, p. 31; *Trans. Inst. Min. Eng.*, vol. lxviii., 1924–5, p. 310.

The average consumption of brattice cloth in U.S.A. is 1,697 sq. yds. per 100 tons of coal raised. (*Report on Brattice Cloth, U.S.A. Bureau of Mines.*)

Regulators.

Regulators.—If A be the area in sq. ft. of a regulator which is required to produce as much resistance to the flow of air as a roadway of length l, perimeter p, both in feet, a area in sq. ft., K = Murgue's coefficient of friction as above when the quantity of air is expressed in thousands of cubic feet per minute and the pressure in inches of water-gauge, then $A = 0{\cdot}4\sqrt{\frac{a^3}{K\,p\,l}}$. The regulator should be placed behind the last man in the split.

Safety Lamps.

Until recently it was held that a lamp gauze should be of wire $\frac{1}{70}$ in. diameter and contain 28 meshes to the linear in. (784 meshes per sq. in.); recent researches by the Miners' Lamps Committee have resulted in recommending wire $\frac{1}{100}$ in. diameter with 20 meshes to the linear in. (400 meshes per sq. in.).

See 'Mine Lighting,' by J. W. Whitaker, 1928.

Table showing Average Candle-power of various Lamps (*a*) at Beginning, and (*b*) at End of Ten Hours' Burning with five different Oils.

Lamp.	No. 1 Oil.		No. 2 Oil.		No. 3 Oil.		No. 4 Oil.		No. 5 Oil.	
	(*a*)	(*b*)	(*a*)	(*b*)	(*a*)	(*b*)	(*a*)	(*b*)	(*a*)	(*b*)
Marsaut	0·73	0·63	0·75	0·64	0·74	0·63	0·72	0·64	0·72	0·64
Donald	0·50	0·37	0·49	0·43	0·48	0·43	0·48	0·44	0·48	0·42
Mueseler	0·54	0·42	0·49	0·40	0·46	0·41	0·49	0·40	0·49	0·38
Davy	0·54	0·44	0·83	0·69	0·78	0·70	0·82	0·75	0·81	0·72
'Geordie'	0·46	0·36	—	—	—	—	—	—	—	—
Clanny	0·78	0·72	—	—	—	—	—	—	—	—
Scotch Davy	0·59	0·48	—	—	—	—	—	—	—	—
Deflector	0·96	0·90	—	—	—	—	—	—	—	—

Table showing Cost in pence per 1,000 Lamps of one average Candle-power Hour, with different Oils.

Oil.	Marsaut.	Donald.	Mueseler.	Davy.	'Geordie.'	Clanny.	Scotch Davy.	Deflector.
No. 1	11½	22	14½	18	16	11½	17½	11½
No. 2	19½	33¾	32	19	—	—	—	—
No. 3	25½	43	43½	26½	—	—	—	—
No. 4	35	56	59½	42½	—	—	—	—
No. 5	45½	73	75¾	43¾	—	—	—	—

The oils used were as follows :—

Oil.	Specific Gravity.	Composition, per Cent.		Cost per Gallon.
		Mineral.	Vegetable.	s. d.
No. 1	0·820	100	0	0 9
No. 2	0·841	70	30	1 6
No. 3	0·844	50	50	1 9
No. 4	0·865	30	70	2 3
No. 5	0·889	0	100	2 9

Lamps burning benzine of sp. gr. 0·765, costing 1*s.* per gallon, gave the following results :—

	Candle-power.		Cost in pence per 1,000 lamps of one average c.p. hour.
	At Beginning.	At End.	
Wolf	0·99	0·94	16½
Protector . . .	0·72	0·66	25½

Gas Caps.

Gas caps are shown in an ordinary safety lamp with the flame turned down so as just not to show any white light, when introduced into an atmosphere containing small amounts of marsh gas or methane, approximately as follows :—

2 % of marsh gas. Cap ¼ in., just visible except at extreme tip.
3 % „ „ „ ½ in., fully formed.
4 % „ „ „ ¾ in., well defined.
4½% of marsh gas. Cap 1½ in., well defined.
5 % „ „ „ 2½ ins. „ „
5½% „ „ „ 4 ins. „ „
6 % „ „ „ fills the safety lamp.

Gas-testing lamps burning alcohol will show clearly as little as 0·2 per cent. of marsh gas.

Coal Dust.

Coal-dust.—Three ounces of coal-dust per cubic yard of gallery space is sufficient to produce an explosive mixture; excess of coal-dust prevents neither the initiation nor the propagation of an explosion. Coal-dust giving less than 12 per cent. of volatile matter is practically incapable of initiating an explosion. At least 50 per cent. of inert stone-dust is required to prevent the initiation of an explosion. If less than 75 per cent. of the dust passes the 200 sieve it is difficult to initiate an explosion. (*Abstracted from the Lievin experiments.*)

The temperature of ignition of coal-dust is about 1,000°C. Mixed with air and introduced into a heated vessel the temperature may be as low as 590°; there is very little lag. (*Taffanel and Le Floch.*)

The General Regulations provide that in all coal mines except anthracite mines and except mines of which the floor, roof, and sides of the road are naturally wet throughout, every road or part of a road shall be treated with incombustible dust in such a manner as to ensure that the dust on the floor, roof, and sides throughout shall never contain more than 50 per cent. of combustible matter, or else they shall be treated with water in such a manner that the dust is always combined with 30 per cent. (by weight) of water. It is provided that the percentage of incombustible dust required by this regulation may be reduced by an amount equivalent to the percentage of water present. The incombustible dust shall contain not less than 50 per cent. of material capable when dry of passing through a sieve with 200 meshes per linear inch. If more than 50 per cent. of dust is used, the percentage of fine material in such dust may be reduced proportionately, but shall not fall below 25 per cent. Representative tests must be made by the management at intervals of not less than one month, and the results posted at the pit head. The method in which the tests are to be made is fully prescribed by the Statutory Orders.

Rescue Apparatus.

There are three main types of rescue apparatus :—

(1) The supply of oxygen is maintained by means of compressed oxygen carried in steel cylinders.

(2) The supply of oxygen is obtained from liquid air.

(3) Oxygen is chemically generated from peroxides, usually sodium-potassium peroxide.

In (1) and (2) the expired carbonic acid is absorbed, usually by caustic soda, in (3) the reagent that evolves the oxygen also absorbs the carbonic acid.

Particulars of some forms are as follows :—

Apparatus.	Weight.	Oxygen Capacity.
	Lbs.	Cubic Feet.
Pneumatogen (Type II.)	14½	5·8
Draeger	36–39	8·9
Shamrock	36	9·7
Fleuss	31	8·0
Weg	About 30	5·3
Proto	37	8·5
Meco	39½	8·5
Salvus	16	3·0
Aerolith	22	?
Aerophor	30	?

It is estimated that 120 cubic inches of oxygen per min. is ample for the severest exercise; a man at rest requires about 15 cubic inches, and climbing hills 90 cubic inches per min. The Aerolith uses liquid air containing about two parts of oxygen to one of nitrogen; it is stated that one gallon is sufficient for three hours' work.

The Aerophor holds 10 lbs. of liquid air, equal to about 125 cubic feet.

The following has been found to be a man's average consumption of oxygen per minute, gas dry, at N.T.P.

Rate of doing Work.	Breathing.			
	Air.		Oxygen.	
	Litres.	Cub. in.	Litres.	Cub. in.
At rest	0·36	22	0·32	19·5
3,000 ft.-lbs. per min.	1·12	68·3	0·97	59·2
6,000 " "	1·74	106·2	1·54	95
9,000 " "	2·33	142·2	2·16	131·8

The most economical rate of walking when travelling a flat road and wearing a rescue apparatus—*i.e.* the rate at which a maximum distance can be covered for a given supply of oxygen—is about 2½ miles per hour.

('*Second Report of the Mine Rescue Research Committee of the Department of Scientific and Industrial Research.*')

When the wet-bulb temperature is much above 80°F., a man cannot remain long at work; he can stand a wet-bulb temperature of 120°F. for about five minutes. (*J. S. Haldane.*)

To Detect Carbon Monoxide in the Air.—A Technical Paper (No. 11) of the United States Bureau of Mines suggests the following simple test capable of detecting as little as 0·03 per cent. of carbon monoxide in the air. One or two drops of blood from a pricked finger are diluted with water until this assumes a buff-yellow colour. Some of this discoloured water is placed in two 100 cubic centimetre test tubes, and one of the tubes is to be taken into the mine, where one-half is to be poured away at the place where the air is to be tested, the mine air taking the place of the water poured away. The tube is then to be corked, taken to the surface, and to be shaken for about ten minutes. If the air in the test tube contains CO, the pink colour of the liquid will indicate its presence on comparing the tube with the other test tube filled with the normal fluid. More recently the 'Hoolamite' detector has been used extensively. This consists of iodine pentoxide dissolved in fuming sulphuric acid, the changes of colour in which indicate the amount of carbonic oxide present; it is said to be able to detect 0·07 per cent. A carbonic-oxide recorder has also been developed, which works by recording the rise in temperature in 'Hopcalite.' This consists of a catalyst combined with oxide of manganese which oxidises carbonic oxide to carbonic acid; the heat developed by this reaction is a measure of the amount of carbonic oxide oxidised. (See 'Devices for Detecting Dangerous Gases in Mine Air,' by J. T. Ryan. '*Trans. Amer. Inst. Min. Met. Eng.*' vol. lxxv. p. 599.) Mr. Ivon Graham has shown that carbon monoxide is a normal constituent of the air of coal mines over 1,500 ft. deep, being produced by the slow oxidation of carbonaceous material at the coal face, in the roads, and in the goaf; it is usually under 0·005 per cent., but in a few cases 0·012 per cent. has been found.

('*Trans. Inst. Min. Eng.*,' Vol. lix., 1920, p. 222.)

Spontaneous Combustion.

Work in the Doncaster Coal Owners' Laboratory has shown that the liability of a coal to spontaneous combustion in the goaf depends, other things being equal, upon its capacity for absorbing oxygen. It appears that a coal, 100 grammes of which, ground to 200 mesh, absorbs less

than 200 c.c. in 96 hours at 30°C., is only liable to gob fires when the working conditions especially favour the latter. Coals liable to fire absorb over 300 c.c. in the same time. Presence of pyrites especially when finely divided, increases the liability to spontaneous combustion.

Fire Extinction.

Investigations have been made recently into the scientific side of the mud-jet method of fire extinction in colliery workings, and it has been decided that the material of the mud, some of which should be of a lime and some of a clay character, should consist of sifted earth of 5 mm. mesh, boiler ash, dust from the cleaning of blast-furnace gas, and even combustible material, such as the refuse of coal washing, can be added to make up bulk without any risk should nothing more suitable be at hand. In the majority of instances the mud tank can be so placed as to operate by gravity, but steam or compressed air pressure can be used. It has been found that if a tank is placed at a height of, say, 15 yards, it will force the mud a horizontal distance of 150 yards, even allowing for fairly sharp bends in the conducting pipes.

Electric Signalling in Coal Mines.

A circular issued by the Home Office points out that while there is no difficulty in constructing bells and relays that will comply with the requirements laid down—i.e. which will not 'spark'—most of those in ordinary use in coal mines do not comply with the provision of General Regulation 132; and it states that in mines to which this regulation applies steps must be taken without delay to bring into compliance with the regulation any existing installations which do not already comply, while no new apparatus must be installed except such as does strictly comply. The fact is emphasised in the report that with the usual type of bell and magnitude of battery at present employed, nearly every break-flash that occurs when the bare signal wires are separated would ignite a mixture of firedamp and air containing between 7·5 and 9·5 per cent. of methane, were such a mixture to surround the wires at the time.

Relays, although not requiring for their action so great a battery power as bells, have owing to their higher inductance a lower 'igniting-current' at which the break-flash on the signal wires becomes dangerous. At the present time, with few exceptions, the battery power used on relays is in excess of that sufficient to give a dangerous break-flash at the signal wires. The commonest kinds of relays are dangerous when used with a battery of more than four wet Leclanché cells, or with any current in excess of the current given by such a battery.

The report shows in detail how the bare wire system of electrical signalling as commonly employed can be rendered quite secure from any danger of the ignition of inflammable gases by the break-flash at the signal wires or at the contacts of the signalling instruments.

In order to secure a safe bare-wire signalling system it is necessary in the first place that the battery power employed on any one circuit should be limited, and, in the second place, that the signalling instrument, whether bell or relay, should comply with certain requirements set out in the report.

The maximum pressure allowed by the General Regulation (No. 134 (a)) for a signalling circuit is 25 volts, and from the point of view of safety a pressure of 25 volts is not excessive, provided that, as is usual, wet Leclanché cells (quart size) are employed.

Systems in which the battery used consists of dry primary cells or secondary cells, or in which continuous or alternating current generators are employed, can be rendered safe if, in addition to the use of anti-sparking devices, sufficient non-inductive resistances are permanently installed in the circuit.

Both bells and relays should have flame-tight covers (preferably locked), and should be so constructed and maintained that when included in a circuit with a battery at 25 volts the break-flash produced, when bare signal wires are separated after giving a signal, is incapable of igniting an 8 per cent. methane air mixture.

ORE DRESSING AND COAL WASHING.

(For details see 'The Dressing of Minerals,' by Henry Louis, from which the following is mainly abstracted.)

The operations included under the head of dressing comprise all the operations between the extraction of the crude mineral from the mine and rendering it merchantable, thus forming a link between the operations of the miner and the metallurgist. By common consent, gold milling is generally looked upon as a dressing operation, though it is doubtful whether cyanidation should also be included.

The various operations consist of:—
Hand picking or sorting.
Comminution (coarse breaking or crushing and fine breaking or grinding).
Sizing (preliminary or rough sizing and final or fine sizing).
Classifying.

Separation by differences of specific gravity, magnetism, electric conductivity, or surface tension.

Usually a more or less complicated series of operations is required, especially in the case of complex ores, and these are usually indicated by means of flow sheets.

Let the weight of feed in any concentration process be A and its assay a per cent., the weight and assay of concentrates be B and b and of tails C and c respectively, then the

$$\text{Enrichment ratio} = \frac{b}{a} = \frac{\%\ \text{Recovery}}{\%\ \text{Concentration}}.$$

$$\%\ \text{Concentration} = \frac{100\text{B}}{\text{A}} = \frac{100(a-c)}{b-c}.$$

$$\%\ \text{Recovery} = \frac{100b(a-c)}{a(b-c)} = \frac{\text{B}100b}{\text{A}a}. \quad \text{A} = \frac{\text{B}(b-c)}{a-c}.$$

$$\%\ \text{Efficiency of concentration} = 100\,\frac{\text{B}}{\text{A}}\left(\frac{b}{a} - \frac{100-b}{100-a}\right) = 100\,\frac{a-c}{b-c}\left(\frac{b}{a} - \frac{100-b}{100-a}\right) =$$

$$100\,\frac{\%\ \text{Recovery} - \%\ \text{Concentration}}{\%\ \text{Waste in feed}}.$$

(*R. T. Hancock.*)

Sorting is performed either on floors or on picking belts or tables. An ordinary coal picking belt may be of iron plates 4 ft. wide, 50 to 100 ft. long, travelling at 40 to 65 ft. per minute. Pickers stand on both sides, 7 ft. to 15 ft. apart; each picker can treat 1–3 tons of coals per hour.

Power required to drive, 2½ to 5 h.p. Upkeep, about 50*l.* per annum.

Rubber-covered canvas belts about 32 ins. wide are also used, and will carry 25 to 50 tons per hour at speeds of 30 to 60 ft. per minute.

Circular tables are 3 to 4 ft. wide, 10 ft. to 18 ft. in diameter, revolving with a peripheral speed of 40 to 60 ft. per minute, and can deal with 30 to 70 tons of mineral per hour.

Comminution.—Rough breaking is done by jaw breakers, either reciprocating like those of the Blake, Dodge, etc., type, or gyrating like the Gates type. The former have been built to take pieces up to 5 ft. by 7 ft., requiring 300 h.p. to drive, and breaking 300 tons per hour. Smaller breakers usually break 1 ton per hour for a consumption of 2–3 h.p. The gyrating crusher is better suited to very large outputs, but cannot take such large pieces, and appears to be rather more economical of power.

Fine Crushing.—Rolls are used for medium and fine crushing. Let r be the radius of the particle to be broken and d the radius of the particles in the crushed product (both considered as spheres); the minimum radius of the roll should be $> 25\,(r-d)$.

For fine crushing to about ⅛-in. mesh the following table shows the general practice :—

Rolls.		Revs. per minute.	Horse-power required.	Cwts. crushed per hour.
Diam.	Face.			
10½ ins.	10½ ins.	130	3	25
12½ ,,	10½ ,,	110	4½	45
16 ,,	10½ ,,	90	6½	65
21 ,,	10½ ,,	75	9½	100
27 ,,	11 ,,	45	12½	145
37 ,,	12 ,,	45	16	200

The peripheral speed should be greater the finer the product required; for grinding to 40 mesh the speed should be 1,000 ft. per minute.

In large plants three or four sets of rolls are placed in series so as gradually to reduce the ore to the desired size. At Mount Morgan, such a plant consisted of eight sets of rolls in two series of four, thus :—

26 ins. diam.,	15 ins. face,	set	⅜	in. apart,	112 revs. per min.
26 ,,	15 ,,	,,	5/16	,, ,,	,,
30 ,,	16 ,,	,,	¼	,, ,,	,,
30 ,,	16 ,,	,,		in contact	,,

The total power consumption was 100 h.p. and the crushing capacity 125 tons per 24 hours crushed to 20 mesh (0·025 in.); the wear on the steel roll-shells was 0·108 lb. of steel per ton of ore crushed.

Stamps.—The ordinary gravitation stamp is used more largely for gold milling than for other purposes, but also for crushing tinstone, etc. The complete stamp consists of the head, shoe, stem and tappet, the weight of which may be up to 1,500 lbs. The relative weights are about 30, 17, 40, 13. It is lifted by a cam forming the involute of a circle; if h be the lift of the stamp, r the radius of the pitch circle of the involute, and α the angular motion of the cam during the lift, $h = \frac{\pi r \alpha}{180}$. The speed is usually about 90 drops per minute.

If n be the number of drops per minute, h the drop in inches, s the number of stamps, and w the weight of a stamp in pounds, the power required is approximately $\frac{nhsw}{300{,}000}$. (For a more exact formula see 'A Handbook of Gold Milling,' by H. Louis.) The average crushing capacity is 2–8 tons per 24 hours for a 1,000 lb. stamp. The Husband atmospheric stamp with two heads requires 35 h.p. to work it, and can crush 40 tons of hard tinstone per 24 hours down to 0·048 in. In modern practice on the Rand stamps are used as preliminary crushers, breaking to ½ to 1 in. and are much heavier than above given.

Steam stamps are used mainly for crushing the copper-bearing rocks of the Lake Superior Mines, down to about ¼-in. mesh. One such stamp will crush 10 tons per hour, the efficiency being about 1·75 tons per i.h.p. per 24 hours.

Ball mills have been made both for dry and wet crushing. The Krupp mill is a dry-crushing mill; the No. 5 size, which is largely used, is 86 ins. diameter by 46 ins. length, makes 20–25 r.p.m., weighs 182 cwts., and a set of balls weighs 18 cwts. At Mount Morgan these mills require 13 h.p. and crush 22·7 tons per 24 hours to 20 mesh (0·025 in.). The balls wear at the rate of 0·725 lb. and the hunch plates at the rate of 0·681 lb. per ton of ore crushed.

Wet crushing ball mills are 60 ins. to 80 ins. in diameter and about 40 ins. long. A charge of balls for such a mill weighs about 2 tons; it is run at 25 to 30 r.p.m., takes from 25 to 35 h.p. to drive it, and will crush 25 to 100 tons down to 0·02 mesh per 24 hours; the water consumption is about 65 gallons per minute.

The tube mill is also used wet or dry, and is in effect an elongated ball mill, 3 ft. to 5 ft. in diameter and 13 ft. to 26 ft. long. A tube mill 4 ft. in diameter, 16 ft. 6 ins. long, crushed in Western Australia 38 tons of sand per 24 hours, 95 per cent. of the product being finer than 100 mesh, requiring 30 h.p. to drive it. Such mills are lined with silex, chilled iron, or hard steel bars, and are usually charged with flints. According to Mr. Davidson, the best number of r.p.m. $= 200 \div \sqrt{d}$, where d is the internal diameter in inches; the weight of flints in pounds should be $N \div 44$, where N is the internal capacity of the mill in cubic feet. Western Australian practice corresponds nearly to $N \div 60$.

The Marathon mill uses rods in place of balls; a mill 3 ft. diam., 7 ft. long, charged with 7,000 lbs. of rods ½ in. to 2 in. diam., running at 30 r.p.m., crushed 18 tons per hour with a consumption of 12·5 h.p., using 42 U.S.A. gallons of water per minute. The feed was below ¼-in. mesh and the product was ground to $\frac{1}{64}$-in. mesh, 14 per cent. being below $\frac{1}{200}$-in. mesh. The wear of the liner plates was 0·14 lb. and of the rods 0·4 lb. per ton of feed.

(*Trans. Amer. Inst. Min. Eng.*, lv., p. 678; lvii., p. 355.)

The Hardinge tube mill has the form of two cones base to base. An 8 ft. Hardinge mill requires 50 h.p. to drive it, and requires about 10 h.p. hours per ton crushed to about 48 mesh.

An 8 ft. × 22 ft. mill crushed 200–250 tons per 24 hrs. using about 56 h.p., with a charge of 5 tons of Danish pebbles wearing at the rate of 2·16 lbs. per ton of feed; with steel balls instead of pebbles 360–400 tons were crushed per 24 hours.

(*Trans. Amer. Inst. Min. Eng.*, lv., p. 678.)

The Marcy mill is a short mill (5–6 ft.) with perforated diaphragm at the discharge end, from 2 ins. to 48 mesh. An 8 ft. by 6 ft. mill driven by a 225 h.p. motor crushes 550 tons per 24 hours at 24 r.p.m., carrying 30,000 lbs. chrome steel balls.

(*C. T. Van Winkle, Trans. Amer. Inst. Min. Eng.*, vol. lix. (1918), p. 227.)

The Huntingdon mill is made in three sizes, 3 ft. 6 ins., 5 ft. and 6 ft. diam.; No. 2 is most used; it weighs about 5 tons; consumes 12 h.p.; makes 60–70 r.p.m., and will crush about 10 tons per 24 hours to about 15 mesh.

The Arrastra is 10 to 20 ft. in diameter, has usually four dragstones, 6–8 cwts. in weight At 6 to 12 r.p.m. a charge of 1 to 3 tons is ground in 2–3 hours.

The Chilian mill has 2–3 rollers, 4 to 6 ft. in diameter, takes up to 75 h.p., and crushes up to 150 tons per 24 hours.

Sizing.—For coarse screening fixed grizzlies or 'jigging' screens are used. For medium and fine screening mostly trommels, 8 ft. to 2 ft. in diameter by 12 ft. to 3 ft. long; the peripheral velocity is usually 50 to 200 ft. per minute. The slopes may vary from 5° to 14°, averaging about 7°. A set of seven trommels, 3 ft. diam., 5 ft. long, will size about 7 tons of ordinary crushed material per hour, and require up to 5 h.p. to drive it. For the finest work, vibrating screens are used. Wedge wire is often used for the screening surface.

Classification.—This term applies to the separation into groups of equal falling particles The terminal velocity of a particle falling freely in water is approximately $V = 2{\cdot}6\sqrt{D(S-1)}$; where V is the velocity in metres per second, D the diameter in metres of the screen mesh through which the particle just passes, and S the density of the mineral.

The above is Rittinger's law, based on Newton's, according to which resistance $\propto$ velocity^2 R. J. Lunnon has recently shown that this law is not strictly true. (See 'Fluid Resistance to Moving Spheres,' *Proc. Roy. Soc. A.*, vol. cx., 1926, and vol. cxviii., 1928; also *Trans. Inst. Min. Eng.*, vol. lxxvii., 1929, p. 66.)

When V is less than 0·5 to 1, according to the substance, the law on which these formulas are based (friction $\propto$ velocity^2 and viscosity negligible) no longer holds good. The particles fall more slowly and obey Stokes's law:

$$v \text{ (in c.m.)} = \frac{g}{18} d^2 \frac{S - s}{\mu}.$$

The minimum radius of a particle that obeys Rittinger's law of terminal velocity is

$$\sqrt{\frac{9\mu^2}{2g(S-1)}} \text{ cm.}$$

where μ is the viscosity (0·0115) and g the acceleration of gravity (981), both in C.G.S. units, S is the specific gravity of the particle, s of the fluid, and d diameter of particle in cm.

Hence log radius in cm. $= \bar{3}{\cdot}92764 - \frac{1}{2}\log(s-1)$.

The following table is calculated from the above formula:—

Sp. gr.	Radius in mm.	Sp. gr.	Radius in mm.	Sp. gr.	Radius in mm.	Sp. gr.	Radius in mm.	Sp. gr.	Radius in mm.	Sp. gr.	Radius in mm.
1·2	0·1528	1·6	0·1004	2·0	0·08465	2·4	0·07567	3·5	0·06237	5·5	0·05128
1·3	0·1265	1·7	0·09534	2·1	0·08201	2·5	0·07395	4·0	0·05870	6·0	0·04950
1·4	0·1149	1·8	0·09118	2·2	0·07966	2·6	0·07238	4·5	0·05575		
1·5	0·1071	1·9	0·08768	2·3	0·07757	3·0	0·06719	5·0	0·05333		

Spitzkasten: the first box should be 2 ft. wide for 10 cub. ft. pulp per minute, or 1–2 cwts. of solid matter; boxes may increase in geometrical progression with factor of 1·5; length of first box about 6 ft., increasing by 3 ft. for each successive box.

Trough classifiers 10 to 20 ft. long, 1 ft. wide at narrow end, will treat 100 tons per 24 hours, with 100 cub. ft. of water per ton of crushed material.

Jigs.—The following represents the general modern practice:—

Diam. of particle .	0·25	0·50	0·75	1·00	1·25	1·50	1·75	2·00 ins.
Length of stroke .	0·4	1·2	1·55	1·8	2·0	2·15	2·3	2·4 ,,
No. of strokes per min.	200	160	145	130	125	125	125	125

CONTINENTAL JIGGING PRACTICE ON ORDINARY LEAD-ZINC ORES.

Three to five compartment jigs; screen surface 20″ × 31·5″.

Size of Particles.	Length of Stroke.	Strokes per Min.	Water per Cwt. treated.	Capacity per Sq. Ft. of Screen Area per Hour.
Inch.	Inch.		Gallons.	Cwt.
0·04	0·2	280	90	0·6
0·046 to 0·06	0·25	280	73	0·9
0·06 to 0·08	0·35	260	60	1·2
0·08 to 0·11	0·50	240	55	1·6
0·11 to 0·16	0·60	220	52	1·8
0·16 to 0·22	0·80	200	50	1·9
0·22 to 0·32	1·00	180	46	2·0
0·32 to 0·43	1·20	160	45	2·1
0·43 to 0·63	1·25	140	54	2·2
0·63 to 0·87	1·60	140	58	3·7
0·87 to 1·25	2·00	140	61	4·3

Jig screens of steel wire are liable to rapid corrosion when pyritic ores are being treated; wire drawn from Ambrac (Cu75Ni20Zn5) has been recommended in such cases.

Round buddle; diam. 18 ft. to 25 ft.; makes 5–10 r.p.m. Capacity 1½–3 cub. ft. per min. carrying 28 to 56 lbs. material; time of filling 3–10 hours.

Slime table; diam. 15 ft. to 25 ft.; ½–1 r.p.m. Pulp 1½–3 cub. ft. per min., carrying 4 to 7 lbs. per cub. ft. Clear water 1½–3 cub. ft. per min. Power required ½–1½ h.

Frue vanner; belt surface 4 ft. by 12 ft.; fall 3 ins. to 6 ins.; travel 6 ft. per min.; strokes 200 per min. Capacity 0·25 to 0·5 cub. ft. pulp per min.; 5 to 8 tons material per 24 hours. Clear water ½ of pulp. Power required ½ h.p.

Lühring vanner; belt surface 3 ft. 6 ins. by 12 ft.; travel 8–12 ft. per min.; 160–180 strokes per min. Capacity 2½–7½ tons per 24 hours; clear water 1–2 cub. ft. per min. Power required ⅓–½ h.p.

Wilfley table; 16 ft. long, 6 ft. wide, tapering to 3 ft. Will take particles up to ⅛ in. 240 strokes per min., ¾ in. long. Capacity 25 to 50 tons per 24 hours. Clear water 1 to 3 cub. ft. per min. Power required 1 h.p.

Coal Cleaning.*

Coal as it comes from the mine is first screened, usually over a 2 ins. to 2¾ ins. screen; the oversize drops on to picking belts, where shale, dirty coal, etc., are picked out. The undersize is taken to a washery, especially when it is required for coke-making. A jigging screen with ¾-in. holes will screen about 1 ton per hour for each 0·75 sq. ft. of screening surface.

Coal is sometimes dry-cleaned in the Purdee spiral separator, in which it is allowed to slide down a spiral track, the clean coal falling over the outer edge, whilst the shale, etc., remains on the track.

When the separation of the clean coal from the accompanying shaly matter is easy, or when the products are not required in the highest possible state of purity, a simple form of washer may be used. Such is, e.g., the Elliott trough washer, capable of treating 5 to 10 tons of small coal per hour with a water consumption of 1,750 gallons per ton of coal washed. The Blackett washer is similar in principle, but the fixed trough with the moving dam-plates is replaced by a revolving drum within which is a spiral baffle-plate; it will treat 10 tons of small coal per hour, requiring 3,000 gallons of water in circulation per ton of coal washed. The Robinson washer works on the principle of a spitzkasten, with upward flow of water and a revolving stirrer; it can treat 15 to 20 tons of small coal per hour.

The purity of the washed coal is usually determined by floating it in solutions of known specific gravity; coal that floats in solutions of sp. gr. 1·35 to 1·45 is generally considered clean coal. The washing characteristics of a coal are best determined by constructing a washability curve or Henry curve. (See *Colliery Engineering*, 1925, p. 341.) The washability of coals is

* See 'The Preparation of Coal for the Market,' by H. Louis, 1928.

generally determined by plotting the proportions of the whole which float in solutions of definite specific gravities against the ash contents of these fractions as a curve, from the shape of which much valuable information as to the washing properties of the coal may be obtained.

Most plants for complete washing make use of jigs (bashes or boxes), and the coarser refuse is sometimes crushed and re-washed. See flow-sheet No. 1., p. 2629.

The Coppée Washery is typical of the more usual form of plant. The unwashed coal, say under 2 ins. to 3 ins., is sized before washing. The washing is carried out in two types of boxes, viz.: (*a*) Small coal felspar boxes, which treat all the coal below ½ in.; (*b*) nut coal type of box, which treats the various sizes below 3 ins. and above ½ in. These sizes vary according to local conditions and the requirements of the market. The nut washers are 8 ft. deep, and the sieves are 5 ft. long and 3 ft. to 5 ft. wide, each washer treating 2·5 to 3·5 tons per hour per foot of width of screen. The more important data affecting the operations of nut washers are shown in the following table:—

Size of Nuts.	Depth of Layer of Shale on Sieve.	Length of Stroke.	No. of Strokes per Minute.	Mesh of Sieve.
½ in. to ½ in.	2½ ins.	2½ ins.	80	$\frac{1}{10}$ in.
½ " " 1 "	3 "	3 "	75	$\frac{1}{8}$ "
1 " " 2 ins.	3½ "	3½ "	70	¼ "
2 ins. " 3½ "	4¼ "	4¾ "	65	$\frac{5}{16}$ "
3½ " " 5 "	5½ "	6¾ "	60	$\frac{3}{10}$ "

The Felspar Washer is built in two compartments, each 4 ft. long, making a total length of 8 ft. by 2 ft. 6 ins. wide, the box being 5 ft. deep; such a washer can wash 7 to 10 tons of small coal per hour up to ½ in. in size. The following are the leading data:—

	1st Compartment.	2nd Compartment.
Thickness of layer of feldspar	2½ ins.	2½ ins.
" " shale	$\frac{3}{10}$ in.	$\frac{2}{16}$ in.
Size of pieces of felspar	$\frac{8}{10}$ in. to 1 in.	$\frac{6}{10}$ in. to $\frac{8}{10}$ in.
Length of stroke	0·55 in.	0·47 in.
No. of strokes per minute	165	165
Mesh of screen	0·55 in.	0·47 in.

A plant to wash 100 tons per hour, 50 per cent. on nut washers and 50 per cent. on felspar washers, would require four of the former, 4 ft. wide, and six of the latter. Such a plant would require in circulation about 135,000 gallons of water, and make-up water amounting to between 2,800 and 3,500 gallons per hour.

When the refuse from the nut washers contains sufficient intergrown coal to be worth treating, it is crushed, generally in a disintegrator, and the crushed material is re-washed on felspar washers.

A coal washery may require from 1·5 to 3·5 h.p. for each ton of unwashed coal treated per hour, varying with (*a*) the total output of the washery (a large washery requiring a relatively smaller h.p. per ton); (*b*) the quantity of interstratified coal, and whether the larger size is crushed and washed together with the smaller size, or whether the smaller size only is re-washed; (*c*) the type and capacity of draining and storage bunkers, and whether apparatus is installed for mixing the various products.

Messrs. Simon-Carves, Limited, build washeries on the so-called Baum principle, in which washing precedes classification. The slack below about 2 ins. is delivered to a bash having an area of about 0·8 sq. ft. per ton per hour; the washed coal then passes to a screen of about ⅜-in. mesh; the over-size is clean coal, and the under-size goes to a second bash having an area of about 1 sq. ft. per ton of coal per hour, the product being clean coal and refuse. The dirt produced in the first washing can be further crushed and washed over again if necessary. A plant to treat 100 tons of slack per hour would, therefore, require a first bash with an area of 80 sq. ft., and a second bash with an area of 100 sq. ft.; the water in circulation would be about 130,000 gallons per hour, and the make-up water required about 2,500 gallons.

The rhéolaveur consists of a series of troughs of steel, cement or wood, sometimes lined with tiles, fitted with a series of pockets with adjustable discharges and, for the coarser sizes of coal, with upward water flows. These are arranged so as to discharge clean coal, refuse or middlings for re-washing as may be required. The plant is relatively cheap, compact and efficient and can treat all sizes from 4 in. to slurry. No general figures can be given as the plant has to be specially designed for each case. It is largely used in Belgium, and altogether about 60 million tons of coal a year are treated by it.

At Grassmoor Colliery a rhéolaveur plant treats 100 tons per hour 0-1 in. in two sizes, 40 tons 0-$\frac{5}{8}$ in., 60 tons $\frac{5}{8}$-1 in., absorbs 105 h.p., uses 140,000 gals. water in circulation.
(*A. Audry, Proc. So. Wales Inst. Eng.*, vol. xli., p. 567.)

(See 'The Rhéolaveur Coal-Washer in Belgium,' by H. Briggs and H. Louis, *Trans. Inst. Min. Eng.*, vol. lxxiv, 1928, p. 285; also 'The Operation of a Small-coal Washer,' by C. E. Pulford, *Trans. Inst. Min. Eng.*, vol. lxxviii., 1929, p. 47.

In the United States small coal is being largely washed on shaking tables of the Wilfley type; such a table will treat 5 to 8 tons per hour of coal under $\frac{1}{4}$-in. mesh, requiring about 1 h.p.

The flotation process has been applied to washing fine coal. (See *Trans. Inst. Min. Eng.*, lxvii. p. 374.) It is being extensively used in Spanish collieries.

The Chance process employs a separating cone worked with a sludge consisting of sand and water; it is said to give excellent results on Anthracite, and is used in the U.S.A. (See *Trans. Amer. Inst. Min. Met. Eng.*, vol. lxx. p. 740.)

One of the principal difficulties in coal washing is presented by the fine coal (say $\frac{1}{30}$ in.), which forms slurry when suspended in water; it consists often largely of fusain, and is therefore non-coking and is often very impure. It may be collected in settling tanks, which, however, occupy much ground space. Most makers of washing plant supply an elevated conical settling tank, from which a thickened slurry can be drawn off from the bottom, whilst tolerably clean water is drawn off from the top. A still better device is the Dorr Thickener, usually between 50 and 100 ft. in diameter, though made up to 200 ft. diameter and about 10 ft. deep; the stirrer makes 4–8 revs. per hr., and 100 sq. ft. of thickener area are allowed for each 25 gals. per min. The Rheolaveur deals efficiently with all but the finest slurry. A recent device is the Kirkless Slurry Separator, which filters the slurry. It is generally considered preferable to blow out the dust or draw it off by an aspirator before the coal goes to the washery if the coal is dry enough. The dust is collected in cyclones, bag filters, etc., and is generally very suitable for dust firing though it may need further grinding for this purpose.

Dry Cleaning.—The American Coal Cleaning Corporation's machine consists of a shaking table with riffles and a porous deck through which a current of air ascends on the principle of the Sutton-Steele table; this cleans dry coal and the finished product contains under 2 per cent. of extrinsic ash. Over 10,000,000 tons of coal were cleaned in this way in 1927 in the U.S.A. (See *Trans. Amer. Inst. Min. Met. Eng.*, vol. lxx. p. 758.)

The largest American plant is the McComas plant at Crane Creek Mines, W. Virginia, treating 225 tons per hour.

This table has been introduced into England by the Birtley Iron Company, Ltd. Its first form, known as the Birtley SJ Table, has been erected in a number of places, one of the first being at Wardley Colliery, Co. Durham. This plant treats coal under $2\frac{1}{2}$ in. and consists of six SJ tables with dust aspirator and baghouse. It cleans to within 2 per cent. of the intrinsic ash leaving about 1 per cent. of free coal in the refuse. Its original capacity was 125 tons per hour, but this has now been increased.

An improved form, known as the Y Table, was introduced about 1926, having double the capacity of the SJ Table, and not needing as close sizing. Quite recently a still newer form known as the V separator, made in two types, the standard V and the super V, has been introduced; the former has about three-fourths of the capacity of the latter, which is as follows on average coal:

2 in. to $\frac{1}{2}$ in.	60 tons per hour
$\frac{1}{2}$ „ „ $\frac{1}{8}$ in.	40 „ „
Below $\frac{1}{8}$ in.	20 „ „

The power consumption for the fan and drive of these machines averages about 0·75 h.p. per hour. The Birtley Company has constructed some eight plants in England, eight more are under construction, and the total operating cost including labour, maintenance, and repairs is given at about 2·25*d.* per ton. (See papers by K. C. Appleyard, *Trans. Inst. M.E.* vol. lxxiii, 1927, p. 404, and World Power Fuel Conference, 1928, No. C 7.)

Another table used recently to some extent in this country is the Static dry cleaner of G. Raw. It consists of a long relatively narrow table, without riffles, but with three skimmers; the bed of the table is, like the last, of perforated material, but a pulsating current instead of a continuous current is introduced into the air-chest beneath the bed. The tables have capacities of from 30 to 50 tons per hr. and the power consumption is 0·85 B.O.T. unit for a ton of coal. Costs including labour, power, maintenance, interest and depreciation, are given at $3\frac{1}{4}$*d.* per ton (see paper by G. Raw and F. F. Ridley, World Power Fuel Conference, 1928, No. C 6). Another dry coal-cleaning table is the Kirkup table, in which the table is at rest, but receives a pulsating air current beneath its perforated bed. The Peale-Davis table is like a Birtley Y table, 40 ft. long, 14 ft. wide, with a capacity up to 250 tons per hr. This is being used at a Yorkshire colliery.

In some cases dry cleaning may with advantage be combined with washing, the coal below, say, ⅝ in. going to a dry-cleaning plant and the coarser coal to a washery; the washed coal after drainage is mixed with the dry-cleaned coal and the product carries a proportion of moisture suitable for many purposes.

(See also 'The Dry Cleaning of Coal,' by H. Louis, *Fuel Economy Review*, 1928.)

Dust is now often removed from coal that is not naturally too wet before it is cleaned by washing or dry-cleaning; it may be screened off by fine vibrating screens or may be taken out by an air current either on the screens or on the dry cleaning tables or else in a dust extractor; the dust is collected by cyclones and bag-houses and is often used for dust-firing. Wedge wire is often used for the fine screens. The Waring dust collector is being used successfully. (See *Min. Mag.*, Dec. 1928.)

Vibrating screens have a tightly stretched screening surface kept in rapid vibration and set at a tolerably steep angle; the Hummer, Hayle, Overstrom, etc., screens differ mainly in the mechanism by which the vibration is produced.

For a good comparison of different methods of coal-cleaning see paper by Wheeler and Chapman in report of the Coal-Cleaning Conference at Edinburgh, July, 1927, pub. by Soc. Chem. Ind.

Mr. Norman Kemp has applied *X*-ray analysis to the products of the coal washery.
(*Trans. Inst. Min. Eng.*, vol. lxvii. p. 59.)

Magnetic Separation.

Maxima of magnetic susceptibility of minerals by volume in C.G.S.

Soft iron		400
Magnetite (pure crystallised)	Piedmont	3·12
Magnetite	Hey Tor, Devonshire	1·44
Magnetite	Altenfjord, Norway	0·27
Magnetite	Lake Champlain, U.S.A.	0·234
Magnetite (altered carbonate, impure)	Bettwys Garmon, Carnarvonshire	0·06
Red Hæmatite	?	0·00073
Red Hæmatite (crystallised)	Cumberland	0·00017
Specular Hæmatite	Nova Scotia	0·00106
" "	?	0·0005
Brown Hæmatite	?	0·00042
Brown Hæmatite (pure crystallised)	Nova Scotia	0·00011
Franklinite	New Jersey	0·0037
"	" "	0·00253
Ferrous Sulphide	Artificial	0·064
Spathic Ore	?	0·000559
Clayband	?	0·00069
Impure Carbonate Ore	Northamptonshire	0·00056
Ilmenite	India	0·00147
Monazite	Travancore	0·000069
Zircon	Ceylon	0·00000055
Pleonaste	?	0·000102
Gahnite	?	0·000054

Common non-ferrous minerals, *e.g.* mica, quartz, felspar, calcite, have susceptibilities < 0·000001. 'Strongly' magnetic minerals have susceptibilities > 0·001. 'Feebly' magnetic minerals between 0·001 and 0·0001. (*After Prof. Ernest Wilson.*)

Magnetic separators: (1) For minerals of high magnetic susceptibility:—

Dry working, *e.g.* Wenström: drum 30 ins. diam., 24 ins. face. Capacity 5–7 tons per hour; will take lumps up to 7 lbs.; 30 r.p.m. Current 15 amperes at 120 volts.

Wet working, *e.g.* Gröndal: 24 ins. wide, 24 ins. diam., 100 r.p.m. Capacity up to 3·5 tons per hour. Current 10 amperes at 110 volts.

(2) For minerals of low magnetic susceptibility:—

Dry working, *e.g.* Rowand-Wetherill. Belt 18 ins. wide, three electro-magnets taking respectively 5, 15, and 30 amperes at 110 volts. Capacity, 0·25 to 1 ton per hour.

Wet working, *e.g.* Ulrich.

THE RAPID MAGNETTING MACHINE CO., LTD., MAGNETIC SEPARATORS.

(The Rapid Magnetting Machine Co., Ltd., Birmingham.)

When choosing a magnetic separator the greatest possible care should be exercised. The test method, which is, without exception, adopted by mineral mining concerns, is to send a reasonable size *normal* sample of the material to be treated, accompanied by information likely to be useful, as, for instance, the quantity to be treated per hour, voltage of current (D.C.) available, whether to be self-contained with feed attachment, space available, etc., etc., and inquirers should insist on the magnetic products being returned for examination for cleanliness.

Specification should be compared with specification, even after material has been tested, as a machine efficient as regards the actual extraction may be of such unsuitable design as to cause a constant loss of material being treated; and, where iron is to be removed as an impurity, guarantees of *efficient* output should also be obtained.

For feebly magnetic minerals the magnetic separators have to be of a special type. For such minerals the 'Rapid' (Thompson-Davies) Patent Ore Separator (dry-working) machine has been designed; this machine is 4 pole, 3 disc, 15-in. belt; the maximum current on each pole is 2½ amps. at 110 volts capacity. The machine will deal with about 3 tons of wolfram-tin per day, monazite sands 5 tons per day, but the capacity varies according to the characteristics of the ore.

Flotation Processes.

These may be classified as follows:—

I. *Film Flotation*, utilising differences of surface tension on particles at an air-water interface—*e.g.* De Bavay, Macquisten processes.

II. *Oil Flotation*, utilising differences of surface tension in an oil-water medium where the particles are buoyed up by oil—*e.g.* First Elmore process.

III. *Adhesive Processes*, utilising differences of surface tension between particles and oil or grease, causing differential adhesion to oiled or greased surfaces—*e.g.* Murex, Cattermole, greased-plate processes.

IV. *Froth Flotation*, utilising differences of surface tension at gas-water interfaces, where the particles are buoyed up by bubbles of gas:—

(*a*) Where the bubbles are produced by chemical action—*e.g.* Potter, Delprat processes.

(*b*) Where the bubbles are produced by releasing the air dissolved in water—*e.g.* Elmore vacuum process.

(*c*) Where the bubbles are produced by mechanical means—*e.g.* Minerals Separation, Callow, Janey, K. & K. (Kraut & Kolberg), Rork, Kleinbentink, Ekof, processes.

(See 'Froth Flotation: its Commercial Application and its Influence on Modern Concentration and Smelting Practice,' Walter Broadbridge, *Trans. Inst. M. M.*, vol. xxix. p. 205, 1920. 'A Contribution to the Study of Flotation,' H. L. Sulman, *Ibid.*, p. 44. 'The Concentration of Ores by Flotation,' H. L. Sulman, *Bull. Inst. M. M.*, Aug. 1930. 'Flotation Reagents,' B. W. Holman, *Bull. Inst. M. M.*, Nov., 1930.)

COSTS OF FLOTATION MACHINES.

Tons capacity for 24 hours.	First Cost per ton of daily capacity.		Operating Costs per ton daily capacity.	
	Mechanical Agitation.	Pneumatic Agitation.	Mechanical Agitation.	Pneumatic Agitation.
25	$30–50	$25–40	$0·20–0·75	—
50	25–45	12–20	—	$0·20–0·75
100	20–40	8–12	0·35	0·35
250	15–35	6– 8	0·28	—
1,000	12–25	4– 6	0·17	0·17

(Ralston.)

Reagents used:—

Gangue modifying:—Acids, alkalies, silicates carbonates, and other salts.

Froth producing:—Soluble oils, essential oils, cresylic acid, amyl alcohol, etc.

Froth stabilising:—Insoluble oils, petroleum, etc.

According to figures collected by the United States Bureau of Mines, a total of 50,073,450 tons of ore was treated in the United States by the flotation process in 1927. By far the greater portion, 40,881,768 tons, consisted of copper ores. The remainder was made up of complex lead-zinc, lead, zinc, copper-iron, and miscellaneous ores. A total of 220,514,373 lbs. of reagents was consumed in the treatment by flotation of all classes of ores. The bulk of this consumption was lime, of which 169,926,145 lbs. were consumed. Pine oils constituted the greater portion of the frothing reagents used, accounting for 5,064,320 of a total of 6,583,151 lbs. Appreciable amounts of cresols were also used. Of the oils used as collecting reagents, coal-tar creosotes and coal tar made up 2,655,352 of a total of 3,508,993 lbs. Other oils used as collecting reagents were wood-tar creosote, crude oils, and petroleum products, blast-furnace oils, water-gas oils and tars, and miscellaneous and reconstructed oils. Ethyl xanthates and di-thio-phosphoric acids were used as collecting reagents to the extent of 3,319,639 lbs. and 1,932,996 lbs. respectively, and made up the great bulk of chemicals used for this purpose. Other chemicals used were higher xanthates (amyl and butyl), thio-carbanilide, alpha-naphthylamine, thio-ureas, and oleic acid. The acids and alkalis used in addition to the great consumption of lime were composed of sulphuric acid, hydrochloric acid, sodium carbonate, sodium bicarbonate, sodium hydroxide, barium carbonate, and cement. Other inorganic reagents used included sodium sulphide, calcium and barium sulphides, copper sulphate, cyanides, sodium sulphite, sodium silicate, zinc sulphate, sodium dichromate, tri-sodium phosphate, aluminium sulphate, sulphur, calcium chloride, and sodium chloride. Some quantities of glue and starch were used as protective organic colloids.

(*The Chemical Age*, July 6, 1929.)

Surface Tension of Liquids against Air.

	Dynes per sq. cm.			Dynes per sq. cm.
Water (at 0° C.)	75·8	Water with 0·14 per cent. cresol	(at 23° C.)	71·5
Water (,, 20° C.)	75·1			
Phenol (,, 41° C.)	37·0	Water with 0·14 per cent. amyl alcohol	(,, 23° C.)	68·4
Cresol	34·8			
Olive oil	32·0	Water with 0·134 per cent. saponine	(,, 23° C.)	48·7
Petroleum	27·7			
Turpentine	25·6	Water with eucalyptus 'solute'	(,, 20° C.)	44·6

Dilute sulphuric acid reduces the contact angle and hysteresis range of that angle far more for quartz, silicates, and oxides than it does for metallic sulphides; oiling the latter greatly increases the contact angle and the hysteresis range.

(*H. L. Sulman*, 'A Contribution to the Study of Flotation,' *Trans. Inst. Min. Met.*, vol. xxix, 1920.)

The Inspiration Consolidated Copper Company's plant at Miami, Arizona, treats 15,000 tons in 24 hours. The ore contains 1¾ per cent. of copper, mainly in finely disseminated sulphides. Flotation reagents used, 1¾ lbs. per ton of ore. Recovery, 85 per cent. of total copper, 90 per cent. of copper present as sulphide; concentrates contain 20 per cent. of copper.

(*Trans. Amer. Inst. Min. Eng.*, lv., 1916, p. 576.)

Selective flotation is very largely employed for the treatment of lead-zinc ores in the Western States and in Mexico. (See 'Selective Flotation as Applied to Canadian Ores.' C. S. Parsons, *Canad. Min. Journ.*, 1927, vol. xlviii, p. 468; also 'The Trend of Flotation,' Weinig & Palmer, *Colorado Sch. of Mines Quart.*, 1926, vol. xxi., No. 2.)

Finely ground coal may be froth-floated from shale, using 1 lb. of cresol and ½ lb. of paraffin, per ton of coaly shale; the shale contains about 65 per cent. of ash, the clean coal about 8 per cent. Petassic Zanthate is also used.

ELECTROSTATIC SEPARATION is used in some special cases—*e.g.* for concentrating rosin blende, molybdenite, and graphite (Blake-Marsden and Huff processes).

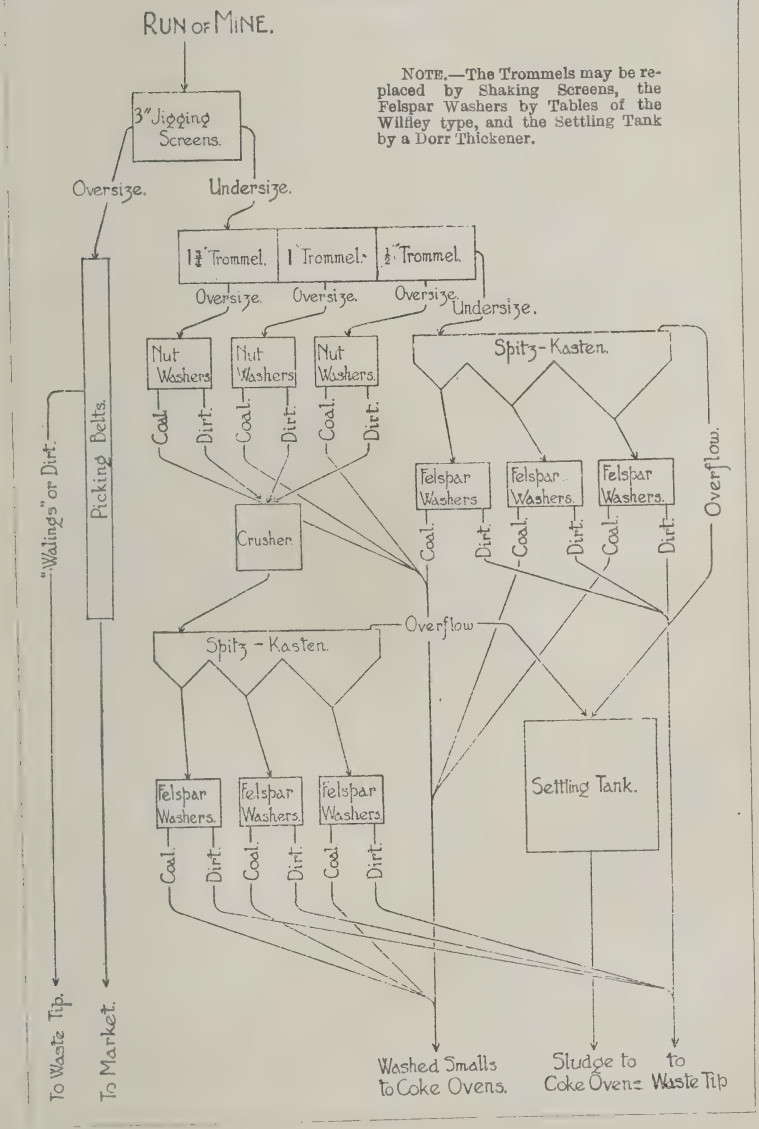
DRY SEPARATORS, using currents of air instead of water, are used under exceptional conditions.

FLOW SHEETS.

The diagrams given on pp. 2629–2637 are specimen flow sheets, illustrating the sequence of operations in certain typical cases:—

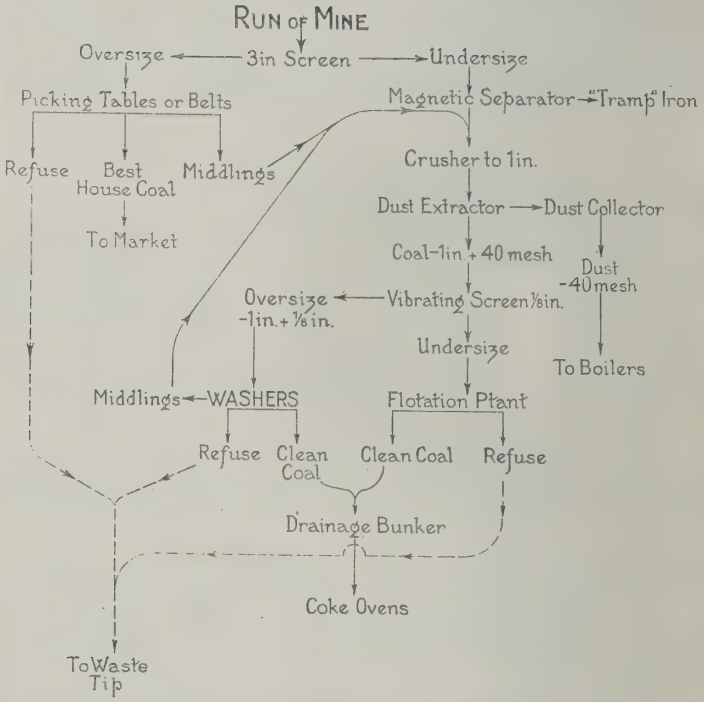
I and II. Coking coal. III. Dry cleaning of coal. IV. Low grade magnetic iron ore. V. Quartz carrying cupriferous pyrites. VI. Lake Superior copper ore. VII. Lead ore. VIII. Tinstone containing wolfram. IX. Copper ore by Froth Flotation.

FLOW SHEET I

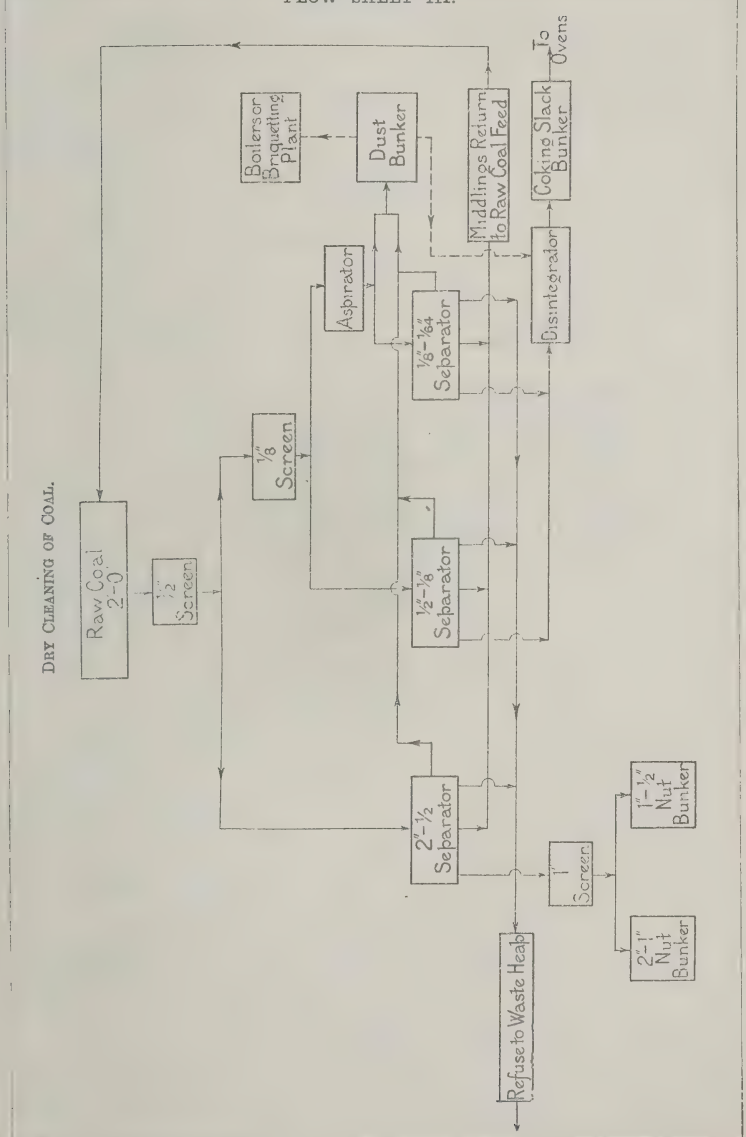

RUN OF MINE.
3" Jigging Screens.
Oversize.
Undersize.
NOTE.—The Trommels may be replaced by Shaking Screens, the Felspar Washers by Tables of the Wilfley type, and the Settling Tank by a Dorr Thickener.
1¾" Trommel.
1" Trommel.
½" Trommel.
Oversize.
Oversize.
Oversize.
Undersize.
Nut Washers.
Nut Washers.
Nut Washers.
Spitz-Kasten.
Coal.
Dirt.
Coal.
Dirt.
Coal.
Dirt.
"Walings" or Dirt.
Picking Belts.
Felspar Washers.
Felspar Washers.
Felspar Washers.
Overflow.
Crusher
Coal.
Dirt.
Coal.
Dirt.
Coal.
Dirt.
Overflow.
Spitz-Kasten.
Felspar Washers.
Felspar Washers.
Felspar Washers.
Settling Tank.
Coal.
Dirt.
Coal.
Dirt.
Coal.
Dirt.
To Waste Tip.
To Market.
Washed Smalls to Coke Ovens.
Sludge to Coke Ovens.
to Waste Tip

FLOW SHEET II.

(ALTERNATIVE TO SHEET I.)

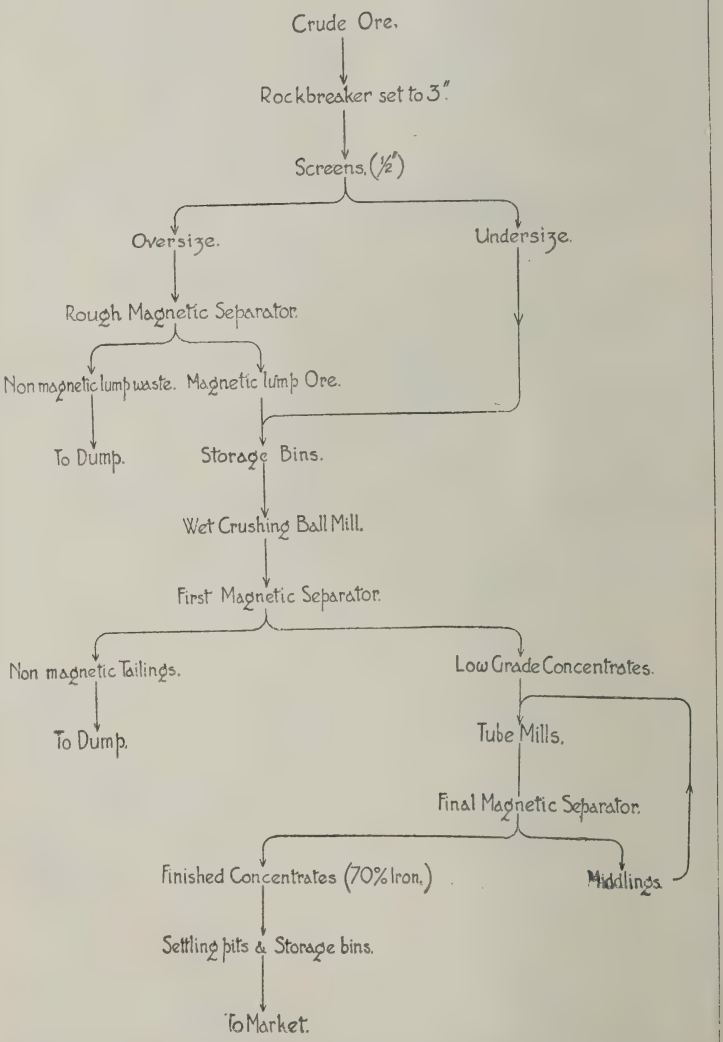


FLOW SHEET III.

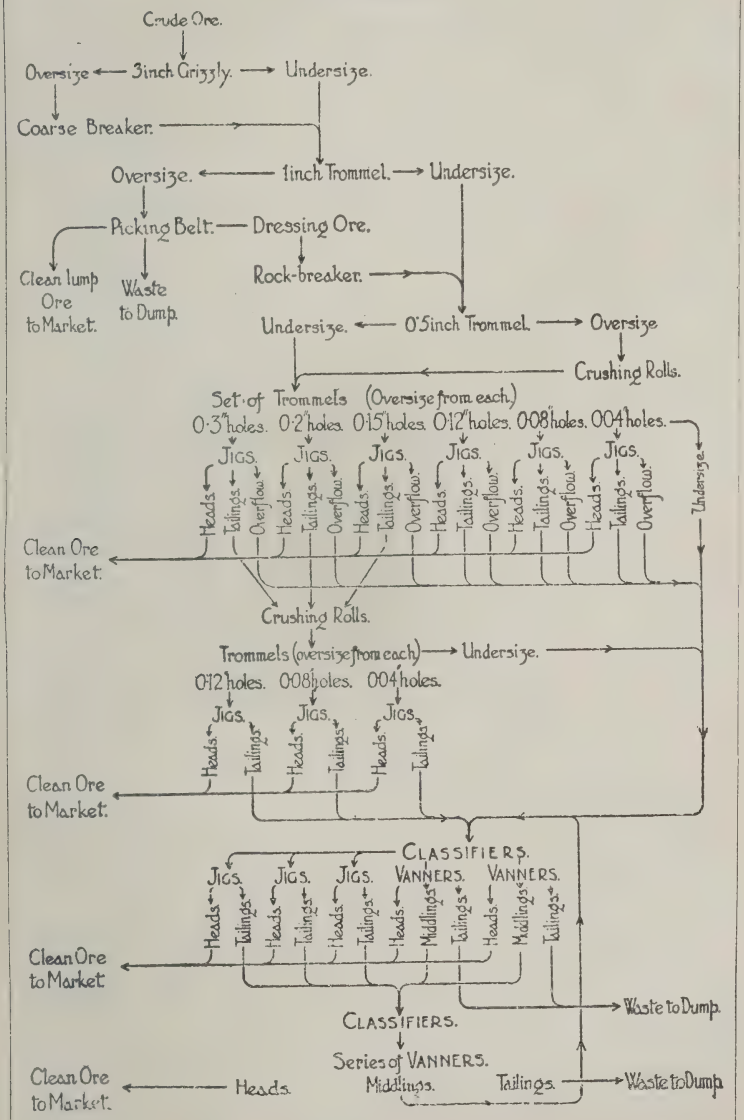


FLOW SHEET IV.

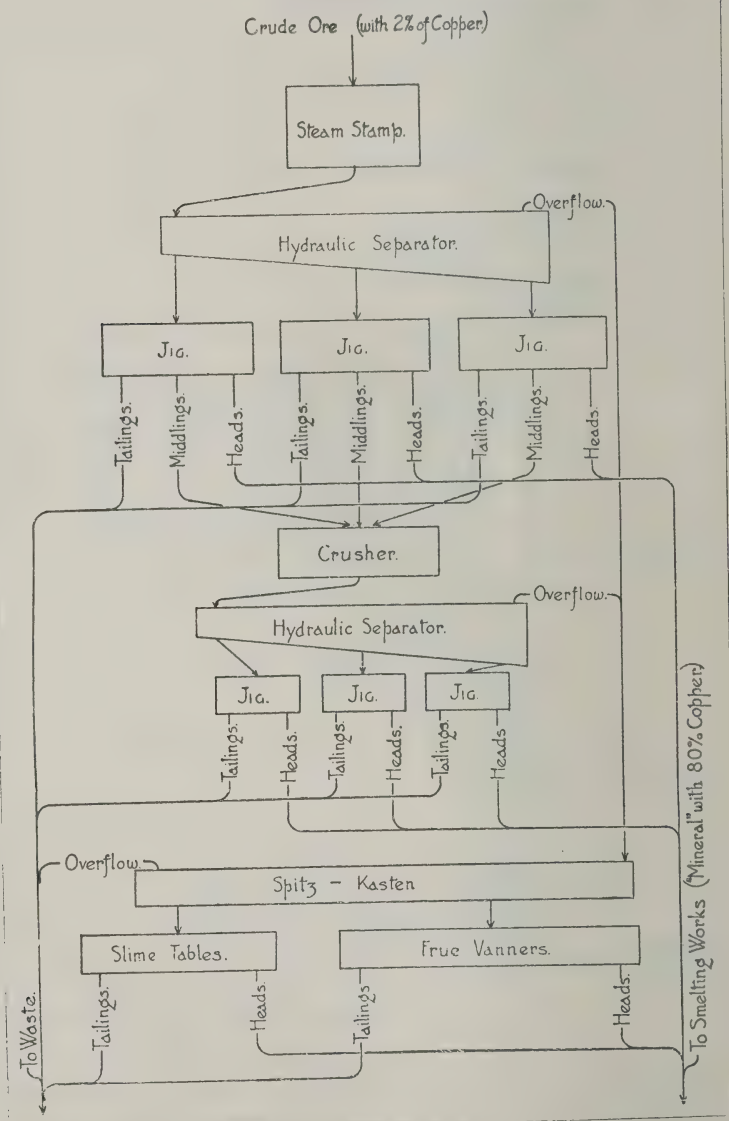
LOW GRADE MAGNETIC IRON ORE. (with about 25% IRON)



FLOW SHEET V

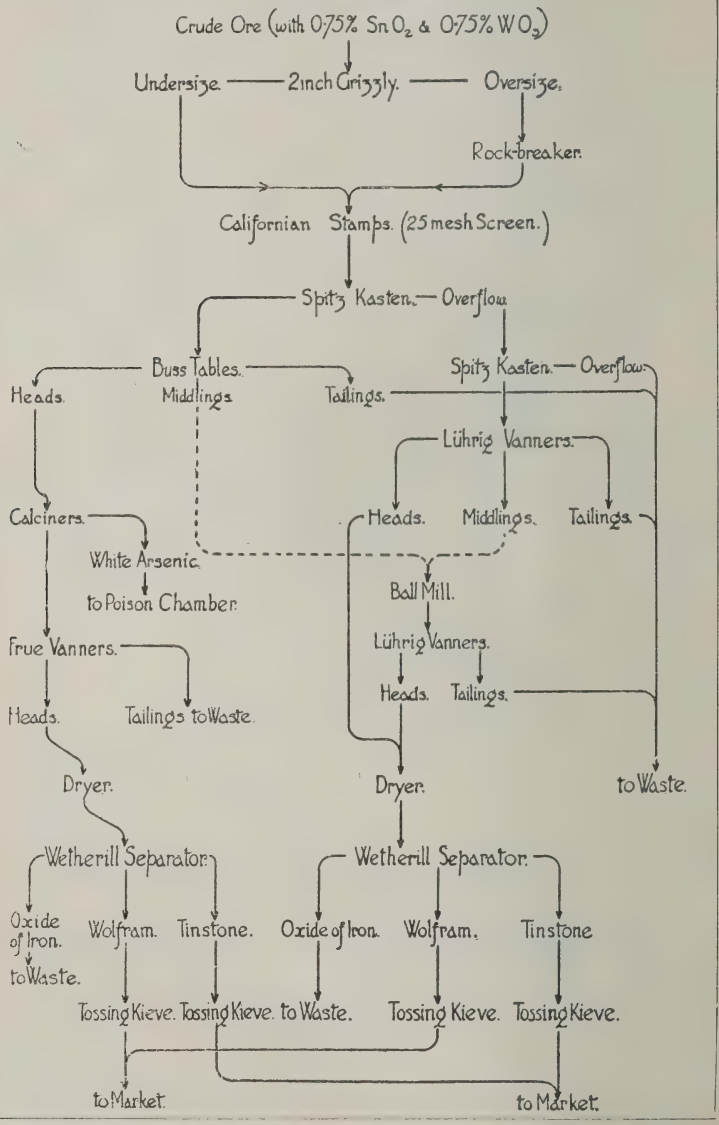

Crude Ore.
Oversize
3 inch Grizzly.
Undersize.
Coarse Breaker.
Oversize.
1 inch Trommel.
Undersize.
Picking Belt.
Dressing Ore.
Clean lump Ore to Market.
Waste to Dump.
Rock-breaker.
Undersize.
0·5 inch Trommel.
Oversize
Crushing Rolls.
Set of Trommels (Oversize from each)
0·3" holes.
0·2" holes.
0·15" holes.
0·12" holes.
0·08" holes.
0·04" holes.
Undersize
JIGS.
Heads
Tailings
Overflow
Clean Ore to Market.
Crushing Rolls.
Trommels (oversize from each)
Undersize.
0·12" holes.
0·08" holes.
0·04" holes.
Clean Ore to Market.
CLASSIFIERS.
JIGS.
VANNERS.
Middlings
Clean Ore to Market.
Waste to Dump.
CLASSIFIERS.
Series of VANNERS.
Middlings.
Tailings.
Waste to Dump.
Clean Ore to Market.
Heads.

FLOW SHEET VI.



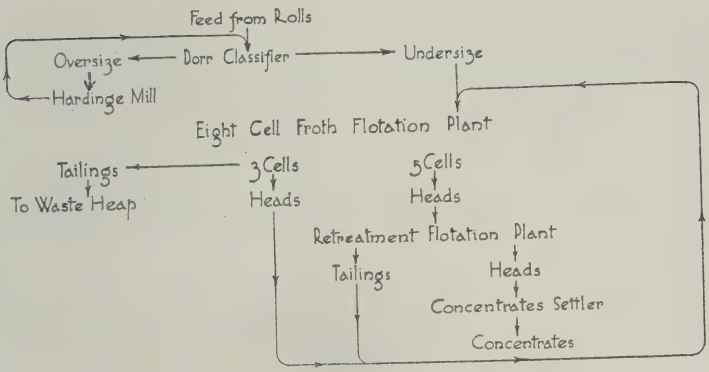
FLOW SHEET VII.

FLOW SHEET VIII.



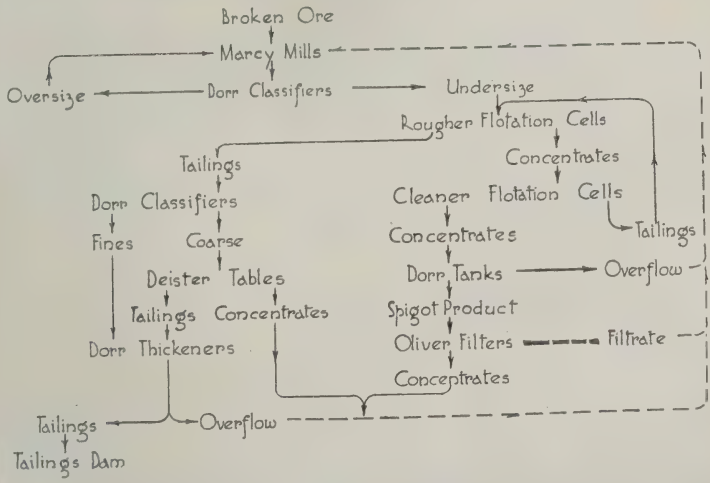
FLOW SHEET IX.

Where the Frothing Agent is added in the Flotation Boxes.



FLOW SHEET IXa.

Where the Frothing Agent is added during Grinding.



GOLD MILLING.

With the improvements in the cyanide practice and also with the exhaustion of the rich free milling surface ores of various camps, which rendered such improvements necessary, have come changes in the process of treating gold ores. Instead of the simple Californian practice of crushing by 900 lb. stamps to about 40 mesh, amalgamating either in or outside the battery, concentrating the sulphides on vanners or shaking tables, chlorinating the concentrates, and sometimes treating the tailings by cyanidation, the 'all-sliming' process has come into force.

As practised on the Rand, the ore is screened through grizzlies with bars 2¾ ins. apart, the oversize going to a sorting belt, thence to a crusher. The ore passes into a feeder, thence to stamps weighing 2,000 lbs. per head, where it is crushed to pass a 10-mesh discharge screen. The pulp is classified in cones and the coarse product delivered to tube mills where it is slimed. The product of the tube mills, together with the overflow from the classifying cones, is delivered to the amalgamating tables. From the tables it is again classified, the sands being cyanided direct, and the slimes treated by air agitation, then filter-pressed or vacuum-filtered.

The most vital changes from the old practice to the new have been: the use of the heavy gravity stamp merely as a preliminary coarse crusher for the tube mills; the adoption of the tube mills for fine grinding and the employment of the tall, air-agitation tanks and of filters.

The practice is modified in each district, the modification depending largely upon the nature of the ore. In some camps silver ores are being successfully treated by fine crushing and cyaniding.

CYANIDE PROCESS.

This process may be conducted in a number of different methods, which may be classified as follows:—

I. Coarse crushing and direct cyanidation (exceptional).

II. Fine crushing with:—

A. Simultaneous cyanidation.

B. Subsequent cyanidation without classification.

C. Subsequent cyanidation, preceded by classification into (*a*) sand, (*b*) slime.

(*a*) Sand cyanidation, generally by percolation.

(*b*) Slime cyanidation, performed by

1. Decantation.
2. Agitation (mechanical or pneumatic) and filtration.
3. Direct cyanidation in filters.

D. All sliming methods with simultaneous or subsequent filtration.

(*H. Louis*, '*Handbook of Metallurgy*,' i. p. 1031, 3rd edn., Schnabel and Louis.)

Filtration methods may be classified as under:—

I. Suction filters, divided into:—

(*a*) Filters forming a thin cake, continuous acting.
(*b*) Filters forming a thick cake, intermittent.

II. Pressure filters, divided into:—

(*c*) Ordinary filter presses.
(*d*) Sluicing filter presses.
(*e*) Filtering chambers or cylinders.

III. Centrifugal filters.

(*S. J. Young*, '*Trans. Amer. Inst. M.E.*,' xlii., 1911, p. 752.)

Precipitation is performed by zinc shavings, zinc-lead shavings, zinc dust, aluminium, electrolysis.

An average cyanide plant may be roughly estimated to cost complete $1,000 per ton treated per day in the Western States. An average cost of cyaniding, including crushing, is now about $1.50 per ton, but varies within very wide limits.

Stamps.

The duty of stamps depends upon the nature of the ore, the weight of the stamp, its speed and drop, the size of the aperture of discharge screen, and the height of the discharge. The following table gives the duty per stamp in various districts :—

DUTY OF STAMPS.

Mine or District.	Weight of Stamp.	Drop in Inches	Number of drops per Min.	Screen Aperture.	Discharge in Inches.	Tons Crushed per head per 24 Hrs.	Remarks.
California . .	900 lbs.	7	90	900 per sq. in.	4	3	Gold Quartz
Melones, Cal. .	1,000 ,,	7	104	20 mesh	—	4·41	
Tonopah Mill, U.S.A.	1,050 ,,	6½	104	slot ·04 in. wide	2½	5·09	Gold and Silver
Montana-Tonopah, U.S.A.	1,050	7	100	20 mesh	3	3·3	Gold and Silver
Tonopah Belmont, U.S.A.	1,050 ,,	6	104	24 mesh	—	3·7	,,
Goldfield, U.S.A..	1,050 ,,	7	108	12 mesh	—	6	,,
Komata Reefs Mine	1,030 ,,	6½	100	4½ holes per in.	4	5	Gold
Luipaards Vlei Rand	1,629 ,,	8½	98·6	200 holes per sq. in.	4	9·6	,,
Simmer & Jack East	1,550 ,,	8	96	200 holes per sq. in.	3½	8·3	,,
Waihi Grand Junction	1,110 ,,	7	105	5 to 10 mesh	1½	7·6	,,
Oroya Brownhill.	1,100 ,,	7½	108	10 mesh	2	6·48	,,
Ivanhoe . .	1,192 ,,	7½	104	225 mesh	2	5½	,,
Gt. Fingall . .	1,150 ,,	8	106	12 mesh	3	7	,,
Mysore Gold Mine, Mysore, India	1,150 ,,	7·5	96	1,200. 625 and 400 per sq. in.	3	3·63	,,

It is claimed that the new 2,000 lb. stamps of the Goldfield (S.A.) are capable of crushing 26 tons of ore per stamp per 24 hours by introducing an elimination screen to 'bypass' the fine material from the rock bins direct to the tube mill and using screens with aperture of ·284 inch.

Amount of water varies on the Rand, from 5 to 10 tons of battery water to one of ore, depending upon the duty. In Western Australia the amount of water required is 4 to 8 gallons of water per stamp per minute.

Power per stamp.—The 2,000 lb. stamp requires 5 h.p. per stamp. In Nevada, for 1,250 lb. stamp 3·5 h.p. per stamp is used. At the Goldfield Con. in Nevada with 140 stamps, fine grinding and cyaniding 1,500 h.p. is supplied to the mill or 1·73 h.p. per ton of ore milled.

COST OF EQUIPMENT OF GOLD MILLS.

The following is the comparative cost of gold mills, including all machinery and buildings .—

Mill.	Number of Stamps.	Cost of Machinery and Plant.	Cost per Stamp.
Rand Mill	200	£334,000	£1,670
New Kleinfontein	200	420,480	2,102
Wit Deep	200	346,210	1,731
Robinson Central Deep . .	100 and Tube Mill	234,013	2,340
Great Fingall	,,	193,745	1,937
El Oro	,,	160,000	1,600

In Nevada, the cost of complete plants doing fine grinding and slimes filtering is 1,400*l.* a stamp.

Tube Mills

are used as secondary crushers following the stamps. On the Rand they are most efficient when fed with ore that has been crushed through a screen the apertures of which are ·27 in. or say ¼ in. They are here 25 ft. in length by 5½ ft. in diameter, and one tube mill crushes the product of 10 stamps. Each tube mill requires 100 h.p. to drive it. In New Zealand tube mills are 16 ft. by 4 ft. revolving at 29 r.p.m., and driven by 25 h.p. motor. The feed is 1½ parts of water by weight to 1 part of ore. In 22 months a tube mill ground 28,500 tons of hard quartz, 50 per cent. of which passed through a 200-mesh screen.

At Pachuca 90 h.p. is used for a 5 ft. 6 ins. by 22 ft. tube mill. Output 140 tons per day through a 60-mesh screen.

At Waihi Grand Junction tube mills 19 ft. 3 ins. long by 4 ft. 8 ins. diameter require 42 h.p. Those 16 ft. 3 ins. long by 4 ft. 3 ins. diameter require 25 h.p. They consume 2·89 lbs. of flint pebbles per ton of ore crushed at stamps. They are run at 27 r.p.m.

Chilian Mills.

Chilian mills are employed either for crushing preliminary to amalgamation, in which case the gold is partly amalgamated in the mill itself, or they are used in lieu of the tube mill as fine grinders following the stamps.

For crushing ores for direct amalgamation the so-called slow speed Chilian mills are well adapted. Machines of this type are extensively employed in Russia and Siberia. They are expensive in first cost, but the power, labour, and repair costs are very low. Furthermore they are notable for their simplicity, requiring little close or skilled attention.

A slow speed Chilian mill was employed in California on hard close quartzose gold ore. It ran at 8 to 10 r.p.m. It had four runners each weighing 1,200 lbs. travelling on a steel die 3 ins. thick with 5 ins. face and 7 ft. outside diameter, 4½ tons of rock in a tank above and sustained by the axles giving additional crushing weight. Twenty tons of ore per 24 hours were crushed with a 4 in. discharge, and 88·1 per cent. extraction obtained, 0·85 per cent. of the ore passed through a 100-mesh screen. One man per shift handled the mill. Cost of ore crushed 6*s.* per ton. Gasoline power was used consuming under heaviest load 2·17 gallons of distillate per hour. (*A. McLaren.*)

The standard type of Chilian mills used in Russia is 7 ft. to 10½ ft. in diameter with 2 runners 5½ to 6⅛ ft. in diameter, and from 7 to 12 ins. in width. The speed is 11 r.p.m. The discharge is 8 to 10 ins. Feed is about 1 in. and discharge screen slotted. The water is 6·5 to 10·1 times the ore in weight and duty is 16 to 24 tons per 24 hours. The power required is 7 to 12 h.p.

Rapid-running Chilian mills were used as fine grinders after the stamps in the Goldfield Consolidated mill in Nevada on fairly soft ore. There 140 stamps were in operation and six 6-ft. Chilian mills crushing from 4-mesh to 16-mesh were installed. These were operated at 32 r.p.m. Their capacity was 75 tons per 24 hours and the power required 35 h.p. each. The cost of crushing by these mills is given at 5*d.* per ton of ore milled. (*J. W. Hutchinson.*)

At Cripple Creek a 6-ft. Chilian mill running at 33 r.p.m. reduces 4 tons of ore per hour, 60 per cent. of which will pass a 150-mesh screen. The feed passed a ⅛-in. aperture.

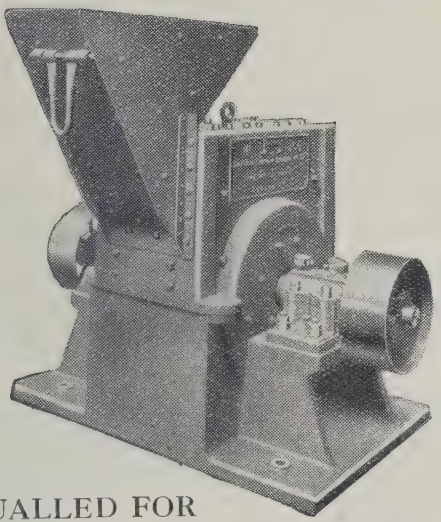
Rolls.

Where a relatively small tonnage is to be crushed coarse, as for example when coarse concentration is employed or where the ore is porous and is cyanided coarse, it is the practice to use rolls. They are cheaper than gravity-stamps in first cost, repairs, labour, and power, and are not complicated by screens.

In South Dakota the ore, a hard, close-grained Cambrian quartzite, carrying from 5 dwts. to 7 dwts. per ton, is crushed by rolls. The ore is fed to two 16- by 36-in. rolls used as cracking rolls and two others of the same size are used as finishing rolls. The final product goes through a ⅛-in. mesh direct to the cyaniding vats. The rolls require 100 h.p. The milling and cyaniding cost is 2*s.* 2*d.* per ton and 14,800 tons of ore are treated monthly.

—THE PATENT—

'LIGHTNING'
CRUSHER AND GRANULATOR.



UNEQUALLED FOR
GRANULATING MATERIALS TO $\frac{3}{4}$" DOWN.

PRODUCT REMARKABLY CUBICAL.

Also adapted for

SEMI-FINE GRINDING TO $\frac{1}{8}$" AND DOWN.

Telephone No.: Clerkenwell 6298.

Codes used: Bentley's 1st & 2nd ABC 6th Edition.

Telegrams & Cables: 'Agenciador, London.'

The Patent Lightning Crusher Co., Ltd.,

14a Rosebery Avenue - - - - LONDON, E.C.1

F. 2640

THE

"MACHINERY MARKET"

ESTABLISHED 1879

A Commercial Engineering Journal
Published every Friday
Price 3d.

USEFUL TO

Every Business Man interested in Machinery.
Every Maker wishing to sell his Products.

GUARANTEED CIRCULATION

The Circulation of the "Machinery Market" is guaranteed. The minimum is 6000 copies weekly.

WIDE RANGE OF SUBSCRIBERS

Subscribers to the "Machinery Market" represent practically every industry in which machinery is employed.

SERVICE DEPARTMENTS

Free Questions and Answers Department. An Inspection Department. Deposit Payment System (whereby buyers can deposit purchase price with this Journal until they are in receipt of the goods ordered).

SALES SUPPLEMENT

Containing the largest list of new and second-hand machinery for Sale or Wanted of any Engineering journal.

WRITE FOR OUR FREE BOOKLET

LONDON:

"MACHINERY MARKET" OFFICES

146A QUEEN VICTORIA STREET, E·C·4

Telegrams: WADHAM CENT LONDON *Telephone:* City 1643 (2 lines)

Ball Mills.

Krupp ball mills are used extensively in Kalgoorlie where 44 are at work. Size No. 5 will crush up to 45 tons of ore per day consuming 25 h.p. No. 8 will crush 90 tons per day requiring 50 to 60 h.p. About 45 per cent. of the product from a 25-mesh screen will pass 150-mesh. It is charged with 1 or 2 tons of steel balls and revolves from 21 to 25 times per minute. They work well on either wet or dry ores. The wear and tear of liners, hunch and scoop plates cost 4·64*d.* per ton milled.

Milling Costs.

The following are the milling costs of different stamp mills :—

Mill.	Stamps.	Tons Crushed Monthly.	Cost of Milling per ton.	Rate of Wages per day.	Remarks.
Tonopah, U.S.A.	100	15,000	12*s.*	18*s.*	Silver Gold ore
Goldfield Con., U.S.A.	140	29,500	9*s.*	18*s.*	Silver Gold ore
Ivanhoe	100	16,600	8*s.* 9*d.*	12*s.* to 14*s.*	Bromo-cyanide treatment of slimes
Oroya Brownhill	50	8,950	17*s.*	12*s.* to 14*s.*	Bromo-cyanide treatment of slimes
Gt. Boulder	—	14,800	12*s.* 4*d.*	12*s.* to 14*s.*	Dry crushing with Griffin mills. Roasting.
Melones, Cal.	100	12,000	1*s.* 1½*d.*	10*s.*	Cheap power, labour and supplies
New Unified (Rand)	—	10,000	1*s.* 4*d.*	—	Milling only
Simmer East	—	—	2*s.* 6*d.*	—	Milling only

On the Rand, cost of milling of those companies using tube mills varies from 4*s.* 10*d.* to 8*s.* 8*d.* per ton of ore.

In California, with free milling ores and cheap water power, milling costs average from 2*s.* to 4*s.* per ton. In Alaska, with large tonnages of soft, free milling ore treated daily and no cyaniding or fine grinding the costs are about 1*s.* per ton of ore.

Horse-Power per Ton of Ore Treated per Day.

Type of Mill.	Mesh.	Horse-power.
Stamps and vanners	20 to 40	0·75 to 1·0
Coarse concentration	10 to 20	0·5 to 0·8
Combination-stamp	16 to 30	1·5 to 1·75
Chloridising stamp (dry)	to 16	2·0 to 2·5
Chloridising stamp (wet)	to 40	4·0 to 4·5
Magnetic separator	—	0·25 to 0·5
Cyanidation (dry) roll-crushing	20 to 30	0·5 to 0·8
Cyanidation (wet), stamp crushing and sliming	to 80	0·75 to 1·5

(*J. A. Barr*, 'Testing for Metallurgical Processes.')

Standards adopted by the Institution of Mining and Metallurgy.

1 oz. of fine gold = value 85*s.*
1 ton = 2,000 lbs. (avoirdupois).
1 miner's inch = a flow of 1½ cubic ft. of water per minute.
1 sluice head = „ 60 „ „
1 gallon of water = 10 lbs. at 34°.

Temperatures are to be expressed in centigrade.

Screen apertures and diameters of wire are equal, giving 25 per cent. screening area.

GOLD AND SILVER TABLES.

It is the practice of Continental and South American engineers to report the values of ore in grammes per metric ton. English engineers generally employ the method of quoting values in grs. and dwts., in some cases per long ton of 2,240 lbs., in others per short ton of 2,000 lbs.

TABLE for Conversion of Grammes per Metric Ton into Ducts, etc., per Long and Short Ton.

Grammes per Metric Ton.	Per Short Ton.		Per Statute Ton.		Grammes per Metric Ton.	Per Short Ton.		Per Statute Ton.	
	Dwts.	Grs.	Dwts.	Grs.		Dwts.	Grs.	Dwts.	Grs.
1	0	14	0	15·7	6	3	12	3	22
2	1	4	1	7	7	4	2	4	14
3	1	18	1	23	8	4	16	5	5
4	2	8	2	15	9	5	6	5	21
5	2	22	3	6	10	5	10	6	13

TABLE for Conversion of Dwts. per Ton into Grammes per Metric Ton; the equivalent of Dwts. per Short Ton is given in the first column, per Statute Ton in the second.

Dwts.	Grammes per Metric Ton.		Dwts.	Grammes per Metric Ton.	
	Dwts.	Grs.		Dwts.	Grs.	Dwts.	Grs.
1	1	71	1	53	6	10	29	9	18
2	3	43	3	06	7	12	00	10	71
3	5	14	4	59	8	13	71	12	24
4	6	86	6	12	9	15	43	13	28
5	8	57	7	65	10	17	14	15	31

Alluvial or Placer Mining.

Deposits of auriferous or tin-bearing alluvial gravels are termed placers, alluvial drifts, or more simply alluvials. They are classified as shallow or superficial and deep-lying, sometimes called deep leads. Deep leads are accumulations of gravel from extinct or ancient rivers. They are mostly capped by igneous flows, as in California and Victoria, though sometimes they may be buried under recent gravels or drift, as at Lightning Creek, B.C., and the Lena Goldfields in Siberia.

The methods of mining used to exploit deposits of this character resemble pillar and stall in some cases and long wall in others. In California the gravel beds can be attacked by means of adits which also serve as drains for the large amount of water present. When the topography of the ground permits adits being used, the cost of mining is low, as at the Hidden Treasure, California, where the costs average 4*s.* 6*d.* per cub. yd. of gravel washed. In the Deep Leads of Victoria, where the pumping charges were unusually large (eleven million gallons per day being raised from one mine), the total cost per cub. yd. was 12*s.* to 15*s.* At the Lena Goldfields the total cost is from 20*s.* to 30*s.* per cub. yd. of gravel washed.

Superficial Deposits.—All gravel under 60 feet in thickness may be termed shallow or superficial gravels. Surface gravels are worked by the following methods:—

Panning.—A miner can pan ¼ cub. yard to 1 cub. yd. of gravel per 10-hour shift, depending on character of ground and nearness to water; 100 pans per day is a good day's work. Gravel in place is usually estimated at 3,000 lbs. per cub. yd., and the standard sheet iron pan 16 ins. diam. at top and 10¾ ins. diam. at bottom will hold 20 lbs.; 150 pans full will make a cub. yd. of dirt.

Rocker.—A man can rock ordinarily from 2 to 3 cubic yards of gravel per day, and save, in practical mining, 90 per cent. of the gold. In alluvial examinations the rocker plays a most important role, and is used in conjunction with the pan for determining the gold contents of any parcel of gravel. When used for this purpose, the design of the rocker differs slightly from that of the ordinary miner's rocker. A well-designed rocker, carefully employed, should save practically all the gold. The plans of most rockers, as published from time to time, are those of the ordinary mining rocker, which is too short, and set on too great a grade for use in saving fine gold. If much fine gold is present in the gravel, it will be advisable to use a strip of common Brussels carpet, laid along the floor of the rocker. This carpet is held in place by cleats and so can be easily removed in cleaning up.

Long Tom or short sluice can handle 3 to 10 cub. yds. of dirt per day, depending on the amount of water available and the character of the dirt.

Hand Sluicing.—Capacity of a sluice box depends upon the amount of water, grade of sluice, deposition of tailings, and character of gravel. No capacity can be given. Grade of sluice varies from 2 ins. to 21 ins. per box (12 ft.). An ordinary sluice box is made from 2-in. planks, and is 12 ins. wide by 12 ft. long.

Grade and Duty of Sluices.

Grade and Duty of Sluices.—The relation of actual width to the best ratio of quantity may not be very significant, but there is a notable increase in efficiency in passing from the smallest widths to the larger, and boxes less than 1 ft. wide are to be avoided, especially if the gravel is coarse. Treating 30- to 40-mesh sand on a 1 per cent. grade (say 1½ in. per 12 ft. box), one sluice head of water requires a channel 2 ft. wide to enable it to attain maximum duty. On a 2 per cent. grade, half a sluice head requires 2 ft. of width, the quantity of water required for maximum duty with a fixed width diminishing with increasing grade, but at what rate there are insufficient data to determine. Apparently the best width required for constant quantity of water varies directly as the grade. Coarser material requires a greater depth of water, and for 4 to 6 mesh gravel on a 2 per cent. grade, 1 ft. of width per sluice head is indicated. Variations up to 50 per cent. in the water make little difference to its duty, consequently the actual position of the optimum is not easily determined. Naturally the capacity of the sluice itself is correspondingly varied, and the mode of transportation and the saving are doubtless affected in any case.

The rate at which duty varies with slope is the resultant of so many variables that no law can be stated for all cases. In the neighbourhood of the best conditions the Utah experiments support the approximate rule that duty (water, width, and material being constant) varies as the 1·75th power of the grade. A single experiment with natural gravel on low grades by the U.S. Geol. Survey gave the 1·84th power. (*R. T. Hancock.*)

Good modern practice provides about 1 ft. of sluice-box width for every 2 cub. yds. of average material treated per hour, and 3,000 gallons of water per cub. yd. treated on a grade of 6 ins. to 12 ft.

Besides the ordinary feeding of sluice box by means of hand shovelling, many modified methods of feeding and disposing of tailings are employed. In Russia and Siberia the favourite method in use is to transport the gravel to and from the sluice box (here greatly modified from the plain open sluice box) by the employment of horses and small carts. This is a most expensive system and the yardage treated is small. Other variations are: the use of wheel barrows, tramming by cars, handling gravel by means of derricks, digging gravels by means of steam shovels, and loading into cars or direct into sluice, feeding sluice by means of scrapers operated by horses or steam power (drag-line scrapers).

The methods of preventing accumulation of tailings are to deposit them in running water or to remove them by means of drags, belt or link elevators, or hydraulic elevators.

HYDRAULIC MINING.

Hydraulicing proper is the method of mining by which jets of water under high pressure are thrown against a gravel bank, thereby undermining the gravel which falls, and is carried by the current into the sluice. The necessary pressure is obtained either by natural means, *i.e.* by conveying the water from some source having a greater elevation than the working face, or by pumping directly into the supply pipe by powerful pumps. In the former case water is conveyed to the working by means of pipes, flumes or ditches. Ditches are generally most convenient and economical for conveying water for long distances, but sometimes the nature of the ground is such (either too porous or too rocky and difficult to excavate) that a flume becomes cheaper. When lumber is abundant wooden flumes are built, but in many cases, especially in the tropics, where wood rots rapidly, sheet steel flumes are preferable. Steel flume per foot, including rivets, angle iron, etc., ready for transport, costs from 4*s.* to 11*.*; the sizes vary from a cross-section of 1·28 sq. ft. to 20·48 sq. ft. From the ditch or flume the water is fed through a strainer into a pressure box. From here, a pipe leads to the Y's, which in turn lead to the monitors or giants.

Monitors or Giants.

Monitors or Giants.—These are appliances consisting of a base for attachment to the pipe line and a long conical reducing piece, arranged with one or more joints so that it can be rotated in any direction horizontally and through a very wide angle vertically. The smaller or conical end is fitted with a nozzle butt to which different sizes of nozzles may be attached as circumstances may require. Modern giants are fitted with deflectors, attachments on the butts whereby the giant is turned or deflected by means of the force of the water itself. The high-pressure giants also have ball-bearing joints.

Nozzles from 2 ins. up to 10 ins. are used, working under heads or pressure from 100 up to 600 ft. or more.

The amount of gravel which can be washed down will depend upon its nature, upon the volume and pressure of the stream of water issuing from the nozzle, and is also dependent in no small degree upon the skill of the operator who handles the giant.

The following gives the approximate amount of gravel washed under various heads :—

Effective Head in Feet.	Size of Nozzle.	Flow in Cub. Ft. per Minute.	Approximate amount of Gravel in 24 hours in Cub. Yds.	Weight of Giant. Lbs.	Price of Double Jointed Ball Bearing Giant.
					£
100	2 ins.	104·88	160	120	28
200		148·32	230		
300		181·61	280		
400		209·82	320		
100	4 ins.	420·06	650	225	55
200		594·00	930		
300		727·56	1,130		
400		840·12	1,300		
100	6 ins.	944·58	1,460	335	66
200		1335·72	2,000		
300		1634·08	2,540		
400		1889·16	2,920		
100	8 ins.	1679·32	2,600	600	97
200		2375·46	3,600		
300		2909·58	4,500		
400		3239·70	5,300		
100	10 ins.	2624	4,080	690	107
200		3711	5,760		
300		4545	7,050		
400		5248	8,140		

(*Hendry.*)

Hydraulic Monitors and Pelton Wheel Jets.

J. J. Garrard (*Trans. I.M.M.* 1917) gives a table of discharges in cubic feet per minute for different sizes of nozzles and various heads derived from the formula : Delivery in cub. ft. per min. $= 3{\cdot}75\, d^2\sqrt{P}$, where d = diameter of nozzle opening in ins. and P = gauge pressure in lbs. per sq. in., or effective head $\times 0{\cdot}4326$. An equivalent formula given by him is $2{\cdot}468\, d^2\sqrt{h}$, where h = effective head in ft. The factor 2·468 allows for a nozzle contraction coefficient of 0·94 whereas Hendry's figures in the above table are based on the theoretical value 2·624, and make no such allowance.

Costs.

The costs per cub. yd. were at North Bloomfield 2·05*d.* per cub. yd .and at La Grange 3*d.*

In British Columbia one and a half million cub. yds. were mined and washed at a working cost of 0·97*d.*

In Northern California placer ground containing only 1·26*d.* per cub. yd. has been profitably worked. The amount of gravel washed was 1,251,399 cub. yds., yielding 6,330*l.* The profit obtained was 821*l.*

Hydraulic and other Elevators.

When a hydraulic mine possesses an insufficient slope of bed rock for moving material (the lightest grade permissible being 220 ft. to the mile) other methods of disposing of the gravel 'piped' must be employed. Where water under pressure is cheap and abundant the hydraulic elevators are employed. They are of the pipe type—either Hendry, Evans or Campbell and the Giant Elevator.

The pipe elevator operates on the principle of an injector. Water under pressure is discharged through a nozzle set within a steel jacket, creating a vacuum, and causing water and sand and gravel fed into intake to rise to a height corresponding to 10 per cent. of the head under which the water is applied.

Three sizes are made: 10-inch throat 3½–4½ inch nozzle, using 300 to 500 ins. of water; 12-inch throat 4½–5 inch nozzle, 500 to 750 ins. of water; 15-inch throat, 750 to 1,000 ins. They weigh about a ton and cost 250*l.*

In South Oregon, U.S.A., at the Weimer placer mine, two hydraulic elevators are employed to handle the gravels. One of them has a lift of 39 ft., and operating with a head of 330 ft., uses 11 cub. ft. of water per sec., besides taking all water from two No. 2 Giants. The second elevator has a lift of 9 ft., a head of 125, and uses 18 cub. ft. of water per sec. besides all the water from the first elevator. This makes a final discharge of 40 cub. ft. per sec.

Giant Elevators.

Besides undermining and cutting the gravel it has been discovered that gravel can be effectively 'driven' up a slope or incline to a raised line of sluices by means of the power of water under pressure. An elevator based on this principle is termed a 'stacker.'

In California a stacker was operated as follows: A chute was built sloping upwards at 30°. This was 4 ft. wide by 24 ft. long and constructed of 1½-inch planks. Riffles 2 ins. apart and with a drop of 2½ ins. between were laid across the bottom. These were covered by an iron grizzly. At the foot of this chute a 3-foot sump was dug in which was set up a hydraulic elevator with a 12-inch throat and worked by a 5-inch giant. This elevator, working under a 90-foot head, elevated to 16 ft. vertically the seepage, fine and water from two field-giants.

Twenty feet in front of the stacker was installed the 3-inch drive giant. Two 4-inch field-giants wash the gravel to the chute and the drive giant forces it up the stack, the fine material falling through the grizzly and down the riffles to the elevator, which raises it to the sluice-boxes.

The field-giants handled 600 cub. yds. per 24 hours, and the gravel could easily be raised to 25 ft. vertically by means of the drive giant. The cost of handling material exclusive of equipment was under 2½*d.* per cub. yd. (*S. S. Smith.*)

Ruble Elevator.

The Ruble elevator is constructed on the above principle, but it is more portable and does away with the hydraulic elevator. It is also self-contained, as the sluice-box is part of the machine.

In Idaho, U.S.A., this elevator has been tried with excellent results (*see* Porter, T.A.I.M.E., 1909, for a detailed description of this elevator). The following points are claimed to its merit: 1. It handles larger rocks with less water than other types of elevators. Handles boulders up to 1 ton in weight. 2. It dispenses with the boulder crew, either at the mine, ground, sluice-box or dump. 3. It saves the gold and the gold-saving department may be protected by lock and key. 4. The gravel is picked up in close proximity to the giants, the gold immediately extracted, and the boulders and other waste material dropped back on bedrock previously worked off, instead of being transported a long distance to the dump. It makes its own dump.

Mechanical Elevator.

In regions where water is neither cheap nor abundant, in handling gravel or tailings, resort must be made to some form of mechanical elevators. A mechanical elevator was erected some years ago on Bonanza Creek, Alaska. This was made of a frame of steel, 49 ft. high, mounted on wheels. The gravel was elevated by a bucket line of the close-connected type, 76 buckets in all, each with a capacity of 3 cub. ft. each. These discharged at a rate of 26 per min. directly into a sluice-box. The pit was drained by means of a 12-inch centrifugal pump, assisted by a hydraulic jet. The power was 200 h.p. in all, 35 for the bucket line and 165–175 for the pumps. The actual capacity per day was 1,500 yds. (*T. A. Rickard*). The first cost of this machine was high, and in actual running it proved costly in operation and repairs. Another machine of the same type, but employing a heavy dredge bucket, is being erected in Eastern Canada. No results of actual working are yet to hand.

Centrifugal Pumps.

Centrifugal Pumps as a means of elevating gravel and tailings have been extensively employed in Australia. It is claimed that with wood fuel at 7*s.* per ton of 50 cub. ft. and European rate of wages paid, to elevate tailings with pumps should not cost more than 2½*d.* per cub. yd. of solid. Thompson & Co., of Victoria, are the makers of gravel pumps suitable for this class of work. Their pumps vary in size from 3½ ins. to 16 ins. The capacity of solids per hour from 7 to 150 cub. yds. The indicated horse-power from 2½ to 583. The combined suction and delivery head from 10 feet to 100. An installation of this kind possesses the advantage of being cheaper to erect and easier to move than the belt or bucket type of elevators mentioned. However, where there is an undue amount of large stones present in the gravel, the latter type is preferable.

Pipe Lining.

Three rubber-lined pipes, each of 9 in. internal diameter and length of about 20 ft., have been in continuous operation on the discharge line from an 8-in. gravel pump at the Sungei Bari Mine, Malaya, for a period ranging from 18 months from the date of installation of the first pipe. These pipes have been subjected to a continuous flow of gravel, amounting to about 17,000 cub. yds. of solids per month, and a working pressure of about 17 lb. per sq. in. The three pipes, although differing somewhat in design, are all alike regarding the successful results of the rubber applied to them, there being little or no evidence of wear or abrasion on the rubber after this lengthy period of operation. The lining is of crepe rubber about ½ in. thick. The life of a lap-welded mild steel pipe ⅜ in. thick on the same line, with frequent turning to ensure even wear, does not amount to an average of seven months. ('*Mechanical World*,' Jan. 21, 1927.)

Dredging for Alluvial Minerals.

(Contributed by Harold E. Fletcher, A.I.N.A., M.I.Mar.E.).

The most important form of alluvial mining is dredging, for the recovery of gold, platinum, and tin. The greater part of the platinum recovered in Siberia and Russia is by dredging, and that of the tin in the Malay States. When all conditions requisite for successful working are present, when the ground has been properly proved * and a well-designed and effective dredge installed, then this type of mining is the safest and surest of all forms of mining investment.

Careful systematic boring is necessary to get true values. Dredging properties call for certain conditions, and unless these exist the success of the enterprise suffers. The conditions are :

(1) The presence of the mineral, which should be evenly distributed throughout the pay dirt.
(2) Sufficient ground to warrant the installation of at least one dredge.
(3) Soft bedrock, free from pinnacles.
(4) Absence of large boulders.
(5) Absence of permanently frozen ground.
(6) Absence of great amount of clay.
(7) Absence of heavy floods.
(8) Surface of an even contour, not too rough and broken.
(9) The ground should retain sufficient water to float dredge (50 miner's ins.), or available water supply from a river.
(10) The economic conditions, *i.e.* labour, transport, climate, vary inversely with amount of mineral in ground.

The life of a well-designed dredge may be calculated at from twelve to fifteen years, provided it is kept in good repair. Repairs for the first three years would be normal (excepting accidents), such as the replacing of bucket pins and bushes, ladder roller spindles and bushes, screen plates, tumbler bearings, and usual power plant replacements, and later on in the life of the plant tumblers and shafts, main gears, buckets, repairs to pontoon, etc.

Broad flood plains or ancient lake gravels form more suitable dredging ground than does deposit situated in narrow rocky gulches. Frozen ground cannot be dredged without preliminary thawing. Dredging of gravels lying in rapid rivers is risky, as there is always danger of losing the dredge by sudden floods. Gravels from 10 to 80 ft. below water and 20 ft. above have been successfully dredged.

The modern dredge consists of a hull or pontoon of wood or steel, upon which is mounted the digging device, a bucket ladder capable of being raised and lowered, the washing plant comprising screen, tables or launders, the pump, of the centrifugal type for water supply to screen and tables, the stacker or elevator to dispose of the tailings well astern of the dredge, and the power plant. The buckets raise the ground and discharge into the revolving perforated screen, the fines pass through the perforations on to the tables, and the stones and large material pass through the screen to the tail chute and on to the tailings stacker. The dredge can cut its own passage. The cutting depth and manœuvring lines are worked by special multi-barrelled winches, controlled by one man, the winchman.

'Suction' dredges are used at times; they are only successful in light and free ground with level bedrock. The power required has been found to exceed that of the bucket dredge of equal output. With the suction dredge it has been found difficult to control the surges of water that sweep the tables when the spoil will not draw easily. This type of dredge has not given good results on tin in the Straits of Malay.

'Bucket' dredges are classified in two types, the New Zealand and the American; the latter has been evolved from the former to suit conditions existing in California and other parts. The principle is the same, but the American type is associated with heavier construction, larger buckets of the close-connected type instead of open, spuds (adjustable piles) instead of a headline, the raised pilot (or central control house), and other modifications. Both types have their advocates and both their utility.

The New Zealand type was associated with open-connected buckets, moderate power, simple engine for driving all units and light construction, but to-day this type is built of fairly large capacity and heavier construction. In its smaller sizes for lighter ground its first cost is in its favour, also for prospecting work or proving a dredging property; also where difficulties of transport prevent the use of the heavier type of dredge.

Transport.—Dredges are generally shipped in pieces for re-erection, the steel pontoons being riveted and launched at site. The heaviest parts to transport are the main engine castings, the boiler shell, the tumbler castings (mostly steel), crown wheel castings, winch barrels, screen ends, etc. The 'Tumblers' weigh from 15 cwts. in the small to 15 tons in the large sizes; boilers stripped, from 3 to 10 tons; watertube boilers, the heaviest part being the steam drum, from 7 to 60 cwts.

Hulls.—These have to be strongly designed to stand the stresses. Wood pontoons are not entirely satisfactory and generally have to be replaced during the life of the machinery; they suffer damage by ice in cold countries, and are difficult to build to stand stresses, and much heavier than steel.

* See *Alluvial Prospecting*, by C. Raeburn and H. B. Milner. 1927.

Power.—Where wood and coal are plentiful, steam is cheapest. In Malay the Rawang coal (8,000 B.Th.U.) is used in water-tube boilers with chain grate stokers; steam pressure 160 lbs., and 100° superheat. Where oil is available at reasonable cost, it can be burnt under boilers or used in Diesel or semi-Diesel engines, and these have proved more economical than steam.

A gold dredge with 9 cu. ft. close-connected buckets is running in South America with semi-Diesel engines, also smaller dredges in Burma, etc. Full Diesel engines require the services of a specially skilled engineer. Several of the recent tin dredges have Diesel-electric generating plant on board running electric motors. Producer-gas plant and engines are used in Burma, Malay, Africa, etc., running on charcoal, driving direct or driving generators to supply current to the several motors. Where electric current is available and power can be transmitted to the dredge by cable at reasonable cost, and transformed on the dredge to the required voltage, it is in favour of low running costs, but it is generally accepted that it does not pay to build a special generating plant unless the ground will carry four or more dredges. Waterfall power to generate current is also being used.

The horse-power installed on gold and platinum dredges varies according to the capacity of bucket, dredging depth, etc. The distribution of power is generally as follows:

Bucket chain . .	33% of total power.	Screen . . .	10% of total power.
Water-pump . .	30% ,, ,,	Elevator . . .	12% ,, ,,
Winch . . .	15% ,, ,,		

On tin dredges, where larger water supply is necessary and higher water pressure, the pump power proportion is larger. On large dredges a separate winch is often fitted to deal with the bucket ladder.

Gold Dredges.

Larger types of American design are used in America, Siberia, etc., in addition to the open-connected type. Gold is saved on tables covered with cocoanut matting and expanded metal; the gold by reason of its heavy specific gravity settles in the mats, the dirt being washed down the tables overboard, the mats being taken up at periods and washed. With properly designed tables and adjustable water supply, quite fine gold can be saved. Platinum is easier to save owing to its heavier specific gravity.

Tin Dredges.

Of recent years these have increased in size considerably. The difficulty is not in raising the ground, but in treating it efficiently; the tables require to be of large area and the water supply larger than for gold dredging. Tables or 'launders' are generally from 3 to 4 ft. wide and up to 80 and 100 ft. long, wood riffles are fitted, and the wash agitated gently by hand-raking or mechanical means to allow the tin to sink through the wash and settle on the bottom of the launders. Owing to the large area of the tables the pontoons are of larger dimensions than for gold dredges; tables up to 7,000 to 8,000 sq. ft. area in two tiers are in use. Owing to the difficulty of carrying such tables, some of the latest dredges have been designed with jigs and classifiers, or a combination of riffles, launders, jigs, and classifiers, which will allow of better treatment and the saving of finer tin. The use of jigs allows of better saving adjustment. With skilled attention, up to 92 per cent. of the boring values have been obtained. Clay in the ground in both gold and tin dredging presents great difficulty to the dredge designer, and to break it up and release the values it contains call for special arrangements, such as puddlers, high water pressure, etc.

Where large boulders or heavy clay is met with a 'tray' connected bucket chain is sometimes used, the trays being castings equal to the base of the buckets and contributing in output each about 25 per cent. to 30 per cent. of the bucket capacity.

The design for tin dredges is tending towards the American type in regard to the design of the driving gear, buckets, tumblers, winches, etc. The latest dredges have close-connected buckets of cast manganese steel and nickel-chrome steel pins, cast manganese tumblers, etc.

There has been a vast amount of capital put into tin dredging in the Malay States and Siam, the number of dredging plants reaching towards 200. Some of these companies have paid 25 per cent. dividend for a number of years.

The largest English-built dredges have close-connected buckets of 12 cu. ft. capacity, a large proportion 8 and 9 cu. ft. The open-connected dredges range from 6 to 15 cu. ft. buckets.

Dredging depth ranges from 30 to 80 ft., and dredges have been built to dig 120 ft.

Output varies according to bucket capacity and the rate at which the ground can be efficiently treated. The open-connected bucket is run from 14 per minute for the smaller to 12 per minute in the larger sizes. The close-connected bucket from 28 to 18 per minute.

Maximum output is calculated:

$B \times S \times 60 \div 27 =$ cu. yds. max. per hour.

B = capacity of buckets in cu. ft.

S = number of buckets delivering per minute.

27 = number of feet per cubic yard.

Roughly, 1 cu. yd. is calculated 1½ tons.

To obtain the estimated output, one-third is deducted from the maximum, but this is not always reached; buried timber and large boulders tend to decrease output. The removal of barren overburden (to reach pay dirt) and difficulties of cleaning up bedrock lessen the profitable dredging time. A further factor is time lost in cleaning up the tables or launders to recover the gold or tin. Working time varies; many dredges have averaged over 600 hours per month.

Working costs vary with difficulties of ground, cost of fuel, and labour.

Comprised in above are European and native labour, fuel, stores, spares, camp upkeep, passage money and leave pay, management expenses, depreciation of plant, insurance.

The aggregate cost of running a small dredge does not vary greatly from running a larger one, and ground that is too low in value to give a return on a small dredge could be made to pay with a dredge of large output. With favourable conditions, dredges on easy ground have been run for as low as 1·5*d.* per cu. yd. treated. Large tin dredges in Malay 3·13*d.* to 4·0*d.* per cu. yd. treated.

TIN DREDGES.

Bucket Cap.	Type.	Monthly Output in cu. yds.	Working Hours per Month.	Dredging Depth.	Pontoon.	Steam. H.P.	Net Weight.
				Ft.	Ft. Ft.		Tons.
10	Open-con.	57,350	592	50	150 × 34	250	500
11	,,	89,000	550	50	150 × 40	320	700
12	,,	85,000	560	60	165 × 42	375	750
13	,,	78,000	—	60	165 × 45	380	820
15	,,	99,000	600	35	140 × 46	* 400	—
5½	Close-con.	89,000	651	45	150 × 45	320	900
6	,,	Est. 90,000	600	50	155 × 45	375	—
8	,,	,, 120,000	,,	60	160 × 46	400	1,200
9	,,	,, 130,000	,,	60	170 × 52	450	1,300
12	,,	,, 150,000	,,	60	180 × 52	500	1,600

* Gas engines.

The difference in output is due to difficulties in working the ground.

MALAY TIN BASIS.

1 Kati = 1⅓ lbs. 100 Katis = 1 Picul = 133⅓ lbs.
16·8 Piculs = 1 ton. Straits Dollar = 2*s.* 4*d.*

Value of tin in ground varies from 0·33 to 2·0 Katis.
Native labour costs 1*s.* 9*d.* to 2*s.* 3*d.* per day. Engineers, 4*s.* 6*d.* to 5*s.* per day.
Native coal, 24*s.* per ton. Wood, 25*s.*
Fuel oil, 90*s.* per ton. In bulk, about 82*s.*

See 'Suction-Cutter Dredging for Tin in Malaya,' G. A. More. 'Some Notes on Malayan Bucket Dredges,' E. J. Vallentine, *Bull. Inst. M. M.*, Dec., 1929.

NEW ZEALAND TYPE DREDGERS (GOLD).

Size of Buckets.	Estimated Capacity. In cu. yds. Monthly.	Digging Depth below Water-Line	Weight including Hull.	Price, f.o.b. London.	Horse Power.
Cu. Ft.		Ft.	Tons.	£	
3	30,000	20	100	5,000	35
4	40,000	20	180	9,000	—
5	50,000	26	220	13,000	56
6	60,000	40	280	16,000	—
7	74,000	32	350	20,000	180

The Californian or American dredge is costly, very heavily built, with close-connected buckets, digs with spuds, and till lately built with wooden hulls.

For properties where the transport is high, as in parts of Siberia, and where heavy repairs are difficult, a well-built New Zealand type of dredge, with ample stock of spares, is preferable. Indeed, its choice is imperative in some cases, for the high transport costs to some properties exclude the use of the ponderous American type of dredge. On the other hand, cheap transport, access to machine shops, and with heavy or stiff semi-cemented gravel, or large deep deposit of fair digging gravel (the latter permitting the handling of enormous daily yardages) are conditions favourable to the use of the American dredge.

CALIFORNIAN TYPE DREDGERS (GOLD).

Size of Buckets.	Estimated Capacity per month. In cu. yds.	Digging Depth below Water-Line.	Weight exclusive of Hull.	Price Erected.	Horse Power.
Cu. Ft.		Ft.	Tons.	£	
3	45,000– 60,000	36	350	22,000	180
5	90,000–110,000	36	520	30,000	220
7	110,000–150,000	35	576	34,000	300
8½	150,000–185,000	48	1,000	65,000	560*
13	250,000–280,000	40	1,100	90,000	415

* This includes 100 h.p. for pump for two monitors or giants.

SECTION XXXIX

PART II

MINERAL VALUATION.

(By Professor Henry Louis, M.A., D.Sc., A.R.S.M., M.Inst.C.E., etc.)

General Principles.

Under the head of Mineral Valuation are included two somewhat different valuations, namely, that of a mineral property, whether worked or not, and that of a mine as a going concern; the difference arises from the fact that the proprietor of the minerals does not as a rule work them himself, but disposes of them, usually for a royalty, to the mine-owner who works them. In cases where the mine-owner is also the proprietor of the minerals, it is generally more convenient to value the mine, the royalty derivable from the ownership of the minerals being included as a part of the profit derivable from the mining operations.

In either case it is necessary to determine the present value of a number of annuities, the latter being the annual revenue derived from the royalties or the profits, as the case may be.

The amount of the royalty per ton is generally known or can be closely approximated, being the price paid or to be paid for the mineral in its unsevered condition as it lies in the deposit. (For a full discussion of the nature of royalties see *Trans. Inst. Min. Eng.*, vol. lxxiv. 1927–8, p. 277.)

The profit per ton must be estimated from the probable cost of working and the probable value of the mineral in its marketable condition.

The tonnage to be wrought annually must be estimated, having due regard to all the conditions of the deposit, and should be taken to be the amount that can be extracted, so as to produce the maximum total profit over the period under consideration. It is obvious that a deposit worked out with excessive rapidity will yield no profit when the interest on the capital outlay required to so work it exceeds the profit derivable from working; on the other hand, if worked excessively slowly, the standing charges may exceed such profit. Between these extremes lies the rate of economic working; it will generally be found that there is a considerable range of working rate, within which the economic results are practically uniform, and the best rate within that range is usually determined by market conditions or other purely commercial considerations.

The probable total quantity of mineral that can be economically produced from the property must next be calculated, meaning thereby the amount which is as likely to be exceeded as to be fallen short of; it is obvious that the quantity that can actually be produced from any property can never be known until the property has been exhausted, when the information is naturally valueless. Hence the question becomes one of probabilities, never one of actually ascertained figures, and must always be considered as subject to the laws of probability in the mathematical sense. The quantity thus estimated divided by the estimated or proposed annual output determines the probable life of the property; if the latter is held under mineral lease, the period of which is less than the probable life, the length of the lease is substituted for the probable life in the calculations.

Sometimes not only the tonnage but also the assay value of the mineral deposit enters into the calculation, and in either case it is never a certainly known figure, but always the most probable figure, that has to be employed. Thus, where the value of a mineral deposit depends upon its thickness and its assay value, let the ascertained most probable thickness be T units and the ascertained most probable value be V units; as neither are known with certainty, these figures should be written $T \pm t$ and $V \pm v$ respectively, where t and v are the respective probable errors; then since the probable error of the product VT is $\sqrt{T^2v^2 + V^2t^2}$, the value of the deposit will be a function of $VT \pm \sqrt{T^2v^2 + V^2t^2}$. (See *Mineral Valuation*, by H. Louis, p. 128.)

In the following formulas, the annual income = annual tonnage × royalty per ton, or annual tonnage × profit per ton, as the case may be, will be designated by a.

The number of years of life of the property or of the lease will be designated by n.

The problem then becomes that of determining in the first instance the present value (P) of a series of annuities of £ a for a term of n years.

Generally the ordinary compound interest formulas apply, which show that X pounds invested at a rate of interest i per unit will amount in n years to $X(1+i)^n$, and conversely the present value of Y pounds payable at the end of n years, allowing interest at rate i, is

$$\frac{Y}{(1+i)^n}.$$

The main difficulty in all mineral valuations lies in determining the proper rate of interest to allow. The rate of interest in operations in which there is practically no risk, that is to say where the probability of the payment of capital and interest in full is practically unity, may be taken to be the rate obtainable by investment in such securities as Trustee Stocks. In all mining operations the risk is always greater, or in other words the probability of payment is less, hence a higher rate of interest is demanded to cover the risk. If i be the safe rate of interest the risk rate may be expressed by ri, where r is a number greater than unity, depending upon the conditions of the case, and which may conveniently be described as the risk factor; it may also be defined as the reciprocal of the fraction representing the probability of success, and is therefore greater the more hazardous the nature of the enterprise. In practice the risk factor ranges from 2 to 5, being lowest in well-proved bedded deposits, the uniformity of which has been determined by experience, and highest in veins or masses in which values are known to be very irregularly distributed. It is less where money is invested merely in the purchase of a mineral property than when it is invested in a mining enterprise, because royalties form a first charge on revenue and are almost invariably secured on the assets of the mining enterprise, whereas the latter can take only remainder profits, which are totally unsecured.

The determination of a proper risk factor is the principal problem that the mining engineer has to solve in valuations.

Present Value.

Let $ri = s$, so that s is the rate of interest *per unit* that is required in any given case.

Some writers call i the 'accumulative' and s the 'remunerative' rate of interest; the term 'risk rate' will be adhered to here in order to emphasise that the high rate is required in compensation for the risk that the investor runs. Of course i is governed by the money market, and is the rate of interest paid on investments in. Government stocks and similar first-class securities; it will be noted that s varies directly with i.

The value of the mineral property or mine *at the moment of commencing profitable operations* is therefore P, the present value of n annuities of £ a. This should obviously be equal to the present value of sums of £ a payable at the end of each of the first, second, third, etc., years up to the end of the nth year, and this sum

$$=\frac{a}{s}\left(1-\frac{1}{(1+s)^n}\right).$$

It is sometimes useful to remember that, if A is the present value of £ a due at the end of n years,

$$P=\frac{a-A}{s}.$$

The expression for present value given above is often spoken of as Inwood's formula. Calculations are easily made by this formula.

There is yet another method of looking at the problem. The annuity exceeds the simple interest on the present value by an amount equal to $a - Ps$ in each year; if this difference be allowed to accumulate at rate s for n years it will amount to

$$\frac{a}{s}\cdot\frac{(1+s)^n-1}{(1+s)^n}=P,$$

that is to say, to the original capital, as it obviously should do, because the purchaser of an annuity, spending each year only the interest on the capital invested, should get back his capital at the end of the period, because this operation would be equivalent to investing the capital at the given rate of interest during the given term of years.

This last method of looking at annuities has given rise to another method of calculating them, known from its originator as Hoskold's method. He considers that the risk rate of interest s should be allowed upon the capital, but that the difference between the annuity and the interest $(a-Ps)$ should be looked upon as accumulating at a safe rate i because it is not necessarily invested in the risky mining operation. This view gives rise to the formula

$$P=\frac{a}{\frac{i}{(1+i)^n-1}+s}.$$

This value may be taken out from any ordinary tables as follows :—

$$\frac{(1+i)^n-1}{i}$$

is the amount of £1 at interest i for n years, and the reciprocal

$$\frac{i}{(1+i)^n-1}$$

is the sinking fund for the repayment of £1, *i.e.* the sum to be set aside annually and allowed to accumulate at interest i in order to amount to £1 in n years. These values are obtained from tables. The denominator of the fraction in Hoskold's formula can be taken out by adding the risk rate to the sinking fund for £1 at interest i for n years.

This method is often used, but is not one that fairly answers all the tests of a true annuity formula. Nevertheless, it is used by valuers on the grounds that the difference between the annuity and the simple interest cannot, in fact, be invested at the risk rate s, but must be invested at the safe rate i, and that this method, therefore, more correctly represents what would obtain in actual practice.

Hoskold's method depends upon considering only one out of the various methods in which the true annuity formula may be looked at; obviously the two can be made to give the same result by selecting suitable risk factors in each case. Thus if r is the risk factor in Inwood's formula, and r' in Hoskold's method, the present values will be the same whichever formula be used, when

$$r' = \frac{r(1+ri)^n}{(1+ri)^n-1} - \frac{1}{(1+i)^n-1}.$$

It follows that r' must be greater than r; in other words Hoskold's method allows a rate of interest higher than the stipulated risk rate, and hence puts a lower present value on a given annuity. It need hardly be said that both his method and the ordinary method provide equally for the replacement of the original capital, if suitable sums are set aside in each case out of the annuities received.

Hoskold's is the generally accepted method of dealing with mineral valuation; it is, however, obvious that year by year a portion of the original purchase price (or present value) is withdrawn from the speculative investment and set aside to form the sinking fund for its repayment, whilst the risk rate of interest is still being paid upon the whole of the original purchase price. It might, however, fairly be stipulated that this risk rate is only to be paid upon such portion of the original capital as has not yet been redeemed by the sinking fund, which accumulates as before at the rate i per unit, whilst the portion of the capital thus withdrawn from the risky investment, though not yet really returned to the inventor is credited with interest at the rate j per unit (of course i and j may in some cases be the same). On the basis of these stipulations, the following formula results :—

$$P = a\,\frac{(1+s-j)^n-(1+i)^n}{s-(j+i)+s[(1+s-j)^n-(1+i)^n]}$$

(*H. Louis*, '*Trans. Inst. Min. Eng*,' lxx., 1925, p. 368.)

Since an investor in mines usually stipulates that he shall receive the risk rate of interest on his investment throughout the whole life of the mine, the Hoskold formula, though possibly less equitable, is generally applicable.

Increasing and Decreasing Values.

The above methods of valuing mineral properties tacitly assume that the mine will forthwith commence to produce at its full rate of output and will continue to do so until the mineral is worked out; this is, of course, not the case in practice, a mine only reaching its full output after several years, and this gradually declining again as the deposit approaches exhaustion. There would be no difficulty in allowing for this in the valuation, but as a rule it is neglected; as has already been pointed out, the figures on which the valuation is based are only probable, not actual figures, so that there is no use in endeavouring to attain minute accuracy in these computations.

There is obviously no difficulty in introducing these refinements if it is considered desirable; the computation is made as above for the estimated period of full production deferred during the period of production at a lower rate than the full output; the income expected in each year of output less than full output is calculated and its present value is determined, and all these present values are added together to obtain the total present value of the property.

The author has recently pointed out that this formula demands a risk rate also on the portion of the capital no longer at risk, and suggests the following :—

$$P = a\,\frac{(1+s-i)^n-(1+i)^n}{s-2i+s[(1+s-i)^n-(1+i)^n]}.$$

(See *Trans. Inst. M.E.*, *loc. cit.*)

If preferred, calculations can also be made directly, whether the annuity increase or decrease in arithmetical or in geometrical ratio. Let c be the income in the first year and let it increase or decrease by b each year, so that the income for the second year is $c \pm b$, for the third year $c \pm 2b$, etc., during n years. Then the present value of this annuity on the Inwood method is :—

$$c\left(\frac{(1+s)^n-1}{(1+s)^n s}\right) \pm \frac{b}{s}\left(\frac{(1+s)^{n-1}-1}{(1+s)^{n-1}s}\right) \mp \left(\frac{(n-1)b}{s} \cdot \frac{1}{(1+s)^n}\right).$$

These values can be taken out from the tables : the first expression is, as before, the present value of £c for n years at rate s ; the second is the present value of £$\frac{b}{s}$ for $(n-1)$ years at rate s, and the third is the principal which will amount to £$\frac{(n-1)b}{s}$ in n years at rate s.

If Hoskold's method be used, the present value is :—

$$\frac{c+\frac{b}{i}}{\frac{i}{(1+i)^n-1}+s} \mp \frac{nb}{i-s+s(1+i)^n}.$$

The first part of this expression is the present value of an annuity of £$\left(c+\frac{b}{i}\right)$ calculated for n years by Hoskold's method ; in the second expression $s(1+i)^n$ is the amount of £s at interest i for n years.

Again, let c be the income for the first year and let it increase geometrically by a ratio k, so that the income for the second year is ck, for the third ck^2 . . . etc., during n years. Then the present value of the annuity on Inwood's method is

$$\frac{c}{(1+i)^n} \cdot \frac{k^n-(1+i)}{k-(1+i)}, \text{ or, if } \frac{k}{1+i} = f, \text{ it is } \frac{c}{1+i} \cdot \frac{f^n-1}{f-1}.$$

On Hoskold's method the present value becomes :

$$c \cdot \frac{(1+i)^n-k}{1+i-k} \cdot \frac{i}{i+s[(1+i)^n-1]}.$$

All the above calculations are based upon the hypothesis that the annuity is paid at the end of each year ; in practice royalties are mostly paid quarterly, and mining dividends at least half-yearly. To meet this point, for n substitute $2n$, and for s substitute $\frac{s}{2}$ when payments are made half-yearly, these becoming $4n$ and $\frac{s}{4}$ respectively when payments are made quarterly.

Dead-rent.

In computing the present value of mineral royalties 'certain rent' or 'dead-rent' has sometimes to be taken into account. Mineral leases usually stipulate for a definite dead-rent, which may be in addition to the variable royalty, or may be a merging dead-rent, that is to say that the amount of royalty to be paid is determined by the tonnage which is in excess of that which corresponds to the dead-rent at the agreed royalty rate. As the dead-rent is payable in full, whatever the results of the mining operations, its present value must be calculated not on the mining risk rate of interest, but on a rate obtained by multiplying the standard safe rate by a factor depending only on the financial stability of the lessee. This consideration may be neglected when the dead-rent is very small compared with the expected annual royalty, but must be taken into account when it is considerable, the present value of the whole royalty, in the case of an additive dead-rent, or of the excess of royalty, in the case of a merging dead-rent, being computed as above and added to the present value of the dead-rent.

Relinquishment.

When, as is often the case, the lessee has power to relinquish the lease after a given number of years or at stated intervals, there is a risk that the lease may be cancelled because the mine proves unprofitable or because the deposit is exhausted, and the rate of interest applicable to the dead-rent during the definitely fixed period is estimated as above, but it must be multiplied by a risk factor representing the probability of such relinquishment in computing the present value of the dead-rent during the successive periods. Here, again, the difficulty of the problem consists entirely in determining the proper probability factor.

Deferment.

In opening a mine there is always a period of development during which no mineral is produced, or at any rate none is produced profitably, whilst the main capital expenditure in the way of shafts, adits, plant, machinery, etc., is incurred. The present value at the commencement of the profit-producing period as found above must therefore be looked upon as deferred for the requisite number of years and the actual value determined. If V be the actual value at the commencement of mining operations and d the number of years occupied in development,

$$V = \frac{P}{(1+s)^d}.$$

The risk rate of interest must be taken, because the capital is being spent and is subject to the full risks of the mining operation.

The annuity formula and deferred value formula may be combined, thus :—

$$V = \frac{a}{s} \cdot \frac{(1+s)^n - 1}{(1+s)^{(n+d)}}.$$

It is, however, better as a rule to make the computation in the two stages, as above.

Mr. O'Donahue has proposed a method of calculating deferments that has some analogy with Hoskold's method of computing the present values of annuities, being also a two-rate method. He considers that simple interest at the risk rate s should be allowed on the actual value V, whilst the interest assumed to fall due accumulates at the safe rate i, whence

$$V = \frac{P}{1 + \frac{s}{i}\left[(1+i)^d - 1\right]}.$$

Since interest is never in fact paid nor payable on the actual value, this method cannot be recommended. This formula and the ordinary deferment formula may be connected as follows :— Let r be the risk factor in the ordinary method and r' the risk factor when O'Donahue's method of deferment is employed ; then

$$r' = \frac{(1+ir)^d - 1}{(1+i)^d - 1} \text{ and } r = \frac{\sqrt[d]{1 + r'[(1+i)^d - 1]} - 1}{i}.$$

In some cases, however, where two different rates are involved, another method must be adopted. In the most general case, it is required to determine the present value, V, of a mine that will take d years to develop, during which time no profits are earned, and which will then yield a profit of a per annum during n years, it being a condition that the original capital shall bear interest at the risk rate s, but that all accumulations can only be invested at the ordinary investment rate i. At the end of the n years the investor is entitled to receive back his capital plus the accumulations of interest (Vs per annum) at the rate i or

$$V + Vs\,\frac{(1+i)^{n+d} - 1}{i}.$$

This must be equal to the amount of an annuity of a per annum for n years, or

$$\frac{a}{i}\left[(1+i)^n - 1\right].$$

Equating :—

$$V = \frac{a\left[(1+i)^n - 1\right]}{i + s\left[(1+i)^{n+d} - 1\right]}$$

This is the same result as follows from the method of calculating deferred annuities, published by Mr. George King in his 'Theory of Finance.' Further, if $d = 0$, the above formula becomes the ordinary Hoskold formula given above.

Capital Expenditure.

From the present value, V, as determined by any of the above methods, the capital necessary to bring the mine into full work must be deducted in order to obtain the value of the mineral property ; if the capital expenditure is spread over several years, the present value of this deferred expenditure must be computed at the ordinary investment rate i, and the sum thus found deducted from V.

DEFERRED DEVELOPMENT.

It occasionally happens that the valuation is required of a mineral property which it is not intended to commence to develop for a number of years; in this case the present value must be calculated as the present value of a sum V, the payment of which is deferred for the given period at a rate of interest *i* per unit. The risk rate of interest must not be taken into account, because there is no mining and therefore no mining risk. If the value of V has been properly determined, making all due allowance for mining risks, the problem resolves itself into determining what capital sum must be invested in safe securities in order to produce the sum V at the time when mining operations are to be commenced. Special attention is drawn to this point, because mistakes are often made in confusing the rate of interest required during the development period and the rate required during the period preceding development, the latter depending on ordinary actuarial considerations and not on the risks incidental to mining.

Investment Value.

Finally, in comparing investments in mines or in mineral properties with other investments of capital, the effect of Income Tax as well as of Mineral Rights Duties must be taken into account. Mines and minerals both constitute wasting assets, but income tax is charged upon the gross revenue derived therefrom, hence not only on the true revenue, but also on that portion of the gross revenue that ought to be looked upon as forming a sinking fund set aside to replace the original capital at the end of operations; mineral rights duties specifically affect only mineral properties and thus constitute an additional charge upon these. In both these respects, therefore, these investments are at a disadvantage compared with ordinary investments, a fact that requires due consideration in computing mineral valuations as a basis for the investment of capital.

ADDENDUM.

Income Tax.

It should be borne in mind in making annuity calculations that capital can only accumulate at the gilt-edged rate *i* less income tax. Thus the accumulative rate when the gilt-edged rate is 4½ % and income tax is 4*s.* 6*d.* in the £ is 3.4875 %. When there are no tables in existence corresponding to such rates, all calculations should be made by logarithms.

Cardox.

Cardox.—This is an explosive of quite a different type; it consists of a steel shell filled with liquid carbonic acid, which is converted into gas by the action of the heat of a mercury fuse, thus generating great pressure. The steel shell is blown out of the shot-hole and can be re-charged. (*Trans. Inst. Min. Eng.*, vol. cxxxi, 1931, p. 9.)

SECTION XL

PART I

METALLURGY (pp. 2659–2758)

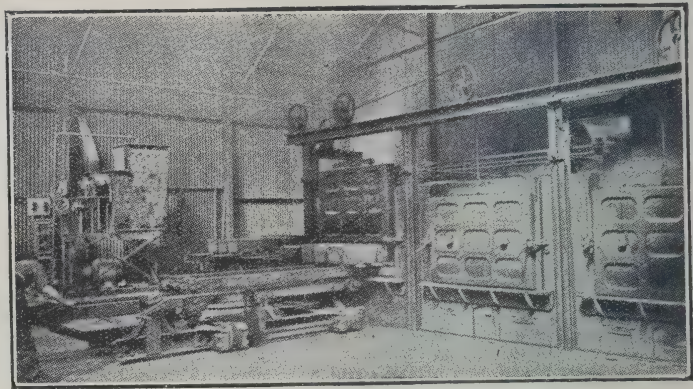
By Professor Owen W. Ellis, M.Sc. (B'ham.), Assoc.M.Inst.C.E.

(With contributions by Ernest W. Birch, A.M.I.A.E., F. J. Cook, L. C. Harvey, Colin Livingston, Professor Dempster Smith, M.B.E., M.Sc., M.I.Mech.E., and others.)

PART II

WELDING AND CUTTING (pp. 2759–2793)

By D. Richardson, Wh.Exh., A.M.I.Mech.E., Capt. R.A.F.



Producer Gas-Fired Three-Chamber Furnace with Charging Machine, Quenching Tank and Oil Cooler.

FURNACES

for	*to operate on*
Forging	Coal
Billet Heating	Coke
Reheating	Oil
Sheet Mills	Producer Gas
Tube Works	Town's Gas
Wire Mills	Electric
Vitreous Enamelling	etc. etc.
All phases of Heat Treatment	
etc. etc.	

Also Gas Producers, Charging Machines, Oil Coolers, and other Auxiliary Plant. Uniform Heating, Low Fuel Consumption, Increased Production, Improved quality of Product.

The INCANDESCENT HEAT CO. LTD.

SMETHWICK, BIRMINGHAM.

Telegrams: 'Repeat, Birmingham.' Telephone: Smethwick 276.

SECTION XL

PART I

METALLURGY

By Professor Owen W. Ellis, M.Sc. (B'ham.), Assoc.M.Inst.C.E.

(With contributions by Ernest W. Birch, A.M.I.A.E., F. J. Cook, L. C. Harvey, Colin Livingston, Professor Dempster Smith, M.B.E., M.Sc., M.I.Mech.E., and others.)

FURNACES.

CLASSIFICATION OF FURNACES.

FURNACES IN WHICH THE CHARGE AND THE FUEL ARE INTERMINGLED.

1. Hearth Furnaces.

(*a*) With blast. Forge for smithing. Grate for zinc-white manufacture.

(*b*) Without blast. Heap and stall for coking and roasting. Trough for liquation and drossing.

2. Shaft Furnaces.

(*a*) With forced draught. Blast furnaces for smelting ores of iron, copper, lead, etc. Cupolas for iron-foundry practice.

(*b*) With induced draught. Only rarely employed.

(*c*) With natural draught. Furnaces for smelting readily fusible ores. Kilns for calcining, roasting, etc.

FURNACES IN WHICH THE CHARGE AND THE FUEL ARE SEPARATE.

1. Reverberatory Furnaces (Radiation).

(*a*) With forced draught. Furnaces for the refining of metals and the smelting of ores.

(*b*) With natural draught. Furnaces for the roasting of ores. Furnaces for the manufacture of wrought iron, etc.

2. Crucible, Retort, Muffle, etc., Furnaces (Conduction).

(*a*) With solid, liquid or gaseous fuel.

(*b*) With electricity.

FURNACES IN WHICH THE CHARGE SUPPLIES ITS OWN FUEL.

(*a*) Preheated furnaces with solid charge. Kilns for ore-roasting. Beehive coke ovens.

(*b*) Preheated furnaces with liquid charge. Converters for refining, etc.

(*c*) Non-preheated furnaces with solid fuel. Blast-roasting furnaces with up and down draught.

ELECTRIC FURNACES HEATED BY RESISTANCE EFFECTS

(*a*) The charge is included in the electric circuit. Héroult aluminium furnace. Kjellin induction furnace.

(*b*) The charge is heated by means of an electrically-heated conductor.

ELECTRIC FURNACES HEATED BY ARC EFFECTS.

(*a*) Direct heating. The charge forms one or both poles of an electric arc.

(*b*) Indirect heating. The charge is enclosed in a space heated by an electric arc.

ELECTRIC FURNACES HEATED BY COMBINED RESISTANCE AND ARC EFFECTS.

Girod, Greaves-Etchells, Héroult, etc., steel furnaces.

SHAFT FURNACES.

BLAST FURNACES.

(Contributed by Colin Livingston.)

A blast furnace should be designed to suit the materials used in smelting; successful designs can best be accomplished by keeping in view the best practice with particular materials. No general formula for the design of all blast furnaces, for a given output, is possible. The following points are common to all successful blast furnace plants, and should be kept in view, in new construction:—

The aim of all blast-furnace work is the production of a good marketable iron at the lowest cost. Low costs rest chiefly on low fuel consumption; low fuel consumption is governed by the quantity of fuel, dryness and temperature of blast, with correct furnace lines for the ores smelted.

Blast temperature is entirely one of stove capacity; it is essential that stove capacity should be on the generous side.

Output will depend on capacity of furnace, diameter of hearth, and will vary according to materials used, quantity and pressure of blast, dry or wet blast, temperature of blast; blast pressure should be able easily to penetrate to centre of furnace.

Bosh line angles are found to vary very little; short steep bosh lines are a decided advantage with most materials. Leading American blast-furnacemen attribute some of their remarkable results, in part, to low steep boshes and large hearths.

Furnaces with hearths 20′ 9″ in diameter are in successful operation.

In general practice, furnace shafts should have a batter of at least 0·625 in. in 12 ins.; where fine ores are smelted this could be considerably increased with advantage. Diameter of bell should be 4 ft. to 4 ft. 6 ins. smaller than diameter of throat.

The furnace should be entirely enclosed in a strong steel shell, from the hearth level up.

All parts of furnace top work should be securely fixed to main shell structure.

In designing blast-furnace details the main considerations should be (1) efficiency, (2) strength, (3) simplicity.

The number of gas offtakes should be such that the gas leaves furnace as slowly as possible, so as to avoid carrying dust over to flues.

The height of furnace varies from 80 to 100 feet; with the materials used, each district has come to fix the height found most suitable for its particular conditions.

Sufficient relief area, by bleeders or explosion doors, should be provided on top of furnace to prevent injury from slips; the relief should, if possible, be directly upwards.

For hearth well and boshing nothing but the best quality of fire-brick should be used. The bricks should not be too large, so that they may be thoroughly burned.

For the upper lining a fire-brick with more wear resisting qualities can be used.

Care should be taken that all bricks when laid are properly bedded and tight built, with fire-clay joints as thin as possible.

In relining a furnace, the bear in old hearth need not be removed, but simply levelled to start building well on.

This will save much hard work and expense—experience will prove that nothing better for a hearth can be put in.

Some means should be taken to preserve the upper lines of furnace from wear where stock from bell strikes brickwork; a satisfactory way of doing this is to insert with brickwork alternate layers of cast iron or steel plates. In normal working, the stock level should be kept just high enough to allow bell to swing freely when full open.

The number of tuyeres is not so important (if sufficient area for output is provided) as equal distribution and penetrative power of blast.

There should always be a reserve of blowing power available after providing for normal maximum requirements.

Where blast-furnace gas is fully utilised and gas economy a necessity, the blowing should be done by gas engines. Where there is a surplus of gas the turbo-blower can be installed with advantage, and will prove an ideal blowing engine. Reciprocating steam-blowing engines are still being installed, but only in cases where conditions make it unsuitable or inconvenient to discard existing plant.

Cooling.—Apart from tuyeres, jumbos, and hearthcooling, any cooling appliances provided for the bosh or upper lining should be carefully considered; external or spray-cooling has the great advantage that leakage of water into the furnace is prevented, or rather more easily detected. The trouble caused by water leakage from blocks inserted in the building is considerable, and can never be fully known; it is often when the damage is done the leak is located. The latest practice is to spray-cool the bosh and sometimes the entire outside shell; by this means a much thinner brickwork can be used, the furnace preserves its original lines longer, and the lining has a longer life.

Stoves.—Should be as simple in design and equipment as possible, with ample facilities for cleaning and inspection, and have a good chimney draught.

Using unwashed gas, chequers should not have less than 9-in. by 9-in. openings; with washed gas smaller chequer openings can be used, giving greatly increased heating surface for same size of shell, and higher blast temperatures and lower chimney temperatures for same gas consumption. Openings 3½ ins. × 3½ ins. with 2-in. thick walls have proved satisfactory in practice.

In latest practice for heating stoves the gas and air is forced by special fan into combustion chamber, the stove burns more gas, which gives higher and more regular temperatures, with a saving up to 25 per cent. in stove construction for a given production.

Preheating the blast before entering the stove proper is in successful operation with increased average temperatures.

It is decided economy to clean blast-furnace gas, as better results are obtained.

It is no advantage to go beyond 1,550° F. in stove heats, any fuel economy being more than off-set by stove repairs.

Better results with certain ores will sometimes be obtained by running on lower blast heats; this may seem to conflict with theory, but in blast-furnace work it is often the case that sound theory follows successful practice.

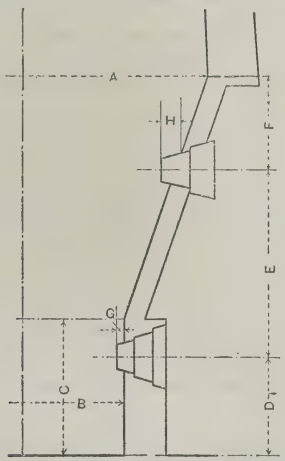


FIG. 1—Bosh Tuyere.

Uniform ore mixtures and quality of fuel and flux should be kept in view, rather than frequent changes of ore mixture, with varying qualities of fuel and flux. It is an advantage to break hard dense ores, especially magnetites.

The approximate saving in lbs. of coke per ton of iron by increase in blast temperature is as follows:—

1,000°	to	1,100° F.,	50 lbs.
1,100°	„	1,200° „	40 „
1,200°	„	1,300° „	30 „
1,300°	„	1,400° „	20 „

Dry Blast.—Average results with dry blast give a saving of 10 per cent. in fuel consumption with 10 per cent. increase in output, more regular grade of metal, and more regular working conditions.

A blast furnace in its working and results responds readily to improper treatment; proper regulation of flux has much to do with this, and the formation of a correct slag is all important.

The slag should not be too basic or limey; this causes the furnace to become stocky and gob up. If the slag is too acid the furnace will drive harder, but trouble is sure to result with grade and quality of pig.

When the lime content of slag is high it will prove profitable to make such slag into cement; the slag form from furnaces making Bessemer iron will generally be found suitable for this purpose. Most blast-furnace slags can be easily made into first-class building bricks.

The slightest indication of irregular furnace working should be promptly attended to and put right; irregular conditions will probably show first in change of slag or irregular settling of stock. The appearance and degree of fluidity of the slag is the furnaceman's best indication of working conditions.

Irregular working at tuyeres is generally caused by water leakage from jumbos and tuyere blocks. It is quite possible for tuyeres to be shut by water leakage, in spite of very careful watching.

No tuyere in a regular working furnace will close while the blast is on except by water leakage.

Irregular working at tuyeres is sometimes caused by an alteration in burden; in this case every tuyere will give trouble, which will continue till the trouble has been put right.

The best way to open tuyeres that may have been hard closed by slips, irregular working, or water leakage, is by the application of oxygen; the apparatus used is simple and cheap: the worst stuck tuyere can be opened in a few minutes.

Bosh tuyeres, it is claimed, increase the output of a furnace 20 per cent. with a slight increase in fuel consumption and more regular working. Fig. 1 and the 'Table of Bosh Tuyeres' give sizes and position of bosh tuyeres from actual practice.

TABLE OF BOSH TUYERES.

(*See Fig.* 1.)

Furnace.	A	B	C	D	E	F	G	H	Tuyeres. Bosh No.	Tuyeres. Bosh Dia.	Tuyeres. Main No.	Tuyeres. Main Dia.
	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	in.	in.		in.		in.
1	20 0	12 6	8 0	5 6	9 0	6 0	6	4	5	4	8	5
1 (*a*)	18 6	11 6	7 6	5 3	5 5	9 4	6	4	6	4	8	5
2	21 0	13 0	—	6 6	10 7	2 6	9	—	6	3	12	5
3	19 10	11 0	—	5 6	6 6	12 9	9	5	5	4 to 3½	9	66
3 (*a*)	19 10	11 0	—	5 6	7 6	11 9	3	5			9	6
4	20 0	12 6	—	7 10	10 7½	10 10½	9	6	4	6	8	5
5	19 0	11 0	—	5 3	9 5	3 10	6	4	5	4	8	5½
5 (*a*)	21 9	12 6	—	7 0	10 9½	3 11½	6	4	5	4	10	6
6	19 8	11 0	7 3	5 3	9 4	5 10	6	2½	4	2½	8	5
6 (*a*)	19 8	11 0	7 3	5 3	11 0	4 2	6	3	4	4	8	5

(*J. S. Hollings.*)

FLUE DUST.

The treatment of flue dust has developed into a distinct branch of blast furnace work; the true solution of the problem is to keep the dust in the furnace, thereby increasing production and reducing costs. Flue dust is caused chiefly by (1) fine ores; (2) soft fuel; (3) hard driving; briquetting or sintering gets over the first, but furnace design will need to be adapted to surmount the second and third.

For a comprehensive schedule of blast-furnace data see ENGINEER'S YEAR-BOOK, 1927, pp. 2580–2591.

CUPOLAS.

BLAST-QUANTITY AND PRESSURE IN CUPOLA WORKING.

An average of 120—in terms of quantity of air in pounds, multiplied by $\sqrt{}$ pressure in ounces is required to melt one ton of ordinary phosphoric iron.

The maximum output of metal melted per hour is more nearly comparative to diameters of cupolas than areas of same.

The maximum output of a cupola will be obtained when the pressure and quantity of the blast in ounces and pounds per minute respectively are so adjusted that $\frac{W\sqrt{P}}{D} = 330$, D being the diameter of the melting zone in feet. Any further increase in the product $W\sqrt{P}$ results in excessive oxidation of the metal in the cupola.

Iron melted in a cupola is affected in hardness directly as the pressure of blast used, so that the total area of the tuyeres employed should be regulated according to the class of iron being melted, small areas being used when greater hardness is required.

The height of the cupola from the top tuyeres to the charging-door sill should *not be less* than two and a half to three times the diameter across the melting zone.

The coke consumption in a well-designed cupola need not be more than 0·10 lb. per pound of ordinary phosphoric iron melted, when the coke contains from 90 to 92 per cent. of carbon.

Centrifugal Fans for Cupolas and Smith's Fires.

To give the high pressure required to supply air to Cupolas and Smith's Fires the fans must be of the 'centrifugal' type and suitably designed. They usually have smaller inlets and outlets than those used for ventilating, as pressure, not volume, is the essential feature.

Care is required in the choice of a suitable fan, as the efficiency varies within wide limits with the design.

Centrifugal Fans for Cupolas, Smith's Fires, etc.

No. of Fan.	For Cupolas.				For Smith's Fires.		
	Dia. of Cupola.	Tons per Hour.	Speed r.p.m.	Power B.H.P.	No. of Tuyeres.	Speed r.p.m.	Power B.H.P.
1	—	—	—	—	3	3,825	$\frac{3}{4}$
2	18	1	4,100	$2\frac{3}{4}$	6	3,050	$1\frac{1}{2}$
3	18	$1\frac{1}{2}$	3,150	4	10	2,440	$2\frac{1}{2}$
4	24	$2\frac{1}{2}$	2,950	7	15	2,280	$4\frac{1}{2}$
5	36	$3\frac{1}{2}$	2,600	10	20	2,030	$6\frac{1}{2}$
6	42	$4\frac{1}{2}$	2,230	14	28	1,810	10
7	48	6	2,050	20	37	1,510	13
8	54	8	1,800	26	50	1,400	18
9	60	10	1,600	35	60	1,250	24
10	66	13	1,450	42	80	1,080	32

The duct to the cupola or fires should be made at least as large as the fan outlet; in long lengths it should be made larger, so as to reduce friction, which increases the power to no purpose.

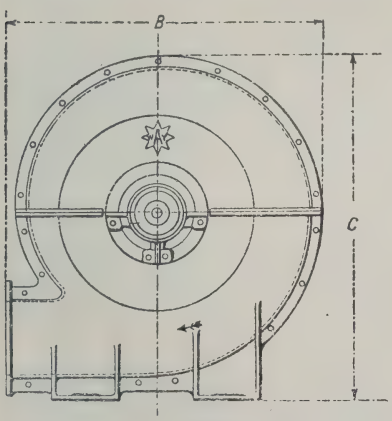


Fig. 2.

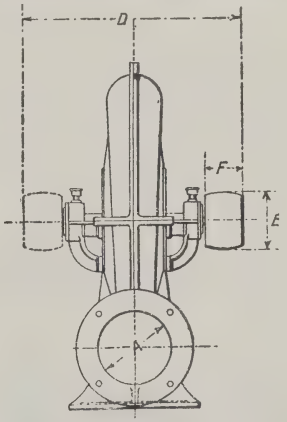


Fig. 3.

If the ducts are below ground cast iron is the best material; sheet-steel ducts $\frac{1}{8}$ in. thick, with flanged joints, are also very suitable, especially if carried above ground. If earthenware pipes are used great care must be taken to lay them straight, with good, sound, and permanent joints.

Approximate Dimensions in Inches.

No.	A.	B.	C.	D.	E.	F.
1	4	17	19	14	3	2
2	5	21	24	17	$3\frac{1}{2}$	3
3	6	26	29	20	$4\frac{1}{2}$	$3\frac{1}{2}$
4	7	30	33	22	5	4
5	8	34	37	24	6	4
6	9	39	43	27	7	$4\frac{1}{2}$
7	10	44	49	30	8	5
8	12	50	55	34	9	5
9	14	58	63	38	10	$5\frac{1}{2}$
10	16	67	73	44	12	6

(*Oswald Stott.*)

Velocities and Volumes of Air at Various Pressures

Pressure.		Velocity of outflow. Corresponding ft. per sec.	Volume through $1\frac{1}{4}$ in. tuyere = 1·2 sq. in. cu. ft. per min.	Volume through 10 sq. ins. of Orifice cu. ft. per min.	Air h.p. with 10 sq. ins. Orifice.
Ins. Water Gauge.	Ozs. per sq. in.				
$\frac{1}{2}$	0·3	46·4	23·2	193	0·015
$\frac{3}{4}$	0·45	56·9	28·4	237	0·028
1	0·6	65·6	32·8	273	0·043
$1\frac{1}{4}$	0·9	80·4	40·2	335	0·079
2	1·2	92·8	46·5	387	0·122
3	1·8	113·4	57	473	0·223
4	2·4	130·8	65·5	545	0·343
5	3	145·8	73	608	0·478
6	3·5	160	80	667	0·630
8	4·7	183·9	92	766	0·965
10	5·9	205·2	103	855	1·35
12	7	224·5	112	935	1·77
14	8·2	241·6	121	1,007	2·22
16	9·4	258·2	129	1,075	2·71
20	11·8	287·0	143	1,195	3·77
24	14	313·0	157	1,304	4·93
28	16·5	336·8	169	1,404	6·18
32	19	357·9	180	1,492	7·51

DETAILS OF DISCHARGE NOZZLES, ETC.

Size No.	Diam. Discharge Nozzle.	Overall Sizes. Length.	Breadth.	Height.	Melting Capacity Tons per hour.	Cupola Diam.	Speed of Fan.	Pressure inches Water.	B.H.P.
	Ins.	Ins.	Ins.	Ins.		Ins.			
0	4	17½	14	18	½– ¾	15	4,800	10	1½
1	4⅞	20	16½	21	¾– 1¼	16	4,700	12	2½
2	5¾	23	18	24	1¼– 1¾	18	4,100	12	3¾
3	6¼	27	21	28	1¾– 2½	20	3,700	14	5¼
4	7⅛	31½	26	33	2½– 3½	26	3,200	14	7½
5	8⅞	37½	29½	39½	3½– 4½	29	2,900	16	12
6	10¼	43	34	45½	4½– 6	35	2,750	16	15
7	12	50¼	41	52	7 – 9½	41	2,300	18	27
8	13⅞	58	46	59	10 –14	53	2,100	18	40
9	16	67	51½	69½	13 –17	70	1,800	20	50
10	18½	76½	61	80½	17 –23	80	1,600	20	66

(*Sturtevant.*)

FORGE AIR PRESSURE.

The air pressure for a forge commonly varies from 2 to 4 ozs. per square inch, with an average of about 3 ozs. A pressure of 4 ozs. at the blower has been recommended when the average number of fires does not exceed ten, and a pressure of 5 ozs. when the number is more than twelve. Small forges with the blower close to them are adequately supplied with 1½ ozs. pressure. If the blower is some distance away and a long discharge pipe with many bends leads to the forge, even though the latter be small, it may be necessary to carry 3 ozs. pressure or more, to overcome the friction in the air ducts. The volume of free air required by the average forge fire is about 140 cubic feet per minute. The exhaust fan for a blacksmith shop should have a capacity approximately four times that of the forge blower, and should operate under a pressure of about ¾ oz.

REVERBERATORY FURNACES.

In a reverberatory furnace the material to be heated is contained in a long horizontal or slightly sloping hearth, the heat being supplied by flames and hot gases which travel over it. A considerable amount of heat is radiated on to the hearth from the arched roof of the furnace. The proportions of the various parts of the furnace are important, as if the roof is too high the gases will not 'lick' the hearth, but will travel close to the roof. Care must be taken to have a sufficient volume of gases in the furnace, as otherwise the upper part of the hearth will be too cool. There is a strong tendency to open the exit damper too wide. When in constant use, a reverberatory furnace heated by producer gas is more economical than one heated by solid fuel.

KILNS.

A kiln is a furnace which is filled almost completely with the articles to be heated. Usually the flames and hot gases from the fuel circulate among the goods and raise them to the required temperature by conduction. When contact with the kiln gases would spoil the goods, they may be enclosed in fireclay cases or a *muffle furnace* may be used.

MUFFLE FURNACES.

A muffle furnace is one in which the articles or materials to be heated are protected from direct contact with flames and hot gases from the fuel, but the term is sometimes used loosely for any kind of furnace heated by town's gas instead of solid fuel. The latter kind of furnace is extensively used for annealing, normalising, case-hardening, tempering, forging, and other heat treatments of iron, steel, and the non-ferrous metals.

The following are figures from actual practice :—

Operation.	Load.	Duration.	Cub. ft. ton.
Annealing H.S. steel . . .	3½ tons	8 hrs.	3,000
,, steel	3½ tons	15½ hrs.	6,000
,, C.I. parts packed in filings in boxes	2,000 lbs.	3 hrs.	3,360

CRUCIBLE MELTING FURNACES.

(L. C. Harvey.)

'Lift-out Crucible' or Pit Type Furnaces.—These are supplied where requirements of molten metal are small, or in cases where it is preferable to melt in relatively small capacity crucibles. *Natural Draught Coke Furnaces* can be either connected up to one main flue and chimney or can be provided with smaller collecting flues, each flue having a chimney to serve a small number of fires, or again, each fire can have its own small stack.

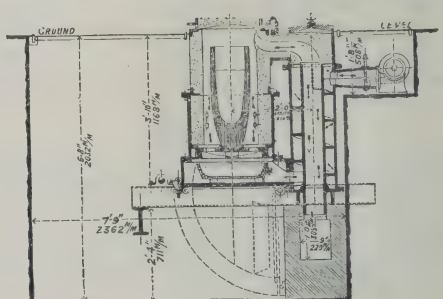


FIG. 4.—Furnace for Coke and Forced Draught. Sectional view, showing ground level.

Forced Draught Coke Furnaces (fig. 4) can be similarly arranged as regards provision for exhausting the waste gases, and as they are not dependent upon chimney pull the heights of stacks are considerably less than those required for natural draught furnaces.

It is usual to supply air from a blower or fan of sufficient capacity to feed any number up to the whole of the plant. Forced draught offers the advantages of (*a*) independence of atmospheric conditions, and, therefore, absence of variation in chimney pull; (*b*) quicker heating qualities of furnaces; (*c*) a greater degree of heat intensity than is possible under all conditions in natural draught furnaces.

Oil-fired Furnaces.—These can be fitted with burners to work with compressed air at 20-30 lbs. per square inch, depending to a large extent upon the temperature desired, the higher pressure effecting a more perfect atomisation of heavy grades of fuel, or with low pressure burners which run with a pressure of only ¾ lb. per square inch for all temperatures up to that necessary for melting mild steel and nickel.

The former require an air compressor, which is a relatively expensive machine, and may develop 5 to 10 h.p. per burner, whereas the latter needs only the ordinary open high-speed fan, and the burners will take 1½ to 2½ h.p. according to size of the furnace, etc. Almost any grade of commercial fuel oil can be used with either type of burner, and this is fed to the burners by gravity.

Producer Gas can be applied to lift-out crucible furnaces if these are made in the multiple form, viz. several crucibles in one common melting chamber, or several crucibles in separate chambers, but exhausting into one air or gas heating regenerator.

For ordinary low-power producer gas, 135 to 175 B. Th.U. per cubic foot, is essential to preheat the air or gas supply to about 300° C. for efficient melting of copper alloys; for iron and higher temperature melting, both the gas and air should be supplied at about this temperature.

When using high-power gas, such as water gas of roughly 280 B.Th.U. value per cubic foot no preheating is required for melting temperature up to about 1,400° C.

Tilting Furnaces *for Coke Fuel* are always of the forced draught pattern, for one of their chief features, quick heating, would be greatly impaired if worked on natural draught. This type of furnace is more efficient than the pit type, and it has the important advantage of melting in greater bulk than can be handled in a lift-out crucible—up to ½ ton of metal at one charge—and the further advantage of making it possible to mechanically tilt the furnace on a fixed front trunnion, so that the metal can be poured direct into ladles or moulds. In the portable pattern the body portion holding the crucible can be transported by means of an overhead crane to the point at which the molten metal is wanted.

Furnaces for Oil Fuel.—The remarks made above in connection with pit furnaces apply also to the tilting pattern. The economy in oil when melting, say, ½ ton of metal in one crucible with one burner, will be very marked, compared with, say, five burners for five 200-lb. size pit furnaces, and the consequent saving in power for air blowers in equal proportion.

Producer Gas can be applied to tilting furnaces as for pit furnaces, but tilting furnaces being of greater capacity than the latter it becomes possible to run single furnaces with producer gas, the heat in the waste gas being utilised to warm the incoming air or gas supply. For copper alloys it is usual only to preheat the air.

ELECTRIC FURNACES.

The classification of electric furnaces has already been dealt with (p. 2659). It remains to treat in detail of these furnaces and their application industrially.

Electric Furnaces Heated by Resistance Effects.

Those furnaces of this type in which the charge is included in the electric circuit (fig. 5) are of indirect interest only to the engineer. They are employed largely in the manufacture of aluminium, carborundum, calcium carbide, etc.

Those furnaces of this type in which the charge is heated by means of an electrically heated conductor find extensive employment in laboratories, and are coming into more general use for the annealing and thermal treatment of steel and the melting and heat treatment of non-ferrous alloys.

The smaller furnaces, such as are used in laboratories, are often made after the style of gas-heated muffle furnaces, but in place of gas jets, wire of platinum or incorrodible alloy, heated by the passage of an electric current, is wound round the body of the chamber. Another type consists of a tube of carbon, cut into a spiral or helix, through which the current flows.

Discussing furnaces of the larger type, T. F. Bally states that a 150-kW. annealing furnace heating 1,000 lbs. of steel to 900° C. in one hour consumes 300 kW.-hours per ton. With regard to melting in a shop working 24 hours per day, the following costs were recorded :—

	s.	d.
Electricity, 400 kW.-hours per ton at ½*d.* . . .	16	8
Metal loss, 1 per cent. zinc at 5*d.* per lb. . . .	8	4
Labour, 1 man per 600 lbs. per hour at ¼*d.* per hour .	5	0
Renewals and repairs	4	2
Total per ton of brass melted	£1 14	2

These furnaces are provided with separate heating elements of carbonaceous material, which do not come into contact with the charge, but are made incandescent by the current passing through them, and thus heat the interior of the furnace chamber.

Induction furnaces (figs. 6 and 7), which may be classed under this head, find employment in the manufacture of steel. They are able to hold their own against other types of steel-melting furnaces on account of their steady current consumption and low power consumption in the melting of cold stock, but suffer from certain disadvantages of a well-defined character. The heat being generated in the metal and only conducted to the slag, renders it impossible to work a fluid basic slag, with the result that desulphurisation of the charge is not so efficiently carried out as in the case of furnaces of the resistance or arc-resistance type. Nor can dephosphorisation be so perfectly attained, since the design of these furnaces militates against ready and complete removal of the oxidising slag needed for the removal of phosphorus. A further disadvantage arises as a result of the continuous circulation of the molten metal in the bath which leads to the erosion of the lining. This, in addition to the corrosive effect of the slag, leads to high figures for lining repairs.

Were the induction furnace of more flexible character and efficient design, it should find wide application in the melting of brass containing as high as 50 per cent. zinc. At present this type of furnace is being experimented with in the direction of non-ferrous alloy melting.

Electric Furnaces heated by Arc Effects.

Only furnaces in which indirect heating of the charge (fig. 8) is carried out are of interest in this connection. The Stassano furnace is the most striking example of this type. The power consumption of arc furnaces is high, and fluctuations in the value of this factor are, when the furnace is being employed in the fusion of cold stock, of high magnitude.

In the melting of copper alloys the indirect arc furnace may be employed economically in large units. It has also been used for melting brass containing not more than 10 per cent. zinc.

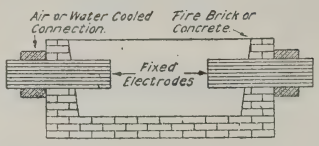


FIG. 5.

Cross Section of Resistance Furnace. Current passes from one electrode through the charge to the other electrode.

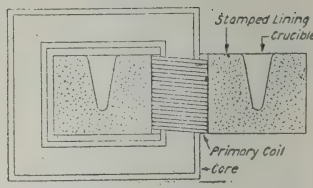


FIG. 6.

Cross Section of Single Ring Induction Furnace.

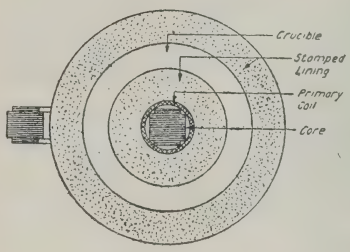


FIG. 7.

Cross Section in Horizontal Plane of Single Ring Induction Furnace.

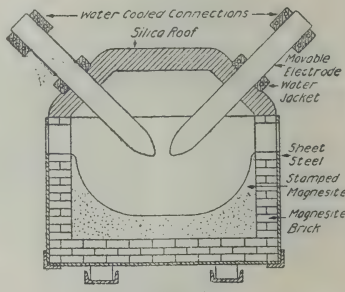


FIG. 8.

Cross Section Indirect Heating Arc Furnace. Current arcs from one electrode to the other above charge.

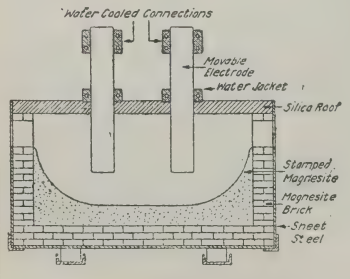


FIG. 9.

Cross Section Direct Heating Arc Furnace. Current arcs from one electrode to charge and from charge to other electrode.

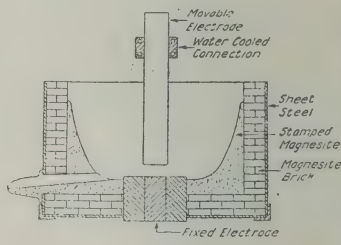


FIG. 10.

Cross Section of Smothered Arc Furnace. Current arcs from the electrode to the charge in which the electrode is plunged and the current passes through the charge to the lower fixed electrode.

Electric Furnaces heated by Combined Resistance and Arc Effects.

By far the largest number of electric furnaces in use industrially are of this type. In this class of furnace an arc is struck between the extremities of vertical electrodes and the liquid or solid furnace charge (fig. 9). In many furnaces of this type negative electrodes are built into the furnace bottom or sides so that current is forced to flow through the bath (fig. 10). In both cases the bath is heated by (i) the heat of the arc and (ii) the heat produced by the resistance of the steel and slag to the passage of electric current.

The electrodes, which are of graphite or amorphous carbon, are capable of movement in a direction parallel to their axis and are controlled, either by hand or by electrical device, so as to retain the arc at as constant a length as may be desired either for melting, refining, heating, etc., of the metal. In certain cases the practice of completely immersing the electrode in the molten charge—thus utilising the resistance effect of the electric current only—is in vogue.

For making steel the arc resistance type of furnace is that chiefly used, the charge generally consisting of scrap steel. The composition of electric steel can be very accurately controlled, and the proportions of sulphur and phosphorus in the final product can be reduced to mere traces. These and similar furnaces are largely used for making small castings in steel, malleable iron, and cast iron, their capacities ranging from 1 to 20 tons ; 6-ton furnaces are very common. Other well-known furnaces of this class are the Girod, Nathusius, Keller, Gronwall, Electro-metals, and Greaves-Erchells. While steel melting and refining, and the production of steel and iron castings, occupy the vast majority of electric furnaces, all the foregoing types can be, and have been, used for melting brass, copper, etc., though it is questionable whether their use for such purposes can be considered as expedient.

Arc-resistance furnaces for steel melting are most frequently lined with basic refractory materials. Acid-lined furnaces are, however, employed, chiefly to melt down high-grade raw materials. The necessity for utilising such materials, for which high prices must be paid, renders the mere melting process a costly one. Further, as the treatment of molten steel on an acid hearth leads to an appreciable gain in those elements, sulphur and phosphorus, most to be avoided, the process is still further limited in its application.

The process employed for the production of steel in basic-lined arc-resistance furnaces is determined by (i) the stock on hand and (ii) the product desired.

Should the stock on hand be of fair quality, *i.e.* containing contents of sulphur and phosphorus not above 0·06 per cent., the process may consist of (i) the application to the charge of a basic oxidising slag and (ii) the conversion of this slag, by means of suitable additions of coal dust and lime, into a reducing slag, which is allowed to remain in contact with the molten steel during the operations of teeming from the furnace and of casting from the ladle. Such additions of ferro-alloys as are required to bring the steel up to the standard required are made when the slag is as completely reducing in character as is possible under the conditions imposed by practice.

If the stock on hand is of poor quality—*i.e.* scrap containing high (from 0·06 to 0·20 per cent.) sulphur and phosphorus—or if it is desired to specially refine stock of fair quality, the first slag—the basic oxidising slag—is removed from the surface of the steel in the furnace. The removal of this slag implies the removal of the major portion of the phosphorus from the steel, and unless the removal be very complete a return of phosphorus to the steel may be expected, the content of phosphorus returning to the steel under these conditions being roughly proportional to the amount of slag allowed to remain upon the charge. Occasionally it may be necessary to apply a further oxidising slag to the metal in the bath ; generally, however, one slag only is made. The first slag having been removed, deoxidation and desulphurisation are effected by the medium of a slag composed largely of calcium silicate. The 'physic' required to provide for carbon, manganese, etc., in the finished steel may be added immediately after the first slag has been removed. Generally speaking, the additions are made at a later stage in the process. The complete deoxidation of the metal is made evident by the appearance of the slag, which, the above conditions prevailing, decrepitates spontaneously and is practically white when cool.

Melting and refining carried out on a basic hearth results in a practically complete loss of carbon, silicon and manganese, and a very appreciable loss of sulphur, phosphorus and such elements as chromium, molybdenum, etc.

The possible uses of the electric furnace in the steel foundry have been tabulated by J. H. Hall :—

Melt scrap . . .	Impure : eliminate phosphorus and sulphur. Pure : get hot and cast. Alloy steels.
Melt pig and more or less scrap. Eliminate C, Si, Mn.	Impure : eliminate phosphorus and sulphur. Pure : get hot and cast.
Treat hot metal . . .	From cupola : eliminate C, Si, Mn, S and P. From acid Bessemer or open hearth : eliminate S and P. From basic open hearth : eliminate S and P.
Melt pig and scrap and treat cupola metal	Eliminate C, P and S without loss of Mn and Si. (Greene's process.)

The arc-resistance furnace proved to be of value during the war in the preparation of high-grade pig iron to replace the Swedish pig iron so much sought after for the preparation of high-grade steels.

According to E. K. Scott, a 10-cwt. steel-melting furnace running 9 charges a day will do as much as 36 crucible-steel furnace holes, each accommodating two 56-lb. pots, and can be operated by one man and an ammeter boy, whereas the crucible furnaces require 15 men, most of them skilled. The electric furnace, with its transformer and switch-gear, occupies one quarter as much space as the crucible plant, and costs half as much to instal. It can deal with a charge consisting entirely of turnings, millings, etc., whilst in the crucible-steel process only a very limited quantity of turnings can be used. The renewal of the furnace linings costs less than that of crucibles for the same output. Consequently the electric-steel furnace has in a number of cases been adopted in Sheffield in place of crucible furnaces.

The chief **losses** in electric furnaces are due to radiation from the walls and conduction through the electrodes. Since the radiation losses increase as the square and the capacity as the cube of the linear dimensions of furnaces, it is obvious that large installations are more economical than small ones. Conduction losses can be reduced to a minimum by suitably designing the electrodes.

Although, with coal at 14*s.* per ton and electrical energy at ½*d.* per kW.-hour, the latter costs twenty-eight times as much as the former per British thermal unit purchased, the efficiency with which the electrical energy is applied is so much higher that the difference is largely neutralised, and in the case of crucible steel is completely wiped out. In addition, the labour cost is greatly reduced with electric furnaces and there is a considerable economy of space and buildings, while the steel produced is of higher quality and uniformity than by any other method.

A useful method of calculating heat flow through the furnace walls, etc., has been developed by Carl Hering on the lines of Ohm's law. If—

W = heat flow in watts ; T = drop in temperature in degrees C. ; R = thermal resistance in 'thermal ohms,'

then,

$$W = T/R.$$

A thermal ohm is that thermal resistance which requires a drop of temperature of 1° C. to produce one watt of heat flow and = 4·2 gramme-calorie units ; 1 gramme-calorie unit of thermal resistance = 0·24 thermal ohm.

Assuming that the electrodes are of uniform section, and that no lateral loss of heat takes place—conditions which are only approximately fulfilled in practice—Hering finds that the total loss of heat through the cold end of the electrode equals the sum of the loss by heat conduction alone (*i.e.* when no current is flowing) and half the T^2R loss. This combined loss is least when the loss by heat conduction alone equals half the T^2R loss ; the total loss then equals the T^2R loss, and no heat will be conducted out of the furnace by the electrodes. This minimum loss is dependent only on the material of the electrodes, the current, and the temperature, but not on the absolute dimensions ; it fixes the ratio of cross-section of electrode to length. In the case of a furnace working at 1,400° C., the cool end of the electrode being at 100° C., the length of the electrode 20 ins., and the current 10,000 amperes, the following dimensions were given by Hering's method of calculation :—

Carbon, 12·5 ins. diameter, loss of power in electrode 23·4 kW. ; Graphite, 7·1 ins. diameter, loss of power 15·1 kW. ; Iron, 4·8 ins. diameter, loss of power 4·6 kW. (*Stansfield.*)

F. T. Snyder states that in the case of a furnace melting 1,000 tons of steel a month, the cost of renewing the electrodes, plus the cost of the heat conducted out of the furnace by them, represents an expenditure of some 1,200*l.* a year per electrode. Hence the type of furnace having the fewest electrodes has an advantage in this respect. By using the metal bath as one terminal of the arc, a furnace can be operated with one carbon electrode per arc. The furnace lining becomes conducting at very high temperatures, and this fact limits the voltage between two electrodes passing through the same roof to a maximum of 110 volts to avoid serious leakage of current. The bulk of the cost of renewing refractory linings is due to roof wear round the electrodes, and for the size of furnace above mentioned can be kept down to 800*l.* per annum per electrode. The higher the power input the quicker is the heating and the greater the efficiency of the furnace. With three electrodes the limit of power is soon reached, owing to lack of room to space the electrodes sufficiently far apart to prevent current leakage through the roof lining, but with a single electrode there is no such limit to the power input.

The power, etc., provided in connection with Héroult furnaces used by the United States Steel Corporation are given in the following table, which applies to cold charges ; if the charge is put into the furnace in a molten condition about half this power suffices :—

POWER REQUIRED FOR ELECTRIC FURNACES.

Size of Furnace, tons.	Transformer capacity, kVA.	Amperes, per phase.	Square inches Copper, per phase.
1	375	2,250	3
2	600	3,600	5
3	750	4,500	6
4	900	5,400	7
6	1,200	7,200	9
10	2,000	12,000	16
15	3,000	18,000	24

The voltage used in arc furnaces depends upon the current and the length of the arc. In three-phase furnaces it ranges between 75 and 100 volts, and in single-phase furnaces between 150 and 200 volts. In resistance furnaces the voltage at starting, with a cold charge, may be 400 volts, decreasing as the charge heats up to 35–60 volts. The consumption of the electrodes varies widely according to the conditions obtaining, from $3\frac{1}{2}$ to 20 lbs. per ton of steel produced. The furnace bottoms last from 1 to 3 years, the walls from 100 to 300 heats, and the roof from 50 to 200 heats. To make ordinary carbon steel of crucible quality with a 6-ton furnace requires from 550 to 750 kW.-hours per ton, starting cold. In smaller furnaces the input per ton is greater. The following figures for a Rennerfelt furnace are due to C. H. Vom Baur.

ENERGY REQUIRED IN TREATING VARIOUS METALS.

	kW.-hours
Melting red brass	168
„ pure copper	197
„ white iron	290
„ gray iron	325
„ 80 per cent. ferro-manganese	441
„ and refining steel scrap on a basic bottom	700

The power factor of the arc resistance furnace varies from 85 per cent. to more than 90 per cent. : that of the resistance type from 90 to 97 per cent. ; that of the induction furnace varies from 50 to 60 per cent. only.

The efficiency of a Héroult furnace was tested with a charge consisting of 5,733 lbs. steel scrap, 430 lbs. iron ore, 346 lbs. lime. The input was 215 kW. for the first hour, at 110 volts, and afterwards 342 kW. The total time taken to melt the charge was $5\frac{1}{4}$ hours, and the total energy input for melting was 1,680 kW.-hours. The ratio of heat utilised to heat supplied (the efficiency) was 76·5 per cent. The time required for the purification and carburisation of the steel was 3 hours, and the result was 5,161 lbs. of tool steel. The total consumption of energy was 2,580 kW.-hours, or 0·171 e.h.p.-year per ton of steel.

The leads from transformers to electric furnaces should be as short and direct as possible, and close together, as the voltage drop due to their self-inductance and the very large currents is considerable, resulting in a low power factor. The conductors should be wide thin bars, or tubes, to prevent loss of energy due to the tendency of alternating current to flow in the skin of the conductor.

ELECTRICAL RESISTIVITY OF MATERIALS, OHMS PER INCH CUBE.

Material.	Cold.	Hot.	Temperature ° C.
Graphite	0·00034	0·00031	100–2000
Carbon	0·0018	0·0014	100–2000
Silicate slag	∞	1 to 2	Molten
Fire bricks (various)	∞	8 to 24	850
		24 to 300	815

(*Stansfield.*)

FURNACE CONSTRUCTION.

The following are points requiring special attention in the construction of furnaces :—

1. Sections of furnaces such as are exposed to differing physical and chemical actions should be united in such a manner as to allow of free and independent movement and as to be subject to independent condition.

2. Such sections as are exposed to high temperatures should be made thin and should, when possible, be water-cooled; such sections as are exposed to low temperatures should be made of such a thickness as will reduce thermal losses due to radiation and conduction to a minimum.

3. The interiors of furnaces should be constructed so as to resist mechanical, physical and chemical wear. With this end in view choice must be made of suitable materials of construction and of efficient design. In this connection note should be made of the following factors :—

(*a*) The action of heat.

(*b*) The erosive and corrosive actions of flame, slag, etc.

(*c*) The mechanical wear due to abrasion by the solid charge, etc.

4. The exteriors of furnaces should be supported and strengthened in such a way as to allow of the free expansion and contraction of the body. The foundations of all furnaces must be of a strong and rigid character.

MELTING POINTS OF REFRACTORY MATERIALS.

	° C.
Alumina (pure)	2010†
Bauxite	1820†
Bauxite brick	1565 to 1785†
Bauxite clay	1795†
Chromite	2180†
Chromite brick	2050†
Fireclay brick—Highest grade	1843*
General grade	1650 (average)*
Ditto	1649 (average of 41 samples)†
Ganister	1700 to 1800
Kaolin	1735 to 1740†
Magnesia brick	2165
Silica—Pure	1750†
Pure	1830*
Pure, with 14·5 per cent. alumina	1690*
Pure, with 63·0 per cent. alumina	1890*
Brick	1700 to 1750†

The melting points have been considered to be those temperatures at which the material in question was first seen to flow easily.

FURNACE EFFICIENCY.

The thermal efficiency of furnaces may be stated either absolutely or relatively :—

$$\text{Absolute furnace efficiency} = \frac{\text{Net thermal effect produced}}{\text{Calorific power of the fuel}}$$

$$\text{Relative thermal efficiency} = \frac{\text{Net thermal effect produced}}{\text{Actual calories realised}}$$

Owing to the large variation in the actual number of calories realised in individual furnaces it is general to employ the former of the above ratios. In order that a determination of the absolute efficiency of a given furnace may be made, a knowledge of the heat-absorbing and the heat-producing factors that are likely to be encountered in practice, of which a number are tabulated below, is required.

(*A*) Heat-absorbing Factors.

1. Heat absorbed by fume, gases, etc.
2. Heat removed by such solids and liquids as are periodically or continuously removed from the furnace.
3. Heat lost by radiation and conduction.
4. Heat absorbed owing to endothermic reactions within the furnace.

(*B*) Heat-producing Factors.

1. Heat supplied to the furnace by the combustion of fuel either inside or outside the furnace.
2. Heat added by such solids and liquids as are periodically or continuously charged into the furnace.
3. Heat gained owing to exothermic reactions within the furnace.

The thermal efficiency of the furnace is represented by the ratio of the heat-absorbing factors to the heat-producing factors.

* Boudouard.

† Kanolt—'Iron and Coal Trades Review,' vol. lxxxv. p. 309.

Heat Economy by Insulation.

A considerable reduction in the loss of heat by radiation (see *A*, 3, above) may be effected by building a ' wall ' of insulating bricks (p. 435) between the lining and the exterior of the furnace.

Latent Heat of Fusion of Metals.

The latent heat of fusion is the number of gram calories required to convert one gram of substance from solid into liquid without change of temperature :—

Mercury	2·8 calories.	Platinum	27·1 calories.
Lead*	5·5 ,,	Sodium	31·7 ,,
Bismuth*	10·2 ,,	Palladium	36·3 ,,
Cadmium*	10·8 ,,	Manganese*	36·7 ,,
Tin*	13·8 ,,	Antimony*	38·9 ,,
Potassium	15·7 ,,	Copper	41·0 ,,
Gold*	15·9 ,,	Nickel*	56·1 ,,
Iron	(See note).	Cobalt*	58·2 ,,
Silver*	26·0 calories.	Aluminium*	94·0 ,,
Zinc*	23·0 ,,		

NOTE.—The latent heat of fusion of iron as determined by Campbell (and confirmed by Desch) from freezing-point depression data is 20·1. C. Brisker (1908), as a result of an examination of Carpenter and Keeling's freezing-point determinations of the iron-carbon system, has arrived at the value 31·6 for this element. Grumer gives values of 23 and 33 for gray and white cast iron respectively. Wüst obtained the value 49·4 for this factor.

Heat and Electrical Energy required for Fusion of Metals.

The following table (after Stansfield) gives the heat and electrical energy required to melt various metals, per lb. :—

	Calb.†	Watt-hours.		Calb.†	Watt-hours.
Tin	28	15	Copper	162	85
Lead	16	8	Cast iron (gray)	245	129
Zinc	68	36	Tool steel (1 % C.)	300	158
Aluminium	258	136	Wrought iron	343	181
Brass	130	69			

(See also p. 2671.)

TEMPERATURE CONTROL OF FURNACES—PYROMETRY.

Determination of Calorific Intensity.

The calorific intensity of a substance is the theoretical maximum temperature attained during the perfect combustion of the same in air at 0° C. and 760 mm. pressure, and is inversely proportional to the time occupied in attaining to this temperature. The calorific intensity of a substance may be determined by calculation or by experiment.

CALCULATION.

Let C = the calorific power of the substance,
T = the rise in temperature during combustion,
$w, w', w'', — $ = the weights of the products of combustion,
$s, s', s'', — $ = the specific heats of the products of combustion,

then,

$$T = \frac{C}{ws + w's' + w''s'' + \text{etc. etc.}}$$

for the fuel at 0° C. and 760 mm. pressure.

The calorific intensity of a fuel (by formula) will be seen to be directly proportional to the calorific power of the fuel and inversely proportional to the weights and the specific heats of the products of combustion.

The calorific power can be raised by

(*a*) Superheating the fuel.
(*b*) Superheating the air needed for the combustion of the fuel.
(*c*) Using a minimum of excess air.
(*d*) Increasing the proportion of oxygen in the air.
(*e*) Reducing the proportion of incombustible material in the fuel.

---

* Values after Wüst. *Stahl und Eisen*, 1918, 38, 177.

† *Calb.*—The amount of heat required to raise one pound of water through one degree Centigrade.

Calorie.—The amount of heat required to raise one gram of water through one degree Centigrade.

Expansion Type Thermometers.

The mercurial thermometer is most important for use at about atmospheric temperatures, largely on account of the wide range of temperatures through which mercury is liquid, —39° C. to 357° C. If a high-grade mercurial thermometer has been calibrated at the National Physical Laboratory, it is possible to measure temperatures throughout its range to an accuracy of 0·001° C By the use of inert gases (nitrogen or carbon dioxide) under pressure in the tube of a thermometer, the boiling point of the mercury may be raised to allow temperatures up to 540° C. to be determined, at which temperature the glass softens. Quartz thermometers containing mercury under pressure can be used up to 700° C. By using tin instead of mercury the maximum limit may be raised to 1000° C. Steel tube mercurial thermometers, in which the capillary is attached to a form of a Bourdon pressure gauge, are not easily broken, and may be relied upon up to 540° C.

Fusion Pyrometers.

Observations of the fusion of metals, alloys, salts or mixtures of salts enable the observer to judge the temperature reached in a furnace at the moment of fusion, given that the temperature of fusion of the material employed is known prior to its introduction into the furnace. Fusion pyrometers are most suitable for the registration of maximum temperatures, after the attainment of which the source of heat may be eliminated and the furnace or heated object allowed to cool.

Metals.

The base metals have the disadvantage of ready oxidation ; the precious metals offer only a limited number of definite temperatures.

Alloys.

Only eutectic alloys (see p. 2678) and solid solutions (see p. 2678) are of value. Base metal alloys being subject to oxidation, only precious metal alloys remain as suitable fusion pyrometers. Both the alloys of silver and gold and those of gold and platinum have been employed for pyrometric work, the latter, according to Séger, being satisfactory up to a temperature of 1200° C. only.

Salts.

The following are the melting points of a number of metallic salts that have found use in pyrometric work :—

Salt.	Melting Point.	
Calcium nitrate	561° C.	1042° F.
Strontium nitrate	570 ,,	1058 ,,
Barium nitrate	593 ,,	1099 ,,
Potassium iodide	705 ,,	1301 ,,
Potassium bromide	750 ,,	1382 ,,
Potassium chloride	790 ,,	1454 ,,
Sodium chloride	800 ,,	1472 ,,
Sodium carbonate	849 ,,	1560 ,,
Sodium sulphate	880 ,,	1616 ,,
Barium chloride	960 ,,	1760 ,,
Potassium sulphate	1050 ,,	1922 ,,

Mixtures of Salts.

Under this heading may be included two important types of fusion pyrometers :—

(*a*) Séger cones.

(*b*) Sentinel pyrometers.

Séger Cones.

These are small three-sided pyramids, each about 2½ ins. high and about ½ in. wide at the base. The term 'cone' is rather misleading, as they are more pyramidal than conical. They are made of a finely ground mixture of felspar, china clay, and flint, the composition of the mixture depending on the temperature to be indicated. For the lower members of the series other mixtures are used.

These cones were invented by the late Dr. Séger of Berlin, in order to provide some means of recording the effect of heat on various porcelains. Their shape and composition are such that when heated to their indicating temperature at a reasonably slow rate and under suitable conditions they bend over slowly from the upright position possessed by them when placed in the kiln or oven and gradually form an arch. When the top has almost reached the same level as the base the indicating temperature is reached. A further rise in temperature will cause the cone to collapse and form a shapeless, glazed mass.

It is customary to employ three different cones at a time ; the first will indicate a temperature about 20 degrees Cent. below the one required to be known, the second cone will indicate the desired temperature (such as the finishing temperature of a glaze) and the third cone will serve as an indicator that the required temperature has not been exceeded by 20 degrees. The composition of the cones is such that the difference in temperature shown between two successive cones is about 20 degrees, though they are not quite regular in this respect.

It is important to remember that Séger cones do not serve as strict indicators of temperature in the sense that a thermometer does. If heated too rapidly a much higher temperature may be reached than is normally required to bend a cone, and which they are supposed to indicate. Under ordinary conditions of use, however, it is easy to find which cones correspond to the temperatures or conditions of heating which it is required to control, and Séger cones are, therefore, invaluable when they are rightly used. They serve as excellent indicators of the effect of heat on pottery and similar materials, and as it is this, rather than the mere momentary temperature which is important, they have advantages not possessed by a pyrometer or high temperature thermometer.

They were originally made only at Dr. Séger's laboratory in Berlin, but since 1914 they have also been made at the Pottery School, Stoke-on-Trent, and sold by Messrs. Keeling & Walker, Sutherland Mills, Stoke-on-Trent.

Owing to their cheapness and the difficulty of ensuring their accuracy when made by amateurs it is usually better to purchase the cones rather than to attempt to make them.

The following temperatures are those nominally associated with Séger cones, but, as explained above, it is preferable to refer to the number of the cone rather than to the temperature.

SÉGER CONES AND TEMPERATURES.

Séger Cone.	Deg. Cent.	Deg. Fah.	Séger Cone.	Deg. Cent.	Deg. Fah.	Séger Cone.	Deg. Cent.	Deg. Fah.	Séger Cone.	Deg. Cent.	Deg. Fah.
022	600	1112	07*a*	960	1760	9	1280	2336	29	1650	3002
021	650	1202	06*a*	980	1796	10	1300	2372	30	1670	3038
020	670	1238	05*a*	1000	1832	11	1320	2408	31	1690	3074
019	690	1274	04*a*	1020	1868	12	1350	2462	32	1710	3110
018	710	1310	03*a*	1040	1904	13	1380	2516	33	1730	3146
017	730	1346	02*a*	1060	1940	14	1410	2570	34	1750	3182
016	750	1382	01*a*	1080	1976	15	1435	2615	35	1770	3218
015*a*	790	1454	1*a*	1100	2012	16	1460	2660	36	1790	3254
014*a*	815	1499	2*a*	1120	2048	17	1480	2696	37	1825	3317
013*a*	835	1535	3*a*	1140	2084	18	1500	2732	38	1850	3362
012*a*	855	1571	4*a*	1160	2120	19	1520	2768	39	1880	3416
011*a*	880	1616	5*a*	1180	2156	20	1530	2786	40	1920	3488
010*a*	900	1652	6*a*	1200	2192	26	1580	2876	41	1960	3560
09*a*	920	1688	7	1230	2246	27	1610	2930	42	2000	3632
08*a*	940	1724	8	1250	2282	28	1630	2966	—	—	—

SENTINEL PYROMETERS.

These are small cylinders, measuring about $\frac{1}{2}$ in. by $\frac{1}{2}$ in. and manufactured from salt mixtures of definite melting points. Only such mixtures can be utilised as are formed of salts that neither dissociate nor corrode when in the molten state. Brearley states that ' mixtures of salts, after being melted together and ground to a very fine powder, may be made into an adhesive paste with vaseline, and applied to many purposes for which no ordinary pyrometer is available. In the smith's hearth, for example, a tool may be heated to low redness and smeared near the point with a small portion of a paste whose melting point is, say, 770° C. The tool is then reheated directly in the fire or protected from the coke by a piece of scrap, wrought or cast iron piping, until the white mark left by the thin layer of the salt mixture disappears. This indicates that the steel itself has nearly reached the desired temperature and may be quenched.'

The following are the melting points of a number of mixtures of salts:—

Salt.	Parts by Weight.	Salt.	Parts by Weight.	Deg.
Potassium carbonate . .	50	Potassium chloride	50	580 C.
Sodium chloride . .	30	Ditto	70	625 C.
Ditto	42	Ditto	58	655 C.
Potassium sulphate .	20	Sodium sulphate	80	825 C.
Ditto	30	Ditto	70	830 C.
Ditto . . .	50	Ditto	50	850 C.

Heat Radiation Pyrometers.

In this type of instrument the relation between the effect of radiant heat on suitably chosen thermal elements and the temperature of the radiating body is observed. The Féry Radiation Pyrometer is an example of this class of instrument.

In this pyrometer the rays from the hot body are received on a concave mirror, which can so be adjusted that the rays form a heat image which covers a small sensitive element permanently fixed on a line through the axis of the mirror. The sensitive element consists of a delicately adjusted thermo-couple, which delivers current to a millivoltmeter, calibrated, if desired, to read the temperature of the body, whence the heat rays are emanating, directly. In both cases suitable means are provided for ensuring that the sensitive element is completely covered by the image of the radiant body.

In order that correct results may be obtained when using pyrometers of the radiation type it is necessary that the hot body sighted should be enclosed within a vessel whose walls are approximately at the same temperature as the body itself. If this precaution is not observed the reading obtained will be lower than the true temperature of the body, the degree of inaccuracy being dependent almost entirely on the nature of the surface of the body in question.

Optical Pyrometers.

Approximate estimations of the temperatures of incandescent bodies may be made by visua observation. Numerous colour scales have been proposed, but those due to Howe and Taylor and White are generally accepted as the most accurate. The following table (based on one by Howe) is of value in this connection.

Taylor and White.		Howe.	
Name of colour.	Degrees C.	Name of colour	Degrees C.
—	—	Lowest visible red in the dark	470
—	—	Lowest visible red in daylight	475
Dull red, blood red, low red	566	Dull red	550–625
Dark cherry red	635	—	—
Cherry, full red	746	Full cherry	700
Light red, bright cherry, light cherry	843	Light red	850
Orange	899	—	—
Light orange	941	—	—
Yellow	996	Full yellow	950-1000
Light yellow	1079	Light yellow	1050
White	1205	White	1150

The design of optical pyrometers is dependent on the fact that the intensity of the light rays emitted from an incandescent 'black body' is directly proportional to the temperature of the 'body.'

The Mésuré and Nouel Pyrometric Telescope consists of two suitably mounted Nicol prisms, so adjusted that their planes are at right angles to one another when the scale of the instrument is at zero and between which is placed a quartz plate cut perpendicularly to its axis. When the incandescent body is observed through the instrument it is necessary to rotate one of the Nicols in order to extinguish the emitted light. The angle of rotation is proportional to the thickness of the quartz plate, which is a constant for any given instrument, and inversely proportional to the square root of the wave length of the observed light. The wave length of the observed light being proportional to the intensity of the light, the angle of rotation of the analyser or polariser is a measure of the temperature of the incandescent body.

The Cornu-Le Chatelier Photometric Pyrometer is designed to measure the intensity of the light emitted by an incandescent body by reducing this intensity until it corresponds to that of the light emitted by a standard lamp. The reduction required is inversely proportional to the temperature of the hot body.

The Wanner Photometric Pyrometer likewise compares the intensities of the lights emitted by the incandescent body and a standard body respectively. In this instrument, however, spectra of the two sources of light are, firstly, broken up into two polarised planes at right angles to one another; secondly, adjusted so that only the narrow red bands of each spectrum are transmitted further; and, thirdly, so brought into vision that the intensity of either field, both being in visual contact, can be strengthened or weakened, until they are equally intense, by rotation of a Nicol prism. The amount of rotation, which can be observed on a suitably situated scale, serves to measure the intensity of the emitted light, and hence to define the required temperature.

The Cambridge Optical Pyrometer is, in principle, the same as the foregoing, and is invaluable for temperature measurements between 700° and 4000° C.

Thermo-Electric Pyrometers.

This type of pyrometer is dependent for its action upon the fact that a current is produced within a closed circuit formed of two dissimilar metals when the junction between the two metals is heated, the current being proportional to the absolute temperature of the junction.

The essential features of this pyrometer are two thin insulated rods or wires of dissimilar metals coupled at one of their ends and connected at their other ends to separate leads. The connection between the two metals and the leads, which is known as the ' cold-junction,' is enclosed within suitably designed apparatus that can be retained at a known constant temperature. The leads from the ' cold-junction ' pass to some form of electrical instrument capable of registering or recording changes in the P. D. produced in the now closed circuit when the thermo-couple is heated or cooled.

The number of metals and alloys available for the manufacture of thermo-couples are few. The following are among those most generally employed :—

Couple.	Range.
Platinum—platinum rhodium	Up to 1400° C.
Platinum—platinum iridium	„ 1000 „
Nickel—nickel chromium	„ 800 „
Silver constantan	„ 600 „
Copper constantan	„ 500 „
Iron constantan	„ 500 „

For the estimation of temperatures above the melting-point of platinum the iridium-iridium-ruthenium couple has been suggested, but it is considered unlikely that the same would prove efficient, on account of the known volatility of iridium at such temperatures.

Union between couple *wires* is effected by fusion in the electric arc or oxyhydrogen flame in the case of couples having a platinum base, by soldering or brazing in the case of base-metal couples or by twisting. Union between the wires and the leads can be effected as above or by means of electrical clamps.

The couple wires, when in use, are generally protected by tubes of nichrome, fireclay, silica or glazed porcelain, the last being particularly useful when it is desired to defend the wires against the deleterious action of hot gases. When necessary, protection is afforded these tubes by enclosing them in steel jackets.

Heavier types of thermo-electric pyrometers consist of a *rod* of one metal fitted tightly into one end of a cylinder concentric with the rod and of another metal and insulated from the same by suitable means.

Calibration of thermo-electric pyrometers is effected by immersing the protected couple in substances or the vapour of substances whose melting or boiling points are known. The couple, being subject to like thermal changes as the substance, causes a fluctuation in the current flowing within the closed circuit, which fluctuation is registered by the instrument within the circuit. For example, while solidification of a metal in which the protected couple is immersed is in progress a constant reading is obtained on the instrument. This value is observed, and is recorded on a graph against the melting point of the metal as ordinate. A series of such points may be obtained by employing a number of metals in similar manner to that already described, and, these points being united, a curve is obtained from which the temperature corresponding to any given reading of the instrument may be abstracted.

The following are the melting points on the thermo-dynamic scale used by the Washington Bureau of Standards as standard temperatures in the standardisation of thermometers and pyrometers :—

Mercury	— 38·9° C.	Antimony	630·0° C.	Nickel	1452° C.
Tin	231·9 „	Aluminium	658·7 „	Iron	1520 „
Cadmium	320·9 „	Silver	960·5 „	Palladium	1549 „
Lead	327·4 „	Gold	1063·0 „	Platinum	1755 „
Zinc	419·4 „	Copper	1083·0 „	Tungsten	3000 „

At temperatures of 1000° C. the error is of the order of 0·1° C., at the melting point of platinum 5° C., at that of tungsten 100° C. A further table of other standard temperatures is given in which the following are included :—Normal boiling-point of oxygen 180·0°, sublimation of carbon dioxide in an inert liquid 78·5°, normal boiling point of water 100°, of naphthalene 217·96°, of benzophenone 305·9°, of sulphur 444·6°, freezing point of sodium chloride 801° C.

In the *Physical Review* the melting point of rhodium is estimated to be at 1907 or 1968° C. and that of iridium as 2292 or 2388° C. That of silicon is placed at 1434 or 1452° C. and that of palladium at 1548 or 1576° C. The reason for the two figures is due to the fact that the calculations are based on different estimates of the melting-point of platinum, which has been placed at 1745° C. by some experimenters and at 1789° C. by others.

Electric Resistance Pyrometers.

In these instruments advantage is taken of the change of resistance of a wire with temperature. Platinum wire is generally employed. The essential features of the commercial apparatus are a coil of fine platinum wire, enclosed in a suitably designed protecting tube, a cell or accumulator for the supply of current to the wire and an indicator or recorder on which the temperature may be read off directly.

In the original type of this instrument errors due to the residual strain and impurities in the lead wires entered, but later instruments are designed with 'compensating wires,' whose component parts are equal in resistance to and are so arranged that they are heated to the same temperature as corresponding parts of the lead wires. Changes in resistance in the lead wires are compensated by equal changes in the 'compensating wires.'

The resistance pyrometer (after Callendar) is represented diagrammatically in fig. 11.

P represents the pyrometer coil; AM, KN the lead wires; CL, DL the 'compensating wires'; FK a bridge wire with scale, so arranged that 1 cm. length of the same corresponds to 1° C. rise or fall of the temperature, H the sliding contact piece, and G a galvanometer. AB and BC are wires of equal resistance. DE represents a set of resistance coils. In practice AB, BC and DE can be

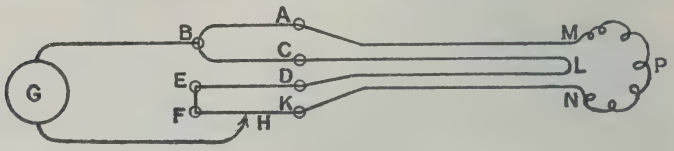


FIG. 11.

replaced by a post office box. Balance is obtained by the adjustment of the resistances in DE and the sliding of the contact piece H. The resistance of the pyrometer coil is given by the sum of the coils DE and the bridge wire scale reading.

Resistance pyrometers are accurate from 0° C. to 1000° C., and are employed industrially wherever reliable measurement is essential. They suffer owing to their high first cost and their fragile character. There are, however, large numbers of this type of pyrometer in use commercially.

METALS AND ALLOYS.

ALLOYS.

There are three well-recognised types of alloys:—

(i) Conglomerates of metals, or of metals and non-metals.
(ii) Homogeneous mixtures of metals, or of metals and non-metals.
(iii) Inter-metallic compounds, or compounds of metals with non-metals.

(i.)

In this type of alloy the component elements or their compounds crystallise out from the molten alloy in a practically pure state, and the addition of one component to the other results —within certain limits—in a lowering of the solidifying point of the resulting alloy. The alloy having the lowest solidifying point in such a series is called the 'eutectic.' Lead-tin alloys are of this type.

(ii.)

In this type of alloy the component metals do not crystallise out from the molten alloy in the pure state, but contain on freezing more or less of each other in solid solution. Copper-nickel alloys are of this type.

(iii.)

Inter-metallic compounds behave similarly to pure metals, crystallising in like manner. Copper and zinc form a compound Cu_2Zn_3, which is extremely hard and brittle.

IRON.

IRON (FERRITE).

Chemically pure iron is not an article of commerce at the present time. Iron has been obtained by electrolytic deposition of a degree of purity greater than 99·9 per cent. iron, while the American Rolling Mills Co. manufacture a commercial product which they claim to contain 99·84 per cent. of iron.

Stead, by extrapolation, has arrived at the following figures for the approximate mechanical properties of pure unworked iron :—

Yield point	9 tons per square inch.	Elongation .	51 per cent. (length/area=8).
Maximum stress	17 ditto.	Reduction in area	84 per cent.

The term ' Ferrite ' is applied to pure iron when it is considered as a microscopical constituent of iron or steel. Pure iron is composed of polyhedral crystallites of ferrite.

Carbon, manganese, silicon, sulphur, phosphorus, arsenic, molybdenum, etc., all have effect on the iron in which they occur and are frequently added with a view to obtain alloys having more or less definite physical and mechanical properties.

INGOT IRON.

' Ingot Iron ' is the generic name of a commercially pure iron produced under patents owned by the International Metal Products Company. The manufacture of the metal in Great Britain is controlled by The Armco International Corporation, which company also undertakes the sale of the metal in various forms throughout the world, through suitable agents and representatives.

Ingot Iron, unlike Puddled Wrought Iron, is produced in the basic open-hearth furnace, and in view of a certain amount of difficulty in finding an acceptable definition of ' iron ' and in differentiating between ' iron ' and ' steel,' it is well to quote the definitions advanced by Dr. Albert Sauveur, who recently received the Bessemer Medal given by the British Iron and Steel Institute Dr. Sauveur says :—

' COMMERCIAL IRON is the element iron as pure as it can be commercially produced.

' INGOT IRON is commercial iron which has been produced in a fluid condition and cast.

' WROUGHT IRON is a ferrous metal which is malleable and which has been produced from a pasty condition.

' STEEL is an alloy of iron and carbon which is malleable, usually containing substantial quantities of manganese.'

'ARMCO ' INGOT IRON is ingot iron produced in the open-hearth furnace containing less than 0·02 per cent. carbon and less than 0·035 per cent manganese.

' Armco' Ingot Iron possesses many special characteristics, some or all of which will appeal to fabricators and users in many industries. The principal points of merit are as follows :—

Chemical Purity.—' Armco ' Ingot Iron contains less than one-sixth of 1 per cent. of impurities considering silicon, sulphur, phosphorus, manganese, carbon, copper, and the gases oxygen, hydrogen, and nitrogen.

Uniformity.—The method of manufacture of Ingot Iron and its freedom from impurities ensure uniformity of analysis ; this, coupled with the fact that the metal is free from slag inclusions, such as exist in even the best puddled irons to a greater or lesser degree, ensures mechanical uniformity.

Corrosion Resistance.—The conclusion that pure iron resists rust is not merely derived from theoretical inference, but represents the definite finding of many observers of the problems of corrosion, after fully investigating generations of service evidence in the form of old hand-wrought irons, such as, for example, the Iron Pillar of Delhi. These old irons were not rustless, but they resisted rust and corrosion to a remarkable degree, by reason of their freedom from corrosion-promoting impurities.

Workability.—Ingot Iron being a dense, ductile, and uniform metal, possesses exceptional working qualities.

Welding Properties.—The superior welding qualities of Ingot Iron are due to its purity and homogeneity. Being of perfectly even texture and density throughout, it welds evenly and easily.

Paint-Holding Surface.—It has been found in service that Ingot Iron sheets require painting less often than other iron and steel sheets. This is partly explained by its velvety surface, which grips and holds paint tightly, and partly by the degasification which the metal undergoes during manufacture, thereby reducing to a minimum occluded gases which in course of time tend to become liberated, and cause flaking of the paint film.

Enamelling Properties.—' Armco ' Ingot Iron constitutes (it is claimed) the best base known for vitreous enamelling, for the following reasons :—

It is chemically and mechanically uniform.
Its velvety surface grips and holds the vitreous coating.
Its degasification during manufacture reduces the risk of pinholes, blisters, and other blemishes to a minimum.
It shows less tendency to warp than other ferrous metals during the vitreous process.

Electrical Properties.—'Armco' Ingot Iron possesses a high magnetic permeability and low retentivity. It is further a better conductor of electricity than steel; in this connection the following is the result of tests undertaken by the National Physical Laboratory :—

Resistivity

By relative mass = 5·43. By relative volume = 6·17.
(Compared with the British standards for soft copper.)

This superiority in electrical conductivity renders the metal specially suitable for telegraph wire, bond wires, conductor rails, etc.

INGOT IRON can be supplied by *The Armco International Corporation* in the form of black and galvanised sheets, plates, billets, bars, shapes, wire, rods, welding rods, cold rolled strip, and welded tubes.

WROUGHT IRON.

Wrought iron is iron low in carbon content, which in course of production is in a pasty condition, owing to the fact of its being worked at a temperature too low to render it fluid. It contains a relatively large percentage of non-metallic impurities in the form of slag, which may be readily seen when the iron is examined microscopically, and which, therefore, serves, among other features, to distinguish this product from steel. The percentage of carbon in wrought iron rarely exceeds 0·15 per cent., and the total of other elements present in the metal is generally less than 0·25 per cent. Wrought iron is not appreciably hardened by quenching.

The fracture of wrought iron is fibrous in appearance and is occasionally characterised by the presence of somewhat large proportions of non-metallic inclusions.

EFFECT OF HEAT-TREATMENT ON WROUGHT IRON.

Condition.	Elastic Limit. Tons per sq. in.	Yield Point. Tons per sq. in.	Ultimate Strength. Tons per sq. in.	Elong. %	Reduction of Area. %	Impact. Ft./lbs.	Brinell No.
As received . .	8·5	9·36	18·9	40·0	64·7	62	67
Quenched from 300° C.	—	6·74	17·1	43·0	64·7	77	65
" " 770° C.	—	9·82	18·5	41·5	64·7	22	67
" " 950° C.	—	8·00	18·8	38·0	63·6	22	73
Quenched from 950° C.; tempered 650° C.; air-cooled . .	—	9·36	18·5	44·0	64·7	71	65

(*Hadfield.*)

WROUGHT-IRON (*or Mitis*) CASTINGS.

If a small amount of aluminium is added to a charge of wrought iron which has been heated until pasty, the iron immediately liquefies, and can be poured into castings having all the properties of wrought iron except fibre, and as sound as if of cast iron. These castings are called 'Mitis' castings because of their softness in contrast with cast-iron castings.

The best results are obtained by using Swedish scrap iron or English hematite iron, that is, materials containing less than 0·1 per cent. of phosphorus, which is a very injurious ingredient if present in much larger quantity. Using a mixture with poorer quality of iron, with phosphorus running up to 0·15 per cent., good results may still be obtained—that is, the castings still compare favourably with ordinary malleable castings. The following mixture for melting has been used :—Mitis scrap, 35 per cent.; hematite muck bar, 35; wrought-iron punchings, 12¾; soft steel scrap (0·1 per cent. carbon), 12¾, white pig iron, 3; ferro-silicon (10 per cent. silicon), 1; ferro-aluminium (6 per cent. aluminium), ¾.

In general it may be said that if iron free from impurities is used very good castings are obtained; if iron is used with a large percentage of phosphorus, proportionately brittle and unsatisfactory castings result.

The ferro-aluminium used should be, for similar reasons, free from any considerable amount of such impurities as generally injure wrought iron.

PIG IRON.

Typical Analyses of Pig Iron, etc.

The following list of analyses is taken from Dr. W. H. Hatfield's book, 'Cast Iron in the Light of Modern Research.' It is published with the kind permission of the author and of the publishers, C. Griffin & Co. Ltd.

West Coast Hematites.

Brand.	G.C.	C.C.	Si.	S.	P.	Mn.	Mn.	Cu.
No. 1 .	3·750	·300	2·60	·02	·045	trace	·50	·04
„ 2 .	3·500	·460	2·40	·03	·045	trace	·50	·04
„ 3 .	3·250	·540	2·10	·04	·045	trace	·50	·04
„ 4 .	2·800	1·000	1·65	·10	·045	trace	·50	·04
„ 5 .	2·400	1·600	1·20	·20	·045	trace	·50	·04
Mottled	1·600	1·950	·90	·25	·045	trace	·20	·04
White .	trace	3·250	·65	·30	·045	trace	·10	·04
	From Native Ores only.						Average Actual Mn and Cu	
Special .	3·75	·300	2·60	·02	·02	0·500		

East Coast Hematites. ('Seaton.')

Brand.	G.C.	C.C.	Si.	S.	P.	Mn.	Cu.
No. 1 . .	3·725	·300	2·50	·02	·05	1·00	·05
„ 2 . .	3·525	·450	2·25	·03	·05	1·00	·05
„ 3 . .	3·150	·560	2·00	·04	·05	1·00	·05
„ 4 . .	2·750	1·000	1·50	·10	·05	1·00	·05
„ 5 . .	2·450	1·550	1·00	·20	·05	·75	·05
Mottled . .	1·500	2·050	·75	·25	·05	·50	·05
White . .	trace	3·150	·65	·30	·05	·50	·05
Special No. 1 .	3·525	·300	2·50	·015	·03	1·00	·05
„ „ 2 .	3·350	·400	2·25	·020	·03	1·00	·05
„ „ 3 .	3·00	·500	2·00	·025	·03	1·00	·05

Scotch Hematites. Welsh Hematites.

These are practically of the same composition as the East Coast Hematites.

Bessemer Basic. ('Carlton.')

G.C.	C.C.	Si.	S.	P.	Mn.
Traces . . .	2·65	·35 to 1·00	·05	2·850	1·50

'Ferro' Alloys.

Brand.	C.	Si.	S.	P.	Mn.
Ferro-Manganese . .	6·640	·610	·028	·180	80·100
Ferro-Silicon . . .	1·300	11·000	·040	·060	2·500
„ (Electro) .	·270	28·520	·040	·040	·400
Silico-Spiegel . . .	1·700	11·000	·040	·049	18·320
„ (Special) .	·690	20·900	·018	·097	55·760
Spiegel	4·500	·900	·030	·070	18·000

NOTE.—Ferro-Manganese is usually 80·00 Mn.
Ferro-Silicon varies from 8·00 to 16·00 Si.
Silico-Spiegel varies from 10·00 to 14·00 and 18·00 to 20·00 Mn.
Spiegel varies from 10·00 to 25·00 Mn.

Scotch Pig Iron. ('Carron.')

Brand.	G.C.	C.C.	Si.	S.	P.	Mn.
No. 1	3·500	·140	2·800	·035	·880	1·450
„ 2	3·370	·250	2·275	·045	·907	1·450
„ 3 Soft	3·280	·180	2·650	·038	·902	1·400
„ 3 Foundry . . .	3·350	·200	2·150	·055	·907	1·311
„ 3 Close . . .	3·170	·280	1·750	·050	·905	1·420
„ 4	3·160	·290	1·570	·060	·905	1·450

Yorkshire Pig Iron. ('Clarence.')

Brand.	G.C.	C.C.	Si.	S.	P.	Mn.
No. 1	3·030	·30	3·170	·03	1·600	·50
,, 3	3·050	·30	2·800	·03	1·570	·60
,, 4 Foundry	2·990	·43	2·240	·06	1·540	·52
,, 4 Forge	2·750	·55	1·850	·10	1·560	·50
Mottled	1·490	1·65	1·270	·18	1·510	·40
White	·100	3·10	·880	·28	1·570	·38

Lincolnshire Iron. ('Redbourne.')

Brand.	G.C.	C.C.	Si.	S.	P.	Mn.
No. 3 Foundry	3·100	·560	2·751	·038	1·326	1·680
,, 4 Foundry	2·816	·751	2·610	·045	1·318	1·589
Grey Forge	2·525	·895	2·508	·056	1·316	1·456
No. 4 Close	2·459	·945	2·452	·059	1·296	1·344
Mottled	1·480	1·520	1·325	·100	1·296	1·130
White	·675	2·285	·450	·155	1·295	1·100

Derbyshire Irons. ('Renishaw.')

Brand.	G.C.	C.C.	Si.	S.	P.	Mn.
No. 1	3·410	Trace	2·939	·036	1·680	·362
,, 2	3·260	1·79	2·146	·034	1·62	·707
,, 3	3·850	·112	2·333	·024	1·581	·572
,, 4 Foundry	3·300	·225	2·099	·053	1·771	·647
Close Forge	2·800	·330	1·819	·079	1·730	·520
Mottled	1·550	1·560	1·200	·200	1·500	·350
White	2·650	·100	·450	·250	1·350	·250
Glazy	3·450	Trace	4·500	·070	1·600	·610

Northampton and District Irons. ('Holwell.')

Brand.	G.C.	C.C.	Si.	S.	P.	Mn.
No. 1 Foundry	3·350	—	2·612	·022	1·190	·490
,, 2 ,,	3·650	—	2·473	·036	1·173	·403
,, 3 ,,	3·850	—	2·379	·041	1·190	·403
,, 4 ,,	3·350	—	2·333	·047	1·255	·389
,, 4 Forge	2·971	—	2·331	·056	1·088	·484
,, 5 ,,	1·750	—	1·376	·062	·929	·245
Mottled	1·540	—	1·261	·225	·970	·217
White	·840	—	1·049	·305	·937	·188
Glazy	2·500	—	4·850	·028	1·285	·532

Welsh Iron.

Brand.	G.C.	C.C.	Si.	S.	P.	Mn.
Grey Forge	3·02	Trace	2·56	·05	1·47	·43
Foundry	3·30	Trace	3·26	Trace	1·75	·34

Yorkshire Cold Blast. Ex. Sir I. Lowthian Bell.

Brand.	G.C.	C.C.	Si.	S.	P.	Mn.
No. 1	3·421	·583	1·708	·073	·630	1·606
,, 2	3·155	·581	1·646	·070	·635	1·472
,, 3	3·361	·393	1·382	·063	·602	1·475
,, 4	3·308	·319	1·381	·081	·602	1·169

Staffordshire Irons (Cold Blast). ('Glazebrook.')

Brand.	G.C.	C.C.	Si.	S.	P.	Mn.
No. 1 . .	3·07	—	1·48	·03	·43	·96
,, 2 . .	3·04	—	1·27	·04	·34	·80
,, 3 . .	3·12	—	1·16	·05	·44	·94
,, 4 . .	3·03	—	·83	·04	·31	·27
,, 5 . .	2·81	—	·57	·06	·29	·13

Staffordshire Iron (Semi-Cold Blast).

Brand.	G.C.	C.C.	Si.	S.	P.	Mn.
No. 3 . .	3·10	·75	1·50	·03	·25	·50
,, 4 . .	2·92	·81	1·22	·04	·49	·75
,, 5 . .	2·78	·51	·93	·03	·30	·68

Malleable Cast Pig Iron.
West Coast Refined.

Brand.	C.	Si.	S.	P.	Mn.
Grey	3·80	1·25	·10	·05	·10
Mottled	3·45	·70–1·00	·10–·25	·05	·10
White	3·20	·60	·30	·05	·10

East Coast Refined. ('S.C.M.')

Brand.	C.	Si.	S.	P.	Mn.
Grey	3·80	1·00	·08	·05	·80
Mottled	3·40	·50–·85	·10–·20	·05	·60
White	3·10	·45	·25	·05	·40

CAST IRON.

Foundry Practice.

1.

The physical and mechanical properties of the products of the cast-iron foundry are almost entirely dependent on the condition of the carbon in these products. The carbon in cast iron can occur either in a free state, as graphite, or in combination with iron, as a carbide of iron, Fe_3C, or with iron and manganese, as a double carbide $(Fe, Mn)_3C$. Foundry practice consists almost entirely of the control of the degree of graphitisation of the carbides present in the cast iron produced.

The graphitisation of these carbides tends to increase—

(*a*) With the temperature.
(*b*) With deceleration of cooling.
(*c*) With the content of silicon, nickel, or aluminium.
(*d*) With the number of graphite nuclei in the iron.
(*e*) With the total content of carbon

Graphitisation is prohibited—

(*a*) By the presence of sulphur, oxygen, vanadium, or chromium.
(*b*) By the presence of manganese under certain conditions.
(*c*) By acceleration of cooling.

It will be seen that by the variation of the above factors an immense number of conditions can be obtained. It is in the scientific control of these that much of the success in foundry practice lies.

2. The Influence of Carbon on Cast Iron.

The strength, hardness, density of structure, etc., of cast iron are dependent almost entirely on the relative quantities of carbide and graphite in the same.

An increase in the ratio $\frac{\text{Carbide}}{\text{Graphite}}$ leads to—

(a) An increase in hardness, brittleness, and density of structure.
(b) A decrease in machinability and ductility.

The strength of cast iron reaches a maximum when the combined carbon content is between 0·80 per cent. and 1·00 per cent.; if the combined carbon content given above is exceeded a reduction in strength, which only terminates when the carbon is entirely in the form of carbide, is witnessed. The following figures, given by Howe, are of value in this connection :—

As the tensile strength increases	From less than 20,000 lbs. per sq. in.	To more than 35,000 lbs. per sq. in.	Or by more than 15,000 lbs. per sq. in.
The total carbon content decreases .	from 3·69 per cent.	to 2·98 per cent.	or by 0·71 per cent.
The graphite content decreases . .	from 3·12 „	to 2·22 „	or by 0·90 „
The combined carbon content increases	from 0·57 „	to 0·76 „	or by 0·19 „

The above table classifies the results of an examination of 180 samples of cast iron taken without selection.

According to Dr. Stead (British Foundrymen's Association Proceedings, 1915), an increase of each 0·1 per cent. of graphite may reduce transverse strength by 2 cwts. and tensile strength by 0·8 ton per square inch.

The fusibility of the iron carbon alloys increases as the total carbon content increases up to about 4·30 per cent.; beyond this point the melting points of the alloys are raised by additions of carbon. Other effects of increased total carbon are decreased tensile strength, increased machinability and decreased chilling power. The last phenomenon is occasioned by the difficulty of retaining carbon as carbide, which naturally increases as the amount of carbon to be thus retained increases. Given suitable means for preventing graphitisation, every increase in the content of total carbon leads to an increase in the hardness of the chilled surface of the alloy in question.

The machining of cast iron containing more than 0·80 per Cent. of combined carbon is practically impossible.

3. The Influence of Silicon on Cast Iron.

Silicon varies from 0·20 to 4·00 per cent. in cast iron, but rarely exceeds 3·50 per cent. or falls below 0·50 per cent. It is present in cast iron, in all probability, as silicide of iron (FeSi), which is dissolved in the pure iron, or ferrite, of the alloy. The presence of this compound tends to strengthen and harden the ferrite. Silicon, however, has the effect of rendering cementite unstable, with the result that there is a tendency towards the formation of ferrite and graphite. Starting with white cast iron, the addition of silicon up to 1·00 per cent. results in an almost proportional decrease in the chilling power of the iron. Again, the addition of silicon to white cast iron leads to an increase in the strength of the resulting irons; this factor reaches a maximum when the content of combined carbon in the alloy is between 0·80 per cent. and 1·00 per cent. As the silicon percentage is further raised so are the combined carbon content and the strength of the cast iron lowered. When the silicon content is above 2·00 per cent. its hardening effect on ferrite is noticeable, iron relatively hard to machine being occasionally improved by a reduction of the silicon content.

The presence of silicon leads to a depression of the melting-point of cast iron; in similar manner the percentage of carbon in the eutectic alloy is reduced from 4·30 per cent. to 3·40 per cent. as the percentage of silicon increases up to 3·00.

Silicon has the effect of reducing the shrinkage of cast iron.

As a means of determining the correct proportions of carbon and silicon in cast iron, the following formula has, according to Mr. F. J. Cook, been used with marked success in practical working :—

$$X = \frac{C}{4{\cdot}26 \times \frac{Si}{3{\cdot}6}},$$

where C = *total* carbon in percentage;
Si = silicon in percentage.

The elements carbon and silicon are in the best proportion when the value of X is as given below :—

For Ordinary work such as pans, grates, easily machined casting and pipes	X = 0·9 to 1
Locomotive cylinder (maximum transverse strength)	„ = 0·83
Steam, gas, Diesel and hydraulic cylinders (maximum tensile strength)	„ = 0·76 to 0·82
Chilled castings	„ = 0·75 to 0·8
Acid resisting	„ = 0·85

4. The Influence of Sulphur on Cast Iron.

Sulphur combines with iron to form a sulphide having a low melting point, which occurs in steel in the form of ovoids, oval laminæ or membranes enveloping the ferrite grains. The presence of this sulphide is the cause of red shortness.

In cast iron this sulphide may modify the properties of the alloy owing to its manner of aggregation, but it is deleterious chiefly on account of its tendency to segregate and its powerful effect on shrinkage, which it increases. Sulphur tends to restrain graphitisation, hence it increases the chilling power of cast iron and may, therefore, be termed a hardening agent. This element should, however, be employed with great caution, if at all, in this capacity, on account of the objectionable properties previously stated. The presence of sulphur makes the molten metal thick and sluggish, giving rise to blowholes, yet this element in fair proportions, say 0·12 per cent., adds considerably to the wearing properties and strength of the metal. In every case, however, sulphurous iron *must* be cast as hot as possible if trouble is to be avoided.

Manganese and silicon are employed to counteract the evil effects of sulphur. Manganese reacts with iron sulphide, with resulting formation of manganese sulphide and liberation of iron. Under certain circumstances manganese sulphide is removable from cast iron, but even when retained its effects are not so deleterious as are those of iron sulphide. Silicon, owing to its strong tendency to induce graphitisation, resists the effect of sulphur in tending to retain carbon in the combined condition; it does not remove sulphur, nor does it prohibit the shrinkage due to sulphur or reduce the tendency towards cracking.

Sulphur is frequently limited by specification to 0·05 per cent., but in the hands of experienced and careful workers as much as 0·20 per cent. can be allowed without ensuing unsoundness. If the sulphur content of an iron is to be low the conditions under which it is prepared must be such that sulphur tends to be eliminated. Hence, coke low in sulphur must be used and suitable fluxes should be employed in the furnace for removing sulphur from the metal, *e.g.* lime, fluor spar or manganous oxide.

5. The Influence of Manganese on Cast Iron.

Manganese and iron unite to form a series of solid solutions whose freezing-point curve lies entirely between the freezing points of the component metals. The freezing points of all the alloys of pure iron and pure manganese are below that of pure iron. The freezing point of cast iron is depressed by the addition of manganese, not only on the above account but also owing to an increase in the degree of solubility of carbon in cast iron, which is brought about by the fact that the affinity of manganese for carbon is greater than that of iron for carbon. The depression of the freezing point reaches a maximum when the percentage of manganese is about 13.

The alloys of iron and manganese (pure) are harder than pure iron. Manganese therefore would, normally, have the effect of hardening the ferrite of cast iron were it not for the fact that its affinity, firstly for sulphur, and secondly for carbon, prohibits its solution in the ferrite.

Manganese, when added to cast iron, combines in the first place with such sulphur as is present in the iron; any excess above that required for the formation of manganese sulphide unites with iron carbide, with the formation of a double carbide of iron and manganese.

The quantity of manganese required to effect the entire conversion of the iron sulphide in the alloy into manganese sulphide is considerably more than that indicated by the equation—

$$FeS + Mn = MnS + Fe.$$

Manganese sulphide is characterised by the following physical properties:—

(1) High melting point—1400° C.;
(2) Insolubility in molten iron;
(3) Low specific gravity;

all of which assist in bringing about the separation of both manganese and sulphur from the molten alloy. Rapid cooling of the molten alloy leads to the more or less complete retention of manganese sulphide in the resulting solid alloy; slow cooling assists in separating the sulphide and bringing it into contact with the slag found on the surface of the alloy when the same is, for example, in the ladle. When the metal does not remain long enough in the ladle, or is not hot enough for the above to take place quickly, the manganese sulphide will be found on the top part of the casting in the form of hard spots, which are difficult to tool. Iron sulphide is deleterious in its action upon iron and steel. Manganese sulphide, though not beneficial, is of a somewhat neutral character. Manganese is, therefore, valuable as a desulphuriser.

Such manganese as is present in cast iron in excess of that required for the transformation of iron sulphide into manganese sulphide combines with the carbide of iron to form a double carbide, as stated above. Manganese is generally considered to induce the retention of the carbon in combination. This is true, but this effect does not make itself felt until the manganese content is well above that usually found in foundry irons. Hence, though hardness is increased by the addition of manganese, the cause is not usually the retention of carbon as carbide.

On the whole it may be said that, owing to (1) the tendency of manganese to produce brittleness and coarse crystallisation, and (2) its activity as a deoxidiser, it is inadvisable to add manganese to cast iron for the sole purpose of increasing hardness and chilling power.

Manganese increases the shrinkage of cast iron.

6. The Influence of Phosphorus on Cast Iron.

Phosphorus combines with iron to form a number of compounds, of which one, that containing three molecules of iron to each molecule of phosphorus, is of common occurrence in cast iron. The presence of phosphorus effects a reduction in the melting-point of cast iron and a reduction in the total carbon content of cast iron.

As the phosphorus content of cast iron is increased up to about 0·40 per cent. so is the strength of the cast iron increased. Further additions of phosphorus appear to have but slight effect on the strength of cast iron until the percentage reaches about 0·75 per cent. Above this point the alloys are liable to be brittle, and unless fluidity is more important than strength 0·75 per cent. of phosphorus should not be exceeded. For thin castings in general, which are not required to be strong, the phosphorus can be increased to 1·5 per cent. or more, but in thin castings such as automobile cylinders, which are required to withstand wear and tear, not more than 0·30 per cent. of phosphorus should be allowed, owing to the slight tendency towards segregation of high phosphide irons.

7. The Influence of Nickel on Cast Iron.

The effect of the addition of nickel to grey iron (carbon 3·6, silicon 1·4) is shown in the following table:—

Nickel. %	Transverse Strength. Tons per sq. in.	Tensile Strength. Tons per sq. in.
0	1·52	10·0
0·5	1·78	13·7
1·0	1·96	14·5
1·5	1·99	14·6
2·0	1·98	14·5

The increase in transverse, tensile, and compressive strength of grey iron due to the addition of from 0·5 to 5 per cent. nickel varies from 10 to 50 per cent., depending on the composition of the iron and other factors. The presence of 0·5 per cent., and preferably up to 0·8 per cent., combined carbon is essential. If by reason of high silicon or total carbon the combined carbon is below the limit given, it should be raised, either by adding chromium with the nickel in the proper ratio, or by reduction of the silicon or total carbon, allowing the nickel to replace silicon in eliminating the excess combined carbon, if any.

The effect of the addition of nickel to a cast-iron mixture is to—(*a*) refine the grain; (*b*) reduce and control chill; (*c*) reduce porosity; (*d*) increase the strength and hardness and simultaneously improve the machinability; (*e*) produce almost uniform hardness from castings of irregular section.

If the iron is partially graphitised in the first place by a small silicon content, then, in general, the graphite present is fine. If, however, the initial silicon content is high enough to give coarse graphite, then the addition of nickel renders the graphite flakes finer; with very high silicon (3 per cent.) the effect is less marked.

Silicon itself exerts a similar effect to nickel on graphitisation, and in a more marked degree. In this respect one part of silicon may be said to equal 2–4 parts of nickel, but the use of silicon for reducing the chill entails progressive impairment of physical properties—more open grain, greater tendency to porosity and internal shrinkage, and decreased hardness and strength. While nickel acts powerfully to reduce the combined carbon to the free carbide point—about 0·9 per cent.—it only acts mildly beyond that point, and does not tend to open the grain and soften the iron when present in excess.

Machinability alone can be improved also, and more cheaply, by the use of higher silicon, but with attendant impairment of the physical properties. Silicon secures machinability by graphitisation alone, and this lowers the hardness. The addition of nickel produces a progressive and uniform hardening of the iron matrix (producing sorbite) instead of through an increase of carbides. As nickel both hardens iron and reduces chill simultaneously it is possible, by the use of 1–3 per cent. nickel, in a casting of varying section to harden the heavy section and at the same time render the light section more machinable. In comparatively heavy sections with high silicon irons up to about 0·6 per cent. chromium may be used.

(*Bureau of Information on Nickel.*)

The Engineer.

Established 1856. Registered as a Newspaper

PROVIDES
THE LATEST AND THE MOST RELIABLE INFORMATION
FOR
ENGINEERS
IN ALL BRANCHES OF THE PROFESSION.

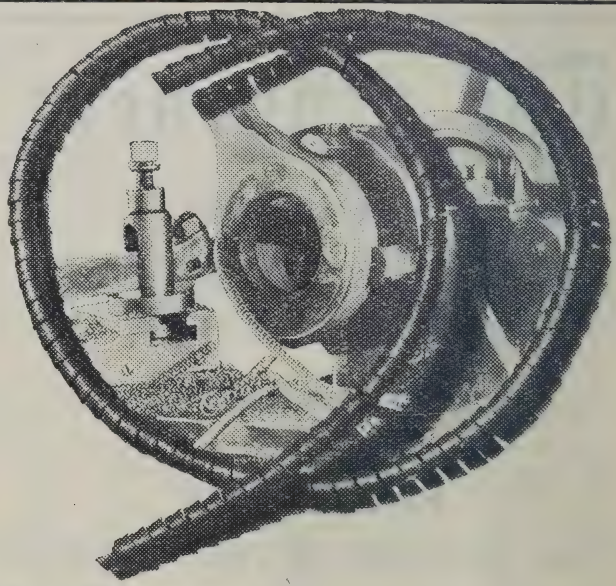
The last issue in each month contains a Supplement, **The Metallurgist,** *a journal devoted to the science of metals, ferrous and non-ferrous.*

Subscriptions

		Per annum
INLAND, YEARLY		£3 5 0
CANADIAN SUBSCRIPTIONS.	Thin paper edition ...	£2 18 6
	Thick paper edition ...	£3 3 0
FOREIGN SUBSCRIPTIONS.	Thin paper edition ...	£3 3 0
	Thick paper edition ...	£3 7 6

'THE ENGINEER' DIRECTORY AND BUYER'S GUIDE

SENT FREE ON APPLICATION.



Reduce machining costs!

WITH Ley's 'Black Heart' Malleable, machining troubles disappear because firstly, there are no hard spots in Ley's malleable, and secondly there is a natural lubricating effect from the temper carbon (which makes the metal as free-cutting as many of the special free-cutting steel bars).

Consult us on malleable machining problems; we can probably tell you at once from our experience exactly what savings you could effect by changing over to Ley's Malleable.

LEY'S

LEY'S MALLEABLE CASTINGS CO. LTD., DERBY.

F.2687

8. The Influence of Chromium on Cast Iron.

The influence of chromium is (*a*) increase in the total carbon content; (*b*) increase in the combined carbon; (*c*) increase in the hardness; (*d*) increase in the transverse strength up to 1 per cent. chromium. The proper use of a small percentage of chromium will enable the amount of combined carbon to be raised and adjusted. In addition, certain combinations of nickel and chromium (*e.g.* 1 per cent. Cr, 3 per cent. Ni) have a greater hardening and strengthening effect than either element alone. The hardening properties of chromium are especially valuable for castings subjected to abrasion. Excessive amounts of chromium produce hard spots of chromium carbide which renders castings unmachinable—therefore, in light sections, it is advisable to use nickel alone. In general, the chromium should not exceed 0·5 per cent., nor the ratio 1 part to 3 parts of nickel.

(*Bureau of Information on Nickel.*)

9. Semi-Steel.

Semi-steel is simply cast iron made from a furnace charge of cast iron to which **mild steel or wrought iron** has been added. Additions **of from 10 to 25 per cent. of steel** are usual, **but 30 and occasionally 40 per cent. are not unknown. As steel is low in carbon, silicon and phosphorus, the final cast iron (for it is still a true cast iron) is low in these elements. The steel should be added as sheared clippings or other small pieces, but not as borings, which take up too much sulphur. Iron appears not to take up sulphur from gases, but only by direct contact with the coke, hence the air furnace gives irons lower in sulphur than the cupola and is specially suited for melting steel with pig irons. Perfectly good results are,** with care, **obtainable from the cupola.** There is, however, a tendency for the more fluid portions of a cupola burden of cast iron and steel to run first. Hence the first tap from the cupola is generally higher in carbon content than succeeding taps.

Tests for Cast Iron.

(Abstract of paper read by Mr. F. J. Cook before the Staffordshire Iron and Steel Institute, January 20, 1917.)

Important tests which are of use as a guide to foundrymen are :—

(1) Keep's shrinkage test;
(2) Transverse test;
(3) Hardness test;
(4) Casting temperature or fluidity test;
(5) Impact test.

(1) **Keep's Test** consists of casting ½ in. square bars between iron yokes, the ends of which are 12 ins. apart. When the bars have cooled down the shrinkage is shown by the difference between the length of the bars and the gap in the yoke in which they were cast; this is usually measured by inserting a taper graduated wedge in the gap and measuring off the dimension of the space.

This test is quite a comparative one, and when the amount of shrinkage which is associated with the most satisfactory castings has been arrived at, any deviation can be rectified by a little addition of silicon if the shrinkage becomes too high and *vice versa* when it is too low.

Under normal conditions a shrinkage of 0·183 in. = 1 per cent. silicon, and, to take the other extreme, 3·5 per cent. silicon is associated with 0·123 in. shrinkage.

(2) **Transverse Test.**—The following table gives average particulars of shrinkage, deflection and breaking load obtained with various classes of iron :—

Class of Material.	Shrinkage.	Deflection.	Breaking load.
	in.	in.	lbs.
Tough strong Iron	0·160	0·25	550
Hard „ „	0·168	0·22	630
Soft Iron	0·123	0·18	423
Mottled Iron	0·207	0·11	375
White Iron	0·226	0·08	250

from which it will be seen that to obtain a big deflection strong irons are necessary.

(3) **Hardness.**—The most handy and reliable form of test is the 'drill test,' as all other methods deal with surface hardness only. The test may be made as follows :—Cast the metal into bars 1½ in. diameter, ¼ in. being shaped off one side to give a flat surface to work upon, and move the outside skin. Drill four holes ⅜ in. diameter, to a depth of 1 in., then the number of revolutions required to drill to this depth (the mean of the four holes being taken) indicates the

general hardness. The angle of the drill should be ground exactly the same for each test, by means of a suitable guide on the grinding machine. It is also essential to keep the same drill for the purpose, and preferably, two drills should be used and two holes drilled with each, which minimises many discrepancies which otherwise might come in. The efficiency of drills is materially affected by the sharpness of their point, and as cast iron is often not homogeneous and may contain more or less hard spots a false result may be obtained owing to the point becoming unduly dulled by striking a hard spot in the early stages of the test. To avoid this a hole $\frac{1}{8}$ in. diameter should be first drilled, which affords clearance for the point of the drill and ensures the full cutting edge coming into operation. It is obviously necessary that the same weight be attached to the drill and also that it runs at the same number of revolutions per minute for each test.

(4) **Casting Temperature.**—Low silicon cast iron is very prone to the effects of casting temperature, and some irons employed for such purpose are also liable to liquid contraction. A handy workshop test for casting temperature with this class of iron consists in making bars of the general dimensions shown in fig. 12. When cold the bars are broken through the line AB, the condition of the fracture giving an indication of the temperature at which

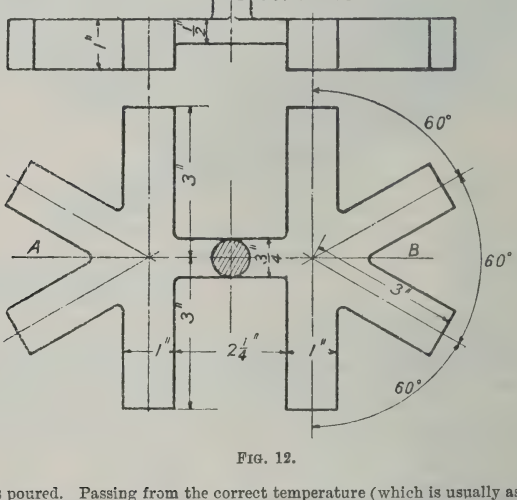


FIG. 12.

the metal was poured. Passing from the correct temperature (which is usually as hot as can be obtained from the cupola) to very low, the following fractures will be observed—perfectly solid and homogeneous, then slight whitish centre, then brown centres of varying sizes and depth of colour as temperature is lowered and then black centre; with the latter is always associated open cavities, and immediately above them is invariably found a gas hole.

For softer cast iron, which can be cast at a lower temperature without suffering the same effects as low silicon irons, a test bar in the form of a wedge is a very useful guide as to the fluidity of the metal. Dimensions of such a test-bar used by the British Chuck and Piston Ring Co. are length 9 ins., width 1 in., $\frac{3}{8}$ in. thick at the largest end tapering to $\frac{1}{32}$ at the smaller, the bars being run from the thick end.

(5) **Impact Test.**—A satisfactory test is on bars 40 mm. square (1·57 in.) supported on knife edges 160 mm. (6·3 ins.) apart, and by dropping a weight of 12 kilos (26·5 lbs.) from a height of 30 cm. (11·8 ins.) increasing the height of drop by increments of 5 cm. (1·9 in.) until such times as the sample breaks, the height at which the bar eventually fractures being taken as the impact figure. Attached to the weight, in such a manner as to strike the bar parallel to the supporting knife edges, and the centre of same, is another knife edge. The face of all the knife edges should be rounded $\frac{1}{16}$ in. radius.

A result of 55 cm. (21·5 ins.) is none too high for the work named, although this is quite a severe test. The maximum attained, as far as is known, has been 80 cm. (31·3 in.).

Analyses and Mechanical Properties of Cast Iron.

TABLE OF STANDARDS FOR BRITISH IRON CASTINGS.

	Steam Cylinders.	Frames.	Flywheels.	General.
Total carbon	2·9	3·3	3·5	3·5
Combined carbon	0·9–0·95		0·37	
Silicon	1·1–1·3	1·6	2·0	1·7–2·2
Manganese	0·6–0·9	0·65	0·4–0·5	0·3–0·4
Phosphorus	0·4–0·6	0·5	0·9	0·4–1·4
Sulphur, under	0·15	0·15	0·15	0·15
Transverse load, cwt.	35–46	34–38	30–33	29–32
,, deflection, inches	0·5	0·5	0·35–0·45	0·3–0·5
Tensile, tons sq. in.	12–14	11–13	11–14	9–11

Transverse tests are made on bars 3 ft. span, 2 ins. by 1 in. rectangular section, 1 in. face vertical.

TABLE OF TYPICAL ANALYSES OF BRITISH IRON CASTINGS.

(*F. M. Thomas, Birmingham Metallurgical Society.*)

Castings.	Combined Carbon.	Graphite Carbon.	Silicon.	Manganese.	Phosphorus.	Sulphur.
Cylinders and hydraulic	·6–·75	2·5–2·8	1·0–1·3	·5–·7	·45– ·7	·08–·12
Heavy machinery	·45–·60	2·8–3·0	1·2–1·5	·6–·85	·4– ·6	·06–·08
General castings	·35–·45	3·0–3·2	2·0–2·25	·8–1·0	1·0–1·3	·06–·08
Ornamental, thin	·08–·15	3·2–3·6	2·5–2·8	·8–1·0	1·3–1·5	·03–·04

TABLE OF STANDARDS FOR AMERICAN IRON CASTINGS.

(*Dr. J. J. Porter's Report to the American Foundrymen's Association in* 1910.)

	Total Carbon.	Combined Carbon.	Silicon.	Manganese.	Phosphorus.	Sulphur.
Air and ammonia cylinders	3·0–3·3	—	1·0–1·75	·7– ·9	·3–·5	U ·09
Motor car cylinders	3·0–3·25	·55–·65	1·75–2·00	·6– ·8	·4–·5	U ·08
Bedplates	—	—	1·25–1·75	·6– ·8	·3–·5	U ·10
Chilled castings	—	—	·75–1·25	·8–1·2	·2–·4	·08–·10
,, wheels	3·5–3·7	·6–·8	·6–·7	·5– ·6	·3–·4	·08–·10
Engine frames	—	—	1·25–2·0	·6–1·0	·3–·5	U ·09
Flywheels	—	—	1·5 –2·25	·5– ·7	·4–·6	U ·08
Gas engine cylinders	3·0–3·3	—	1·0 –1·75	·7– ·9	·2–·4	U ·08
Gears, heavy	Low	—	1·0 –1·5	·8–1·0	·3–·5	·08–·10
,, small	—	—	2·0 –2·5	·6– ·8	·5–·7	U ·08
Hydraulic cylinders, heavy	Low	—	·8 –1·2	·8–1·0	·2–·4	U ·10
,, ,, medium	Low	—	1·2 –1·6	·7– ·9	·3–·5	U ·09
Locomotive cylinders	—	—	1·0 –1·5	·8–1·0	·3–·5	·08–·10
Machinery, cast, heavy	Low	—	1·0 –1·5	·8–1·0	·3–·5	U ·10
,, ,, light	—	—	2·0 –2·5	·5– ·7	·5–·7	U ·08
Pipes	—	—	1·5 –2·0	·6– ·8	·5–·8	U ·10
Pipe fittings	—	—	1·75–2·5	·6– ·8	·5–·8	U ·08
Piston rings	Low	—	1·5 –2·0	·4– ·6	·3–·5	U ·08
Pulleys, heavy	—	—	1·75–2·25	·6– ·8	·5–·7	U ·09
,, light	—	—	2·25–2·75	·5– ·7	·6–·8	U ·08
Steam cylinders, heavy	Low	—	1·0 –1·25	·8–1·0	·2–·4	U ·10
,, ,, medium	—	—	1·25–1·75	·7– ·9	·3–·5	U ·09
Valves, large	—	—	1·25–1·75	·8–1·0	·2–·4	U ·09
,, small	Low	—	1·75–2·25	·6– ·8	·3–·5	U ·08
Water heaters	—	—	2·0 –2·25	·6– ·8	·3–·5	U ·08

NOTE.—U indicates 'under.'

TABLE OF STANDARDS FOR GERMAN IRON CASTINGS.

	Gas Cylinders.	Steam Cylinder. No liner.	Steam Cylinder. With liner.	Beds and Standards	General Machinery	Piston Rings and Special Small Work.
Total carbon	2·9–3·2	3·2–3·4	3·4–3·5	3·4–3·6	3·5–3·8	3·0–3·3
Silicon	·9–1·0	1·0–1·2	1·3–1·5	1·4–1·6	1·5–1·9	1·2–1·5
Manganese	·6– ·9	·6– ·8	·6– ·7	·5– ·7	·5– ·6	·7– ·8
Phosphorus	·1– ·2	·3– ·5	·4– ·5	·5– ·8	·8–1·0	·6– ·8
Sulphur	·08–·09	·08–·09	·08–·09	·08–·09	·08–·10	·07–·08
Copper, under	·05	·05	Trace	·06–·08	·08–·10	·03
Metal thickness, mm.	50–100	35–55	35–55	35–100	10–35	10–40
Tensile strength, tons	16·5–14·0	14·0–11·4	12·7–10·2	11·4–8·9	10·2– 7·6	14·0–11·4
Bending stress, tons	28·6–25·4	26·7–24·1	24·1–20·3	22·2–19·0	19·0–16·5	25·4–22·2
Deflection, mm.	22–7	18–8	15–10	18–8	16–10	25–12
Diam. test bar, mm.	40–100	40–80	40–80	30–80	30–50	30–50
Span „ „	1,000	1,000	1,000	1,000	1,000	1,000
E., lbs. sq. in., about	12,100,000	15,600,000	14,200,000	13,500,000	13,100,000	12,800,000

E—modulus of elasticity.

The above is taken from the practice of a large German engineering firm as published in the German technical press in 1908. The test bars are round, of the varying diameters stated: all cross-breaking loads being on one metre (39·37 ins.) span.

STRENGTH OF CAST IRONS.

Analysis.					Keep's Test.		Transverse Test.	Tensile Test.	Remarks. Suitable for :—
Total C.	Si.	S.	P.	Mn.	Shrinkage on 12" bar.	Chill. In.	1-in. sq. Bar. Cwt.	Tons per sq. in.	
3·25	3·30	·07	1·10	·50	·135	1/64	24	9·5	Thin section pulleys
3·40	1·60	·09	0·80	·65	·160	1/32	30	12·5	Medium-size machinery castings
3·33	1·10	·10	0·80	·60	·185	1/16	25	13	Heavy work. Good bearing quality
3·25	1·80	·08	0·75	·70	—	—	30	13	Cylinder castings
3·25	3·30	·07	1·10	·50	·135	1/64	34 *	17 *	Permanent mould castings

(*Morgan.*)

ACID-RESISTING IRON CASTINGS.

	English.	American.
Carbon graphitic	·75–1·25	—
Carbon total	—	·20–·60
Silicon	14–15	14–14·5
Sulphur	·05–·15	under ·05
Phosphorus	·05–·10	·16–·20
Manganese	2·0–2·5	·25–·35
Melting point ° F.	2550	2530–2550
Tensile strength	6–7 tons per sq. in.	about 25% less than ordinary cast iron
Compressive strength	6–8 tons per sq. in.	7·0

(*Mason.*)

* Annealed at 800° C. for 2 hours.

STEEL.

(Contributed by Ernest W. Birch, A.M.I.A.E.)

The term 'steel' is applied in a general sense to those alloys of iron and carbon whose carbon is present entirely, or almost entirely, in the combined condition and whose total carbon content does not exceed 2·00 per cent. In a special sense the term is applied to alloys of iron and carbon with elements which are generally in excess of the carbon and whose presence considerably modifies the resulting alloys; such alloys are generally referred to as 'alloy or special steels.'

Steel is always contaminated by some or all of the elements—manganese, silicon, sulphur and phosphorus, and the allowable percentages are given in the following specifications. Sulphur and phosphorus should be kept as low as possible owing to the detrimental effect of these elements.

Carbon renders the steel susceptible to hardening, although below 0·2 per cent. steel shows no appreciable hardening effect when rapidly cooled. As the carbon percentage increases, so does the tenacity and hardness, with a corresponding reduction in the ductility and toughness.

Steel is generally graded according to the percentage of carbon present. A general classification is as follows:

Carbon under 0·10 per cent.	Dead soft steel, very mild or extra mild steel, very low-carbon steel.
Carbon 0·10 to 0·25 per cent.	Mild steel, soft steel, low-carbon steel.
Carbon 0·25 to 0·60 per cent.	Medium-carbon steel, medium high-carbon steel, half-hard steel.
Carbon 0·60 to 1·20 per cent.	High-carbon steel, tool steel, hard steel.
Carbon over 1·20 per cent.	Extra-hard steel, very high-carbon steel.

The classification is not rigid, as no attempt has been made to standardise the various grades. A further classification is made with carbon tool steels, the steels being graded according to their 'temper.' (This term has no connection with the term 'tempering,' used in thermal treatment.) The Temper numbers are arbitrarily chosen by the makers, and the following is an example:

Temper No.	Carbon Content.
	Per cent.
1	1½
2	1¼
3	1⅛
4	1
5	⅞
6	¾

The critical points in a straight carbon steel are dependent on the amount of carbon present, and these critical points control the temperature for forging, annealing, hardening, and tempering. The higher the carbon content the lower the critical temperatures and the more sensitive is the steel. For the before-mentioned steels the recommended working temperatures are as follows:

Carbon Content. Per cent.	Forging Temp. ° C.	Annealing Temp. ° C.	Hardening Temp. ° C.
1½	800	720	760
1¼	825	720	760
1⅛	850	720	770
1	875	720	770
⅞	900	750	785
¾	950	770	800

The tempering temperature depends on requirements. The higher the reheating (tempering) temperature, the greater is the loss in hardness, with a corresponding increase in the ductility (see pp. 1063 and 1070).

Carbon Content.	Suitable for:
Per cent.	
1½	Special turning and planing tools.
1¼	The most suitable for turning and planing tools, twist drills and small cutters.
1⅛	For large turning tools, circular cutters, taps, small punches, reamers, screwing dies, and small shear blades.
1	For hot setts, large punches, large taps, cold chisels, etc.
⅞	For chisels, setts, dies, smiths' tools, large shear blades, masons' tools, etc.
¾	For hammers, general dies, and miners' drills.

Having selected the correct quality steel for the particular purpose, it is essential to give the correct heat treatment in order to obtain the best results. The hardening temperatures are given in the previous table. The following table gives the necessary tempering temperature:

Tempering Temperature, ° C.	Suitable for:
300	Springs.
298	Wood saws.
295	Circular saws for metal.
290	Cold chisels for wrought iron.
285	Moulding and planing cutters for soft wood, needles, cold chisels for cast iron.
280	Firmer chisels, saws for bone and ivory, cold chisels and setts for steel, gimlets.
275	Hot setts, axes, and adzes.
270	Augers, pressing cutters, flat drills for brass.
265	Coopers' tools, wood borers, twist drills, stone-cutting tools.
260	Plane irons, gouges, planing and moulding cutters.
255	Moulding and planing cutters for hard wood, punches and dies, cups, snaps, and shear blades.
250	Penknives, chasers, mill chisels and picks, taps, rock drills, screw-cutting dies.
245	Leather-cutting dies, boring cutters, reamers.
240	Bone-cutting tools, milling cutters, drills, wood engraving tools.
235	Paper cutters, planers for iron.
230	Ivory-cutting tools, planes for steel, screwing dies for brass, hammer faces.
225	Light turning tools, steel engraving tools.
220	Scrapers and lathe tools for brass.

HIGH-SPEED TOOL STEELS.

Type.	Commonly called—
A. Low Tungsten (13 per cent.)	'Twist Drill' quality
B. Medium Tungsten (15 ,,)	'Standard' quality
C. High Tungsten (18 ,,)	'Super' quality

USUAL LIMITS OF CHEMICAL COMPOSITION.

Type.	C.	Mn.	P.	S.	Si.	Cr.	W.	Va.
A. English	0·5–0·7	Max. 0·2	*	*	trace	2·5 –3·0	12·0–14·0	trace
American	0·6–0·75	0·10–0·45	0·05	0·06	0·10–0·45	3·25–4·25	11·0–14·0	1·75–2·25 or 0·5–1·25
B. English	0·55–0·65	Max. 0·2	*	*	trace	2·75–3·5	14·0–16·0	nil to 0·5
American	0·6 –0·75	0·10–0·45	0·05	0·06	0·10–0·45	3·25–4·5	14·0–16·0	0·5–2·25
C. English	0·55–0·65	Max. 0·2	*	*	trace	3·0 –4·0	16·0–18·0	0·5–1·0
American	0·55–0·75	0·10–0·45	0·05	0·06	0·10–0·45	3·0 –4·5	16·0–19·5	0·5–1·25

* As low as possible.

HEAT-TREATMENT.

Type.	Preheat. ° F.	High Temperature. ° F.	Temper. ° F.
A	1,500	2,350	1,000
B	1,500	2,300	1,000
C	1,500	2,300–2,500	1,000

Cobalt High-Speed Steels.

Carbon, 0·58– 0·68
Manganese, 0·10– 0·40
Phosphorus, As low as possible
Sulphur,
Silicon, 0·15– 0·45
Chromium, 2·75– 4·0
Tungsten, 13·0 –18·0
Vanadium, 1·0 – 1·6
Cobalt, 1·75– 4·25

Heat-Treatment.
Preheat, 1,500° F.
High heat, 2,350° F.
Temper, 1,000° F.

Molybdenum High-Speed Steel.

Carbon, 0·65
Manganese, 0·23
Phosphorus, 0·02
Sulphur, 0·02
Silicon, 0·42
Chromium, 3·55
Tungsten, 16·0
Vanadium, 1·1
Molybdenum, 0·71

Preheat, 1,500° F.
High heat, 2,350° F.
Temper, 1,050° F.

Tungstenless High-Speed Steel.

Carbon, 0·65
Manganese, 0·55
Phosphorus, 0·027
Sulphur, 0·044
Silicon, 0·48
Chromium, 4·62
Vanadium, 0·68
Cobalt, 4·73
Molybdenum, 4·72

Preheat, 1,500° F.
High heat, 2,000° F.
Temper, 900° F.

CARBON AND ALLOY STEEL SPECIFICATIONS.

The following data are based upon B.E.S.A. Report 5005–1924 and upon the very complete specifications prepared by the S.A.E. (Society of Automotive Engineers, U.S.A.).

The range of steels given covers practically all general requirements. The selection of a given steel for a given part must depend upon an intimate knowledge of a number of important factors, such as the detailed design of the part and the severity of the service to be imposed, whether the part is to be machined or forged, machinability, and the method of manufacture. From the data given, a steel can be selected from a consideration of physical properties, after which the final choice depends upon such considerations as machinability, heat-treatment, etc.

CARBON STEELS.

Reference No.	Carbon Range.	Manganese Range.	Phosphorus. Max.	Sulphur. Max.
1	0·05–0·15	0·30–0·60	0·045	0·05
2	0·10–0·20	0·30–0·60	0·045	0·05
3	0·15–0·25	0·30–0·60	0·045	0·05
4	0·20–0·30	0·50–0·80	0·045	0·05
5	0·25–0·35	0·50–0·80	0·045	0·05
6	0·30–0·40	0·50–0·80	0·045	0·05
7	0·35–0·45	0·50–0·80	0·045	0·05
8	0·40–0·50	0·50–0·80	0·045	0·05
9	0·45–0·55	0·50–0·80	0·045	0·05
10	0·90–1·05	0·25–0·50	0·045	0·05
11	0·45–0·55	0·90–1·20	0·045	0·05
12	0·55–0·70	0·90–1·20	0·045	0·05
13	0·08–0·16	0·60–0·80	0·09–0·13	0·075–0·15
14	0·15–0·25	0·60–0·90	0·06– max.	0·075–0·15

Steels Nos. 1 and 2.—Used for tubing and pressings; they are soft and ductile and will stand much deformation without cracking. Low tensile strength in annealed condition, but the yield point can be raised considerably by cold-working. They do not machine freely and will tear badly in turning, threading, or broaching. Machining properties can be improved by quenching in water from 925° C. Suitable for case-hardening.

Steel No. 3.—Forges well and machines better than No. 1 and 2. Can be used for machined forged, and case-hardened parts where no great strength is required. Can be drawn into tubes and cold rolled, and for cold-worked shapes that do not require deep drawing. When cold worked, yield point 18–35 tons per sq. in. in rounds not over $\frac{1}{2}$ in. diam. or in flats $\frac{1}{4}$ in. thick. Heat treatment will refine the grain after rolling or forging. Particularly suitable for case-hardening.

Steels Nos. 4 and 5.—Suitable for forged, machined, or cold-worked parts requiring higher physical properties than No. 2, also more responsive to heat-treatment and machines better. These steels should not be used for case-hardened parts.

Steels Nos. 6 and 7.—Medium carbon steels possessing good machining properties, and are suitable for small and medium-size forgings. After heat-treatment Brinell range 174–212.

Steels Nos. 8 and 9.—Medium carbon steels suitable for large forgings. Can also be used for a wide range of parts from bar stock. After heat-treatment Brinell range 192–235.

Steel No. 10.—Suitable for leaf springs and certain coiled springs. Heat-treatment: heat to 760–780° C., quench in oil, water, or brine, depending on size and shape of part. Draw to required hardness.

Steels Nos. 11 and 12.—Suitable for helical springs made of round cold drawn wire up to $\frac{3}{16}$ in. diam. No. 11 steel is recommended for wire from 0·025 to 0·100 in. diam., and steel No. 12 for wire 0·100 in. diam. and larger.

Steels Nos. 13 and 14.—Suitable for 'screw stock'; they machine very sweetly and take a good finish, but owing to cold-shortness they are not suitable for parts subject to shock.

STRAIGHT NICKEL STEELS.

Reference No.	Carbon Range.	Manganese Range.	Nickel Range.
15	0·10–0·20	0·30–0·60	0·40–0·60
16	0·10–0·20	0·30–0·60	1·25–1·75
17	0·10–0·20	0·30–0·60	3·25–3·75
18	0·15–0·25	0·50–0·80	3·25–3·75
19	0·25–0·35	0·50–0·80	3·00–3·50
20	0·30–0·40	0·50–0·80	3·00–3·50
21	0·35–0·45	0·50–0·80	3·00–3·50
22	0·40–0·50	0·50–0·80	3·00–3·50
23	0·45–0·55	0·50–0·80	3·00–3·50
24	0·17 max.	0·30–0·60	4·75–5·25

In all cases sulphur max.: 0·045; phosphorus max., 0·040; silicon, 0·15–0·40.

Steels Nos. 15, 16, and 17 are suitable for case-hardening.

Steel No. 18 not recommended for case-hardening except for massive parts, owing to the reduced ductility of the core due to the increased carbon content.

Steels Nos. 19 and 20 are suitable for parts requiring considerable strength and toughness. Not suitable for case-hardening.

Steels Nos. 21 and 22. As the carbon content is higher than for steels Nos. 19 and 20, higher tensile strength and greater hardness can be obtained after treatment, but with a loss in ductile properties.

Steel No. 23 is suitable for large gears, etc.

Steel No. 24 is used for case-hardened parts that require an exceptionally tough core. When it is necessary to machine parts made of this steel after carburising, the nickel content should be kept as near the lower limit as possible.

NICKEL CHROME STEELS.

Reference No.	Carbon Range.	Manganese Range.	Nickel Range.	Chromium Range.
25	0·10–0·20	0·30–0·60	1·00–1·50	0·45–0·75
26	0·20–0·30	0·50–0·80	1·00–1·50	0·45–0·75
27	0·35–0·45	0·30–0·60	1·50–2·00	0·90–1·25
28	0·17 max.	0·30–0·60	3·00–3·50	1·00–1·50
29	0·20–0·30	0·30–0·60	2·75–3·50	0·50–1·00
30	0·35–0·45	0·30–0·60	3·25–3·75	1·25–1·75
31	0·10–0·20	0·30–0·60	2·75–3·25	0·60–0·95
32	0·45–0·55	0·30–0·60	2·75–3·25	0·60–0·95

In all cases sulphur max.= 0·04; phosphorus max. = 0·04; silicon = 0·15–0·40.

In general these steels give higher tensile values and greater toughness than the corresponding straight nickel steels. They are, however, more sensitive to heat treatment.

Steel No. 25 is primarily intended for case-hardening, but can also be used for structural parts with suitable heat treatment. Where these steels are used for gears where considerable strength and a high degree of accuracy are called for, the gears should be normalised before carburising.

Steels Nos. 28 and 31 are suitable for case-hardening when a core possessing great strength and toughness is required.

The remaining steels are suitable for such heat-treated parts, either machined or forged, as require exceptionally high physical properties, and must be capable of resisting severe dynamic stresses in service.

MOLYBDENUM STEELS.

Reference No.	Carbon Range.	Manganese Range.	Chromium Range.	Nickel Range.	Molybdenum Range.
33	0·25–0·35	0·40–0·70	0·50–0·80	—	0·15–0·25
34	0·35–0·45	0·40–0·70	0·80–1·10	—	0·15–0·25
35	0·45–0·55	0·40–0·70	0·80–1·10	—	0·15–0·25
36	0·10–0·20	0·30–0·60	—	1·5–2·0	0·20–0·30

In all cases sulphur max. = 0·045; phosphorus max. = 0·040.

In general these steels have similar physical characteristics to the corresponding nickel and nickel-chrome steels. Steel No. 36 is suitable for case-hardening.

CHROMIUM STEELS.

Reference No.	Carbon Range.	Manganese Range	Chromium Range.	Sulphur Max.	Phosphorus Max.
37	0·15–0·25	0·30–0·60	0·60–0·90	0·045	0·04
38	0·35–0·45	0·50–0·80	0·80–1·10	0·045	0·04
39	0·45–0·55	0·50–0·80	0·80–1·10	0·045	0·04
40	0·95–1·10	0·20–0·50	1·20–1·50	0·030	0·03

Steel No. 37. This steel is primarily for case-hardening. For structural purposes the chromium content of this steel gives a deeper penetration of the effect of heat treatment than can be obtained in a straight carbon steel having a similar carbon content.

Steel No. 39 is an oil-hardening type of steel intended for heat-treated forgings that require greater strength and toughness than are obtainable with straight carbon steels, or with some of the simple alloy steels having a lower carbon content. It also has a wider heat-treating range than the other simple alloy steels.

Steel No. 40 is used chiefly for the races and balls or rollers of anti-friction bearings. The chromium and carbon contents cause a maximum penetration of the effect of heat treatment and give a high degree of hardness.

CHROME VANADIUM STEELS.

Reference No.	Carbon Range.	Manganese Range.	Chromium Range.	Vanadium.	
				Min.	Desired.
41	0·15–0·25	0·50–0·80	0·80–1·10	0·15	0·18
42	0·25–0·35	0·50–0·80	0·80–1·10	0·15	0·18
43	0·35–0·45	0·50–0·80	0·80–1·10	0·15	0·18
44	0·45–0·55	0·50–0·80	0·80–1·10	0·15	0·18
45	0·90–1·05	0·20–0·45	0·80–1·10	0·15	0·18

Steel No. 41 is intended primarily for case-hardening and can be used for structural parts after suitable heat treatment.

Steels Nos. 42 and 43 These steels can be used in place of the nickel and nickel-chrome steels of similar carbon content for heat-treated forgings requiring greater strength, toughness and resistance to fatigue than it is possible to obtain with straight carbon steels.

Steel No. 45 can be used for anti-friction bearings and is used extensively for machine-tool parts, being primarily a tool steel.

SILICO MANGANESE STEELS.

Reference No.	Carbon Range.	Manganese Range.	Silicon Range.
46	0·45–0·55	0·60–0·90	1·80–2·20
47	0·55–0·65	0·60–0·90	1·80–2·20

Sulphur and phosphorus = 0·045 max.

These steels are essentially spring steels, but may be used for gears. In either case the steel must be suitably heat-treated.

HEAT TREATMENT OF STEELS.

In order to obtain the desired physical properties from steels it is generally necessary to subject them to a thermal treatment. The subject of heat treatment embraces the operations of (*a*) normalising, (*b*) annealing, (*c*) hardening, (*d*) cementing, carburising or case-hardening, (*e*) the tempering or drawing of hardened steels.

The nomenclature and definitions relating to heat treatment as given by B.E.S.A. Report No. 5005–1924 are as follows :

(*a*) Normalising. Normalising means heating a steel (however previously treated) to a temperature exceeding its upper critical range, and allowing it to cool freely in air. It is desirable that the temperature of the steel shall be maintained for about 15 minutes, and shall not exceed the upper limit of the critical range by more than 50° Centrigrade.

(*b*) Annealing. Annealing means reheating followed by slow cooling. Its purposes may be

(1) To remove internal stresses or to induce softness, in which case the maximum temperature may be arbitrarily chosen.

(2 To refine the crystalline structure in addition to the above, in which case the temperature used must exceed the upper critical range as in normalising.

(*c*) Hardening. Hardening means heating a steel to its normalising temperature, and cooling more or less rapidly in a suitable medium, *e.g.* water, oil or air.

(*d*) Cementing. Cementing means heating a steel above its normalising temperature in a medium which will increase its carbon content. The core of a case-hardened bar is the interior portion of the bar which is substantially unaffected by the cementing process.

(*e*) Tempering. Tempering means heating a steel (however previously hardened) to a temperature below its carbon change-point with the object of reducing the hardness or increasing the toughness to a greater or less degree.

(*f*) Refining (cemented parts). Refining means reheating a steel to its normalising temperature, and is usually followed by quenching.

See also p. 2758.

Effect of Tempering.

3½ per cent. nickel steel, oil-hardened at 860° C. and tempered at the temperatures indicated.

Tempered at. ° C.	Yield Point. Tons per sq. in.	Ultimate Stress. Tons per sq. in.	Elongation. %.	Reduction of Area. %.	Impact. Test. Ft.-lbs.	Brinell No.
Not tempered	102·0	116·0	10·0	23·0	6	514
200	92·5	101·0	10·5	32·5	5	461
300	83·0	92·5	12·0	40·0	5	415
400	66·4	76·3	15·1	50·0	11	352
500	54·5	63·5	20·0	59·5	50	285
600	44·0	53·0	25·0	64·5	73	241
650	40·0	49·0	27·0	66·0	82	235

Effect of Mass.

The effect of mass must receive serious consideration when dealing with the physical properties of any heat-treated steel.

The hardening operation consists of heating the steel through and just above its critical range in order to obtain the condition of solid solution, and quenching with sufficient rapidity to retain this condition. With large masses it is practically impossible to obtain a uniform temperature throughout the mass on heating, the interior portion always being at a lower temperature than

the outside. On quenching the heat is absorbed from the outside, and in many cases it is impossible, even with the most drastic quenching, to remove the heat sufficiently rapidly to retain the desired structure in the core. Again, with very thin sections, the rate of cooling may be sufficiently rapid if carried out in air. Only by experience can the effect of mass be correctly estimated.

The following table shows the effect of mass on the physical properties of a 5 per cent. nickel case-hardening steel. Bars of varying diameters were given exactly the same treatment and then tested :

Size of Bar. In.	Heat Treatment.	Yield Point. Tons per sq. in.	Ultimate Stress. Tons per sq. in.	Elongation. %.	Reduction of Area. %.	Impact. Test. Ft.-lbs.	Brinell No.
½	Carburised at 860° C. Reheated to 860° C. Quenched in water Reheated to 750° C. Quenched in water	40·0	51·0	27·0	66·1	55	235
¾		34·5	44·6	31·0	68·3	90	212
1		31·0	40·0	35·0	70·0	105	197
1¼		27·8	37·5	36·7	72·2	111	179
1½		26·7	36·2	36·8	73·5	111	174
1¾		26·5	36·1	37·0	73·5	110	174
2		26·2	36·0	37·0	72·5	104	174
2¼		26·0	36·0	37·0	72·0	90	170

Quenching Liquids.

Liquid quenching mediums include water, brine, oils, and special liquids. Soft water, distilled if possible, is used for hardening ordinary carbon steels. Impurities in the water, such as grease or certain acids, are objectionable, as the former are liable to cause uneven hardness by insulating the steel locally with an oil film, and the acids produce brittleness. Hard water is very unsatisfactory because of the scale thrown down when its temperature is raised. Water to which soap or salt has been added is sometimes used to secure quenching rates respectively lower or higher than that of pure water. Cold sea-water or brine produces extreme rapidity of quenching and consequently maximum hardness. Pieces of complicated design cannot be quenched safely in this medium because of the shock.

Oil-quenching mediums are extensively used where extreme hardness is not required, and where freedom from quenching shock is necessary. These include mineral oils, paraffin, fish oil, whale oil, cotton-seed oil, linseed oil, lard oil, and special animal hydrocarbon oils.

Mineral oils are satisfactory where the quenching bath is kept cool by artificial means. If the oil is not cooled there occurs a gradual breaking down and thickening of the oil with heat, the formation of residue, and wide variation in the quenching rate. Paraffin is dangerous unless kept at a temperature well below its flash point.

Fish and whale oils have the disadvantage of offensive odours and are open to the same objections as the seed oils. Cotton-seed and linseed oils become gummy from oxidation. This increases the viscosity and seriously affects the quenching speed. Lard oil is unsatisfactory because of the tendency to become rancid.

Special oils usually obtained by distillation of wool grease are used extensively with satisfactory results. Artificial cooling of these oils is very necessary when handling large quantities of metal continuously.

Carbonate of lime, mercury, milk, glycerine, soda, tallow, and wax are sometimes used for special cases.

(*Millett.*)

Spring Steels.

(*Lea and Heywood.*)

Steel A. Basic. Carbon, 0·69 ; manganese, 0·56. Tempered only.
B. Crucible cast steel. Carbon, 1·23 ; manganese, 0·28. Quenched and tempered.
C. Carbon, 1·35 ; manganese, 0·27. Quenched and tempered.
D. Carbon, 0·84 ; manganese 0·55. Tempered only.
E. Crucible cast steel. Carbon, 1·63 ; manganese, 0·32. Quenched and tempered.
F. Carbon, 1·17 ; manganese, 0·33. Quenched and tempered.

SPRING STEELS.

Steel.	A.	B.	C.	D.	E.	F.
No. of coils	23	31·5	31·5	31·5	11·5	11·5
Mean diam. of coils, ins.	0·607	0·611	0·610	0·620	1·12	1·117
Mean diam. of wire, ins.	0·074	0·071	0·071	0·074	0·115	0·115
Modulus of rigidity. $\times 10^6$ lbs. per sq. in. (mean value)	11·97	10·65	10·56	11·69	11·35	10·95
Ultimate tensile strength. Tons per sq. in.	73·7	88·3	98·5	111·0	107·5	94·0
Ultimate tensile strength. Tons per sq. in. Normalised	31·7	31·5	29·8	42·5	40·5	40·5
Proportional limit in torsion. Tons per sq. in.	24·2	45·2	33·5	34·0	—	22·8
Ultimate strength in torsion. Tons per sq. in.	—	83·6	—	75·4	103·5	—
Brinell number	432	518	496	472	533	492

ANALYSIS OF VARIOUS STEELS.

Type of Steel.	Analysis.								
	C.	Si.	S.	P.	Mn.	Ni.	Cr.	Va.	W.
Boiler plates	0·22	0·32	0·028	0·03	0·52	—	—	—	—
40-ton forging	0·36	0·41	0·035	0·032	0·80	—	—	—	—
Axle steel	0·28	0·28	0·030	0·026	0·90	—	—	—	—
Rails (railway)	0·48	0·36	0·038	0·036	0·72	—	—	—	—
,, ,,	0·59	0·32	0·052	0·023	0·75	—	—	—	—
Tyres, loco.	0·62	0·28	0·02	0·04	0·55	—	—	—	—
,, waggon	0·38	0·19	0·05	0·04	1·20	—	—	—	—
Turbine disc	0·30	0·20	0·05	0·034	0·83	—	—		—
High-tensile carbon steel	0·45	0·26	0·03	0·038	0·56	—	—	—	—
30-ton carbon steel	0·32	0·30	0·048	0·042	0·68	—	—	—	—
40-ton ,, ,,	0·51	0·33	0·040	0·036	0·59	—	—	—	—
Steel tubes for welding	0·20	0·42	0·039	0·027	0·51	—	—	—	—
Piano wire	0·60	0·10	0·022	0·026	0·45	—	—	—	—
Plough steel (for wire ropes)	0·85	0·14	0·01	0·01	0·60	—	—	—	—
Plough steel (for wire ropes)	0·58	0·16	0·032	0·033	0·41	—	—	—	—
Steel rolls	1·08	0·27	0·02	0·025	0·39	—	1·88	—	—
File steel	0·90	0·10	0·01	0·02	0·30	—	—	—	—
,, ,,	1·40	0·25	0·04	0·03	0·80	—	—	—	—
Carbon spring steel	0·89	0·22	0·022	0·024	0·43	—	—	—	—
Silico ,, ,,	0·50	1·72	0·038	0·033	0·68	—	—	—	—
Chrome ,, ,,	0·61	0·40	0·030	0·029	0·79	—	—	—	—
Silico-chrome spring steel	0·53	1·06	0·031	0·022	0·35	—	0·78	—	—
Chrome - vanadium spring steel	0·50	0·36	0·035	0·035	0·60	—	1·0	0·16	—
Silico-manganese spring steel	0·50	1·95	0·04	0·04	0·70	—	—	—	—
Magnet steel	0·82	0·28	0·012	0·014	0·31	—	—	—	5·3
Non-corrodible nickel steel	0·32	0·38	0·026	0·021	0·26	26·7	—	—	—
Nickel-vanadium steel	0·24	0·51	0·035	0·041	0·70	3·5	—	0·15	—
Chrome - vanadium forging	0·26	0·44	0·04	0·038	0·50	—	1·0	0·18	—
High-speed steel	0·72	0·17	0·022	0·018	0·21	—	4·4	—	14·3
Ball-bearing steel	1·12	0·015	0·017	0·019	0·19	—	0·25	—	—
,, ,,	0·95	0·014	0·018	0·019	0·25	—	1·25	—	—
Chrome exhaust valve steel	1·75	0·58	0·048	0·013	1·10	—	10·85	—	—
Nickel-chrome tube	0·30	0·16	0·03	0·03	0·45	1·10	4·0	—	—
Exhaust valve steel	0·37	0·10	0·03	0·02	0·35	—	2·5	0·32	16·53
,, ,, ,,	0·78	0·31	0·03	0·02	0·20	—	2·24	1·03	19·75
Chrome-molybdenum steel	0·40	0·35	0·04	0·032	0·41	—	1·0	0·35	—

Test Results Obtained from Heat-Treated Steels.

Abbreviations used: N = normalised at; WQ = water-quenched from; OQ = oil-quenched from; YP = stress at yield point, tons per sq. in.; US = ultimate stress, tons per sq. in.; E % = elongation per cent. on 2 ins.; R/A % = reduction of area, %; I = impact test, foot-pounds; B = Brinell No.

All tests made on bars 1⅛ in. diameter.

Type of Steel.	Analysis.							Heat Treatment.	Physical Properties.						Remarks.
	C.	Si.	Mn.	S.	P.	Ni.	Cr.		YP.	US.	E %.	R/A %.	I.	B.	
Dead mild case-hardening	0·06	0·01	0·25	0·01	0·05	—	—	N 920° C.	19·0	26·0	41·0	71·0	36	121	
								WQ 920° C.	—	35·0	30·0	67·0	50	167	
								OQ 920° C.	—	30·0	35·0	69·0	90	137	
								Cemented, WQ 920° C.; reheated, WQ 760° C.	—	33·0	33·0	68·0	—	—	Core test
Mild case-hardening	0·17	0·24	0·72	0·041	0·047	—	—	N 920° C.	20·0	31·0	38·0	64·0	80	137	
								WQ 920° C.	—	46·0	22·0	51·0	24	223	
								OQ 920° C.	—	36·0	31·0	65·0	92	170	
								Cemented, WQ 920° C.; reheated, WQ 760° C.	—	40·0	32·0	64·0	43	183	Core test
2% Nickel case-hardening	0·15	0·05	0·61	0·017	0·012	2·60	—	N 880° C.	24·0	32·0	37·0	69·0	95	143	
								WQ 880° C.	—	50·0	23·0	60·0	51	241	
								OQ 880° C.	—	38·0	30·0	70·0	100	179	
								Cemented WQ 880° C.; reheated, WQ 760° C.	—	44·0	28·0	61·0	84	202	Core test
6% Nickel case-hardening	0·17	0·39	0·34	0·014	0·014	6·0	—	N 860° C.	27·0	42·0	33·0	60·0	77	187	
								WQ 860° C.	—	87·0	15·0	49·0	31	388	
								OQ 860° C.	—	82·0	15·0	43·0	26	375	
								Cemented, WQ 860° C.; reheated, WQ 750° C.	—	75·0	17·0	48·0	35	331	Core test

5% Nickel case-hardening	0·07	0·06	0·18	0·02	0·008	5·1	—	N 860° C.	26·0	31·0	39·0	72·0	104	143	—
								WQ 860° C.	—	42·0	25·0	70·0	87	201	—
								OQ 860° C.	—	35·0	34·0	75·0	112	179	—
Mild carbon	0·26	0·30	0·75	0·044	0·05	0·15	—	N 900° C.	24·5	35·5	32·0	59·0	46	159	—
								WQ 900° C.	—	57·0	11·0	30·0	13	—	—
								OQ 900° C.	—	44·0	25·0	55·0	31	207	—
Medium carbon	0·45	0·32	0·78	0·02	0·025	—	—	N 870° C.	27·0	44·0	27·0	54·5	31	192	—
								WQ 870° C.	47·0	67·0	12·0	28·0	14	321	—
								OQ 870° C.	36·0	56·0	20·0	49·0	27	—	—
3% Nickel	0·17	0·05	0·34	0·033	0·023	2·9	0·2	N 860° C.	24·0	32·0	36·0	64·0	89	140	—
								WQ 860° C.	41·0	60·0	15·0	37·0	29	286	—
								OQ 860 C.	26·0	42·0	26·0	61·0	64	192	—
3½% Nickel	0·37	0·17	0·65	0·02	0·03	3·65	—	N 860° C.	27·0	44·0	28·0	56·0	36	196	—
								WQ 860° C.	—	124·0	9·0	25·0	4	514	—
								OQ 860° C.	—	116·0	10·0	22·0	6	495	—
1¼% Nickel-chrome	0·28	0·08	0·40	0·02	0·018	1·52	0·80	N 850° C.	29·0	40·0	33·0	65·0	74	174	—
								OQ 850° C.	45·0	64·0	16·0	34·0	17	302	—
3% Nickel-chrome	0·31	0·14	0·70	0·027	0·030	3·27	0·82	N 820° C.	—	76·0	15·0	40·0	6	341	—
								OQ 820° C.	—	116·0	14·0	37·0	11	495	—
Air hardening nickel-chrome	0·32	0·12	0·46	0·017	0·014	3·70	1·42	Air hardened from 820° C.	80·0	115·0	13·0	40·0	18	477	—
								OQ 820° C.	87·0	91·0	13·0	52·0	17	418	—

The figures given after water- or oil-quenching represent the hardened condition. Intermediate values, ranging from the values given for the normalised condition to the hardened condition, are obtained by tempering, as shown in the table on p. 2697, which gives the values for a 3½ % nickel steel.

CHISEL STEELS.

Class of Steel.	Analysis. C.	Si.	S. Max.	P. Max.	Mn.	Heat-Treatment.	Remarks.
High Carbon	·6–·65	·15 max.	·04	·04	·65–·8	Oil or water quench 830° C., temper selectively	Cheap quality. Gives good service for ordinary grade work
Extra high Carbon	·7–·85	·15 max.	·04	·04	·55–·70	Oil quench 815° C.	Cheap quality. Gives good service for ordinary grade work. Harder and lasts longer
Nickel Chrome— Ni 1–1·5 Cr 0·4–0·6	·45–·55	·15 max.	·04	·04	·55–·70	Oil quench 840° C.	Good and tough for ordinary work. Not very hard
High Carbon Tungsten W 1–1·5	·7–·8	·15 max.	·035	·035	·55–·70	Water quench 805° C.	Very good. Hard and tough. Gives long life for difficult work
Silicon steel	·5–·6	1·2–1·6	·035	·035	·4–·8	Water quench 860° C.	Good
High Speed— Cr 3–3·5 W 15–18 Va 0·6–1·0	·6–·7	·15	·035	·035	—	Harden as for cutting tools	Good for chipping very hard metal
Tungsten Chrome— Cr 1·8–2·5 W 1·8–2·5 Va 0·3–0·5	·4–·5	·15 max.	·035	·035	—	Oil harden 870° C., or water harden 820° C. May be used untempered for very hard work	Excellent. Can be made very hard but tough. Gives long life under difficult conditions

(*Machinery.*)

EFFECT OF NICKEL IN STEEL.

In the annealed condition the average effect of nickel in amounts up to 8 per cent. may be stated as follows:—

Each 1 per cent. of nickel:—

Increases the elastic limit by 1·8 tons per sq. in.
" " tensile strength by 1·9 tons per sq. in.
" " reduction of area by 0·5 per cent.
Decreases the elongation by 1·0 per cent.

For nickel steel with up to 5 per cent. nickel each 1 per cent. of nickel lowers the Ar_1 critical point from 10°–20° C. below those of a corresponding straight carbon steel, and the Ac_3 point from 8°–14° C. The lower the carbon content the greater is the depression of both Ac_3 and Ar_1. With a nickel content of 32–33 per cent. the $Ar_{3\cdot2}$ point is apparently depressed to absolute zero. The addition of nickel also reduces the eutectoid carbon ratio from ·9 per cent. for a straight carbon steel to about ·75 per cent. for a 3½ per cent. nickel steel, and about ·7 per cent. for a 5 per cent. nickel steel.

In the heat-treated condition, the tensile strength, yield point, and hardness are raised with a corresponding reduction in the ductility as compared with a carbon steel. Nickel tends to retard grain growth, thus allowing greater latitude in treating; and a nickel steel may be held above the critical range for longer periods of time without serious injury. On slow cooling, the tendency is to maintain a fine-grained pearlitic or sorbitic structure instead of a full lamellar pearlite.

Nickel further improves the endurance ratio, and this is most marked in the heat-treated condition. Steel with a high percentage of nickel (austenitic nickel steels) have excellent corrosion-resisting properties and resistance to oxidation at elevated temperatures, and are practically non-magnetic.

(*Bureau of Information on Nickel.*)

Effect of Chromium in Steel.

Chromium differs from that of nickel in that it forms a double carbide, resulting in greater strength and hardness in a heat-treated steel. By the use of nickel and chromium, the nickel strengthens the ferrite matrix and the chromium strengthens the carbide constituent, giving superior physical properties than when either element is used alone. The presence of nickel and chromium ensures considerable hardening, with a rate of cooling through the critical range slower than that permissible in the case of a steel with a similar content of either element. This makes nickel-chrome steels especially suited for large sections requiring heat-treatment; giving them deep and uniform hardening power. In carbon steel chromium raises fairly uniformly both the Ac_1 and Ac_3 points. In nickel steel, however, the effect is somewhat different. The Ac_1 point in uniformly raised, but according to available data the Ac_3 point, when the carbon is below ·3 per cent., seems to be actually lower than in a steel of equivalent nickel content, while with higher carbon the Ac_3 point is higher than in an equivalent steel.

Invar.*

Invar is a nickel steel containing about 36 per cent. nickel, together with about 0·5 per cent. each of carbon and manganese, with metallurgically negligible quantities of sulphur, phosphorus and other elements. It is made either in the open-hearth furnace or by the crucible method. It melts sharply at 1425° C. Above some 200° C. to its melting point, invar may be considered to consist of a homogeneous solid solution of iron, nickel and carbon. Below 200° C. and at a temperature dependent on its history and exact composition it undergoes a reversible transformation of such a nature that for any sample the transformation may be incomplete. This condition of thermo-chemical instability gives rise to both slowly changing and quickly changing values of its physical properties—changes which are particularly manifested in the expansion.

Invar can be forged, rolled, turned, filed and drawn into wires, and it takes a beautiful polish, giving an excellent surface on which fine lines may be ruled. In general it should be worked slowly. It will withstand without spotting the corrosive action of water. Its density is about 8·0 grm. per cm.^{3}, its electrical resistivity is of the order of 80 microhm per cm., or about eight times that of pure iron, and its temperature coefficient of electrical resistance about 0·0012 per degree centigrade. It is ferromagnetic, but becomes paramagnetic in the neighbourhood of 165° C.

The mean coefficient of linear expansion between 0 and 40° C. is in the order of one millionth for the ordinary invar, and samples have been prepared with even small negative coefficients; the amounts of carbon and manganese present appear to exercise considerable influence on the expansion. Above 200° C. the expansion of invar is approximately that of Bessemer steel.

Invar is subject to changes in length due to 'after effects' following cooling from a high temperature, and to changes in length following even slight alterations in temperature. Invar also gradually elongates with time, forged and drawn material behaving somewhat differently in this respect, so that there is a determinable, seasonal correction to be applied to its length when used as a length standard. It also shows marked magnetostriction phenomena or changes of length accompanying changes in strong magnetic fields.

Mechanical properties: Tensile strength, 22·5–36 tons per sq. in.; elastic limit, 3·1–13·5 tons per sq. in.; elongation, 40–50 per cent.; reduction of area, 40–65 per cent.; Brinell hardness, 160; scleroscope hardness, 19; modulus of elasticity, 22,500,000 lbs. per sq. in.

Expansion of Nickel Steels (Guillaume).

Mean coefficients of linear expansion between 0° and t° C. Applicable between 0° and 38° C.

Per Cent. Nickel.	Mean Coefficient of Linear Expansion × 10^6.
30·4	$4·570 + 0·01194\,t$
31·4	$3·395 + 0·00885\,t$
34·6	$1·373 + 0·00237\,t$
35·6	$0·877 + 0·00127\,t$
37·3	$3·457 - 0·00647\,t$
34·8 + 1·5 Cr.	$3·580 - 0·00132\,t$
35·7 + 1·7 Cr.	$3·373 + 0·00165\,t$
36·4 + 0·9 Cr.	$4·433 - 0·00392\,t$

* Extract from Circular No. 58, Bureau of Standards.

Coefficient of Expansion of Steels at Elevated Temperatures.

Average Coefficient of Expansion on Heating, $\times 10^{-6}$.

Steel.	Temperature Range. ° C.					
	25–100.	100–200.	200–300.	300–400.	400–500.	500–600.
A	11·1	12·2	14·3	15·8	15·7	16·0
B	11·0	12·3	14·8	15·6	15·9	16·5
C	10·8	11·7	13·5	14·0	14·5	14·4
D	10·4	12·1	13·7	15·9	15·7	16·1
E	10·0	10·6	12·0	12·6	13·5	13·9
F	3·7	8·4	14·1	16·6	18·4	18·8
G	11·0	11·6	13·6	15·1	16·3	16·1

Analysis:

A. C 0·41, Mn 0·64.
B. C 0·35, Mn 0·08, Cr 1·17, Va 0·14.
C. C 0·168, Ni 3·94, Cr 2·5, Va 0·39.
D. C 0·512, Mn 0·42, W 1·58.
E. C 0·3–0·4, Cr 13·0.
F. C 0·14, Ni 34·52.
G. C 1·28, Mn 0·37, Cr 0·19.

(*Bureau of Standards, Circular No.* 433.)

Cobaltron Steel Alloy (Patent).

This alloy (patent) introduced by Messrs. Darwin and Milner, of Sheffield, is made by adding cobalt to chromium-carbon steel. The maximum temperature necessary for the hardening is 1000° C. (tungsten steel requires from 1250° to 1350° for the best results) and hardening is said to be satisfactory when the steel is allowed to cool naturally in still air. Analysis of cobaltron shows that the following elements approximate to the amounts stated below:—

Carbon	About 1·5 per cent.	Molybdenum	About 1·25 per cent.
Cobalt	,, 2·25 ,,	Tungsten	,, 0·25 ,,
Chromium	,, 11·0 ,,		

'Heenan' Patent Oil Coolers for Heat Treatment of Steel.

The continuous introduction of hot steel into oil quenching tanks naturally heats up the oil, which retains heat for a long time, and rapidly becomes too hot to permit of the process being carried on.

Owing to the poor heat conductivity of oil, such arrangements as water-jacketing the tank or inserting pipe coils through which water is circulated are ineffective, the cool surfaces affecting only that portion of the oil in immediate contact therewith, while allowing the main body to remain over-heated.

The 'Heenan' system of oil cooling is to extract the hot oil by means of a suitable centrifugal pump, and pass it through a 'Heenan' oil cooler before returning it to the quenching tank. This oil cooler is built up on lines similar to those of the water cooler described on p. 1232. In effect, oil is spread over a large cooling surface with which cool air from the surrounding atmosphere is then drawn into contact. Circulating water is not used.

This machine has been applied amongst other purposes to quenching processes in connection with the hardening and tempering of gun tubes, rifle barrels, steel tools, gear wheels, springs, aeroplane engine parts, etc.

Chromium Plating of Steel.

Using solutions of 3 grams per litre chromium sulphate and 200–400 grams per litre chromic acid, good adherent deposits of chromium on iron can be obtained. The cathode current density range for good deposits should be from 100–150 amperes per sq. ft.

Best results are obtained using Sargent's solution of 3 grams per litre chromium sulphate, 245 grams per litre chromic acid, and a current density of 125 amperes per sq. ft., the anodes being of chromium.

Chromium-plated iron and steel have a hard, bright, silver-like surface, resist atmospheric corrosion indefinitely, and are resistant to the action of fumes of nitric acid, hydrogen sulphide, and ammonia. They also resist the corrosive action of molten tin, zinc, and brass. Steel does not lose its temper during chromium plating.

(See also p. 2719.) (*Chemical and Metallurgical Engineering, April* 21, 1924.)

STRENGTH OF VARIOUS METALS AT HIGH TEMPERATURES.

(*G. & J. Weir, Ltd.*)

Temperature. °C.	Ultimate Tensile Strength. Tons per sq. in.				
	Mild Steel.	Naval Brass.	Manganese Brass.	Hot-rolled Monel Metal.	Cast Monel Metal.
15	31·0	26·0	29·0	42·5	24·0
100	32·0	26·5	27·5	42·0	23·5
200	33·5	23·4	25·5	40·0	22·0
300	33·6	18·0	13·0	38·0	20·0
400	27·5	6·5	4·5	36·0	18·0
500	15·0	2·5	—	30·0	16·0
600	9·0	—	—	20·0	12·0
700	4·0	—	—	12·0	8·0
800	—	—	—	6·5	—

STRENGTH OF CARBON AND NICKEL STEEL BOILER PLATES AT ELEVATED TEMPERATURES.

(*Pomp and Körber.*)

Steel.	Temperature.							
	200° C.		300° C.		400° C.		500° C.	
	Y.P.	U.S.	Y.P.	U.S.	Y.P.	U.S.	Y.P.	U.S.
Carbon steel 0·06 C	10·2	26·6	7·0	26·0	7·0	18·4	5·2	10·8
,, ,, 0·13 C	12·7	29·8	9·5	28·6	8·3	22·3	5·7	13·4
,, ,, 0·27 C	13·3	32·4	10·8	31·7	10·8	24·2	5·7	15·8
Nickel steel . .	19·7	33·6	17·1	35·5	13·3	26·6	9·5	17·8

Y.P. = Yield point, tons per sq. in.
U.S. = Ultimate strength, tons per sq. in.

CASE-HARDENING.

The case-hardening of steel is dependent upon the fact that above certain temperatures the affinity of iron for carbon is so great that when iron or steel is heated at these temperatures in contact with suitably chosen carbonaceous materials ready absorption of carbon supervenes. The quantity of carbon that can be absorbed by iron or steel under the above conditions varies according to the following factors—

1. The composition of the iron or steel.
2. The temperature at which the process is conducted.
3. The length of time during which the process is continued.
4. The nature of the carburising material.

The Composition of the Iron or Steel.

Case-hardening, in the accepted sense, is rarely practised on steel containing more than 0·20 per cent. of carbon or 0·40 per cent. of manganese. The process is stated by Guillet to be hindered by the presence of nickel, silicon and aluminium, and hastened in the presence of manganese, tungsten, chromium, and molybdenum.

The Temperature at which the Process is Conducted.

The process is best conducted at temperatures above the critical range of the iron or steel treated. The usual temperatures for carburisation vary from 900° C. to 1000° C. The higher the temperature of carburisation the greater the rate of absorption of carbon and the deeper the effect of absorption.

The Length of Time during which the Process is Continued.

This factor can only be determined by experiment. It is generally considered that the best results are obtained when the carbon content of the surface layers of the case-hardened material are of eutectoid composition (about 0·90 per cent. C.).

The Nature of the Carburising Material.

The carburising material may be either solid, liquid, or gaseous. Solid materials are employed more generally than are liquid or gaseous materials.

Solid Case-hardening Materials.

For the purpose of obtaining deep 'cases' the articles to be hardened are packed in a suitably designed box with solid case-hardening substances or mixtures of substances, of which the following are examples :—

Charcoal (animal or wood).
Charred leather.
Crushed bone or horn.
Barium carbonate 40 per cent. and charcoal 60 per cent. (Caron's mixture)
Charcoal (wood) 95 per cent. and soda ash 5 per cent.
Charcoal (wood) 90 per cent. and common salt 10 per cent. (Guillet)

As case-hardening composition is bought by weight, and is required to fill up the case-hardening boxes, the specific gravity is an important consideration. The speed of conduction of heat through various depths of carburising material requires determining by experiment. A good case-hardening material should not diminish in volume appreciably during the carburising operation, and should be free from sulphur and phosphorus.

The box should be of plate iron not less than $\frac{1}{8}$ to $\frac{1}{4}$ inch thick. The lid of the box has two holes pierced in it for drawing testing-pieces out if required. For small articles a piece of wrought-iron tube, plugged or capped at one end, may be used ; the other end is closed by a second plug fastened with an iron pin passing through it and the pipe, the whole being luted with fine clay. The articles being previously finished, except polishing, are put into the box in alternate layers with the carburising mixture, commencing at the bottom of the box with carburising mixture to the thickness of about $\frac{1}{2}$ inch ; upon this a layer of the articles is placed, then another of carburising mixture about one-third the thickness of the first, and so on until the box is nearly full, finishing with carburising mixture about the thickness of the first layer, leaving room every way for the expansion of the articles by the heat, otherwise they will bend each other in the box. The packing completed, the lid is put on, the box luted and placed in a suitable furnace. The contents of the box require to be very gradually and uniformly heated to redness, and retained at this heat for the period required for the depth of caburisation desired. The depth of carburisation depends on the length of time the temperature is kept up. For testing pieces plain pieces of the same kind of iron as the articles may be used. They require to be brightened, and are placed, at the time of the packing of the box, in the central part, in such a manner that they may be readily pulled out through the holes in the lid, either by a piece of iron wire attached or by being made long enough to project through the holes, so that they may be gripped with pliers ; the holes are luted the same as are the other parts. When the articles are sufficiently carburised the box is drawn from the fire, the lid taken off, and the contents are immersed in pure cold water : taken out when cold they are ready for polishing. To prevent rusting, the articles may be dried by riddling in a sieve with dry sawdust, after which they are wiped with a greasy cloth. If the articles be immersed in oil instead of water they will be much tougher but less hard, though sufficiently so for some purposes. It is not necessary to immerse them direct from the box, as it answers equally as well to allow them to remain in the box till cool, and then reheat them in an open fire and immerse them separately.

When the case-hardening is required to terminate at any particular part of the article, the part needed to be soft may be bound with iron wire and cased with loam. This prevents the iron from absorbing carbon at that part. The loam requires to be gradually dried on the article previous to putting in the box, otherwise it will crack. Another method is to cover the part needing protection with a copper tube (this is the most satisfactory method when the shape of the work will allow the use of the tube) or to paint it with a protecting paint (paint can be made from china clay and blacklead or from sodium silicate and sand). A frequent procedure is to carburise an object and allow it to cool in the box, finally machining off the carburised portion before quenching wherever it is desired to keep the surface soft. It is of course necessary to leave sufficient metal on the object before carburising in order to ensure complete removal of all carburised metal before quenching. With long objects, one end of which requires hardening while the other is to remain soft, it is possible to carburise the entire length, then reheat and quench one end only, allowing the other end to cool in air.

For the purpose of superficial case-hardening it is necessary only to heat the iron or steel to be treated to redness, to sprinkle one or other of the following finely powdered compounds or mixtures on the surfaces to be hardened, and to plunge the iron so treated into pure cold water. This method is simple and speedy, but the case obtained is insufficiently thick to resist much wear.

Potassium cyanide.
Potassium ferrocyanide.

Potassium ferrocyanide and potassium bichromate.

Sodium ferrocyanide 90 per cent. and anhydrous sodium carbonate 10 per cent.

Sodium ferrocyanide 80 per cent., anhydrous sodium carbonate 10 per cent., and potassium ferrocyanide 10 per cent.

Liquid Case-hardening Materials.

The immersion of steel articles in a bath of molten potassium cyanide (the fumes of which are of a highly poisonous nature) heated to 850° C. results in the formation of very hard but shallow cases.

Gaseous Case-hardening Materials.

The following gases have been employed for case-hardening iron and steel—illuminating gas, carbon monoxide, acetylene and petroleum vapour. All, according to Brearley, are supposed to be more effective if they are led through a solution containing ammonia, or otherwise are partly saturated with ammonia gas before entering the furnace. By increasing the pressure of the gaseous mixture it is possible to increase the rate of carburisation.

Rapid Case Hardening Methods.

One of the best is to instal a container for sodium cyanide and common salt mixed. These salts used molten are very efficacious, and many small parts can be thus cased without going to the trouble and expense of casing in boxes. The sodium cyanide needs careful handling, but it is a simple matter to arrange for it to be placed into the equivalent of a fume chamber, a place fitted with a fan so that an adequate draught will take away all the poisonous fumes.

The immersion of clean machined steel parts in the mixture will produce a reasonably deep case in the course of $\frac{1}{2}$ to 1 hour, and there is the additional advantage that the parts emerge from the bath quite clean and bright and can be quenched for hardening at once. Some trouble may be experienced in obtaining a suitable container for these molten salts, but the use of one of the nickel-chromium alloys will give excellent results.

Another good method of obtaining a reasonable case on steel consists simply of playing an oxy-acetylene flame on the steel, and a very few minutes at a temperature of 900–1,000° C. will produce an adequate depth of case. The flame must be suitably regulated so that there is no danger of the steel being cut, but an excess of acetylene in the flame will prevent this and will cause the casing to be formed much more quickly.

These methods are those by which small parts may be quickly hardened on their surface and are not intended to supplant the ordinary box carburising.

(*' Machinery,'* May 12, 1927.)

Thermal Treatment of Case-hardened Materials.

Quenching.—If the quenching takes place directly after the carburising operation, the resulting product is liable to possess a coarse structure in both the core and the case. In order to improve the structure it is better to allow the work to cool in the box after cementation, and then reheat it in a separate operation. The reheat may be made bare in the furnace, but with large objects which take long periods to obtain the desired temperature it is advisable to protect them from decarburisation by covering them with charcoal or old case-hardening mixture. This second reheat is most satisfactory at temperatures of about 780° to 820° C. It is claimed that very good results may be obtained by reheating twice, first with a view to refining the core and secondly to refine the case. A sample treatment of this description is :—

1. Carburise at 900° C. and allow to cool in box.
2. Reheat at 900° C. bare, and allow to cool in air.
3. Reheat bare for 20 minutes at 810° C. and quench in water.

Another common treatment which is said to yield fine core structures is to reheat twice and quench after each reheat, *e.g.* :—

1. Carburise at 920° C.
2. Reheat to 900° C. and quench in oil.
3. Reheat to 810° C. and quench in water.

In order to refine the cores of case-hardened articles it is advisable to heat the same to at least 875° C. (unless the carbon content of the core is above 0·20 per cent.) and quench. This treatment refines the core of the article but coarsens the case, hence it is necessary to reheat to a temperature somewhat above the critical range corresponding to the carbon content of the case (775° C. to 825° C.), subsequently quenching the article in water, or in oil.

Quenching in water results in a much more rapid cooling than oil quenching ; the former is necessary for ball races, cams, and all parts which have to be subjected to hard wear and require extreme hardness (80 to 90 scleroscope). Oil quenching of correctly carburised mild steel results in a hardness of 50 to 70 scleroscope. Small objects cool more rapidly, and therefore become harder than large objects undergoing the same treatment.

Tempering of the quenched articles still further improves the properties of the same.

'Nitride,' or Nitrogen Case-Hardening.

The process is based on the fact that certain steels will absorb nitrogen, and such steels undergo a eutectoidal transformation at 1,076° F., which enables an extremely hard non-brittle surface

to be produced. Steels containing 0·5–2·0 per cent. aluminium and 0·5–4·0 per cent. of an element, or combination of elements, having the properties of chromium are used. Ordinary carbon and alloy steels are not suitable.

TYPICAL ANALYSES OF STEEL 'NITRALLOY' (CENTRAL ALLOY STEEL CORPORATION, U.S.A.).

	Nitralloy 'G.'	Nitralloy 'H.'
Carbon	0·36	0·23
Manganese	0·51	0·51
Silicon	0·27	0·20
Aluminium	1·23	1·24
Chromium	1·49	1·58
Sulphur	0·01	0·011
Phosphorus	0·013	0·011
Molybdenum	0·18	0·20
Average Brinell No., annealed	186	157
Average Brinell No., heat-treated core	300	255

PHYSICAL PROPERTIES, ALLOY 'G.' OIL-QUENCHED FROM 1650° F.

Tempered at ° F.	Yield Point. Tons per sq. in.	Ultimate Strength. Tons per sq. in.	Elong. %	R./A. %	Impact. Ft.-lbs.	Brinell No.
800	80·5	100·0	11·0	36·0	12	445
900	73·6	92·4	11·5	37·5	15	415
1,000	70·6	81·5	15·0	50·0	22	363
1,100	61·5	69·6	16·5	57·0	35	330
1,200	53·7	61·6	20·0	60·0	44	285
1,300	46·2	54·1	23·0	62·5	55	226
1,400	36·0	46·6	28·0	59·0	54	200
Annealed at 1,450	30·8	42·4	30·0	67·5	32	186

The nitriding process is carried out in a furnace (preferably electric), the temperature of which must be closely controlled around 950° F. The parts are placed in a gas-tight box provided with inlet and outlet tubes for the circulation of ammonia gas. No packing material is used, individual layers being separated by nickel wire netting. The rate of flow of gas is regulated by a needle valve on the ammonia cylinder, and this must be kept fairly constant during the entire cycle. The process takes from 2 to 90 hours, depending upon depth of case required.

Parts that require to be kept soft can be protected by tinning or other means.

Hardness of Case.—Readings taken with Herbert pendulum and converted into Brinell numbers.

Depth in Inches.	Brinell No.
0·000	1,002
0·005	999
0·010	810
0·015	516
0·020	434
0·025	378
0·030	378
0·035	351
0·040	351

Before the nitriding operation the parts should be heat-treated to impart the desired physical properties to the core. All strains set up by forging, etc., prior to nitriding must be thoroughly relieved by annealing at 1,000° F.

Relation between Hardness Scales.

Scleroscope No.	Brinell No. (3,000 kg.)	Rockwell. $\frac{1}{16}$-in. ball. 100 kg.	Rockwell. Cone. 150 kg.	Pyramid Diamond.*
20	105	60	— 20	·42
30	200	93	15	·30
40	235	100	22	·28
50	300	106	30	·25
60	370	110	34	·22
70	440	112	40	·20
80	525	113	46	·185
90	610	115	48	·17
100	650	—	54	·165
103	700	—	58	·16
106	750	—	60	·15

* Pyramid diamond, 136°. Diagonal dimensions of impression in millimetres.

Ratio of Tensile Strength to Brinell Number.

Material.	Ratio = Tensile strength. Tons per sq. in. ÷ Brinell No.
Heat-treated alloy steels. Brinell range, 250–400	0·21
Heat-treated carbon and alloy steels. Brinell below 250	0·215
Medium carbon steels, as rolled, annealed, or normalised	0·22
Mild steels, rolled, annealed, or normalised	0·23

Tensile strength, tons per sq. in. = Brinell No. × ratio.
The above values do not apply to severely cold-worked or to austenitic steels.
See also page 1100.

Scleroscope Hardness Values of Metal.

Metal.	Annealed or Cast.	Cold-worked.	Chilled or Hardened
Lead	2–4	3–7	
Aluminium	3·5–5	6–12	
Gold, 24–14 carat	5–25	24–70	
Silver	6·5–14	20–37	
Copper	6–8	14–20	
Zinc	8–10	18–20	
Babbitt metal	4–9	—	
Tin	8–9	12–14	
Bismuth	8–9	—	
Brass	7–35	20–45	
Platinum	10–15	17–30	
Phosphor-bronze	12–21	25–40	
Manganese-bronze	16–21	25–40	
Pure wrought iron	16–18	25–30	
Nickel (cast)	13–16	—	
Nickel (wrought)	17–19	35–40	
Mild steel (·05–·15 C)	18–25	30–40	
Cast iron grey (sand-cast)	25–45	—	
Chilled grey cast iron	—	—	50–90
Tungsten	60–70	—	60–70
Tool steel, 1 per cent. C	30–35	40–50	90–105
Tool steel, 1·65 per cent. C	38–45	—	90–105
Vanadium steel	30–50	50–60	50–105
Nickel-chrome steel	35–50	40–60	60–105
Nickel steel	25–30	35–45	50–90
High-speed steel	30–45	40–60	70–100

(Shore.)

Annealing.

To Prevent Oxidation and Scaling during Annealing.

It is sometimes desirable that steel should be thoroughly annealed without any oxidation and scaling whatever taking place. The most popular method of doing this is to simply pack the stampings in sand, but this method is not good, because, unless the sand be extremely fine, air will eventually percolate through the grains of the sand. The following method has been found to give first-grade results, and steel articles may be heated for many hours and come out with a perfectly bright surface. The packing mixture consists of finely powdered burnt lime containing a trace of powdered charcoal. A thick layer of lime is tightly rammed down in the bottom of a box, similar to those used for case-hardening purposes. The articles are placed in position on the lime and a further layer of lime is tightly rammed down on top of them, with a sprinkling of powdered charcoal on top of the lime. Several layers of articles may be packed in this way till the box is full, when it is covered with a lid and luted up with clay. Although this process is not so quickly performed as some, it gives excellent results.

If such articles as small die castings or brass or bronze extruded sections require to be annealed, the articles are dipped in a saturated solution of borax or boric acid in methylated spirit. The spirit evaporates at once, and leaves a sound protective coating of the solid which coagulates on heating and effectively resists the deleterious action of oxidising and other gases at temperatures up to some 750 deg. C., which is as high a temperature as is usually employed for the general heat-treatment of non-ferrous metals and alloys.

The coating of the borax or boric acid is so extremely thin that there is usually no necessity to remove it after heating, but should this be necessary, hot water will suffice.

(*Met.*, '*Machinery.*')

Corrosion of Metals.*

Corrosion of Iron.

Professor T. Turner in his book on 'Iron' says the results of the experiments of the Admiralty Committees which issued reports in 1877 and 1880 led to the conclusion that common iron was less affected by corrosion than best Yorkshire iron, which is in accordance with the statement of Gmelin that phosphorus diminishes corrosion in iron.

In these experiments :—

Common iron resisted corrosion better than Yorkshire iron 9·6 per cent.

Yorkshire iron resisted corrosion better than mild steel, 16 per cent.

In another series of experiments conducted by D. Phillips, in Cardigan Bay, and lasting for seven years, it was found that the average corrosion of mild steel during the whole period was 126 per cent. more than wrought iron.

The independent experiments conducted by T. Andrews also showed that wrought iron corroded less rapidly than mild steel when the cleaned metallic surfaces were exposed to the action of the water.

Iron rusts very quickly when immersed in water from bogs and marshes which contain a large quantity of carbonic acid, derived from decayed vegetable matter.

Iron does not rust in dry air.

Iron subject to vibration rusts slowly. Railway rails in running roads where trains are passing frequently rust much less than those in sidings which are little used. Rails which are kept in reserve and not in use rust more quickly still.

Thin iron rusts more rapidly than thick, because the expansion and contraction owing to changes of temperature throws off the scales more quickly.

In moist air, steel rusts more rapidly than wrought iron, and wrought rusts more rapidly than cast. (See table, p. 174.)

Cast iron in salt water becomes in time soft and porous. The smoother the surface of iron the more slowly it rusts.

Once rust forms the process goes on more rapidly; rust being a porous substance it absorbs moisture and acids from the atmosphere.

Protection of Iron from Rust.

The types of coating applied to iron and steel for their protection may be classified as follows :—

1. Adherent coatings. 2. Compound coatings.

1. Adherent Coatings.

These may be metallic or non-metallic in character. Of methods for the production of metallic coatings of this type the following are the most noteworthy examples :—

---

* See also p. 174.

(*a*) *Galvanising.*—This is a process which has for its object the coating of the material with zinc. Two processes, known respectively as hot galvanising and electro-galvanising, are employed for attaining the desired object.

Hot galvanising consists in the immersion of the object to be protected from corrosion in molten zinc or spelter, the surface of which is largely protected from oxidation by a supernatant layer of ammonium chloride. Modifications of the simple process briefly described in the last sentence are known under the following names, Winiwarter's process, Kufler's process, Bedson's process, Reese's process, Porter's process, etc.

The amount of zinc on modern galvanised sheets varies from 1 to 2·5 oz. per square foot coated. The coating generally contains from 0·25 to 1·00 per cent. of lead and from 2 to 6 per cent. of iron.

The most obvious disadvantages of galvanising are two in number: first, the iron or steel coated has to be raised in temperature for the desired effect to be attained; second, a layer of weak zinc-iron alloy forms at some point intermediate between the true zinc layer and the iron or steel upon which it is designed to rest. This thin lamina of weak alloy, mainly the compound $FeZn_7$, may be insufficient in a majority of cases to cause trouble, but a serious reduction in tensile strength is frequently brought about as a result of the presence of this weak material in fine gauge wires and sheets.

Electro-galvanising (cold galvanising) consists in the deposition of zinc electrolytically upon the object to be protected from corrosion, a soluble zinc salt being employed as electrolyte. The most important process under this head is that of S. Cowper-Coles, wherein an electrolyte of zinc sulphate is used, the electrolyte being continuously regenerated by circulation through tanks containing zinc dust. The methods of cold galvanising associated with the names of Heathfield, Powick, Pettkoff, Rawson, Wagner, Walker, etc., are worthy of note. Particulars of their processes are obtainable in the technical press.

The electrolytic process leads to the formation of an intermediate lamina of only minute thickness. In this particular the coating produced by the electrolytic process is an improvement upon that obtaining as a result of hot galvanising. It is, however, porous, and in this respect is inferior to the zinc coating produced in the dipping process. No marked superiority can be claimed for either of the processes.

Electro-galvanising has made great progress during the past few years, and may now be regarded as an efficient method of rustproofing. By its means a wide range of articles may be coated with a film of zinc which, while ample to protect from oxidation, does not affect ductility or tensile strength. The fact that perfect cleanliness must be achieved before the adherence of zinc can take place prevents dirty patches on the article becoming coated. The main cause of failure with hot galvanised work is the covering up, through the surface tension of molten spelter, of dirty or scaly places, which blister or otherwise fail in a comparatively short time. A commercially protective coating is obtained with a deposit of 4 oz. per square yard of surface. The deposit is practically pure, lead being the chief adulterant, then only in a fraction per cent. Accurately machined parts can be assembled after coating, this overcoming a difficulty ever present in the past. Hot galvanised bolts must have the thread bared before the nut will fit. Cold galvanising coats the thread so that the configuration is not destroyed, and unless the nut is a very tight fit the zinc remains on the thread and thus protects it against corrosion.

Admiralty specifications require a deposit of 0·86 oz. of electro-deposited zinc per sq. foot of surface of the articles treated.

(*b*) *Tin plating* is a process similar to hot galvanising, tin being the coating metal. The protection afforded to iron and steel by tin is less effective than is that of zinc.

(*c*) *Terne plating* consists of the coating of the material to be protected with an alloy of lead containing from 50 to 75 per cent. of tin (generally 70 per cent. tin).

(*d*) *Lohmanising*—in this process the metals to be protected are (i) pickled in an acid bath, (ii) dipped into a salt bath which deposits a metallic salt over the entire surface of the body, (iii) immersed in a bath of molten alloy of zinc, lead and tin heated to a temperature of from 500 to 600° C. The composition of the tin-lead-zinc alloy is varied to suit the material being treated.

(*e*) *Electro-plating*—the coating of iron and steel electrolytically with copper, nickel, etc., is well known.

(*f*) *Copper 'clothing'*—the casting of copper round or the welding of copper to steel and the rolling of the resultant material to the required shape and dimensions has been practised, e.g. Monnot process, Willis process, etc. (vide *Iron Trade Review*, vol. li. 479).

(*g*) *Nickel 'clothing'*—processes similar to the Monnot and Willis processes have been successfully applied to the coating of iron and steel with nickel and cupro-nickel.

(*h*) *Metallic spraying*—the application of liquid metal in finely subdivided form by a spraying machine has been accomplished by M. U. Schoop, whose process, 'the Schoop process,' allows of the coating of iron or steel with an adherent layer of tin, lead or zinc, the material being coated remaining meanwhile at the ordinary temperature. The material under treatment must be scrupulously clean if adhesion of the finely-divided spray is to be perfected (vide *Metal Industry*, vol. xvi. 41).

2. Compound Coatings.

The term 'compound' is applied here to such protective coatings as are produced by chemical action, the ultimate result of which is the formation of a chemical compound, such, for example, as magnetic oxide of iron (Fe_3O_4), iron-zinc ($FeZn_2$), etc. The compounds formed may be, as already noted, either chemical (in the general sense of the term) or intermetallic (in the specialised sense of the term).

Chemical Compounds.

Barff's process consists in passing superheated steam over the objects, which are heated in an air-tight retort to about 260° C.; for the formation of a more perfect protective coating higher temperatures are necessary (650° C.). Articles so treated are said to be 'barffed.' Relatively small objects only can be treated as above. The coating formed consists of the mixed oxides of iron (Fe_2O_3 and Fe_3O_4).

The *Bower-Barff* process is an improvement on the above, and consists in the heating of the objects to be protected to about 900° C., at the attainment of which temperature superheated steam is introduced to the same, which has the effect of producing a coating of the mixed oxide already referred to. This operation having continued for about twenty minutes, carbon monoxide is injected into the retort for a period of from fifteen to twenty-five minutes, during which time the entire reduction of the coating to the condition of magnetic oxide of iron is completed.

The *Gesner process* treats iron and steel at a moderately high temperature (550° C. to 650° C.) in a retort whereinto may be introduced an atmosphere of steam. The preliminary heating occupies twenty minutes; the steam is then allowed to react with the oxide formed in the first operation for about half an hour, when naphtha is introduced and allowed to remain in contact with the material under treatment for about ten minutes. Finally, steam is again introduced. The whole sequence of operations occupies about one hour and twenty minutes.

The *Parker 'dry' process.*—Under a patent assigned to the Parker Rust-proof Company, of Detroit, the formation of an adherent coating of oxide may be obtained by first heating the objects to be treated to about 600° F. and then subjecting them to fumes of the acid metaphosphates of strontium, molybdenum, and tungsten.

The *Hans Renold process.*—Remove all grease or oil from the articles to be treated by suspending in a bath of hot lyco (11 oz. per gal.) for about three minutes, wash in cold running water for a few seconds, then suspend from cathode brass bars by steel hooks in an electrolytic bath (KCN, 98 per cent. pure, 4 oz. per gal. of water), iron plates forming the anodes. Leave here for five minutes, and when a current of 25 amps. at 8–10 volts is passed through the bath all foreign matter on the surface will be removed. Wash off the cyanide by means of hot water to facilitate drying.

Heat the articles to 700° C., then quench in cotton-seed oil by immersing in the main body of the quenching tank for seven or eight seconds before allowing to fall to the bottom.

Dry by heated sawdust and hot-air blast (dried and filtered by passing through calcium chloride and cotton wool).

Insufficient oxidation results in the finished bodies having a grey appearance, due to the atmosphere of the furnace containing too much gas. Blistering is due to overheating.

Villon's process consists in subjecting the iron or steel to be protected to the vapour of sulphur as a result of which treatment the monosulphide of iron is found.

The *Bertand process* requires that the articles to be treated be thoroughly cleaned by sand-blasting or similar means before they are immersed in a bath, such as will lead to the deposition of copper upon the same as a result of reaction between the iron and a suitable electrolyte. The deposition may be achieved by the application of electricity or by more purely chemical means. After thorough washing the articles are heated in an oxidising atmosphere to a temperature of about 600° C., which treatment leads to the formation of a fairly adherent coating of magnetic oxide of iron.

The *De Meictin's process* depends upon the formation of a coating of magnetic oxide (Fe_3O_4) upon iron or steel which is placed as the positive pole in a bath of distilled water, maintained at a temperature of from 70° C. to 80° C., a current of electricity being passed through it. The oxide coating is adherent, provided the current density is not over great, and may be rubbed and polished. This method is of value in the treatment of such articles as cannot be submitted to any processes which might tend to temper or anneal the objects under treatment.

The *Parker 'wet' process* (patent) consists in the immersion of the iron or steel to be protected in a bath of a 1½ per cent. solution of the acid metaphosphates of tungsten, molybdenum, or any of the metals of the third, fourth or fifth group of elements.

The *Coslett process* (*Coslettising*).—Cast iron or steel articles treated by this process are not only rendered rust-proof but improved in appearance.

The articles to be treated should be perfectly clean and free from tin, nickel, or grease.

The best results are obtained on a frosted surface, such as is left by sand-blasting or pickling. Sand-blasting is to be preferred, as any traces of acid render the treatment ineffectual.

The coslettising bath is prepared by heating in a cast-iron or welded steel tank a sufficient quantity of distilled water to easily cover the articles to be treated. When boiling add for every gallon of water in the bath (1) two ounces of phosphoric acid (syrupy sp. gr. 1·5); (2) one ounce of iron filings. This should be carried out in the open or under an uptake, as the reaction results in the formation of disagreeable fumes. When this reaction has ceased the bath is ready for use.

The articles should be placed in the bath as soon as possible after sand-blasting, and should be gently boiled for two hours.

A second tank should be prepared containing distilled water, which should be heated in time to wash the articles as taken from the coslettising bath.

At the expiration of the specified time the articles should be taken from the coslettising bath and placed directly into the hot water and thoroughly scrubbed with a stiff bristle brush.

They may then be dried off, and if heavy will dry quickly themselves, but if light should be placed in an externally heated oven.

As soon as dry and while warm they should be dipped in a mixture of lubricating oil and turps substitute, in the proportion of 2 to 1, and left to drain. When cold, surplus oil may be removed with clean waste or wiper.

To allow for evaporation of bath a mixture similar to that in the bath should be made up and added as required.

To keep the bath at the correct strength it is necessary to titrate a sample of the bath with $\frac{1}{10}$ normal sodium hydrate. To make this test, 10 c.c. of the bath should be placed in a conical flask, a few drops of phenolphthalein solution being introduced as an indicator. The $\frac{1}{10}$ normal sodium hydrate should be added from a burette, until a permanent pink tinge is given to the liquid in the flask. The amount of sodium hydrate dropped into the flask should be noted, and if the bath is correct this amount will be 10 c.c. If more sodium hydrate is used, the bath is too strong and will require diluting with distilled water; if less is used, the bath is weak and will require the addition of phosphoric acid and iron filings in the given proportion.

The appearance of the deposit obtained on the article is a rough guide to the state of the bath. The proper deposit is of a uniform dull black colour, but if the bath is too strong this black coat is covered with a pearly white deposit which will not scrub off. If the bath is too weak the articles appear unevenly covered with a thin grey coat.

It is stated that in the present coslettising process (which cannot be used except under licence from the Coslett Anti-Rust Syndicate) phosphorised zinc, made under special conditions, is used instead of phosphorised iron, the solution otherwise being prepared in much the same manner. This newer process is said to give much better results, and the depth to which the articles are rust-proofed to be considerably greater.

The need for the absolute cleanliness of the articles before treatment cannot be too strongly emphasised, and they should not be handled after cleaning, or before they are finally oiled.

Intermetallic Compounds.

Sherardising.—The articles to be coated are first cleaned and dipped in an acid bath as for galvanising; they are then put in a suitable retort which is, if possible, airtight, and are surrounded with ' blue powder '; the retort is then raised to a temperature of 500° or 600° F.

If the retort cannot be made airtight, about 3 per cent. of finely powdered carbon should be mixed with the zinc to prevent the formation of zinc oxide.

The heat is kept up for a few hours, after which it is allowed to gradually decrease.

A coating is formed which consists mainly of an intermetallic compound of zinc and iron ($FeZn_3$).

Calorising.—A somewhat similar process to the above, in which aluminium powder is utilised as the agent and wherein the resultant protective coating consists chiefly of an intermetallic compound of iron and aluminium.

Stainless Steel.

Stainless steel or rustless iron are not altogether stainless or rustless under all conditions. Moreover, the ' rustless iron ' is really a dead-mild steel containing a certain percentage of chromium, which gives the steel its corrosion resisting properties. Mild stainless steel contains about 11 to 14 per cent. chromium, with 0·1 per cent. carbon and under, the other constituents, silicon, sulphur, phosphorus, and manganese, being all more or less in normal percentages. (Apparently when the cast contains 0·1 per cent. and less of carbon it is called ' rustless iron, and above that percentage ' rustless steel.')

One of the valuable properties of mild stainless steel is its behaviour at high temperatures. When pieces of the material which have been polished are heated up gradually, temper colours

appear on the surface similar to those obtained when ordinary carbon steels are heated up, but in the case of mild stainless steel these temper colours appear at much higher temperatures, as shown by the following table:

Colour.	Mild Stainless Steel.	Ordinary Steel.
	° C.	° C.
Pale straw	290	225
Straw	340	235
Brown purple	390	265
Purple	450	280
Bright blue	530	290
Dark blue	600	315

Up to temperatures of about 825° C., mild stainless steel resists scaling to a remarkable degree, but above this temperature the material begins to scale to some extent.

Hot Rolling and Forging of Mild Stainless Steel.

The most suitable hot rolling temperature for this material is 1050° C. to 1100° C., and it is essential that for rolling and forging the material should be brought up slowly to about 900° C. and then slightly quicker to the rolling and forging temperature. It should be allowed to soak for a long period to ensure a uniform temperature through and through.

For forging, it should be worked quickly and with rapid blows between the temperatures of 1150° C. and 900° C. Below 900° C. to 850° C. it is not so easily deformed, and if any heavy work is put on the material after it has fallen below these temperatures it is very apt to be badly stressed or burst.

For drop stamping, it is advisable to work at a slightly higher temperature, and all stampings must be trimmed while hot, or annealed before trimming on account of the air hardening properties of the steel. Otherwise, if an attempt to trim is made, cracking and splitting will occur, as well as damage to tools.

Annealing.

The critical range of mild stainless steel runs from about 865° C. to 965° C.

To anneal mild stainless steel to obtain the maximum softness, it should be heated up slowly to about 1000° C. and held at this temperature for a period of time in proportion to the mass of metal, and then allowed to cool off slowly in the furnace, making sure that no currents of cold air enter the furnace.

Although steel annealed this way has the maximum degree of softness, it is not in the best condition for machining. The steel in this state is very soft and tough, and so drags and tears when machined, leaving a very rough surface.

To soften the material for good machining properties or cold drawing and cold rolling, it is best to anneal at 750° C. to 800° C., preferably about 780° C., and then to cool in the furnace or draw and cool in the open air.

Mild stainless steel has distinct air hardening properties, so that it should always be annealed or high temperature drawn at 750° C. to 800° C. as stated above, after it has cooled down from a hot rolling or forging temperature, before it can be successfully machined or cold worked.

Cold Rolling and Drawing.

The material will cold roll, cold draw and deep draw, or press very well, but owing to the percentage of chromium in the metal, the depth of each draw, when the steel is being deep pressed or drawn, obtained at each operation is only about half that got when working on mild steel, before the material must be annealed.

In cold rolling and cold drawing, the reduction of one gauge at a pass can be used, but it is found necessary to anneal after every second or third pass.

As with all materials, a distorted surface is always more liable to rust, and the same applies to mild stainless steel, especially if the surface happens to be unevenly distorted.

In cold working, strains are always set up in the material, and sometimes strained parts are not confined to the surface where they can be ground off, but are inherent in the steel and the strains must be removed by heat treatment. These strains are not only liable to cause rust, but are apt to cause the steel to crack if put into service. It is thus evident that it is advisable to anneal or high temperature draw at 780° C. all material which has been cold worked in any way.

Pickling.

The proper pickling of mild stainless steel is a very important factor in the manufacture of articles of this material. If the steel is left coated with scale, or has small pieces of scale left on the surface it will rust wherever the scale is present. It is therefore essential that the finished surface should be free from all scale, roakes, seams, pits, or cracks, or even stamp marks, as otherwise rusting will result.

The acid pickling baths generally used for this material are as follows:

1. 50 per cent. solution of hydrochloric acid used hot at 60° C. to 70° C.
2. 15 per cent. solution of sulphuric acid used hot at 60° C. to 70° C.
3. Ten minutes in a 50 per cent. commercial hydrochloric acid, then transfer without washing to 20 per cent. nitric acid for 25 minutes, both these solutions being worked cold.

Of these acid pickling solutions, No. 3 is the one which has given the most satisfactory results. Other methods which have been used for cleaning this steel are electrolytic pickling in an alkaline bath, which has been more or less successful for bars, etc., and articles of simple shape. Acid pickles containing chemical salts to accelerate the action, have also been used with considerable success.

Grinding.

The grinding of mild stainless steel must be carried out very carefully, as this material is more difficult to grind than ordinary steel. This metal dissipates heat very slowly, and great care should be taken in grinding in order that grinders' 'scorch' is not produced. Grinders' 'scorch' is due to harsh grinding and produces a brownish discoloration on the steel. This discoloration is really a temper colour produced by the heat generated on the surface of the work, and will cause rusting, due to the local distortion set up. Wet grinding is much better than dry grinding.

Etching.

In etching mild stainless steel, the following solutions have given very good results:

1. A saturated solution of ferric chloride in hydrochloric acid to which a little nitric acid has been added. This solution is used full strength.
2. A mixture of hydrochloric acid and nitric acid in the proportion of three to one and used full strength after allowing to stand for about 24 hours.
3. A saturated solution of copper sulphate in hydrochloric acid. This solution is used for light etching.

Physical Properties.

The physical properties obtained from mild stainless steel in the annealed condition, and also in the hardened and tempered state, cover a very wide range; this will be seen from the following table:

TESTS OF TEMPERED BARS.

Tempered at.	Yield Point. Tons per sq. in.	Tensile Strength. Tons per sq. in.	Elongation. Per Cent.	Reduction of Area. Per Cent.	Izod Impact. Ft. lbs.	Brinell Hardness. Kg. per sq. mm.
° C.						
200	66·5	73·0	12·0	37·5	35	340
300	66·0	72·5	12·5	37·0	39	332
400	65·5	72·3	16·0	50·0	38	332
500	59·0	72·5	18·0	52·0	36	340
600	38·0	48·0	22·5	62·0	66	235
700	31·0	41·0	27·0	66·0	80	192
750	28·0	36·6	30·0	69·0	88	174

These tests were made on 'as rolled' and 'annealed' bars quenched from 950° C. in oil and tempered at various temperatures.

Welding.

Mild stainless steel can be welded by either the electrical process or by the oxy-acetylene method. It will not weld by the usual method of heating in a smiths' fire.

In welding, the temperature of the metal at the weld is raised to a fairly high temperature, so that the material on cooling down will air harden and should be annealed before any turning, grinding, or polishing is done.

(*H. S. Primrose, A.M.Mech.E., M.Inst. Met., Paper read before the Manchester Association of Engineers, Friday, December* 19, 1924.)

HARDNESS OF STAINLESS STEEL AT ELEVATED TEMPERATURES.

Temperature. ° C.	Brinell Hardness Number.	
	A.	B.
15	235	440
100	212	420
200	190	414
300	160	345
400	130	360
500	110	345
600	100	175
700	85	89
800	70	70

A. Oil-quenched, 950° C.; water-quenched, 600° C.
B. Oil-quenched, 950° C.; tempered, 250° C.

(*Hadfield.*)

STAINLESS IRON.

Stainless iron is produced in the basic-lined electric furnace. Owing to its greater softness it forges more easily than the harder varieties of stainless material; it works probably as easily as a 40 carbon steel, and hence may be forged, rolled, or drop stamped. Stainless iron does not air-harden so intensively as the higher carbon varieties. It does air-harden, but the degree of hardness is considerably less, being in the range 280–380 Brinell instead of 450–550, as obtained with a stainless steel with 0·3 carbon.

PHYSICAL PROPERTIES OF STAINLESS IRON. ONE-IN. BAR. OIL-HARDENED, 930° C.

Tempered at ° C.	Ultimate Strength. Tons per sq. in.	Yield Point. Tons per sq. in.	Elongation. %	Reduction of Area. %	Impact. Ft./lbs.
—	73	—	12·0	42	28
200	72	—	12·5	38	33
300	72	—	13·0	40	38
400	72	—	16·0	49	39
500	71	59	19·0	53	39
600	50	38	22·0	59	60
700	40	30	28·0	66	80
750	38	28	31·0	68	86
800	56	33	13·0	38	37
Annealed	35	21	37·0	68	80

It may be readily cold-worked, and provided the amount of distortion produced is not too great, the resistance to corrosion is not seriously affected. It can be drawn into wire and tubes. The general character of the resistance of stainless iron to various corroding media is similar to that of stainless steel. It is attacked readily by hydrochloric, sulphuric, and sulphurous acids. It is practically unaffected by saturated or superheated steam, lubricating oils, petrol, benzol, paraffin, greases, etc. It resists oxidation well at elevated temperatures. At temperatures up to 700°–750° C. a series of temper colours are produced, similar to those obtained on ordinary steel, but at much higher temperatures. Above 700°–750° C. up to about 825° C. a polished surface becomes covered with a grey film without, however, losing its polished appearance, and the specimen neither gains nor loses weight appreciably. Above 825° C. it begins to scale appreciably.

(*Monypenny.*)

Heat-Resisting, Non-Corrosive Steels ('Enduro Nirosta').—These steels are austenitic and non-magnetic, and cannot be hardened by heat-treatment, any increased hardness being produced by cold-working. Heat-treatment necessary to put steel in a condition to resist corrosive attack and to restore softness, ductility, and machinability:—Heat quickly to 2,100°–2,150° F., cool rapidly in air or water, depending on section. Do not soak in furnace. For forging, heat slowly to 1,500° F. and soak, heat rapidly to 2,100° F. without soaking.

This material is suitable for all classes of work, including deep drawing. Having the ductility of copper it can be drawn into fine wire.

Average Analysis.—Carbon under 0·16, manganese under 0·5, silicon over 0·5, chromium 17–20, nickel 7–10, phosphorus and sulphur 0·25 max.

Physical properties after water-quenching from 2100° F.:—

Ultimate strength	33·5–42·5 tons per sq. in.
Yield point	13·5–18·0 „ „ „
Elongation (on 2 ins.)	60–55 per cent.
Reduction of area	75–70 „ „
Brinell	135–140
Rockwell (B scale)	75–79
Charpy, ft.-lbs.	110–100
Izod, ft.-lbs.	120–115

STRENGTH AT ELEVATED TEMPERATURES.

Temp. ° C.	Ultimate Strength. Lbs. per sq. in.	Yield Point. Lbs. per sq. in.	Elongation. %	R./A. %
20	105,000	49,000	55	62
200	101,000	45,000	53	66
300	98,000	40,000	50	65
400	91,000	35,000	45	62
500	83,000	28,000	39	61
600	70,000	22,000	30	—
700	55,000	16,000	23	—
800	30,000	11,000	18	—
900	13,000	5,500	24	—
1,000	9,000	4,500	47	—

COPPER.

Copper containing at least 99·95 per cent. of the pure element is produced on a commercial scale in large quantities. The average tenacity of cast copper is about 10 tons per square inch, that of worked copper is about 15 tons per square inch, while Roberts-Austen quotes the following figures for the mechanical properties of slowly annealed, hammered, rolled bars of copper of a high degree of purity :—

Elastic limit	1·16 tons per sq. in.
Maximum stress	13·47 „
Elongation	37·0 per cent.
Reduction of area	61·0 „

Impurities in Commercial Copper.

Arsenic increases the hardness and the tenacity of copper. When present in moderate quantities it increases the percentage elongation of the metal. It improves the rigidity and strength of copper at high temperatures, hence, for fire-box plates, the use of copper containing from 0·3 per cent. to 0·5 per cent. of arsenic is frequently specified. In copper that is to be employed in the manufacture of brass subsequently to be cold-worked, not more than 0·05 per cent. of arsenic should be allowed.

Antimony is similar in action to arsenic, but it should be limited to traces in such copper as is to be used for the manufacture of brass that is to be cold-worked.

Bismuth being the most deleterious element that can be present in copper, owing to its tendency to induce red-shortness, not more than traces of this element should be allowed.

Iron is rarely present in copper in more than traces.

Lead is, after bismuth, the most objectionable impurity likely to be present in copper, although it is of value in metal that is to be rolled into sheets, on account of its softening effect on copper. Not more than the smallest quantities of this element should be allowed in copper for cold-worked brass.

Nickel hardens copper, but is in no way deleterious when present in such amounts as are usual in commercial copper.

Oxygen in small quantity does not have any very radical effect on the mechanical properties of copper. It is important in bringing copper up to the 'pitch,' and in combining with deleterious impurities to form insoluble and infusible compounds, thus separating and distributing these impurities and rendering them harmless.

Sulphur tends to embrittle copper, but is rarely present in refined copper.

Tellurium is similar to sulphur in its action on copper.

Silver, tin, and zinc are sometimes present in copper, but are rarely present in such quantity as to have much effect on the metal.

It has been found practically impossible to cast copper which is mechanically sound and of high electrical conductivity, on account of the porous metal that is obtained. By an addition of boron this can be accomplished. One per cent. of boron suboxide flux—equivalent to 0·08 to 0·1 per cent. of boron suboxide—is added to the copper, and a casting that is commercially sound is obtained.

ZINC.

Zinc is marketed in two forms—as sheet zinc (see Sheet Zinc Trade Gauge, page 11) and as spelter, *i.e.* flat rectangular ingots from 1 to 2 inches in thickness. The tenacity of zinc as cast is about 1 ton per square inch, and its elongation is practically nil. Zinc, after compression under a stress of 20 tons per square inch at 100° C., has a tenacity of about 11 tons per square inch.

It is ductile between 212° F. and 302° F. and can then be shaped as required. But when either above or below these limits it becomes brittle and unpliant, and therefore not adapted for treatment. It melts at about 786° F. if volatilisation is guarded against.

Impurities in Commercial Zinc.

Arsenic is a common impurity in spelter. It does not occur in such quantity as to have a deleterious effect upon the mechanical properties of spelter or of such alloys as contain it. It is, however, an objectionable ingredient in such zinc as is to be employed in the production of hydrogen.

Cadmium is nearly always present in zinc; in electrolytic varieties of the metal it rarely exceeds 0·07 per cent., in other varieties as much as 0·40 per cent. is found. The effect of this element is to harden spelter and to make it more brittle. Cadmium is, on this account, objectionable when the zinc has to be rolled into sheet or to be used for galvanising telegraph wires. In zinc for the manufacture of brass subsequently to be cold-worked as much as 0·20 per cent. of cadmium may safely be allowed.

Copper tends to harden zinc, but is rarely present in quantity sufficient to effect radical change in the metal.

Iron hardens and embrittles zinc. Its presence tends to the formation of dross in the galvanising process.

Such zinc as is to be used in the manufacture of cold-worked brass should not contain above 0·10 per cent. of iron. Electrolytic spelters rarely contain above 0·025 per cent.; as much as 2·5 per cent. of iron is, however, to be found in certain zincs.

Lead is present in certain brands of zinc up to 2·5 per cent. The purer brands of spelter contain up to 0·20 per cent. of lead. The presence of lead is not deleterious where the spelter containing it is to be used for general brass manufacture, but where brass is to be cold-worked or specially treated care must be taken that the spelter used has not too high a content of lead.

Lead in moderate quantity tends to make zinc softer in rolling. When present in quantities in excess of 0·7 per cent. it tends to the production of serious cracks in slush castings of zinc.

Other elements occasionally present in spelter in traces are sulphur and carbon.

TIN.

The London Metal Exchange form of contract (1911) for the purchase of tin recognises two classes of tin—Class A, tin assaying not less than 99·75 per cent. of pure metal; Class B, tin assaying not less than 99 per cent. of pure metal.

The tensile strength of cast tin varies (according to Thurston) from 2,000 to 6,000 lbs. per square inch. The elongation is, for all practical purposes, nil.

The principal impurities in commercial tin are lead, iron, copper, and arsenic. Sulphur, silver, bismuth, and antimony are of frequent occurrence.

NICKEL.

Nickel containing small quantities only of impurities is a commercial article. On account of its readiness to occlude carbon monoxide and other gases, and its affinity for carbon, it is difficult to obtain nickel in a pure state in the cast condition. Parker, experimenting on pure, dead soft, wrought nickel, has arrived at the following figures for the mechanical properties of nickel:—

Elastic limit	6–10 tons per sq. in.
Maximum stress . . .	18–20 „
Elongation . . .	8–15 per cent.

Nickel Plating.

According to the American Bureau of Standards, if nickel plating be made considerably thicker than is necessary or customary on brass parts, the protective value can be materially increased. It is believed that 0·001 in. is the minimum thickness that should be employed.

Impurities in Commercial Nickel.

Carbon is present in nickel up to 0·20 per cent., though it rarely exceeds 0·15 per cent. It has an embrittling effect on nickel when in the form of graphite, but when combined with nickel as nickel carbide it has a strengthening effect.

Cobalt has a strengthening and hardening effect on nickel, but in the quantities—up to 1·00 per cent.—usually found in commercial nickel, this element has but little effect.

Copper is similar to cobalt in its action, but rarely exceeds 0·20 per cent. in amount.

Arsenic should not exceed about 0·025 per cent. in such nickel as is to be subjected to cold-work, or in alloys for a similar purpose.

Iron has an effect on nickel similar to that of cobalt and chromium, but, being rarely present in quantities in excess of 0·75 per cent., its presence is scarcely noticeable.

As much as 0·30 per cent. of siliceous matter can be present in commercial nickel. Its effect on the properties of the nickel is uncertain.

Sulphur is occasionally noted in samples of nickel.

Nickel, if it is to be satisfactory for the manufacture of copper-nickel alloys for cold-working, should be free from all unreduced oxides.

CHROMIUM.

The colour of chromium is silver-white. Nickel has a little more of a yellow tinge to it than chromium. On the other hand, chromium will hold its colour indefinitely. Nickel will turn almost black in a laboratory atmosphere, and chromium will stay bright indefinitely. It is about the only metal known that will stay bright in a laboratory atmosphere. Not even platinum will stay as bright.

Electro-Deposition of Chromium.

In an article in the *Iron and Coal Trades Review* Mr. C. H. S. Tupholme gives the following hints on the electro-deposition of chromium. The cathode used was of iron, the bath at atmospheric temperature, and no mechanical stirring was used. In these conditions the most satisfactory concentration was obtained at 245 grammes per litre of chromic acid together with 3 grammes per litre of chromium sulphate. With this solution standardised the effect of varying the current density over the range between 2·7 and 54 amperes per sq. dm. or 25 to 500 amperes per square foot was tried, and it was found that good deposits were obtained up to 16 amperes per square dm. or 150 amperes per square foot. Beyond this current density the deposits were not so satisfactory, as the deposits were dark in colour and ultimately exhibited a tendency to crack. To obtain a bright deposit the most satisfactory current density was found to be between 100 and 125 amperes per square foot, using the electrolyte at ordinary atmospheric temperature.

ALUMINIUM.

For electrical conductors, and for a few special purposes, a grade of aluminium having a minimum purity of 99·5 per cent. is available. The lower grade, having a purity of about 99·2 per cent., is used for rolling into sheets for the manufacture of cooking utensils, chemical plant, etc. The impurities generally present in primary aluminium pig are: copper, iron, silicon, sodium, zinc, nitrogen, and a typical analysis of high purity aluminium is as follows:

Aluminium	99·5 per cent.
Iron	0·3 ,,
Silicon	0·2 ,,

The mechanical properties of cast aluminium are as follows:

Elastic limit	3–4 tons per sq. in.
Maximum stress	5–7 ,,
Elongation	3–4 per cent.

The tenacity of hard-rolled aluminium can be as high as 18 tons per square inch.

Primary aluminium is not used in the cast condition for constructional purposes. It is used as a deoxidiser in the manufacture of steel, and in the form of powder and dust in the thermit process and in paint. The principal use is in the wrought form, such as sheets, tubes, mouldings,

rods, wire, etc. As commercially pure aluminium has extensive application in the brewing and chemical industries, and also for domestic purposes, it is important to consider the action of acids upon it, which may be briefly stated as follows:

Hydrochloric Acid.—Attacked with great rapidity hot or cold and at all concentrations.

Hydrofluoric Acid.—Rapidly attacked.

Sulphuric Acid.—Attack slow with concentrated acid. Appreciably attacked with hot dilute acid.

Nitric Acid.—Attack very slow, the metal becoming passive (can be used for containers for cold strong HNO_3).

Acetic Acid.—Attack slow with cold acid, increases with use of temperature.

Lactic Acid.—Attacked very slowly (suitable for milk containers).

Oleic Acid.—Practically no effect.

Action of Air.—Attack very slow in fairly pure air, increases considerably in impure atmospheres. (The oxygen forms a thin film of oxide which tends to retard further action.)

Action of Alkalies.—Potassium and Sodium Hydroxide.—Rapidly attacked. Ammonium Hydroxide.—Attack slow. Alkaline Water.—Attacked appreciably by water in which alkalies or soap have been added.

Most metals will alloy with aluminium, but only a few definite alloys (aluminium—copper, iron, manganese, magnesium, nickel, silicon, tin, zinc) are used to a large extent. These include the binary alloys Al–Zn, Al–Cu, ternary alloys Al–Cu–Zn, Duralumin and related alloys, silicon and certain complex alloys. The so-called heavier alloys are the aluminium brasses and bronzes. Certain iron-aluminium bronzes and manganese-aluminium bronzes are also used.

Annealing Sheet Aluminium.

1. Prepare a mixture of ordinary whiting and oil, mixed to the consistency of thick paste. Coat the metal on both sides with this preparation, and place it on a clean fire or in a furnace, allowing it to remain until the oil in the paste becomes ignited. When this occurs withdraw the work from the furnace and allow it to cool gradually. The coat of whiting should then be removed from the surface of the metal by washing and brushing.

2. Heat the metal to a dull red colour, perceptible in the dark, and cool slowly. With this method care must be exercised to get an equal annealing temperature distributed over the entire surface of the work.

3. Heat the sheet aluminium until pine-wood sawdust just glows when sprinkled on it, which indicates a temperature of about 750° F. At this temperature a dry match-stick will just char when rubbed on the metal. (*A. Eyles*, '*Machinery*,' *Aug.* 11, 1921.)

The Protection of Aluminium from Corrosion.

Aluminium and its alloys can be efficiently protected against atmospheric corrosion under the most severe conditions only by the use of an enamel or varnish paint. Oil paints are very inferior in protective power to those referred to above.

MAGNESIUM.

Properties of Pure Magnesium.

	Pure Magnesium.		Pure Aluminium.	
	Cast.	Worked.	Cast.	Worked.
Ultimate strength. Tons per sq. in.	7·5	14·5	5·2	10·1
Yield point „ „	—	0·55	—	5·03
Elongation on 2 in., per cent.	6	6	35	16
Modulus. Lbs. per sq. in.	—	$6{\cdot}74 \times 10^6$	—	$10{\cdot}1 \times 10^6$
Rigidity modulus. Lbs. per sq. in.	—	$0{\cdot}58 \times 10^6$	—	$3{\cdot}45 \times 10^6$
Impact, Izod. Ft.-lbs.	—	4·0	—	24
Endurance limit, range of stress. Tons per sq. in.	—	± 3·48	—	± 4·69
Hardness, Brinell No.	20	42	23	45
Density. Grammes per c.c.	—	1·72	—	2·70
Thermal expansion per ° C.	—	25×10^{-6}	—	24×10^{-6}
Electrical resistivity. Microhms per cm.^{3}	—	4·59	—	2·84
Thermal conductivity (C.G.S. units)	—	0·34	—	0·52

NON-FERROUS ALLOYS.

COPPER AND ZINC.

The alloys of copper and zinc containing over 50 per cent. of copper are generally termed 'brasses.' The mechanical properties of these alloys have been the subject of numerous investigations; in this connection such authorities as Thurston, Charpy, Lohr Guillet, etc., should be consulted.

Below are given the results of Guillet's tests on the brasses. His tests were carried out upon material which was worked, then heated for ten minutes at 750° C. in a salt bath and finally cooled in air.

Alloy Composition.	Yield Point.	Max. Stress.	Elongation per cent. on 100 mm.	Hardness.
Copper.				
100 per cent.	2·09	14·28	52·5	48/49
98 ,,	1·64	14·41	58·0	48/50
96 ,,	2·59	15·27	57·5	·52
90 ,,	3·54	16·43	54·0	55/58
85 ,,	3·35	16·43	58·0	57/58
80 ,,	4·17	18·52	72·0	60/63
75 ,,	4·49	19·47	77·0	60/62
72 ,,	4·69	19·59	77·5	60/61
70 ,,	4·65	19·59	73·5	60/64
67 ,,	4·49	19·02	72·5	56/60
65 ,,	4·69	19·25	73·0	62
62 ,,	6·59	23·06	57·0	80/81
60 ,,	6·95	23·76	50·0	81/82
58 ,,	4·27	25·15	51·0	92/95
56 ,,	7·33	21·06	40·0	121/131

Figs. 13 and 14 (p. 2722) show the relation existing between the mechanical properties of the pure brasses and their copper content.

The casting temperatures of these alloys are of exceptional moment. The following tests show the influence of varying casting temperature on the tensile properties of 40·5 per cent. zinc-copper alloy castings poured from one crucible within a short interval of each other. Chemically the three castings were identical.

Casting Temperature °C.	Maximum Stress Tons per Square Inch.	Elongation per cent.
1,038	12·45	6·0
943	16·28	9·5
934	18·88	15·0

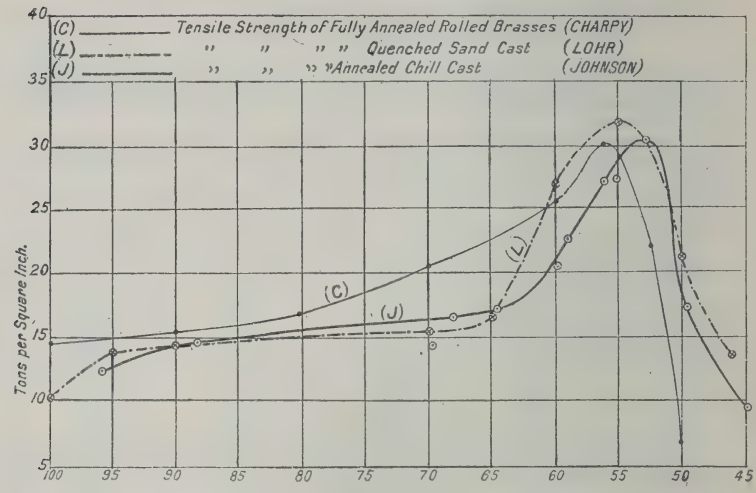
In every case with other alloys similar temperature variations have a similar effect on the mechanical properties. The following results obtained from manganese bronze were obtained by Guillemin :—

Casting Temperature °C.	Maximum Stress. Tons per Sq. In.	Elongation per cent.
1,400	15·2	13
1,300	19·7	45
1,250	21·6	35
1,200	22·9	30

In judging a suitable pouring temperature the appearance of the alloy in ladle or crucible may be used as a guide. With manganese bronzes this appearance is somewhat confused by the aluminium present, for, as is well known, this imparts a characteristic skin to the surface of the molten alloy. The practical effect of this is to make the alloy appear colder and less fluid than is really the case. Careful observation readily overcomes this difficulty.

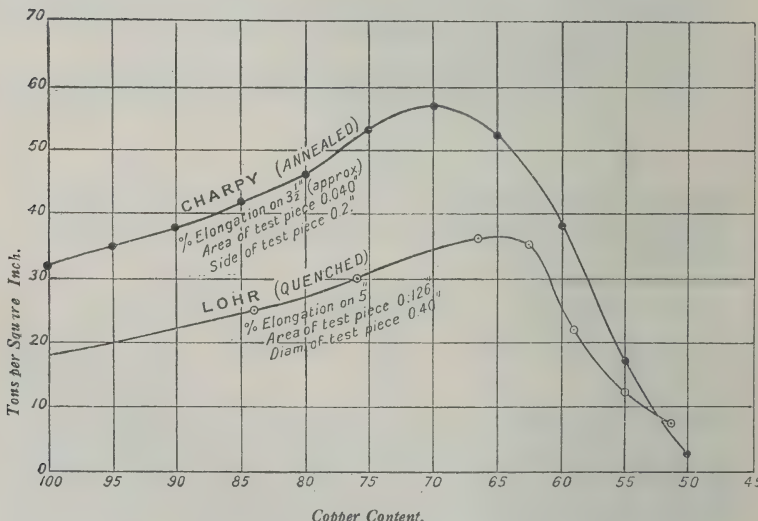
There is little doubt that the pyrometer is, in the hands of careful workers, the best means of determining the temperatures of metal or alloy in the ladle or crucible.

(See also Zinc-Tin Alloys, page 2740.)



Copper Content.

FIG. 13.



Copper Content.

FIG. 14.

Cast Brass.

Cast brass is somewhat variable in composition, but usually contains about 66 per cent. of copper, this percentage being that found in English standard brass.

The metals employed in the production of cast brass are not generally selected with such care as are those used in the manufacture of malleable brass. Lead, tin and iron are common impurities.

Lead in proportions varying from 1 to 3 per cent. is frequently added for the purpose of facilitating machining. It reduces the tenacity and hardness and increases the corrodibility of the resulting alloy, and is liable to segregate in those parts of the castings that solidify latest on cooling.

Tin tends to harden brass and is added in quantities up to 3 per cent., though 2 per cent. is not generally exceeded.

Iron hardens and increases the tenacity of brass.

The presence of more than one of these elements in quantity is to be deprecated, as their combined effects on the mechanical properties of the resulting alloy are likely to be injurious.

The mechanical properties of cast brass of average composition (66 per cent. of copper) and containing only small quantities of impurities or added elements are :—

Yield point, 6 tons per sq. in.; Tenacity, 14 tons per sq. in.; Elongation, 40 per cent.

A number of alloys, adapted for particular purposes, are tabulated below :—

Alloy.	Cu. Per cent.	Zn. Per cent.	Other Metals. Per cent.
Very delicate castings	86·00	14·00	—
Good yellow brass	75·00	25·00	Sn. 2·00
Name plates	86·00	9·00	Sn. 3·00; Pb. 2·00
Small castings (red)	90·00	10·00	,, ,,
Taps, cocks, etc.	78·00	14·00	Sn. 3·00; Pb. 3·00
Fine castings (yellow)	71·00	24·00	Sn. 2·00; Pb. 3·00
Hydraulic pumps	58·00	38·00	Sn. 2·00; Fe. 2·00

Ductile Brass.

The brasses containing upwards of 63 per cent. of copper are ductile and malleable at ordinary temperatures, provided that the metals used in the production of the alloys be of the best quality, and that the alloys be well annealed during working. The most suitable annealing temperatures for ductile brass are between 650° C. and 700° C.

Lead is admissible in alloys that are to be rolled into sheet, but is injurious in alloys that are to be drawn into wire. It should not be present in such quantity as is liable to create trouble at the annealing furnace.

The most ductile of the brasses contains 70 per cent. of copper and possesses the following mechanical properties when fully annealed subsequent to rolling :—

Yield point . 10 tons per sq. in.; Elongation . 60 per cent. on 2 ins.;
Maximum stress . 23 ,, ; Hardness (Brinell) 120.

Hot-forgeable Brass.

The alloys of copper and zinc containing from 59 to 61 per cent. of copper are characterised by their ability to withstand hot rolling at temperatures ranging from 500° C. to 800° C. They are employed in sheet form for the sheathing of wooden ships and in bar form for the manufacture of bolts, fastenings, etc. They are forgeable under the drop hammer and, hence, are frequently used in the manufacture of stampings. In this last connection it is particularly noteworthy that constancy of composition must be attained, on account of the large variations in machining properties accounted for by small changes in the copper content. The alloys in the range 57 to 63 per cent. of copper are forgeable, though not capable of withstanding rolling, in the temperature range already stated, and may be considered here.

All the above alloys are retained in a harder, stronger, and less ductile condition if quenched from temperatures between 750° C. and 800° C. than if allowed to cool slowly. They are liable to spontaneous cracking if worked at temperatures below 600° C., which liability, if thought to be inherent, can be lessened if the alloys are annealed at temperatures between 200° C. and 300° C., subsequent to working. The best temperatures for annealing, with a view to reducing variations in the mechanical properties due to variations in the mechanical and thermal treatment, are below rather than above 650° C. The temperature range (750° C.–800° C.) is, according to Stead, a dangerous one in which to conduct the annealing of hot-forgeable brass.

The average mechanical properties of fully annealed cast 60/40 brass are as follows :—

Yield point . 10 tons per sq. in.; Elongation . 35 per cent.;
Maximum stress . 25 ,, ; Hardness (Brinell) . 85.

Drawn rod of 60/40 composition is obtainable having the following average mechanical properties :—

Yield point	. 28 tons per sq. in.;	Elongation .	. 20–30 per cent. on 2 ins.;
Maximum stress	. 32 ,, ;	Hardness (Brinell)	140.

The last factor appears to be dependent on the diameter of the rod in question, being least when the diameter of the rod is greatest. On account of the almost infinite variety of hot- and cold-working possible in the treatment of these alloys almost infinite variations in mechanical properties are obtainable.

Effect of certain Metals on Brass.

Tin raises the tenacity, increases the hardness and resistance to corrosion and lessens the elongation of the 60/40 alloy, to which it is added commercially to make Naval brass. The latter contains 60–62 per cent. copper, 1–1½ per cent. tin, and the rest zinc. Not more than 2 per cent. of tin should ever be present ; 4 per cent. induces great brittleness.

Iron increases the tenacity and hardness of brass. The difficulty of dissolving iron in brass was first overcome by a special process, the invention of Dick, whose alloy, generally containing 55–57 per cent. copper, 42 per cent. zinc, 1–2 per cent. of iron and traces of phosphorus, is well known.

Iron is now generally added to brass in the form of ferromanganese, an alloy of iron and manganese which is readily dissolved in molten brass.

Manganese is added to brass with a view to improve the mechanical properties of the same. The resulting alloys are divisible into two classes :

(*a*) Alloys in which either no manganese or traces only of manganese are present, *i.e.* alloys to which the element has been added for purposes of deoxidation.

(*b*) Alloys in which appreciable quantities of manganese are retained in solution. These may be subdivided into (i) such as are suitable for rolling into rod and containing up to 0·05 per cent. of manganese, and (ii) such as are suitable for casting and containing less than 60 per cent. of copper, and up to 2·0 per cent. of manganese.

The manganese is introduced into the alloys by the addition of ferromanganese or cupro-manganese.

These alloys are forgeable when heated to temperatures above 750° C. They resist the corrosive action of sea water, and are employed in the manufacture of ships' propellers. Iron, aluminium and tin are generally present in manganese ' bronze.'

' Delta ' Brand Alloys.

The metals and alloys sold by the Delta Metal Co., Ltd. (of London and Birmingham), under their registered trade-mark ' Delta,' are employed for a very great variety of purposes, the composition varying widely according to the requirements and purposes for which each of these metals or alloys is more particularly adapted. Some of them are based upon the introduction and chemical combination of definite quantities of iron and other elements in copper-zinc alloys. The effect of iron on brass has already been noted ; the difficulty of introducing it in definite quantities was first overcome by the late Mr. Alexander Dick, the founder of the Delta Metal Co., Ltd., who adopted the name ' Delta ' (the Greek equivalent of the initial letter of his name) as a trade-mark. In most of the alloys which may be classed under the ' Brasses,' the copper content lies between 57 and 62 per cent., that of zinc between 42 and 38 per cent., whilst some contain iron, phosphorus, manganese, or other metals, up to a total of about 2 per cent.

The following alloys, designated by the trade name ' Delta,' and manufactured by the Delta Metal Co. Ltd., find wide application.

Delta Bronze Nos. I and IV are practically incorrodible, and consequently of very great durability, even when used in strongly corrosive mine-waters, etc. ; they are not attacked by sea-water, and resist acids much better than other copper alloys. On account of their great strength and resistance to corrosion, these alloys are largely used for parts of marine engines and mining and hydraulic plant which require a strength not less than that of the best steel, but at the same time freedom from corrosion—such as pump rods, piston rods, shafts, cranks, plungers, spindles, rams, etc. For propellers and propeller blades, Delta bronze is in great demand, and it is also successfully used for parts of ordnance, etc.

These alloys are malleable bronzes of very great strength, exceeding, even in the cast state, the strength of mild steel. They can be cast with facility, and will produce perfectly sound and homogeneous castings. When heated to a dull red heat, they become soft and highly malleable, and can then be readily forged, stamped, rolled, or pressed, to any extent required. These operations increase the strength considerably, as will be seen by the following copies of tests.

Description of Metal.	Diameter. In.	Area. Sq. in.	Ultimate Strength. Tons per sq. in.	Elongation in 2 ins. Per cent.	Reduction of Area. Per cent.
		Delta Bronze, No. I.			
Cast . . .	0·500	0·1963	41·30	20·0	20·4
Extruded bars .	0·502	0·1979	49·82	26·0	24·9
		Delta Bronze, No. IV.			
Cast . . .	0·611	0·293	23·89	21·0	20·1
Extruded bar .	0·500	0·1963	37·1	27·0	20·0

Extrusion Process.

The process of extruding copper and its alloys, the invention of the late Mr. Alexander Dick, developed and worked by the Delta Metal Co., Ltd., of London and Birmingham, under numerous subsequent new patents and improvements, consists of the pressing or squeezing of heated metals through dies of any desired cross-section, by hydraulic pressure varying according to the section to be produced and the character of the metal operated upon.

Dick's extruded bars have a perfectly smooth and uniform surface, free from cracks or other defects; size and shape are of the greatest accuracy, and owing to their superior finish, machining is reduced to a minimum. The Company are able to produce extruded rods as ordinary round or pinion wire weighing a fraction of a pound per foot per run, to heavy round, square or hexagon rods, or other sections, more or less complicated, weighing as much as half a cwt. and over per foot. The facility of manipulation ensures rapid output, so that sections not regularly kept in stock can be supplied at very short notice.

The Delta Metal Co., Ltd., are not only the original, but claim to be also the largest manufacturers of extruded rods and sections. They supply these in various metals and alloys: in 'Delta' bronzes Nos. I, II, IV, and VII, Heagard bronze, manganese and aluminium bronze, brass, yellow metal, naval brass, etc.

They make a speciality of high-speed turning and screwing brass ('Delta' Brand and 'Dixtrudo' Brand) in every variety of shapes and sections. This material, like the stampings and pressings in brass, bronze, and copper made by the Delta Metal Co., Ltd., is used largely for repetition work and mass production, and supersedes castings.

BULL'S METALS.

(Bull's Metal & Melloid Co. Ltd., Glasgow.)

Bull's Metals are strong, malleable alloys of good casting qualities, and equal to good gunmetal in resistance to corrosion. They are manufactured in two qualities, No. II and No. III.

Alloy No. II is suitable for founding purposes, and extensively used for propellers and propeller blades, parts of guns and gun carriages, requiring the strength of cast steel with great toughness and resistance to corrosion.

Alloy No. III is very malleable and can be rolled, forged, stamped, or otherwise wrought at a red heat; this metal also casts well. Its composition ensures great resistance to corrosion and wear. It is absolutely non-magnetic.

Rolled in round, hexagon, angle, and section bars, or into sheets and plates; forged or stamped Bull's metal is used for pump rods, centrifugal pump spindles, rams, stern shafts, sluice valve spindles, parts of marine and hydraulic machinery generally. Forged propeller blades, propeller studs and nuts. Bolts, nuts, screws, studs, and nails. Parts of torpedoes and torpedo boats, ordnance, ammunition, etc. Condenser plates, strainer plates, doctor blades. Hull plates and angles for launches, torpedo boats, yachts, sailing and rowing boats.

The elastic and ultimate resistance to tensile, crushing, and torsional stresses are greater in the case of wrought Bull's metal than in mild steel, and higher working strains may therefore be used with Bull's metal. Where resistance to the action of water or chemicals enters as an important factor, Bull's metal, whether wrought or cast, becomes specially superior to steel.

TENSILE TESTS OF BULL'S METALS.

Dimensions.	Area. Square inch.	Reduction area at fracture.	Extension.	Ultimate Stress.	Remarks.
Diam. Inch.		Per cent.	Per cent.	Tons per sq. in.	
0·749	0·4406	22·1	23·6 on 8 ins.	31·41	Forged.
0·623	0·3048	37·9	8·1 ,,	39·93	Rolled, hard finish.
0·690	0·3739	37·8	20·0 ,,	25·01	Rolled, medium finish.
0·688	0·3717	30·6	29·0 on 6 ins.	31·47	Cast medium.
0·750	0·4418	24·2	22·0 on 3 ins.	38·68	Cast hard.

'MELLOID.'

(Registered Trade Mark.)

This alloy is a malleable genuine bronze containing no inferior metal; its resistance to corrosion is very great. Its tensile strength in a rolled state when suitably finished is as high as that of mild steel; the strength is not appreciably affected at the temperature of high-pressure steam, whilst copper at such heat is very much reduced in strength and elasticity.

The alloy is perfectly malleable at *all* temperatures, from cold to blood-red heat; it has no brittle stage at a black heat like most metals and alloys, and at a red heat it is much superior in strength and toughness to mild steel under such conditions, and the alloy, under all conditions, is exceedingly tough.

'Melloid' meets the mechanical requirements referring to copper, naval brass, and rolled bronze specified by the Admiralty, the principal railway companies, and consulting engineers.

It is claimed that 'Melloid' will be found superior to all brass alloys, including naval brass, yellow or Muntz metal, and also to wrought copper, wherever a combination of toughness, great strength, and a maximum resistance of corrosion is required.

NICKEL ALLOYS.

Ferro-Nickel Alloys.

These alloys generally contain 25–30 per cent. nickel, 0·3–0·5 per cent. carbon. They are austenitic, and not subject to heat-treatment. They are tough and strong, while some have a low thermal expansivity and are very resistant to corrosion in air, fresh and sea water. They may be forged or rolled, but not so readily as ordinary steel.

Average tensile properties in the normal condition (not heat-treated):—

	25–28 per cent. Nickel.	30–35 per cent. Nickel.	35–38 per cent. Nickel.
Tensile strength, tons per sq. in.	38–41	38–42	45–51
Yield point, tons per sq. in.	15·5–22	18–22	28–35
Elongation on 2 in. per cent.	30–35	30–40	25–35
Reduction of area, per cent.	50–60	40–60	50

Alloys containing about 25 per cent. nickel are used for electrical resistance wire in the construction of rheostats and electrical heaters. Within the range of composition from 20–30 per cent. nickel, the ferro-nickel alloys may be readily obtained in a non-magnetic condition by cooling at normal rates from rolling or forging temperatures.

Permalloy.—A nickel-iron alloy, nickel 78·5 per cent., iron 21·5 per cent., with impurities as low as possible. Heat-treatment. Heat to 200° C. for 1 hour, cool slowly, reheat to 600°C. cool at a definite rate in air. This alloy possesses high magnetic properties, the maximum permeability is largely in excess of the highest values found for silicon steels. The presence of impurities affects the permeability and carbon is especially harmful, but the effect of changes in the heat-treatment are much greater than those due to small quantities of impurities.

Typical composition of Permalloy: nickel, 78·23; iron, 21·35; carbon, 0·04; silicon, 0·03; manganese, 0·22; sulphur, 0·035; phosphorus, trace; cobalt, 0·37; copper, 0·10.

μH values for Permalloy, 50 per cent. nickel alloy, 4 per cent. silicon-iron alloy, and pure iron.

Magnetising Force H. Gilberts per cm.	Permeability $\mu = \frac{B}{H}$			
	Permalloy.	50 % Ni.	4 % Si–Fe.	Pure Iron.
0·05	74,000	69,000	5,000	2,400
0·1	50,000	60,000	7,000	3,600
0·2	30,000	39,000	13,000	10,000
0·4	18,000	23,000	15,500	26,500
0·6	13,000	16,500	14,500	21,500
0·8	10,000	13,000	11,000	16,500
1·0	8,000	10,000	10,000	14,000

Mumetal.—A nickel-iron alloy containing copper and manganese. Its initial permeability somewhat less than that of the binary alloy, but has a higher electrical resistivity which is of value in keeping down eddy-current losses.

The advantage of the nickel-iron alloys are pronounced in very low magnetic fields and are particularly well marked in respect of 'initial permeability'—*i.e.* the permeability in fields whose

strength approaches zero. Ternary nickel iron alloys containing such elements as aluminium, cadmium, chromium, cobolt, copper, magnesium, molybdenum, tantalum, titanium, tungsten, vanadium, etc., have been developed to meet special requirements.

In general, the alloys must be made from as pure metals as possible, the presence in the alloy of certain impurities, such as sulphur and carbon, being very detrimental. Heat-treatment is of great importance, also care must be taken to protect metal from stress or work after treatment. An alloy with 81 per cent. nickel is less sensitive in this direction, and the addition of molybdenum also helps.

Permax.—Nickel-iron alloy containing added elements. It does not possess the high permeability of some of the other alloys, but is remarkably constant in certain of its magnetic properties —*e.g.* under fields varying from 200 gauss to 1 gauss, the coercive force changes only from 0·48 to 0·45.

Invar.—See p. 2703.

Elinvar.—An alloy of 36 per cent. nickel and about 12 per cent. chromium (or its equivalent where small quantities of manganese, tungsten, or carbon are used in conjunction with the chromium additions).

At ordinary temperatures has a practically invariable elastic modulus coupled with a very low thermal expansivity.

Dilver.—Forty per cent. nickel, has a coefficient of thermal expansion approximately that of ordinary glass.

Platinite.—Forty-six per cent. nickel, has a coefficient of thermal expansion equal to that of platinum.

A.M.F. Alloy.—Fifty-seven per cent. nickel, has the same thermal expansion as steel but more stable and less liable to corrosion.

Nickel Silver.

The most harmful impurities are lead (except in rods or castings intended for machining, in which case lead up to 3 per cent. is purposely added to improve the machining quality of the alloy), sulphur, and carbon, and precautions should be taken that these elements are kept within very narrow limits. Tin, iron, and unreduced oxides may also be present as impurities.

Melting operations are carried out usually in coke-fired pit type crucible furnaces. Electric furnaces are also used, and the metal is cast into iron moulds in forms suitable for rolling sheets or rods. The rolling and drawing of nickel silver is carried out at normal temperatures, since none of the ordinary commercial alloys are capable of being hot-worked.

Annealing temperature: Nickel, less than 14 per cent., 600–650° C.
Nickel, 16 per cent. and upwards, 700–750° C.

Average mechanical properties (nickel 18 per cent. and up) cold-worked (rolled or drawn) and fully annealed. Yield stress, 15 tons per sq. in. Ultimate stress, 26–30 tons per sq. in. Elongation on 2 ins., 35–40 per cent. Reduction of area, 45–50 per cent. With nickel less than 18 per cent.: Yield stress, 12–14 tons per sq. in. Ultimate stress, 23–26 tons per sq. in. Elongation on 2 ins., 50–55 per cent. Reduction of area, 50–55 per cent.

The mechanical properties of nickel silver are not as important as other physical properties —*i.e.* whiteness of colour and brilliance of polished surface, resistance to corrosion.

The alloy possessing the finest silvery whiteness is that produced from a specification of approximately equivalent weights of nickel, copper, and zinc. This composition is only suitable for castings. It is not so important commercially as the alloy coming next to it known as B.B. and containing 30 per cent. nickel. In general, the colour varies directly with the nickel content, those alloys of highest nickel content giving the whitest colour and most lustrous polish.

Results of research show that the tendency of commercial nickel silver to 'burn' is increased (*a*) by increasing the nickel content, (*b*) as the ratio of the zinc to the copper is increased, and (*c*) with the amount of impurities present.

The following alloy is stated to be possessed of the best all-round properties:—

Copper, 46 per cent.; Nickel, 39 per cent.; Zinc, 20 per cent.

Platnam.

Consists of 54 per cent. nickel, 33 per cent. copper, and 13 per cent. tin. It is used for the manufacture of valve seats and valves which have to withstand the action of superheated steam.

Nickel Chromium.

These alloys have a composition which varies considerably according to the purpose for which they are required, but in those of commercial utility the nickel content lies within the limits of 60 per cent. and 80 per cent. and the chromium content beetwen 12 per cent. and 22 per cent.

The balance consists mainly of iron, with small percentages of manganese, silicon, aluminium, and carbon, other elements being present only as traces of impurity.

A typical alloy has the approximate composition of nickel 66 per cent., chromium 20 per cent., iron 10 per cent., the balance consisting of the above-mentioned minor components.

Physical Properties.

Tensile strength in cast state . . .	25 to 30 tons per sq. in.
Melting-point	1300° to 1350° C.
Density	·29 lbs. per cu. in.
Specific heat	·11

These alloys have two outstanding properties which have great commercial value, namely: their strength is well maintained at high temperatures, and they are chemically inactive under a wide range of conditions even at high temperatures. Thus they resist the attack of furnace gases to a marked degree, and unless sulphur is present in unusually large quantities, their life at temperatures up to 1200° C. is very considerable.

The actual life of articles made of this alloy is greatly dependent upon conditions, but a period approaching 10,000 hours at such temperatures is by no means unusual.

The above properties make the alloys eminently suitable for the manufacture of such articles as pyrometer sheaths, case-hardening boxes, furnace parts, retorts, muffles, in which rôles they have been of great assistance to engineers in overcoming difficulties due to the severe conditions of temperature and corrosion which modern works practice produces.

Such alloys have long been recognised, when drawn into wire or rolled in strip form, to be extremely valuable for such electrical purposes as windings for electric furnaces and stoves.

The alloys are frequently of service to chemical engineers for use with corrosive fluids at normal temperatures, and have found application in the manufacture of such articles as acid pumps and valves subject to attack by acids and acid vapours.

An alloy designated the 'Emperor' chrome nickel alloy is manufactured by Messrs. Sutcliffe, Speakman & Co., Ltd. and is of great durability.

A large number of pyrometer tubes, case-hardening boxes, and similar utensils made of the alloy are in use in some of the principal works of the country.

Heat- and Acid-Resisting Alloys.

The following table gives the analysis of some typical commercial alloys:—

Alloy No.	C.	Mn.	Si.	Cr.	Ni.	Co.	Al.	Cu.	W.	Mo.	Fe.
1	0·94	1·1	1·9	21·0	68·0	—	—	—	—	—	7·0
2	0·27	0·06	3·92	29·1	62·4	—	—	—	—	—	4·9
3	0·09	0·02	0·46	15·6	81·2	1·56	0·42	—	—	—	0·8
4	0·50	0·79	0·80	13·68	5·75	—	—	—	—	—	27·3
5	—	0·98	1·04	21·07	60·7	—	1·09	6·42	2·13	4·7	0·76
6	1·13	2·2	0·63	15·5	64·9	—	—	—	—	—	15·8
7	1·37	1·90	0·49	8·5	40·0	—	—	5·5	—	—	42·0
8	0·60	1·88	1·17	13·2	59·9	—	—	—	—	—	23·2

Alloy No. 1 has the most general useful analysis. Silicon rather higher than necessary. Narrow freezing range solidification complete about 1325° C.

No. 3 alloy is suitable for the manufacture of wire, but is expensive. The cobalt probably due to impurity in the nickel. Melting range, 1230–1390° C.

No. 4 alloy is rather poor, suitable for resistance wire.

No. 5 alloy, generally known as Parr's Alloy, developed specially for the manufacture of Mahler bombs.

No. 6 alloy is an average alloy. The carbon is high, showing that a low-grade ferro-chrome was used, and the manganese is also unnecessarily high.

No. 8 alloy is a poor type of alloy. Iron far too high and nickel very low. The copper should not be added to an alloy intended for general purposes; it is useless in a heat-resisting alloy.

Maximim tensile strength of alloy No. 1 in cast state, 25–30 tons per sq. in.; alloy No. 4 in rolled sheets; yield stress, 28 tons per sq. in.; ultimate strength, 50 tons per sq. in. Elongation, 8 per cent. Reduction of area, 12 per cent.

Practically all the alloys are capable of being softened and rendered more ductile by heat-treatment, and in that respect behave as most non-ferrous alloys—*i.e.* they become more ductile when quenched. (*Kayser.*)

Acid-Resisting Alloys.

The corrosion-resisting properties vary according to their composition, but if the silicon alone does not exceed 2 per cent., and the combined carbon, silicon, and manganese 4 per cent., the properties of these alloys are very similar. Nickel-chromium alloys offer a perfect resistance to the corrosive action of the atmosphere in the presence of either hard or soft water, or sea-water. In the case of stainless steel it is necessary that it be correctly heat-treated and free from surface scale or pit marks.

No heat-treatment is necessary for nickel-chromium alloys, and it is not necessary even to polish or grind the surface.

ACID-RESISTING ALLOYS.

Reagent.	Temperature. ° C.	Nickel-Chromium Alloys.	Stainless Steel.
Acetic acid, 20 per cent. solution	15–20	No attack visible	Attacked
	100	General etching	Attacked
Citric acid, 30 per cent. solution	15–20	No attack	Marked corrosion in 24 hours
	100	Stained yellow	Marked corrosion in 24 hours
Nitric acid, 1·20	15–20	Etched	Etched
	100	Attacked	—
Sulphurous acid	15–20	Discoloured	Rapid attack
Dry battery mixture	15–20	Slight pitting	Marked corrosion
	100	Pitting	Marked corrosion
Acetic anhydride	15–20	No visible attack	Badly pitted in 5 hrs.
	100	No visible attack	Badly pitted in 5 hrs.
Sulphuric acid, 50 per cent. solution	15–20	General attack	Rapid attack
	100	Badly corroded	Rapid attack
Phosphoric acid, 75 per cent. solution	15–20	No visible attack	Attacked
	100	Stained	General attack
Ammonium sulphate, 50 per cent. solution	15–20	No attack	General attack
	100	Brown stain	General attack

(*Kayser.*)

Platinite.

Platinite is an alloy of iron, nickel, and carbon, which is used extensively for scientific instruments for standard measurements of length because of its peculiar quality of being practically non-expansive when heated to high temperatures. It is also used in incandescent electric lamps where the wire connections fused into the glass must have so small a coefficient of expansion as not to cause fracture in the glass. This alloy contains about 46 per cent. of nickel and 54 per cent. of iron, with a very small percentage (0·15 per cent.) of carbon.

Monel Metal.

(C. & J. Weir, Ltd., Cathcart, Glasgow.)

This is a natural nickel alloy, and which, after refining, has an average composition of nickel 67 per cent., copper 28 per cent., iron 3 per cent., manganese 2 per cent.; a small percentage of carbon may also be present. In appearance it is very similar to pure nickel and takes the same high finish, but has a slightly softer and somewhat more silvery lustre. It is as strong, tough, and ductile as steel, and under corroding influences far superior to copper, gun-metal, and bronze. It machines readily, and can be rolled, drawn, cast, forged, soldered, brazed, and welded.

Average tensile properties: Hot-rolled metal, 30–42 tons per sq. in., according to size and treatment. Ordinary cast metal, 19–25 tons per sq. in., according to character of casting.

Monel metal resists sea-water, alkalies, and some dilute acids, but is not suitable where nitric acid is present. One of the most valuable properties is its tensile strength at elevated temperatures. For example, at a temperature of 400° C., a forged rod has a tensile strength of over 35 tons (see page 2705).

Average Physical Properties.

Melting point, 1360° C.
Pouring temperature of castings, 1500° C.
Forging temperature, 1040–1100° C.
Annealing temperature, 700–900° C.
Specific gravity, 8·82.
Weight per cub. in. = ·323 lbs.
Shrinkage allowance for castings, $\frac{1}{4}$ in. per foot.
Coefficient of expansion, 0·0000137 per ° C.
Electrical resistivity, 48 michroms per cm.^{3}
Electrical conductivity, 3·4–5 per cent. of copper.
Heat conductivity, $\frac{1}{15}$th that of copper.
Brinell: Cast metal, 120–140.
Hot-rolled sheets and rods, 150–190.
Cold-drawn bar, 200–217.

Monel metal in the rolled condition is magnetic at ordinary temperatures, but becomes non-magnetic when heated from 100–150° C. The magnetic change is reversible, the metal becoming magnetic on cooling. Hysteresis tests show that under a magnetising force (H) of 100 C.G.S. units the number of lines per cm.^{2} (B) is 2,460. The remanence is found to be 990, and the coercive force 1·1.

Casting.—Melting practice, also gates and risers similar to that used for steel. The care required for moulding and ramming more nearly resembles that for non-ferrous metals. Core must be collapsible to allow free shrinkage of the metal.

Metal can be melted in graphite crucibles, using (*a*) oil-fired pit furnaces; (*b*) oil-fired clay brick-lined reverberatory furnaces; or (*c*) in clay brick-lined or basic-lined electric arc furnaces. All ladles and crucibles used for transferring the metal should be preheated to conserve the heat in the metal.

Forging.—Practice similar to that for steel. The metal is harder under the hammer than mild steel and approaches closer to the nickel steels in behaviour. Oxide scale formed is strongly adherent and will not free itself even under hammer blows with steam and salt. Metal becomes hot-short just above maximum forging temperature.

Annealing should be carried out in a reducing atmosphere. General practice is to anneal in a cast-iron box covered with fine charcoal and a cover sealed with fireclay.

ALUMINIUM.

Aluminium Brass.

Aluminium acts as a deoxidiser and is added to cast and hot-forgeable brass, for the purpose of improving their mechanical properties, in quantities varying from 0·3 to 1·5 per cent.

Typical tests of cast aluminium brass, containing 38 per cent. of zinc, 2 per cent. of aluminium, and 2 per cent. of lead, are :—

Yield point	18 tons per sq. in.
Maximum stress	34 ,,
Elongation	12 per cent.

Melting Points of Commercial Brasses.

(*Gillett & Norton.*)

	Copper.	Zinc. (as desired).	Tin.	Lead.	Copper.	Zinc. (by analysis).	Tin.	Lead.	No. of Tests.	Melting Point.
Red brass	85	5	5	5	—	—	—	—	8	970°C.
Low-grade red brass	82	10	3	5	81·5	10·4	3·1	5·0	4	980°C.
Half yellow, half red	75	20	2	3	75·0	20·0	2·0	3·0	3	920°C.
Cast yellow brass	67	31	—	2	66·9	30·8	—	2·3	4	895°C.
Naval brass	61·5	37	1·5	—	61·7	36·9	1·4	—	5	855°C.
Manganese brass	Copper 56, Zinc 41, Iron 1·5, Tin 0·9 per cent. Aluminium 0·45, Manganese 0·15 per cent.								6	870°C.

COPPER AND TIN ALLOYS.

Gun-metal.

The alloys of copper and tin containing upwards of 86 per cent. of copper are possessed of high tenacity and elasticity. Those containing from 89 to 91 per cent. of copper are the most important and are known as 'gun-metals.'

The addition of tin to these bronzes results in an increase in the hardness of the resulting alloy. Gun-metal is rarely manufactured from specially selected metals and frequently contains lead, zinc, etc. The Admiralty specification for gun-metal is as follows :

Copper	88 per cent.
Tin	10 ,,
Zinc	2 ,,

the specified tests for this material being

Maximum stress	14 tons per sq. in.
Elongation	7½ per cent.

Gun-metal containing 90 per cent. of copper and 10 per cent. of tin should give the following tests :—

Yield point	7 tons per sq. in.
Maximum stress	15 ,,
Elongation	15 per cent. on 2 ins.

Bell-metal.

The alloys of copper and tin containing from 75 to 86 per cent. of copper are used in the manufacture of bells ; iron, zinc and lead are frequently added to the alloys either for cheapness or variety of tone.

Effect of Certain Elements on Bronze.

ZINC.

Gun-metal, Bearing Metal, etc.

The alloys of copper, tin and zinc are rarely, if ever, free from elements such as lead, iron, etc. These alloys may be divided into two classes :—

(i) Alloys containing from 7 to 12 per cent. of tin, up to 6 per cent. of zinc, and up to 2·5 per cent. of lead. These alloys, among which may be noted gun-metal to the Admiralty specification, are used in the manufacture of castings, such as cocks, stuffing-boxes and pump barrels, and for bearings which are subject to but little friction.

(ii) Alloys containing from 12 to 20 per cent. of tin, up to 5 per cent., though rarely more than 2·5 per cent., of zinc and up to 5 per cent. of lead. These alloys are used in the manufacture of heavy bearings, e.g. railway wagon and locomotive axle boxes. The majority of alloys of this class contain from 14 to 16 per cent. of tin and from 1·5 to 2·5 per cent. of zinc.

Zinc combines with the oxides and free oxygen of the bronzes, producing greater fluidity in the molten state, and hence freedom from ' pinholes ' in the solid state.

LEAD.

Bearing Metal, etc.

The best bronzes should contain not more than 0·15 per cent. of lead, on account of its deleterious effect on the mechanical properties of these alloys generally.

Bearing metals having a bronze matrix and containing up to 40 per cent. of lead are in common use. A usual composition for bearings of this type is :—

Copper	64 per cent.
Lead	30 ,,
Tin	5 ,,
Nickel	1 ,,

The nickel is added to prevent the liquation of the lead. It increases the rate of cooling of the alloy and thus assists in the retention of the lead in the finely divided form.

Lead does not alloy with bronze, but separates out of the molten mass, as the same cools, in the form of globules. The lead bronzes may, therefore, be considered to consist of a network of bronze whose meshes enclose the lead content of the alloys in question.

PHOSPHORUS.

Phosphor Bronze.

Phosphorus acts as a deoxidiser and is added in small quantities for the purpose of deoxidation to copper. Not more than 0·15 per cent. is allowable.

The term ' phosphor bronze ' is applied to the alloys grouped below.

(i) Bronzes to which phosphorus has been added as a deoxidiser and containing either no phosphorus or merely traces of the element.

(ii) Bronzes containing less than 9 per cent. of tin and up to 0·25 per cent. of phosphorus. These alloys are forgeable and are relatively non-corrodible. They are employed for rolling and drawing into wire and rod, and in the manufacture of general castings; in the latter case lead is occasionally added in quantities up to 5 per cent. The forgeable alloys of this group generally contain less than 5 per cent. of tin.

(iii) Bronzes containing from 8 to 14 per cent. of tin and from 0·25 per cent. to 1·00 per cent. of phosphorus. These alloys are employed for casting purposes in the manufacture of machine parts likely to be exposed to great friction. Lead up to 5 per cent. is added in certain cases.

Phosphorus is added to these alloys in the form of phosphor copper or phosphor tin.

The tensile strength of phosphor bronze varies from 14 to 18 tons per square inch, and the elongation from 6 to 10 per cent.

'Delta' Brand Alloys.

(The Delta Metal Co. Ltd.)

Delta Bronze No. V. is an improved bearing metal of great strength and durability, and comparatively low price. It is particularly suitable for reducing friction at heavy pressures and high speeds, and is used for worm wheels for large cranes and other heavy machinery, hot neck rolls, locomotive slide valves, axle box, coupling, and connecting rod bushes, railway and carriage bearings, bushes, sliding surfaces, and other parts of machinery subject to heavy frictional wear. The metal is easy to tool, and requires no special precautions in casting.

Another metal used in lieu of phosphor bronze or gun-metal is 'Delta' Bronze No. VI., which produces fine, close-grained castings of great strength.

Melting Points of Commercial Bronzes.

(Gillett & Norton.)

	Copper.	Tin. (as desired).	Lead.	Zinc.	Copper.	Tin. (by analysis).	Lead.	Zinc.	No. of Tests.	Melting Point.
Gun-metal	88	10	—	2	—	—	—	—	4	995°C.
Leaded gun-metal	85·5	9·5	3	2	85·4	9·7	3·0	1·9	6	980°C.
Leaded bronze	80	10	10	—	—	—	—	—	3	945°C.
Bronze with zinc	85	10	—	5	84·6	19·4	—	5·0	4	980°C.

COPPER AND ALUMINIUM ALLOYS.

Aluminium Bronze.

The alloys known as the 'aluminium bronzes' are solid solutions of copper and aluminium containing not less than 88 per cent. of copper. Of these alloys that containing 10 per cent. of aluminium is the strongest, though the alloys containing less than this percentage are of industrial importance. These 'bronzes' are of use, firstly, on account of their valuable mechanical properties and, secondly, on account of their resistance to corrosion. They are malleable both when hot and when cold; the alloys containing between 8 and 12 per cent. of aluminium are affected in a marked degree by heat-treatment.

Following are tabulated the mechanical properties of some of these alloys :—

Percentage of Aluminium.	Yield Point. Tons sq. in.	Max. Stress. Tons sq. in.	Elongation. Per cent. on 4 in.	Remarks.
2·5	—	20·0	40·0	Cast.
5·0		25·0	55·0	Cast.
10·0	18·0	35·0	25·0	Cast in chill.
10·0	18·0	46·0	25·0	Cast rod reduced 25 per cent. by forging hot. Annealed.
10·0	15·0	40·0	30·0	Rolled bar.
10·0	40·0	50·0	3 0	Rolled bar quenched at 900° C.

The following are tests of the 'R3' and 'R4' aluminium bronzes, manufactured by the British Aluminium Company :—

R3 Bronze.—Average breaking load, 34·04 tons; average elongation, 21 per cent.; maximum breaking load, 37·6 tons; maximum elongation, 35 per cent.; minimum breaking load, 32·8 tons; minimum elongation, 12·5 per cent.

R4 Bronze.—Average breaking load, 41·8 tons; average elongation, 9·75 per cent.; maximum breaking load, 43·7 tons; maximum elongation, 13 per cent.; minimum breaking load, 39·8 tons; minimum elongation, 6·5 per cent.

COPPER AND MANGANESE ALLOYS.

Manganese Bronze, Manganin.

The term 'manganese bronze' is applied to copper-zinc alloys of 60/40 composition to which small quantities of manganese have been added. There are, however, alloys of copper and manganese of industrial application which rightly should be admitted under this heading. The alloys of copper and manganese containing up to 5 per cent. of the latter element are harder and tougher than is copper. Those containing from 2 to 4 per cent. of manganese are frequently employed in situations where high temperatures are existent, on account of their retention of their mechanical properties under these conditions.

The following figures represent the results obtainable by stamping certain of these alloys at a red-heat (*circa* 800° C.) :—

Copper.	Manganese.	Maximum Stress.	Elongation. Per cent.
97	3	18·5	32
96	4	21·0	60

Manganin is the name applied to an alloy of copper, manganese, nickel and iron, characterised by low electrical conductivity and almost zero temperature coefficient. It is employed in the construction of electrical resistance coils. The composition is somewhat variable, but the following figures will serve to indicate its approximate analysis :—

Copper . .	80 to 84 per cent.	Nickel	up to 12 per cent.
Manganese . .	4 to 15 ,,	Iron	Difference.

COPPER AND NICKEL ALLOYS.

Cupro-nickel, Constantan, etc.

The alloys of copper and nickel containing upwards of 50 per cent. of copper are of industrial importance. The alloys may be divided into three classes.

(i) Alloys containing up to 5 per cent. of nickel and having the property of resisting high temperatures without undue deterioration. Of these alloys those containing from 2 to 3 per cent. of nickel have been employed in the manufacture of locomotive firebox tubes. Following are the mechanical properties of the 2 per cent. alloy :—

Maximum Stress.	Elongation. Per cent.	Remarks.
45·0	5·0	Hard rolled.
30·0	45·0	Hard rolled and annealed at 650° C.

Those of the 5 per cent. alloy are very similar in character.

(ii) Alloys containing from 15 to 25 per cent. of nickel and valuable on account of their ability to withstand cold working.

For the best results with these alloys only the purest metals should be employed, as small quantities of such elements as iron, silicon and sulphur are detrimental to the malleability and ductility of the resultant alloy. The maximum percentages of impurities allowable in nickel for this high-class work should be 0·050 per cent. of arsenic, 2 per cent. of cobalt, and 1·5 per cent. of other metallic impurities. The metal should be free from all unreduced oxides. Cupro-manganese, 25–50 per cent., is commonly employed as a deoxidiser in the manufacture of these nickel copper alloys, upwards of 0·25 per cent. being added to charges containing 30 per cent. of scrap. Following are the mechanical properties of the 20 per cent. alloy :—

Maximum Stress.	Elongation. Per cent.	Remarks.
40·0	5·0	Hard rolled.
20·0	35·0	Hard rolled and annealed.

The alloy containing 25 per cent. of nickel has been successfully employed in the manufacture of firebox plates.

(iii) Alloys having low electrical conductivities and whose temperature coefficients of electrical conductivity are practically nil. The 40–45 per cent. nickel alloy, known as 'constantan,' can be drawn into wire, and is employed in the construction of resistance coils and base-metal thermo-couples. Another alloy of this type, containing only 25 per cent. of nickel and 0·5 per cent. of iron, is known as 'nickeline.'

Copper and Silicon.

Alloys of copper and silicon are used for electrical purposes and in the manufacture of telephone and telegraph wires. The effect of silicon is to increase the tensile strength, without seriously affecting the conductivity.

	Tensile Strength (as Cast). Tons per sq. in.
Electrolitic copper	10·8
,, ,, + 0·25 per cent. silicon	14·5
,, ,, + 0·53 ,, ,,	15·5
,, ,, + 1·05 ,, ,,	17·4

ALUMINIUM AND ZINC ALLOYS.

The alloys of aluminium and zinc are of importance as light alloys, their specific gravities varying uniformly from 2·7 to 3·3 as their zinc content increases up to 30 per cent., which may be considered the limiting percentage for those alloys of this series that may be termed 'light.' All those alloys containing upwards of 70 per cent. of aluminium are forgeable at temperatures between 300° C. and 400° C.

These alloys do not allow of their use in casting into intricate shapes, but for all plain casting purposes they are quite satisfactory. Under certain conditions these alloys corrode rapidly; the use of high-grade spelter in their manufacture eliminates this defect. The machining properties of these alloys which are, at all times, superior to those of the pure metal, improve as the zinc content increases.

Below are tabulated certain of the results of tests reported by Messrs. Rosenhain and Archbutt to the Alloys Research Committee of the Institution of Mechanical Engineers.

Zinc. Approx. Per cent.	Condition of Alloy.	Yield Point. Tons per sq. in.	Max. Stress. Tons per sq. in.	Elongation. Per cent. on 2 ins.
5	1	2·70	5·20	16·0
	2	2·80	6·60	29·0
	4	4·30	7·48	33·6
11	1	6·40	9·41	8·0
	2	5·00	10·24	16·0
	3	9·42	13·78	33·0 (on 1 in.)
15	1	9·60	11·14	2·0
	2	5·80	11·69	8·5
	3	11·61	17·92	31·0 (on 1 in.)
	4	6·80	16·38	33·0
20	1	10·00	13·07	0·7
	2	7·80	13·87	4·0
	4	17·30	22·64	20·5
25	1	nil	16·26	1·0
	2	9·10	13·50	3·5
	3	20·24	24·03	27·0 (on 1 in.)
	4	25·00	27·09	16·5
30	1	11·70	16·63	1·5
	2	10·30	17·92	3·0

1. Sand-casting.
2. Chill-casting
3. Hot-rolled bars, ¾ in. diam.
4. ditto 1¼ in. diam.

The addition of copper to certain of the aluminium-zinc alloys improves the mechanical properties of the same. The average results of certain tests on chill-castings of the aluminium-zinc-copper series are given below.

Zinc. Per cent.	Copper. Per cent.	Yield Point. Tons sq. in.	Max. Stress. Tons sq. in.	Elongation. Per cent.
19	1	9·5	14·9	5·5
19	2	12·5	16·5	5·7
20	2	10·4	14·0	3·3
20	3	11·5	16·8	4·5
20	4	10·4	15·3	3·8

ALUMINIUM AND COPPER ALLOYS.

Like the zinc alloys, the commercial alloys are not simple binary alloys, but generally contain iron and silicon, and may also contain Mg, Mn, Ni, Zn, etc.

Al.	Cu.	Use.
93–91·5	7–8·5	General castings.
92	8	General castings, tendency to be brittle.
96	4	General castings, less brittle than 92 : 8 alloy; suitable for rolled sheets, sand-cast cooking utensils.
91–89·5	9–10·5	Light alloy pistons } American automobile practice.
89–86·5	11–13·5	Manifolds }
90	10	Die castings. Permanent mould castings.
88	12	
50	50	'Hardness' or rich alloys; not used for constructiona purposes, but used in the foundry for the purpose of introducing copper into aluminium or aluminium into copper.
60	40	

(See 'Aluminium Bronze,' p. 2732.)

ALUMINIUM CASTINGS

OF EVERY DESCRIPTION, SAND AND DIE, FOR ALL PURPOSES AND ALL TRADES

We have over 30 years experience with this metal and its alloys, which we place at your service

Send us your enquiries
Keenest prices quoted

On A.I.D. Approved List

ROBERT W. COAN, LIMITED

The Aluminium Foundries,

COAN HOUSE, DUNCAN ST., LONDON, N. 1

Telephone: Clerkenwell 4466 (4 lines)
Telegrams: Krankases, Isling, London

F. 2734

THE IRONMONGER

HARDWARE AND METAL TRADES ADVERTISER.

ESTABLISHED 1859.

PUBLISHED WEEKLY

28 ESSEX STREET, STRAND, LONDON, W.C.2

For over seventy years The Ironmonger has been the leading newspaper published in the interests of buyers of hardware, metals and mill furnishings.

Its news service is invariably up-to-date and its information reliable, while its weekly record of current prices of staple commodities gives to its pages a world-wide reputation as a valuable buyers' guide for the purchasing departments of engineers, shipbuilders, collieries, etc. etc.

All this information is obtainable for the cost of a year's subscription, £1, which sum includes a copy of The Ironmonger Diary and Hardware Buyers Guide, published each December.

To anyone interested a specimen copy of The Ironmonger will be sent on application accompanied by trade card.

The specific gravity increases with increase of copper from 2·7–2·85 for sand-cast alloys for 0–8 per cent. copper. The specific gravity of sand-cast alloys is less than that of chill castings, and that of rolled and drawn alloys is approximately equal to chill cast. In sand-cast alloys the strength increases with increase of copper at the expense of the elongation. In the hot-rolled condition the tensile strength increases up to about 6 per cent. copper and then decreases. The physical properties of these alloys are indicated by the following figures :

92–8 alloy. Ultimate strength, 18,000 lbs. per sq. in.; elongation, 1–1·5 per cent.

97–3 alloy (forged). Yield point, 16,000 lbs. per sq. in.; ultimate strength, 24,000 lbs. per sq. in.; elongation, 4 per cent.

96–4 alloy (rolled, annealed). Ultimate strength, 15,000–28,000 lbs. per sq. in.; elongation, 34–16 per cent.

Tensile strength of hot-rolled bars, 21,000–37,000 lbs. per sq. in. (for 1–6 per cent. copper); elongation varies from 30–18 per cent. For cold-drawn bars—tensile strength, 29,000–45,000 lbs. per sq. in. (for 1–4 per cent. copper); elongation, 14·5–6 per cent.

ALUMINIUM AND SILICON ALLOYS.

A striking property of these alloys is their low contraction, due to the silicon expanding on freezing. Yield exceptionally sound sand castings. Iron should be a minimum, as it has a detrimental effect. These alloys may be forged and both hot and cold formed, and, in their raw condition, are very malleable at the higher temperatures. In addition, they may be welded and will resist all ordinary corrosion, being unaffected by atmospheric conditions. The strength is largely dependent on the actual alloy used, but, in general, is somewhat higher than that of cast iron, and somewhat lower than that of brass. Much higher strengths are, of course, possible in the case of 'modified' alloys. A tensile strength as high as 36,000 lbs. per sq. in. is obtainable, together with a high elastic limit.

These alloys appear under trade names of 'Wilmil' in England, 'Alpax' in America and France, and 'Silumin' in Germany.

Method of preparation of these alloys has a great effect upon their properties. (1) Normal alloy, made by simply melting together aluminium and zinc. (2) Modified alloy, made by direct electrolytic reduction or by the addition of an alkali fluoride or with sodium or potassium to the melted normal alloy.

The silicon in (1) appears as relatively large crystals, and in (2) in high dispersion.

For 'normal' castings—composition, 95–90 Al, 5–10 Si.

For 'modified' castings—composition, 90–85 Al, 10–15 Si.

The simple Al–Si alloys are difficult to machine, and Al–Cu–Si alloys have been developed to overcome this.

OTHER ALLOYS OF ALUMINIUM.

Magnalium.

Magnalium is the name applied to a series of alloys having an aluminium-magnesium base, containing from 1·5 to 10·0 per cent. of magnesium and varying proportions of other elements, *e.g.* copper, tin, nickel, lead.

The table below gives the analyses of certain of these alloys, together with the uses to which they are generally put.

Use of Alloy.	Magnesium.	Copper.	Nickel.	Tin.	Lead.
Strong castings . .	1·60	1·76	1·16	nil	nil
General ditto . .	1·60	1·76	nil	traces	traces
Rolling and drawing .	1·58	0·21	nil	3·15	0·72

'Vickers' Duralumin.'

(James Booth & Co. (1915) Ltd.)

'Vickers' Duralumin' is the trade name applied to a series of quaternary alloys of aluminium, copper, manganese and magnesium, containing over 90 per cent. of aluminium and possessed of high tenacity and hardness. The presence of magnesium confers the property of cumulative hardening on certain aluminium alloys—*i.e.* subsequent to suitable thermal treatment the hardness of the alloy increases slowly to a maximum, which is attained after several days.

Following are typical analyses of 'Vickers' Duralumin':—

Copper.	Manganese.	Magnesium.	Silicon.	Iron.	Aluminium.
3·62	0·35	0·87	0·62	0·045	Diff.
3·00	1·00	2·00	—	1·000	Diff.
3·60	0·40	—	0·50	0·600	Diff.

The specific gravity varies between 2·77 and 2·84. The melting point is about 650° C. 'Vickers' Duralumin' can be rolled, forged, and drawn when hot or cold. According to the purpose for which it may be required, the composition can be modified and the treatment so arranged that the material is susceptible to forging, rolling, and drawing, or it may be arranged that the greatest hardness, with corresponding smaller extension, can be produced. The following table gives a summary in which the properties of 'Vickers' Duralumin' vary by such modifications in the composition and treatment.

Description of the Alloy.	Specific Gravity.	Yield Point. Tons per sq. in.	Max. Stress. Tons per sq. in.	Elongation. Per cent.	Contraction.	Brinell Hardness.
H	2·77	12·06	22·25	23	34	98
681d	2·84	16·5	29·20	18	26	125

These values are for soft rolled plates of 7 mm. thickness.

'Vickers' Duralumin' is supplied in sheets, bars, wires, strips, forgings, stampings, various sections and tubes.

BEARING METALS.*

The requisite properties of good bearing metals are hardness, to resist wear; strength, to bear compression; and plasticity, to allow of ready adjustment of the supported body on the bearing.

Two types of bearing metal are known:—

(*a*) Bearing metals in which the hard constituents of the alloy are distributed indiscriminately throughout a soft matrix.

(*b*) Bearing metals of spongy character; the network of the sponge being moderately hard and tough, the meshwork being soft and plastic.

These types can be grouped according to composition as follows:—

(i) Brasses and bronzes containing compounds of copper with tin, phosphorus, etc.

(ii) Bronzes containing appreciable quantities of lead.

(iii) Alloys having a tin base and containing compounds of tin with antimony, copper, etc.

(iv) Alloys having a lead base and containing compounds of tin with antimony, etc.

(v) General alloys possessed of the characteristics required of a bearing metal.

Of the above, types (i) and (ii) have been considered under the heading of 'Brass' and 'Bronze' (*q.v.*), types (iii) and (iv) are known as 'white' metals, while the majority of the alloys of type (v) can be termed 'white metals.' Of types (iii), (iv) and (v) it has been observed that metals whose atomic weight divided by their specific gravity gives a high number are good anti-friction metals, while the reverse is the case for metals which have a low quotient. Hence the use of lead and antimony for bearing alloys. In using white metal for bearings the best practice is to use the bearings wholly of white metal, discarding the cast-iron shells and using up the old bearings with the addition of a further small quantity of the more volatile constituents. White metal must be melted in a clean vessel and at as low a temperature as possible. Dross and scum in the metal tends to rapid heating in service. For the metals to be properly alloyed the high as well as the low melting points of these must be reached, but care must be taken not to overheat the alloy. For a thorough alloying a temperature of about 500° C. (930° F.) and less has been found to be sufficient. The molten metal should be stirred with a stick of pine, and a layer of charcoal always kept on the surface to prevent oxidation. The rate of cooling affects the size of the crystals and also the wearing quality. As a general rule cooling too slowly and pouring too cold produces a coarse crystal formation which is liable to heat quickly in use. In the case of the tin-antimony alloys it has been found that the best size for the tin-antimony compound cubes is 0·25 mm. edge. Pouring too hot on the other hand results in a soft metal. The shell should be heated to a temperature of 100° C. to 150° C. (212° F. to 300° F.) before pouring the metal into it, as this not only prevents blow-holes, but reduces the tendency of the lining shrinking away from the shell. The bearing should not be disturbed while the lining is solidifying, since vibration tends to enlarge the crystals and produce brittleness. The matrix of the lining should be just stiff enough to support the crystals, and the latter should be as numerous as is consistent with these, lying clear of each other and thereby causing brittleness. A Brinell hardness of 23·5 for an alloy with a lead base and 30 for a tin base has been found to give excellent results.

* See also Bearing Pressures, page 961.

ALLOYS HAVING A TIN BASE.

Tin-Antimony Alloys.

These alloys, which are of type (*a*) (p. 2736), are the basis of Britannia metal. The following table indicates the uses of certain of these alloys.

Tin.	Antimony.	Use of Alloy.
86	14	For seats of stop-cocks.
80	20	For plugs of stop-cocks.
79	21	Metal for bearings.
67	33	Very hard alloy, which forms the extreme limit of the ordinary tin-antimony alloys (*Hiorns*).

Babbitt Metal.

This alloy, whose composition is as follows:—Tin, 83·3; Antimony, 11·1; Copper, 5·6; is used on important bearings under heavy loads, such as crank-shaft, crank-pin, and cross-head pin journals.

Several of the French railways use the following alloy, which is low in tin:

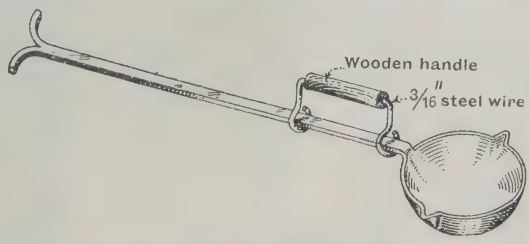
Tin, 67; Antimony, 22 to 11; Copper, 11 to 22.

In cast-iron boxes, for light work: 50 parts tin, 5 parts antimony, 1 part copper; for heavy work: 46 parts tin, 8 parts antimony, 4 parts copper. In brass boxes: 64 parts copper, 8 parts tin, 1 part zinc. The bearing to be lined with the metal is first tinned with the Babbitt metal mixed with twice its weight of tin.

Babbitt metal should be stirred thoroughly from the bottom of the pot at frequent intervals when being used, to prevent segregation of the constituents. If the pot is used only occasionally, the Babbitt should be covered with sawdust to prevent the formation of dross.

Pouring Babbitt Metal.

A convenient auxiliary handle that can be fitted to any Babbitt ladle is made from a short length of wooden bar, such as a piece of broom handle, drilled through for a piece of heavy wire or



¼-in. round rod, which is bent around the ladle handle as shown, so that it is loose enough to slide up and down readily.

(*H. Moore*, '*Power*,' *Aug.* 21, 1928.)

Babbitt Metal, Plastic Metal, and Phosphor Bronze.

These metals are supplied by the Delta Metal Co. Ltd., in various grades; phosphor bronze is supplied in three qualities, A, B, and C. The A quality is a hard bearing metal specially suitable for worm-wheels, bushes, crank pins, bearings, etc.; B quality phosphor bronze is a bearing metal more suitable for general purposes; whilst the C grade is adapted for large pinions, worms, etc. The Company also supply various grades of Babbitt, plastic, and other lining metals, for various speeds and pressures.

Admiralty White Metal.

Tin, 86·0; Antimony, 8·5; Copper, 5·5.

ALLOYS HAVING A LEAD BASE.

Lead-Antimony Alloys.

These alloys should not contain more than 18 per cent. of antimony if they are not to be brittle. Such as do contain upwards of 18 per cent. of antimony must be given suitable support. The following table indicates the increase of Brinell hardness numbers of these alloys with increase of antimony.

Antimony. Per cent.	Brinell Hardness Number. 10 mm. 100 kilos.
2	9 approx.
5	12 ,,
10	16 ,,
15	17 ,,
20	21 ,,

The 25 per cent. antimony alloy is used by the Eastern Railroad of France as a lining for cross head slides, rod brasses and axle-bearings.

'Magnolia.'

Lead, 80; Antimony, 14; Tin, 6.
Lead, 79; Antimony, 20; Tin, Copper, etc., 1.

Other Ternary Alloys having a Lead Base.

Following is a list of ternary alloys of lead, tin and antimony, showing the mechanical properties of the same :—

No.	Lead (by difference).	Tin.	Antimony.	Yield Point. Tons per sq. in.	Tenacity. Tons per sq. in.	Elongation per cent. $l = 4\sqrt{a}$	Compression. Load in Tons required to compress to Half Length.	Brinell Hardness Number. 10 mm. ball. 200 kilos.
1	88·0	4·1	7·9	2·14	2·41	1·5	12·7	15·2
2	86·5	8·6	4·9	2·14	3·22	3·0	N.O.	15·1
3	81·8	13·1	5·1	2·68	3·35	2·0	N.O.	16·7
4	76·8	14·8	8·4	2·81	3·08	1·0	N.O.	18·0
5	72·6	22·1	5·3	N.O.	2·41	—	8·9	16·8
6	65·7	28·7	5·6	2·95	3·75	1·5	N.O.	15·1
7	85·2	4·6	10·2	4·01	6·17	10·5	12·6	23·2
8	82·0	8·9	9·1	3·86	6·43	13·0	11·0	25·4
9	75·1	14·7	10·2	3·75	6·43	5·5	N.O.	26·4
10	71·0	18·6	10·4	4·01	5·09	1·5	9·0 C.	23·2
11	66·8	23·0	10·2	3·75	5·36	1·0	N.O.	24·0
12	79·6	4·5	15·9	3·08	5·22	4·0	N.O.	25·6
13	77·1	8·6	14·3	Broken in machining			11·6	31·0
14	66·4	19·0	14·6	3·48	5·36	—	10·7 S.C.	32·0
15	66·0	18·9	15·1	3·48	5·36	1·5	9·7 S.C.	27·6
16	70·1	4·6	25·3	Broken in machining			12·5 S.C.	26·6
17	68·8	9·1	22·1	4·55	4·55	—	9·8 *	37·0
18	64·1	13·9	22·0	4·28	5·09	—	10·0 *	35·6
19	72·3	4·5	23·3	3·48	3·48	—	8·8 *	27·8
20*	65·0	10·0	25·0	Broken in machining			8·3 *	33·6
21	64·2	5·9	29·9	3·75	5·62	1·0	6·5 *	28·8

N.O.—Not observed. C.—Cracks in compression test-piece.
S.C.—Slight cracks in compression test-piece. * Failed without compression to half length.

The results contained in this table are considered sufficient to merit the following conclusions: (1) that the effect of the presence of the tin antimony compound in these alloys is such as to render the same brittle and weak; (2) that the general mechanical properties of the lead-tin-antimony alloys containing less than about 15 per cent. of tin are improved by the addition of antimony in quantities not exceeding about 10 per cent.

Hughes recommends the following alloy for metallic packings :—
Lead, 70 ; Antimony, 13 ; Tin, 17;
while the following alloys are used as lining by the Eastern Railroad of France :—

	Lead.	Antimony.	Tin.
1 . .	70	20	10
2 . .	80	8	12

1. For eccentric straps and bronze slide valves.
2. Special alloys for metallic rod packing.

Ternary alloys of lead, antimony and copper are also in use. These alloys should contain from 15 to 25 per cent. of antimony and not more than 10 per cent. of copper.

Anti-Friction Alloys.

Delta Anti-Friction Alloys.

(The Delta Metal Co. Ltd.)

Delta White Anti-Friction Metals, Nos. IX and IXa, are used for lining bearings, bushes, etc., and are particularly suitable for heavy bearings such as those of marine engines, for railway trucks, dynamos, etc., as they do not wear, heat, or cut the journals ; they are also cheap.

The application of the metals presents no difficulties whatever ; the parts required to be lined should be thoroughly cleaned, and when the metal is melted (which may be done in a ladle, over a slow fire, taking care not to overheat it) it is applied in the usual way. The melting point is about 450°C.

Fry's Anti-Friction Alloys.

(Fry's Metal Foundries, Ltd.)

These anti-friction alloys cover the whole range of white metals employed in engineering practice ; commencing with lead-free alloys of high tin content with sufficient antimony and copper to provide necessary hardness and wearing qualities combined with great toughness (as in the O1 brand) which are used for special service, as in the big end bearings of engines for aircraft or high-powered cars, and ending with the finest quality of lead-base alloys of the Magnolia class.

The series includes bearing metals suitable for every class of service, heavy and light loads, high speeds and slow-running machinery, and plastic metal for emergency and special services. The high-tin alloys are self-tinning. All but the first alloy in the series have a structure consisting of a matrix of tough and durable alloy with cuboid antimony-tin crystals regularly embedded throughout. The first is a tin-antimony solid solution with crystals of copper-tin and copper-antimony constituents interspersed in dendritic or needle-shaped formation. The constituent of dendritic crystal form is actually present in all well-balanced alloys of this character, adding greatly to their toughness and durability, and also serving to ensure the even distribution of the cuboid antimony-tin constituent. Finally, hard and highly resistant zinc-tin rock crusher bearing metal of great endurance can be furnished.

The following table shows a selection of bearing metals for particular purposes and for design work :

Constituents, per cent. A = Tin; B = Antimony; C = Copper; D = Lead.					Tensile Test.		Brinell Hardness. Number. 10 mm. Ball 500 Kilos.	Compression Test. Tons per sq. in.	
No.	A	B	C	D	Tons per sq. in.	Elongation per cent. in 2".		Yield Pt. 0·001 in.	Compressed to half length.
1	93	3·5	3·5	—	5·12	11·6	24·9	3·569	14·732
2	86	10·5	3·5	—	6·65	7·1	33·3	4·372	17·232
3	83	10·5	2·5	4·0	5·60	—	34·5	4·284	17·640
4	80	11·0	3·0	6·0	5·70	—	32·1	4·640	17·500
5	60	10·0	1·5	28·5	5·04	—	27·1	3·696	12·856
6	40	10·0	1·5	48·5	4·58	—	21·8	3·660	11·284
7	20	15·0	1·5	63·5	5·48	—	31·3	4·016	12·212
8	78	11·0	11·0	—	6·36	—	37·0	4·550	17·856
9	5	15·0	—	80·0	4·69	2·8	24·9	3·590	13·356

Miscellaneous Metals for Bearings and Engine Parts.

The following special mixtures are stated to be satisfactory for the duty mentioned:—

For Lining.	Tin.	Lead.	Zinc.	Antimony.	Copper.	Bismuth.
Dynamos—high speed	88	—	—	8	3·5	0·5
Marine engines	77	17	—	3	3	—
Eccentrics	5	78	—	15	2	0·25
Submerged bearings	40	48	—	10	2	—
Main bearings	34	44	—	16	6	—
Slides, thrusts	63	—	30	2·5	2·5	—
Railway trucks	42	—	56	—	2	—
Axle-boxes (by analysis)	74·22	13·50	1·80	6·55	3·60	—
Loco and carriage bearings (Italian Railways)	38	37	—	25	—	—
Carriage bearings (German Railroad)	20	60	—	20	—	—
Bearings for rock-breaking machinery	10	—	80	10	—	—
Loco. bearings (A. Allan & Sons)	5·5	35	—	—	59·5	—
Cross-head slides and rod bearings (Eastern Railway, France)	0	65	—	25	10	—
Axle bearings and connecting rods for heavy engines (Eastern Railway, France)	83·3	—	—	11·12	5·5	—
Eccentric straps and slide valves (Eastern Railway, France)	10	70	—	20	—	—
Passenger and freight car bearings (A. Allan & Sons)	4·5	45	—	—	50·5	—
Loco. bearings (Dewrance)	33	—	—	45	22	—
Loco. and wagon axle boxes (Fenton's alloy)	14·5	—	80	—	5·5	—
Bearings, sleeves, slides and guide gibs (U.S. Navy)	12·5 to 14·5	1	2·5 to 4·5	—	82 to 84	—
Pivot bearings (Hoyle's alloy)	46	42	—	12	—	—
Lining, bearings and crank pin bearings (moderate speed) (Parsons' white brass)	68	0·5	30·5	—	1	—
Metallic rod packing (Eastern Railway, France)	12	80	—	8	—	—
Piston packing rings	—	50	—	—	50	—
	13	1	1	—	85	—
Pump and pump chambers and gun-metal valves	10	1·5	1·5	—	87	—
Cog wheels	9	25	—	—	56	—
	10	10	—	—	80	—
Anti-acid metal (by analysis)	78·84	14·75	—	Trace	3·70	—
Plastic metal	80	10	—	1	8	1
Genuine Babbitt (hard)	80	—	—	10	10	—
" " (No. 2)	83	—	—	9	8	—
Universal bearing metal	6	78	—	16	—	0·25
Anti-friction castings	24	—	80	—	4	—
Bearings for high-speed machines	—	—	14 to 18	—	82 to 86	—
" " "	17	—	—	77	6	—
" " "	18·5			76	5·5	

ZINC-TIN ALLOYS.

Zinc and tin form a series of eutectiferous alloys, some of which are of value as bearing metals. The alloys, in common with all bearing metals, are very susceptible to differences in casting

temperature, as may be gathered from the following table, which gives in detail the effect of varying the temperature of casting an alloy of 70 per cent. of zinc and 30 per cent. of tin :—

Casting Temperature.	Maximum Stress.	Elongation. Per cent.	Stress to compress to half length.
525° C.	5·09	3·0	7·7
500° C.	6·17	3·0	13·6
475° C.	6·56	8·0	15·4
450° C.	7·37	17·5	14·8
425° C.	7·37	35·5	15·0
400° C.	7·24	20·0	15·8

The above figures are for chill castings.

The alloys of zinc and tin containing from 25 to 75 per cent. of zinc possess, when in the chill cast condition, tenacities varying from 6·0 to 6·5 tons per square inch, percentage elongations varying from 30 to 45, and are compressible to half-length under stresses (calculated on the original areas of the test-pieces) varying from 13 to 17 tons per square inch. The eutectic melts at about 200° C. and contains 16 atomic per cent. of zinc.

In order that these alloys may approximate more closely to type A (*q.v.*) bearing metal, the zinc-tin series containing no relatively hard constituent, copper and antimony are frequently added. The addition of one or both of these constituents results in the formation of a bearing metal in which the hard compounds of antimony and zinc, etc., are indiscriminately distributed throughout the eutectiferous matrix. The following are analyses of certain alloys of this type :—

Zinc.	Tin.	Copper.	Antimony.	Use of Alloy.
80	14·5	5·5	—	Fenton's alloy for locomotive axle-boxes.
69	26·0	5·0	—	Propeller bush.
88	2·0	8·0	2·0	Bearing metal for general use.

In this connection it may be noted that ternary alloys generally are far preferable to complicated mixtures and usually fulfil all the requirements of good bearing metals.

A French report on new anti-friction metals includes trials of alloys consisting chiefly of aluminium, cadmium, magnesium, and especially of zinc. Such alloys were designed to reduce as far as possible the use of copper, tin, lead, and antimony, and the most satisfactory results are stated to have been given by a compound of 63·3 per cent. of zinc, 21·3 per cent. of tin, 12 per cent. of lead, and 3·3 per cent. of copper.

FUSIBLE METALS.

With the following combinations of metals, which form 'eutectiferous' alloys, it is possible to prepare an alloy having a melting point lower than that of any member of the series.

Series.	Alloy with Lowest Melting Point (Eutectic).	Melting Point.
Bismuth-lead	57·4 per cent. Bismuth	124·5° C.
Bismuth-tin	56·7 ,, ,,	137° C.
Bismuth-cadmium	60 ,, ,,	146° C.
Cadmium-tin	52 ,, Cadmium	177° C.
Lead-tin	36·5 ,, Lead	180° C.
Cadmium-lead	17·4 ,, Cadmium	247·3° C.

By the addition of one or both of the residual metals to any of the above combinations, systems of ternary and quaternary alloys, having individual members with melting points lower than those of all other members of these systems and less than those of the eutectics given above, may be obtained. These ternary and quaternary alloys are the true 'fusible metals.'

The alloys of lead, tin and bismuth are of value for the following purposes :—

(i) The manufacture of casts of delicate objects that would be damaged if subjected to high temperatures. The following alloys are recommended for this purpose :—

Bismuth	59·2	50·0	50·0	50·0
Lead	27·3	31·2	30·0	25·0
Tin	13·5	18·8	20·0	25·0

These alloys expand considerably on cooling and are valuable also for the taking of impressions from dies, medals, etc.

(ii) Fusion pyrometers. Parkes and Martin recommend the following fusible metals as pyrometers for the tempering of steel tools. The article to be treated is placed on a plate of the alloy, which is heated till it melts. The tool, being then in the required condition, is quenched in water.

Table of Melting Point of Bismuth-Lead-Tin Alloys.

Bismuth.	Lead.	Tin.	Melting Pt.	Bismuth.	Lead.	Tin.	Melting Pt.
50·0	31·2	18·8	94° C.	16·6	33·2	50·2	158° C.
47·0	35·5	17·7	98° C.	16·0	36·0	48·0	155° C.
42·1	42·1	15·8	108° C.	15·3	38·8	45·9	154° C.
40·0	40·0	20·0	113° C.	14·8	40·2	45·0	153° C.
36·5	36·5	27·0	117° C.	14·0	43·0	43·0	154° C.
33·3	33·3	33·3	123° C.	13·7	44·8	41·5	160° C.
30·8	38·4	30·8	130° C.	13·3	46·6	40·1	165° C.
28·5	43·0	28·5	132° C.	12·8	49·0	38·2	172° C.
25·0	50·0	25·0	149° C.	12·5	50·0	37·5	178° C.
23·5	47·0	29·5	151° C.	11·7	46·8	41·5	167° C.
22·2	44·4	33·4	143° C.	11·4	45·6	43·0	165° C.
21·0	42·0	57·0	143° C.	11·2	44·4	44·4	160° C.
20·0	40·0	40·0	145° C.	10·8	43·2	46·0	159° C.
19·0	38·0	43·0	148° C.	10·5	42·0	47·5	160° C.
18·1	36·2	45·7	151° C.	10·2	41·0	49·5	161° C.
17·3	34·6	48·1	155° C.	10·0	40·0	50·0	162° C.

(iii) Soft solders. For the union of metal to glass, *e.g.* in the manufacture of paraffin lamps, the following alloy has been suggested :—

Lead, 40·0; Bismuth, 33·4; Tin, 26·6.

(iv) Fusible plugs.

The above alloys are nearly all of a brittle character. For the manufacture of ductile fusible alloys the use of cadmium in place of bismuth is recommended.

The following table includes the best known of the other fusible metals :—

Alloy.	Cadmium.	Lead.	Tin.	Bismuth.	Melting Pt.
Fusible alloy . .	12·5	25·0	12·5	50·0	55·5° C.
Lipowitz's alloy .	10·0	26·6	13·3	50·0	60–68° C.
Woods' alloy . .	15·4	30·8	15·4	38·4	71° C.
Fusible alloy . .	34·5	27·5	10	27·5	75° C.
,, ,, . .	6·2	34·5	9·3	50·0	77° C.
,, ,, . .	25	25	50	—	86° C.
,, ,, . .	16·6	—	33·3	50·1	95° C.
,, ,, . .	11·1	—	33·3	55·6	95° C.
,, ,, . .	25	—	25	50	95° C.

Spence's Metal.

This compound is obtained by melting the three sulphides of iron, zinc and lead with sulphur. The product is a dark grey mass of great tenacity, small power of conducting heat, a specific gravity of 3·4, a melting point of 160° C. In congealing it expands like bismuth and type metal, and resists in a remarkable degree the action of atmospheric influences, alkalies and acids, even of aquaregia, its surface scarcely being affected after having been exposed to the action of the latter for four weeks. Its property of expanding in congealing, and therefore filling completely all depressions of the mould, makes it particularly available for castings. If the compound is poured on a plate on which the impression of the hand has been made, the cast will show all the lines and the pores of the palm.

The metal has been used for jointing gas and water pipes.

For further particulars see Patent Specification No. 2706, 1879.

Type-Metal.

Metal.	Tin.	Antimony.	Lead.
Monotype	5% to 10%	15% to 19%	Remainder
Founders' Type for Hand Composition	10% ,, 18%	20% ,, 30%	,,
Slug Casting Machines—			
Linotype, Intertype, and Ludlow .	2·5% ,, 5%	10% ,, 13%	,,
Stereotype—			
Flat Plates	5% ,, 10%	15% ,, 18%	,,
Rotary Plates	6% ,, 10%	15%	,,

The addition of Copper to type metals as an extra hardener is generally discontinued. Small quantities creep in during repeated remelting. Quantities above 0·12 per cent. are apt to give trouble in machine composition work.

Non-Magnetic Alloys.

These alloys are used as substitutes for steel wherever it is employed in the works of watches. The composition of the alloys varies from 45 to 75 parts of palladium, 15 to 30 parts of copper, 20 to 25 parts of silver, with sometimes the addition of small proportions of iron, steel, nickel, gold and platinum. These alloys are claimed to be unoxidisable in moist air, to preserve their elasticity indefinitely, not to vary sensibly with changes of temperature, and to remain uninfluenced by proximity to dynamos.

Another non-magnetic alloy, recently brought out by Messrs. Ostermann & Lacroix, of Geneva, Switzerland, is made of from 30 to 40 parts of gold, 30 to 40 parts of palladium, 0·1 to 5 parts of

rhodium, 10 to 20 parts of copper, 0·1 to 5 parts of manganese, and the same proportions of silver and platinum. The copper and manganese are first mixed, after which the other articles are added; or all the metals may be put into a crucible at the same time, the manganese being used for the bottom layer.

A non-magnetic variety of 'Bull's Metal' is manufactured and sold as No. III. and is being adopted for funnels and other erections of warships where the material must possess the mechanical properties of steel without its magnetic influence.

The Delta Metal Co. Ltd., supply under their 'Delta' Brand non-magnetic metals (bronze and brass) to meet special requirements.

GENERAL ALLOYS.

Alloy.	Copper.	Zinc.	Tin.	Nickel.	Lead.	Antimony.	Bismuth.
Babbitt's metal	3·7	..	89	..	..	7·3	
Bell metal (bronze)—							
Musical bells	87·5	..	12·5				
Gongs	81·6	..	18·4				
House bells	77	..	23				
Church bells	69	..	31				
" "	80	5·6	10·1	..	4·3		
Brass, red	87	13					
" common	75	25					
" wire	67	33					
" yellow, fine	66	34					
" Muntz metal	60	40					
" sheathing metal	55	45					
" red	86	11·1	2·9				
" rolled	74·3	22·3	3·4				
" white	69·8	25·8	4·4				
" yellow	67·2	31·2	1·6				
" statuary	91·4	5·5	1·4	..	1·7		
Britannia metal	..	..	25	..	..	25	
When fused, add	..	..	..	..	..	25	25
Bronze, gun-metal, small	93	..	7				
" medals	93	..	7				
" mathematical instruments	92·2	..	7·8				
" gun-metal, large	90	..	10				
" wheels, valves	90	..	10				
" machinery bearings	87·5	..	12·5				
" lathe bushes	80	..	20				
" telescope mirrors	66·6	..	33·4				
" rivets	64	..	36				
" very tenacious	88·9	2·8	8·3				
" common	84·3	5·2	10·5				
" " hard	79·3	6·4	14·3				
" machinery bearings, hard	77·4	7	15·6				
" speculum metal	50	21	29				
Chinese white copper	40·4	25·4	2·6	31·6			
German-silver	33·3	33·4	..	33·3			
Alloy that expands in cooling	..	..	..	..	75	16·7	8·3
Pewter, best	..	..	86	..	..	14	
" common	..	..	80	..	20		
Printing type	..	..	..	..	80	20	
Type and stereotype plates	..	..	..	..	69	15·5	15·5
White metal	7·4	7·4	28·4	..	..	56·8	

COPPER AND COPPER ALLOYS.*

Alloy.	Condition.	Yield Point. Tons per sq. in.	Ultimate Strength. Tons per sq. in.	Elongation. %	Reduction of Area. %	Brinell No.	Remarks.
Copper, 99·9%	(*a*) Electrolytic, annealed at 200° C.	3·8	16·9	50	50	40	Shearing strength, cast copper. 3,000 lbs. per sq. in.
	(*b*) Not annealed, 96% reduction	—	30	0·8	64·5	—	Modulus of elasticity (Copper).
	(*c*) Hot-drawn, 64% reduction	—	20·9	4·3	70·5	—	Lbs. per sq. in. (*a*) Electrolytic $17·4 \times 10^6$ (*b*) Cast $11·0 \times 10^6$ (*c*) Hard-drawn $17·6 \times 10^6$
Copper, 99·6% Commercial	Cast	4·5	11·3	20	60	80	
	Rolled hard (40% reduction)	9·0	22·5	5	8	94	
	Annealed at 500°C.	—	15·6	50	60	42	
	Cold-drawn (50% reduction)	16·5	22·5	9	—	—	
Brass (red) Cu 90, Zn 10	Sand-cast	—	13·0	22	—	—	Compressive strength (Brasses).
	Hard-rolled	—	24·6	5	—	60	
	Soft-rolled	—	16·5	40	70	47	Cu 90 / Zn 10 / Lbs. per sq. in. 30,000
							80 / 20 / 39,000
							70 / 30 / 60,000
Brass (yellow) Cu 68, Zn 40	Sand-cast	9·8	20·5	15	22	—	60 / 40 / 75,000
	Hard-rolled	20·0	31·2	30	50	—	50 / 50 / 110,000
							Modulus of elasticity. Average, 13×10^6 lbs. per sq. in.
Bronze Cu 97·7, Sn 2·3	Cast	3·8	12·5	20	—	—	Compressive strength (Bronzes).
	Rolled	4·8	21·4	55	75	—	Cu / Sn / Lbs. per sq. in. 97·7 / 2·3 / 34,000
							90·0 / 10·0 / 56,000
							80·0 / 20·0 / 118,000
							70·0 / 30·0 / 150,000
							Modulus of elasticity $10–15·5 \times 10^6$ lbs. per sq. in.
Bell metal Cu 90, Sn 10	Cast	4·5	14·3	1·5	—	—	

PHYSICAL PROPERTIES OF METALS AND ALLOYS.*

Metal or Alloy.	Condition.	Yield Point. Tons per sq. in.	Ultimate Strength. Tons per sq. in.	Elong. %	Reduction of Area. %	Brinell No.	Remarks.
Cobalt, 99·7%	Cast	12·4	14·7	—	—	—	
	Annealed	14·1	16·5	—	—	—	
	Wrought wire	—	39·7	3·0	—	—	
Stellite . .	—	—	—	—	—	512	Co 59·5, Mo 22·5, Cr 10·8 Fe 3·1, Mn 2·0, C 0·9, Si 0·8

* Bureau of Standards, Washington.

PHYSICAL PROPERTIES OF METALS AND ALLOYS—*continued.*

Metal or Alloy.	Condition.	Yield Point. Tons per sq. in.	Ultimate Strength, Tons per sq. in.	Elong. %	Reduction of Area. %	Brinell No.	Remarks.
Gold, 100%	Cast	—	11·3	25·0	—	—	
	Hard-drawn	—	16·5	—	—	—	
Gold-copper Au 90, Cu 10	Hard drawn	—	29·0	—	—	75	
Lead, Commercial	Cast	—	0·8	—	—	—	
	Hard-rolled	—	1·5	—	—		
	Soft-rolled	—	1·1	—	—	—	
	Hard-drawn	—	1·4	—	—	—	
Nickel, 98·5%	Cast	10·6	17·0	5·7	6·1	76	
	Wrought	8·0	19·0	11·0	—	83	
	Hard-rolled	—	41·0	11·0	—	—	
	Hard-drawn	—	69·4	—	—	—	
Palladium	Hard-drawn	—	17·4	—	—	—	
Platinum	Hard-drawn	—	23·6	18·0	—	—	
	Drawn, annealed	—	15·6	50·0	—	—	
Silver, 100%	Cast	—	17·8	—	—	—	
	Hard-drawn	—	22·9	—	—	59	
Ag 75, Cu 25	Hard-drawn	—	49·0	—	—	—	
Tantalum	Hard-drawn	—	58·0	—	—	—	
Tin, 99·8%	Cast	—	1·8	35·0	—	14	
	Rolled	0·72	2·4	—	—	—	
	Hard-drawn	—	4·5	—	—	—	
Tungsten, 99·2%	Swaged rod	—	96·0	4·0	28·0	—	
	Hard-drawn	—	263·0	—	65·0	—	
Zinc	Cast, fine	—	5·4	—	48·0	—	
	Rolled	1·3	12·0	—	—	—	
	Hard-drawn	—	7·1	—	—	—	
Aich's metal, Cu 60, Zn 38·2, Fe 1·8	—	25·5	—	—	—	—	
Delta metal	Sand-cast	—	20·0	10	—	—	
Cu 57, Zn 42, Fe 1	Hard-rolled	—	26·8	17	—	—	
Sterro metal	Cast	—	27·0	—	—	—	
Cu 55, Zn 42·4, Fe 1·8, Sn 0·8	Forged	—	34·0	—	—	—	
	Hard-drawn	—	37·0	—	—	—	
Leaded brass	Cast	—	14·7–17·4	30–26	35–30	—	
Cu 60, Zn 35, Pb 5·3	Annealed sheet	—	18·7	50	—	—	
	Hard sheet	—	27·2	30	—	—	
Manganese-bronze	Sand-cast	13·4–15·6	31·2–33·5	30–22	32–25	109–119	
Cu 58, Zn 39, Mn 0·05	Chill-cast	14·3–16·5	33·5–35·7	32–25	34–28	119–130	
Nickel silver Cu 60·4, Zn 31·8, Ni 7·7	Cast	6·9	16·0	40·0	42	46	
German silver Cu 61·6, Zn 17·2, Ni 21·1	Cast	8·4	18·3	28·5	25	80	
Naval brass	Sand-cast	7·0	19·0	30·0	32	—	
	Rolled and annealed	16·5	27·6	25·0	37	—	
Tobin bronze Cu 58·2, Zn 39·5 Sn 2·3	Cast	11·2	26·8	—	—	—	
	Rolled	24·1	35·2	35·0	40	—	
Victor bronze Cu 58·6, Zn 38·5, Al 1·5, Fe 1·0, Va 0·03	Cold-drawn	35·7	41·0	11·5	29	—	

Properties of Non-Ferrous Alloys at Elevated Temperatures.

Alloy.	Analysis.
A. Muntz Metal	Cu 58·96, Zn 39·77, Sn 0·56, Fe 0·14, Pb 0·67.
B. Extruded Sceptre Brass.	Cu 61·7, Zn 35·6, Fe 1·44, Al 1·05, Pb 0·17.
C. Drawn Manganese-Bronze	Cu 56·9, Zn 40·28, Sn 0·75, Ni 0·21, Fe 0·82, Al 0·19, Mn 0·19, Pb 0·66.
D. Sand-Cast Manganese-Bronze	Cu 58·6, Zn 36·9, Sn 0·08, Ni. trace, Fe 1·46, Al 0·01, Mn 1·88, Pb 0·06.
E. Sand-Cast Phosphor-Bronze	Cu 85·38, Zn 1·01, Sn 12·55, Ni 0·11, Fe 0·02, Pb 0·61, P 0·24.
F. Chill-Cast Phosphor-Bronze	Cu 90·25, Sn 9·47, Pb 0·17, P 0·17.
G. Nickel Copper	Cu 97·79, Ni 2·02, Fe 0·08.
H. Naval Brass	Cu 60·95, Zn 35·04, Sn 1·3, Fe 0·11, Pb 0·20.
J. Sand-Cast Gun-Metal	Cu 85·5, Zn 3·29, Sn 10·2, Pb 0·12.

Brinell Hardness at Different Temperatures.

Alloy.	Temperature. ° C.			
	18.	250.	350.	450.
A	126	107	63	18
B	133	111	71	26
C	114	99	61	15
D	132	101	62	19
E	98	82	77	52
F	77	75	64	42
G	171	—	142	141
H	96	88	61	17
J	86	69	63	58

Tensile Strength at Different Temperatures. (Tons per sq. in.).

Alloy.	Temperature. ° C.			
	20.	250.	350.	450.
A	31·5	23·0	12·6	3·9
B	—	—	—	—
C	34·4	22·8	11·7	2·7
D	32·5	23·0	12·6	4·1
E	13·5	13·5	9·7	9·4
F	—	—	—	—
G	38·8	35·8	36·0	31·0
H	—	—	—	—
J	15·8	16·1	10·3	5·2

(*Aitchison.*)

SOLDERS AND SOLDERING.

The following notes are published with the kind permission of the Editor of 'The Metal Industry.' (See 'The Metal Industry,' vol. 9, No. 4.)

Points to be Observed for Successful Soldering.

Attention should be paid to the following points if success is to be attained in soldering or brazing :—

1. In the preparation of the solder, uniformity of composition should be aimed at. Readily volatile or oxidisable constituents should be retained.

2. In the choice of the solder, consideration should be taken of the fusibility of the solder. The melting point of the solder should be less than that of the metal to be soldered. It should be noted, however, that the more nearly the melting point of the solder approaches that of the metal to be soldered the better is the union obtained, although the more difficult is the operation of soldering. It should be considered whether the solder will mingle perfectly with the surfaces to be united. One having mechanical and physical properties as nearly as possible approximating to those of the metals to be united should be used. One whose mechanical and physical properties will not be subsequently modified by any treatment—thermal or otherwise—should be employed.

3. In the operation of soldering, thorough mechanical and chemical cleanliness of the surfaces to be united must be obtained, e.g. by scraping, filing, grinding, etc., and by dipping in suitable chemical reagents. A source of heat both smokeless and sufficiently hot to raise the temperature of the sections to be united above the melting point of the solder is necessary. A suitable flux must be used.

Fluxes.

Fluxes are used to protect the surfaces of the metal to be united from the oxidising action of the atmosphere during the process of heating. The presence of any non-metallic materials, such as oxides or carbonates, on the surfaces to be joined, effectually prohibits perfect union. As the process of heating a metal in the air always leads to the formation of a film of oxide on its surface, means must be employed to protect the surface. Use is made of readily fusible salts and of oils that are non-volatile at the temperature of soldering for this purpose. For hard-soldering borax is practically the only flux employed. It is applied as a paste with a brush to the surfaces of small articles, and as a powder to those of larger articles. For soft-soldering a variety of materials is used, the following list giving the most important of these, together with their uses.

(i) Zinc chloride is used for the soldering of sheet metal, copper alloys and bright iron. It is poisonous and corrosive, and is on these accounts unsuited (*a*) for the soldering of tins destined to contain food, and (*b*) for the production of delicate work, e.g. electric fuses.

Zinc Chloride Solution.

Kill 2 quarts hydrochloric acid with all the zinc it will take up. Add a quart of water or add part until zinc is fairly dissolved. A quart of glycerine which has been mixed with a quart of alcohol is then added to the solution.

This fluid is used for all kinds of soldering, especially with greasy and dirty connections and iron. It is claimed that glycerine prevents rust, which plays havoc with other soldering solutions containing hydrochloric acid. (*Street Railway Journal.*)

Zinc Chloride Paste.

1 lb. vaseline or petrolatum ; 1 oz. (fluid) saturated solution of chloride of zinc.

The use of petrolatum instead of vaseline is recommended. While they are identical in composition, the name 'vaseline' is registered as a trade-mark, and commands a higher price on this account. Petrolatum is much cheaper. The chloride of zinc solution is made by dissolving as much zinc in strong muriatic acid as it will take up. An excess of zinc should be present, and all the acid neutralised. This will form a thick, oily solution. The petrolatum and chloride of zinc are mixed and thoroughly incorporated by means of a mortar and pestle or by vigorous stirring. The advantage of this soldering paste lies in the fact that it does not spatter and is not corrosive.

(ii) Ammonium chloride is used as a flux when soldering copper and iron.

(iii) Ammonium phosphate is used when treating with tin, zinc, copper and brass ware. It is non-corrosive and non-poisonous, but is somewhat refractory.

(iv) Hydrochloric acid is employed when dealing with zinc and zinc-coated ware.

(v) Lactic acid is useful in connection with copper and its alloys, but has the disadvantage of producing a tarnish on the surfaces adjacent to the soldered surfaces.

(vi) Venice turpentine and Gallipoli oil are used (separately) when soldering pewter or Britannia metal.

(vii) Russian tallow is employed when soldering heavy lead work.

(viii) Palm oil is employed in the treatment of light lead work.

(ix) Resin is an advantageous flux when subsequent corrosion is likely to be deleterious. It is used on soft alloys and tinware.

Hard Solders.

The hard solders may be divided into two groups, each of which will be considered separately in succeeding paragraphs—brazing or 'spelter' solders and silver solders.

Brazing or 'Spelter' Solders.

The basis of the brazing or 'spelter' solders ('spelters' they are commonly called) is brass. As the percentage of zinc is increased in the brasses so does their utility as solders for the more readily fusible metals and alloys increase. There is, however, a limit beyond which the addition of zinc to these alloys serves no useful purpose, owing to the fact that the resulting alloys are, on account of their brittleness and weakness, of no practical value. This limiting value is at about 45 per cent. of zinc. The particular alloy which is in most general use contains 50 per cent. of zinc and melts at about 875° C. (1607° F.). It is employed in brazing general brass work, which is possessed of melting points varying from about 890° C. (1634° F.) to about, 1000° C. (1832° F.). For brazing metals, etc., having higher melting points than the above brasses containing a less percentage of zinc are used. Thus, for the soldering of general ironwork the brass containing 35 per cent. of zinc is used, while for the brazing of copper and light ironwork that having 40 per cent. of zinc is employed. The most malleable solder contains some 30 per cent. of zinc, but its refractory nature somewhat subtracts from its value.

For certain purposes, the above alloys, owing to their yellow colour, are prohibited. Occasionally—e.g. in the brazing of nickel silver and iron—whiter alloys are required. For this purpose alloys containing nickel are employed, of which the following are examples :—

Copper per cent.	Zinc per cent.	Nickel per cent.
37·50	50·00	12·50
35·00	57·00	8·00

Other white solders, for use in the brazing of brass and more fusible than 'spelters,' contain a percentage of tin. Two such solders are given below :—

Copper per cent.	Zinc per cent.	Tin per cent.
43·34	6·66	50·00
57·50	25·00	17·50

The British Engineering Standards Association has issued a Specification (No. 263—1926) covering two grades of brazing solder. The copper contents for the two grades range from 53 to 55 per cent. and from 49 to 51 per cent. respectively, maximum limits being specified for the tin, antimony, arsenic, bismuth, iron and lead permitted in the alloy. A table is included as an appendix to the specification giving a range of grain sizes of granular solder to which it is recommended that the material be ordered.

Silver Solders.

Silver solders are employed for soldering gold, silver, copper, iron, and their alloys, the flux generally used being borax, though with very hard solders—*i.e.* solders having high melting points—powdered glass is sometimes of value.

Such as are suitable for general soldering and brazing, and containing an excess of base metal, are tabulated below :—

Copper per cent.	Silver per cent.	Zinc per cent.	Uses.
90—95	10—5	—	Solder for thin sheets of mild steel.
55·0	45·0	—	Alloy for instrument-makers ; tough and fluid.
50·0	30·0	20·0	Solder for small brass work.
46·0	38·5	15·5	Solder for bronze and nickel silver.
43·0	9·0	48·0	Solder for general work on copper alloys.

Tin-Lead Solders.

Tin and lead form a series of alloys, of which only those containing upwards of 35 per cent. of lead are employed as solders. These alloys may be divided into two classes, those that solidify at practically one definite temperature and those which are pasty over a considerable range of temperature before they become completely solid. The former are used as tinmen's solders, the latter as plumbers' solders, their value to each of these craftsmen being dependent on the properties associated with their solidification already mentioned. The diagram on page 2750 indicates the ranges of composition of the two types of solder already mentioned.

It will be seen that fine tinmen's solder solidifies at a definite temperature of 180° C. (356° F.) while coarse tinmen's solder commences to solidify at about 209° C. (408° F.), passes through a pasty stage, and finally completely solidifies at 180° C. (356° F.). Soft solder is the name given to the alloy containing 50 per cent. of lead, whilst the fine plumbers' solder, used in making 'wiped' joints, which commences to solidify at about 250° C. (482° F.), contains 66·6 per cent. of lead. A solder frequently used for the union of tinplate contains about 90·0 per cent. of lead, and hence has a very marked pasty range of temperature.

The appearance of the tin-lead solders affords some idea as to their value. The surface of good solder should be smooth and even, and rectangular castings of the same should have a distinct furrow along the centre of their upper surface. The presence of zinc, though only traces of this metal be present, causes the free surface of the solder to become minutely crystalline.

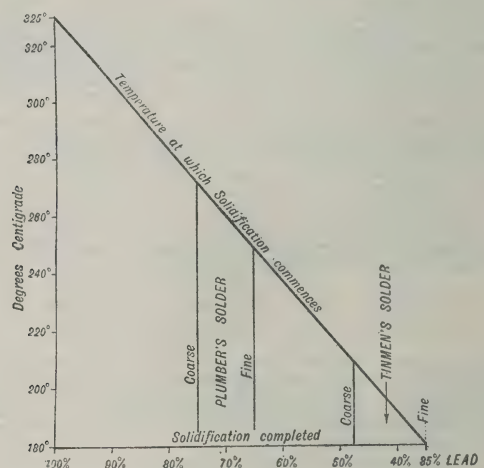


Diagram illustrating the approximate Range of Solidification of the Lead-Tin Alloys.

Bismuth, copper, and silver do not affect the surface condition of solders when present in quantities varying from 0·2 per cent. in the case of the last metal to 1·0 per cent. in the case of the first, while antimony up to 2·0 per cent. improves, rather than otherwise, the surface condition. The presence of any of these elements leads to a reduction of the melting point of the solder.

To Prevent Solder Sticking to Work.

To prevent hot solder sticking to work, mix common whiting or cold-water paint with wood alcohol, and paint the part that is to be protected. The hot solder will not stick, even if the piece is held in the melted alloy. Water will do as well as alcohol to mix the paint; but alcohol is the most convenient, inasmuch as it can be used without waiting for the paint to dry. If water is used the paint must be thoroughly dry, as otherwise the moisture will cause the solder to fly.

Bismuth Solders.

These solders are alloys of bismuth, tin and lead, and are in general more fusible than the tin-lead solders. The useful alloys contain from 20 per cent. to 65 per cent. of lead, from 12 per cent. to 55 per cent. of tin, and from 12 per cent. to 55 per cent. of bismuth. The alloys containing the higher percentages of bismuth are the most fusible of these solders, which are employed for general pewtering. The solder containing about 50 per cent. of bismuth and 20 per cent. of tin has the lowest melting point of the series, its value being 96° C. (205° F.).

Aluminium Soldering.

Success in soldering aluminium depends on the effective removal from the metal of the microscopically thin film of oxide always present on the surface. When measures are taken to deal with this film the main difficulty of soldering is removed. Three different types of soldering are employed, which may be distinguished by the terms hard soldering, soft soldering, and reaction soldering.

Hard Soldering.

In this process the solder consists of an alloy of aluminium having a melting point between 500 and 600° C. Many such alloys exist but the silicon alloy, containing 10 to 13 per cent. of silicon, is undoubtedly the best. The oxide is removed by means of an alkaline halide flux, such as is used for aluminium welding. At the temperature at which the soldering is carried out the flux is melted and rapidly attacks the oxide, permitting the melted solder to come into contact with clean aluminium and alloy with the surface. In carrying out the process a gas blowpipe is used as heating medium, but apart from this and the high temperature required, the process does not differ from the ordinary soldering of brass. The flux is applied on the end of the solder stick, which is melted up and flows readily, sweating the parts together. It is necessary to see that the edges to be soldered are properly hot, and in some cases it will be found convenient to heat them first until they are hot enough to melt the solder stick when the latter is rubbed on them. Certain manufacturers supply silicon alloy solder in the form of a tube with the flux contained inside.

Hard soldering with a silicon alloy solder is thoroughly to be recommended as regards ease of application, strength, and permanence. Unlike soft soldering, the joint is capable of withstanding the action of boiling water or steam without protection.

Soft Soldering.

In this process the solder melts at a comparatively low temperature, and it is this type of work which has given rise to the widespread view that aluminium is difficult to solder. The reason is that no satisfactory flux is available which will attack the oxide at the low temperature of working, so that the oxide must be removed by mechanical means. After a preliminary cleaning, the metal is heated until the solder melts upon it. The molten solder will not adhere but it can be made to do so by scraping through it with an old hacksaw blade or other form of scraper to break up the oxide film. Once the film is broken the oxide cannot reform under the solder, and alloying takes place. When the surface is fairly well covered with molten solder the adhesion is improved by rubbing with a wire scratch brush while still molten, thus breaking up any remaining traces of oxide. After such 'tinning' the parts can be sweated together in the ordinary way. Fluxes are sometimes supplied with these solders but these consist largely of stearin or resin and are of little assistance.

The composition of the solder is not of great importance. It usually consists of zinc and tin with or without small additions of other ingredients. The best compositions as regards permanence are those with high percentages of zinc, though these are of higher melting point and are therefore not so readily applied.

Reaction Soldering.

This is a newly developed process which is particularly interesting. The solder is a chemical mixture containing halides and zinc chloride, which is spread on the parts to be jointed and heated by a blowpipe to about 200° C. A chemical reaction takes place which results in the deposition of pure zinc in a molten condition on the aluminium surfaces to be joined. The zinc flows readily between the edges and alloys readily with the aluminium, forming an excellent joint. Such joints are much more permanent than joints made by the ordinary soft soldering process.

Cold Solder (Amalgam Solder).

By shaking finely-divided copper with mercury in a little dilute sulphuric acid an amalgam can be prepared which may be used to unite joints at ordinary temperatures. The amalgam is formed into pellets, which are heated till mercury begins to exude from the surface. These pellets are wiped free from mercury and are then rubbed to a soft paste in a mortar. The surfaces to be united are first treated with sodium amalgam (sodium 2 per cent.), the soft paste of copper amalgam being then applied. The surfaces to be joined are then tightly pressed together, the union being completed in about three hours.

Soldering Aluminium Bronze.

To solder aluminium bronze with ordinary soft (pewter) solder : Cleanse well the parts to be joined from dirt and grease. Then place the parts to be soldered in a strong solution of sulphate of copper, and place in the bath a rod of soft iron, touching the parts to be joined. After a while a copper-like surface will be seen on the metal. Remove from bath, rinse quite clean, and brighten the surfaces. These surfaces can then be tinned by using a fluid consisting of zinc dissolved in hydrochloric acid in the ordinary way with common soft solder If the surface is not coppered an aluminium alloy must be used.

Soldering Cast Iron.

(1.)

If the part of iron to be soldered is cast iron that is hard and thin, it should be polished on an emery wheel and made clean and bright. Then dip it in potash water, after which dip it for an instant in clean water, and wash quickly with undiluted hydrochloric acid of the ordinary strength ; go over it with powdered resin and solder made from half tin and half lead. This must be done quickly, before the surface has time to dry.

(2.)

File the surface clean, and wash as before ; wipe it over with a flux made of sheet zinc dissolved in hydrochloric acid until it is surcharged, or is a saturated solution and has been diluted with its own quantity of water ; then sprinkle powdered sal ammoniac on it, and heat it on a charcoal fire until the sal ammoniac smokes. Dip it in melted tin, and then remove, and rap off the surplus tin.

CASTING AND FOUNDING.

Moulding-sand.*

(1.)

1 cubic foot sharply burnt and pulverised pure fire-clay ; 60 cubic inches sugar ; 2 quarts water ; ½ quart paraffin oil. No lime or charcoal must be included in the mixture. Thoroughly mix. Suitable for castings of ingot iron.

(2.)

93 parts pure quartz sand ; 7 parts clay free from lime ; the latter acts as a cement for the former.

(3.)

For small castings :—Red sand, 3 parts by measure ; coal dust (finely sifted), 1 part by measure.

(4.)

For large castings :—Red sand, 3 parts by measure ; coal dust (finely sifted), 1 part by measure ; chopped straw, or dried horse manure, ½ part by measure.

The latter makes the sand light and porous, facilitating the release of the gases. As the sand becomes impoverished by being burnt, the coal dust is renewed from time to time. If paraffin is used to damp the mould, black sand will bind and can then be dried as when red sand is used.

Moulding-sand which has partially lost its cementing power may have it restored by mixing with fresh sand.

Green, or floor sand that has ' gone dead ' (dehydrated) is frequently revived with Birmingham or Mansfield sand. Bunter sand, and also Barrow sand, have been found very good for the same purpose.

When the metal touches the clay bond the clay is undoubtedly killed, but not so the ferruginous bond, and long-life sand is generally rich in the latter.

Ferruginous bond in moulding sand is usually the cause of the production of a smooth, sound surface to the casting, particularly in the case of steel. In non-ferrous practice it is usual to ensure the smoothness of the skin by painting the surface of the mould or ' facing ' it with a layer of finer sand (facing sand) or a wash.

The washes used for painting the mould in non-ferrous work have usually either china clay or a graphitic substance as base. Graphite (plumbago) is itself used at other times. ' Scotch blacking,' which is a mixture of charcoal and graphite, is employed.

A suspension of the powder in water may be assisted by the addition of a little borax or molasses.

---

* For further information on Moulding Sands see ' Sands and Crushed Rocks,' by *A. B. Searle* (*Oxford University Publications*).

Cores.

For the making of 'cores' in British foundries a 'compo' moulding sand is made from ordinary sand such as Mansfield and broken fragments of graphite crucibles ground and mixed.

The admixture of the coarser material ensures good venting of cores; at times the sand alone is used.

Oil-bonded cores (where the oil is generally a commercial linseed, such as 'Baltic seed oil'), unlike those used in steel founding, have not met with much success in non-ferrous founding at home, but are adopted extensively in America, where a fairly coarse high-silica sand is the base. Sands from Lumberton and Howell, N.J., are so used. (*Horobin.*)

Moulding Sands.

Sand.	Moisture Content. %	Green Bonded Strength.		Traverse Strength. Lbs.	Permeability. Secs.
		Riddled. Ins.	Milled. Ins.		
Green facing sand .	7·6	2·3	2·4	—	120
Dry facing sand .	8·5	2·5	2·8	3·0	99
Black . . .	7·5	1·8	—	0·75	141
Red . . .	10·1	2·3	—	2·25	86

(*Morgan.*)

Core Sand Mixtures.

Mixture.	Moisture Content. %	Transverse Strength. Lbs.	Green Bonded Strength. Ins.	Permeability. Secs.	Remarks.
Ordinary core sand	8·2	2·25	2·3	106	Not free venting, but can be handled in the green state
Sea-sand 20 parts to 1 part linseed-oil	5	65	1·2	46	Cannot be handled in undried state.
Sea-sand 40 parts to 1 part linseed-oil	5	28	1·1	43	Suitable for most ordinary work
Leighton Buzzard 20 parts to 1 part linseed-oil	5	53	1·1	28	Binds better with treacle than linseed-oil
Leighton Buzzard 40 parts to 1 part linseed-oil	5	22	0·75	23	

Testing Moulding Sands.

Methods of testing moulding sands are described in 'Sands and Crushed Rocks,' by A.B. Searle (Oxford University Publications). A rough-and-ready test for voids and porosity consists in putting 50 c.c. of water in a glass measuring cylinder and 50 c.c. of the sand (previously passed through a 12-mesh sieve) into a graduated glass cylinder holding 100 c.c. The water is added to the sand and the cylinder with its contents is violently shaken until the water and sand are thoroughly mixed. The mixture is allowed to settle and the total volume is then measured. The reduction in volume is a measure of the volume of voids or pores in 50 c.c. of the sand.

By watching carefully, the time taken for the grains of different sizes to settle may also be noted.

To Repair Castings by Burning on.

A piece of machine-framing, the necks of rolls, or a standard which has been broken or found defective, may be repaired as follows: First cut away the defective parts down to the sound metal; build a coke fire round the part of the casting which is to be repaired until it is brought to a bright red-heat, then dust over the surface of the cut metal with powdered glass or borax. Then apply a hollow loam mould of the desired part to the casting, properly secured in position and provided with a hole for the exit of the metal. Pour very hot liquid cast iron into the mould, and allow it to flow away until the cut surface of the original metal of the casting can be felt with an iron bar to have become soft and pasty by contact with the hot liquid iron. Then stop the exit hole and allow the metal in the mould to set. If the operation has been properly performed the casting should, when struck, ring with the same sound as a single good casting, thus showing that the old and new metal are perfectly united.

Sealing up Cracks in Castings and Cementing Joints.

Cracks in the walls of pressure vessels may be patched by bolting on a plate with a thickness of sal ammoniac and iron borings beneath it. Many an external firebox sheet has been patched in this way.

A cast-iron cylinder for a hydraulic press in which the iron was so porous that, under 2,000 lb. pressure, the water ran down the outside in rivulets, has been sealed by filling the cylinder with water containing a large quantity of sal ammoniac and letting it stand for a couple of days. At the end of that time the cylinder was perfectly tight.

The sal ammoniac treatment has been used in railway shops for stopping cracks in bushing cylinders. In one instance an 18 × 22 in. cylinder was patched that way. It was an old cylinder that had been bored out and a cast-iron bushing inserted. A coat of white lead was spread on the outside of the bushing and after it was in place it was stayed to the cylinder wall by drilling holes through the wall and bushing and screwing in brass plugs. Some time later a crack running lengthwise between the plugs in the outer shell opened up for about 15 in. This was made tight by removing the piston, closing the ends of the cylinder and pouring a sal ammoniac solution through the ports until the cylinder was full. This was allowed to stand for ten days, then the cylinder was drained and refilled with a weak solution of muriatic acid and allowed to stand again for about ten hours. The purpose of the muriatic-acid solution was to complete the work begun by the sal ammoniac. It was presumed that the acid would cut away the coating of rust produced by the sal ammoniac, and that the particles would gravitate toward the pressure. And apparently this is what happened.

White lead has been used with advantage in making up threaded pipe joints and when pressing the brasses into locomotive driving boxes, placing driving pins in the wheels, etc. In the latter case the white lead acts as a lubricant while the job is being done, and later on, after it has set, it serves to cement the part pressed in. For many uses red lead is preferable to white, and next to litharge, is best for holding anything. If a threaded pipe joint is made up, say a 2 in., with white lead, after a year's time it will still be possible to remove the fitting, but if the same fitting is made up with red lead, and it is desired to take it off after the lead has hardened, the fitting will have to be broken unless sufficient heat can be applied to cause it to loosen from the pipe.

There is no cementing effect when graphite and oil is used on a pipe thread. With graphite and oil on a thread a fitting can be made up tighter than would otherwise be possible. Also, the graphite is squeezed compactly into the interstices in the joint and its very compactness holds it against blowing out under pressure, but it does not anchor the fitting. Therefore, no greater stress is necessary to remove the fitting at any time than was used to put it on.

Litharge will make a threaded pipe joint absolutely leak-proof against any kind of fluid and against any pressure that the pipe will stand.

(*Power, July* 1, 1924.)

Pattern Composition

A composition that has many advantages over wood for making small patterns can be made as follows:—With hot water mix into a thick paste three parts by volume of starch, one part ground glue, two parts fine resinous sawdust. The sawdust should not be added until the starch and glue have been dissolved by the water. After the ingredients are thoroughly mixed, heat the whole to 190° F., and continue the heating until the whole becomes a hard mass, then allow to cool, and remove from the receptacle. The resulting composition is a strong, hard, horn-like substance, that can be machined, sand-papered and varnished just as can wood. The principal advantages of this composition over wood lie in the fact that it has no grain, and therefore turned and complicated patterns made from it do not have to be built up or glued together, and that for the same reason it is easier to turn and machine, and offers a smoother surface when finished. It is also more fireproof than wood and not so readily affected by atmospheric changes.

(*American Machinist.*)

Shrinkage of Castings.

The allowance for shrinking in casting should be for each foot in length :—

	Parts of an Inch.		Parts of an Inch.
For cast-iron pipes	·125 = $\frac{1}{8}$	Brass	·17 = $\frac{3}{16}$
„ beams and girders	·1 = $\frac{1}{10}$	Lead	·31 = $\frac{5}{16}$
„ cylinders, large	·094 = $\frac{3}{32}$	Zinc	·25 = $\frac{1}{4}$
„ small	·06 = $\frac{1}{16}$	Copper	·17 = $\frac{3}{16}$
In thick breadth	·156 = $\frac{5}{32}$	Tin	·25 = $\frac{1}{4}$
In thin breadth	·156 = $\frac{5}{32}$	Bismuth	·156 = $\frac{5}{32}$

Cleaning Castings.

A large proportion of the loose sand on castings may be removed in the 'tumbling' process, but the adhering sand and scale of oxide, which soon destroy the cutting edge of the tools, are best reduced by a pickle; the following have met with general approval.

Sulphuric Acid Pickle.—Ordinary commercial sulphuric acid and water in the proportion of 1 of the former to 30 of the latter. In this case the castings are required to remain in the pickle for about 10 hours, and are then washed with hot water to remove all trace of the acid and prevent rusting.

Hydrofluoric Acid Pickle.—This acid is slower in its action and not so liable to cause corrosion as the above. The castings, however, should be first tumbled in order to remove loose sand (as the sand neutralises the solution very much more quickly than the iron) and then immersed in a mixture containing 1 of hydrofluoric acid to 15 of water. The bath should be maintained at a temperature of 120°F. by a steam coil. The castings are then washed in clean hot water and sometimes in hot water with a small proportion of lime (about 1 lb. of lime to 8 gallons of water).

The baths are best made of oak, wood pinned, with no exposed metal fastenings and well lined with asphaltum. The attendant should be furnished with wooden clogs, strong leather apron and rubber gloves to protect him from the acid.

The acid should be added *very slowly* to the *water* and well stirred in the process. Never add the water to the acid, otherwise the acid vapour will be carried all over the place.

An American factory is using high-pressure water for cleansing castings of sand and cores. A centrifugal pump, driven by a 50-h.p. motor, delivers 250 gallons per minute at a pressure of 250 lb. per sq. in., and the work is carried on in a closed chamber with glass windows. It is said that a casting which previously took a whole day to clean can be finished with the water jet in half an hour.

(See also 'Lubrication Troubles,' p. 988.)

Dips for Castings.

Special for Electro-galvanising.—For preparing work for electro-galvanising the following can be used with advantage, but for no other solution except, perhaps, copper : Add to the dilute acid ½ lb. of sheet zinc to each 25 gals. of solution, adding additional amount occasionally as it works out.

Bright Dip for Iron.—Work that has been pickled in the ordinary way to remove sand and scale, and then dried, usually has a dull-black appearance. If the work is to be polished, the black will be removed and the colour restored in the polishing process; but if it is to be plated, or if it is to be left with parts of iron showing, the work should be run through the bright dip. This will bring it out white, improving the appearance and tending to make it take on a white deposit if it is to be nickelled. This dip is composed of :—

Hydrofluoric acid, 24 oz. Nitric acid, 10 oz. Metallic zinc, 2 oz. Water, 2 gals.

Work that is to be dipped should be strung on copper wire, and swirled around in the solution for a few seconds, rinsed in cold and then very hot water, and dried in sawdust. Articles should not be allowed to stay in the dip any time, as they would be spoiled and the solution ruined.

Satin-finish Dips.—Brass work requiring a satin finish may be dipped for a few seconds into a mixture consisting of :—

(1) Hydrofluoric acid, 1 part; Water, 3 parts;

then dried and run through the bright dip (4) to give it a lustre. If not satined enough the operation may be repeated.

(2) Hydrofluoric acid, 2 parts; Nitric acid, 1 part; Muriatic acid, ½ part; Water, 5 parts. After which the work may be rinsed, dried, and run through the bright dip (4) if necessary.

(3) Hydrofluoric acid, 2 parts; Nitric acid, 1 part; Water, 10 parts.

The following is a *bright dip* for copper, brass, bronze, or German silver :—

(4) Hydrofluoric acid, 4 quarts; Nitric acid, 3 quarts; Common salt, small handful.

Protection of Copper Castings.

The best method for protecting copper castings from oxidation in air, resulting in a coating of verdigris, is for the casting to be done under hot water, with the metal at a very high temperature. This method, largely employed in Japan, forms a coating of red cuprous oxide, which has been known to give protection for upwards of thirty years.

Die-Casting.

Contributed by A. H. Mundey, Assoc.Inst.C.E., F.C.S., M.Inst.Met., of Fry's Metal Foundries Ltd.

Die-castings have been defined as castings produced by pumping metal into metallic dies at a pressure higher than that of the atmosphere. This distinguishes them from the class of castings made by the permanent mould process into which the metal is simply poured, also sometimes referred to as die-castings and on other occasions as chill castings. There is, however, a tendency now to make the phrase 'die-casting' embrace both the methods, and to distinguish the sections as 'gravity die-castings' and 'pressure die-casting.'

The equipment for die-casting includes a melting pot (usually of cast iron), gas-heated, placed near the die-casting machine, for supplying the molten metal to the reservoirs of the caster.

The casting machines are of two general classes, those in which the metal is forced into the die or mould by the action of a plunger or pump, and the other in which compressed air is used for this purpose.

The reservoir attached to the machine is heated by gas, the pump being in direct contact with it; the plunger is actuated, by means of a long lever, by the operator, the metal being forced from the pump chamber through a suitable nozzle into the dies. The moulds are attached to a hinged plate which swings away from the reservoir and pump nozzle, after each cast, to allow the workmen to remove the tang and clean the metal passages.

The swinging plate is securely gripped to the reservoir at the moment of casting.

The mould is held to the swinging plate by means of a toggle joint, and varies in form and design according to the particular castings; it is always at least a two-part mould, but usually is also fitted with elaborate cores and drawbacks and mechanism for working them, which make the production of complex castings possible.

The dies are made in mild steel, alloy steel, and occasionally in cast iron, according to the alloy and character of the work in hand. Sometimes the dies are heated, for other work they are required to be water-cooled, so that no hard and fast line can be set in this respect.

The compressed-air type of die-casting machine consists of an air-tight cast-iron chamber for the metal, suitably heated to keep the metal at the desired temperature. This chamber is connected with the die by one opening, and with the compressed-air line, by another. In the latter opening a valve is fitted with a lever for the sudden release of the air into the metal chamber. This valve also permits of a gradual decrease of the pressure on the metal after it has filled the die. This type of machine may be used on metals cast by the plunger type, but is especially suitable for metals that cannot be cast in the plunger type of machine on account of their high melting-point or corrosive action on the cast-iron plunger.

A die-casting die consists essentially of two parts which are brought together and maintained in perfect register by means of the usual dowel pins.

Internal threads are produced by screwed cores and external ones by removable screwed collars, whilst the holes which are usually drilled in machine-made fittings are in these cases formed by means of retreating cores; the latter may be of almost any shape.

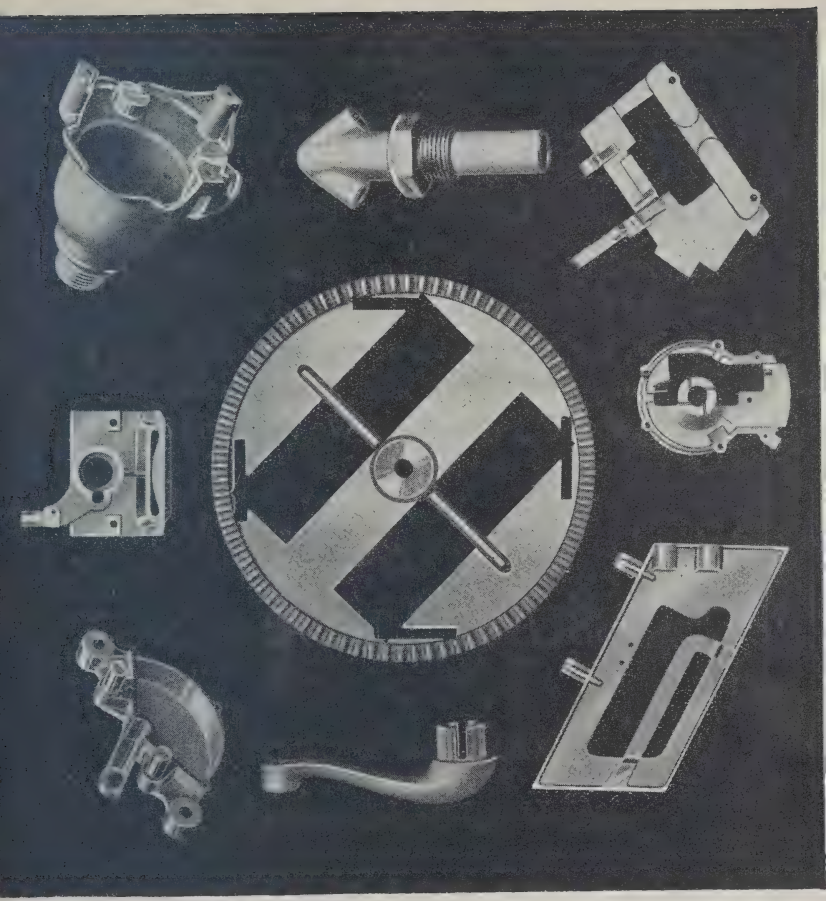
Small cores are operated by levers, large cores by a rack and pinion; they are always locked whilst the casting operation is proceeding. A vital consideration is the proper venting of the die; the position and form of the vents can only be determined by experiment.

Great accuracy is obtainable; it is quite possible to produce castings to a limit of 0·001 inch.

Alloys used for Die Castings.

The alloys generally used are of considerable variation, according to the special purpose and circumstances those of the white-brass or zinc-base group are usually approximately:

86 per cent. zinc.; 7 to 10 per cent. tin; 4 to 7 per cent. copper; with 0·5 to 1·0 per cent. aluminium.



ALUMINIUM ALLOYS

Silicon
Copper
Nickel

ZINC BASE DURA ZINK

High Tensile

'PRANA' PRESSURE DIE CASTINGS

FOR ECONOMY

SPARKLETS LTD.

EDMONTON, LONDON, N. 18

SOLE LICENSEES

OF THE NEW DOEHLER PRESSURE-ALUMINIUM AND BRASS PROCESSES

Telegrams: Spumante Phone London

Telephone: TOTTENHAM 2647

F. 2756

THINK IN TERMS OF DIE CASTINGS

When large numbers of small castings are required **die castings save money.** In particular, the production of pressure die castings in yellow brass by a new process is of great importance to engineers. We will send details on application.

We produce

PRESSURE DIE CASTINGS

in

YELLOW BRASS

ALUMINIUM ALLOYS

ZINC BASE ALLOYS

TIN and LEAD BASE ALLOYS

Fry's Metal Foundries, Ltd.

25-42 Holland St., Blackfriars, London, S.E.1

Telephone: Hop 4720. Telegrams: "Frymetalos, Phone, London."

F. 2757

Increased tin gives greater ductility, whilst increased copper renders the alloy harder.

The tin-base alloys are (*a*) bearing alloys of all the usual grades of white anti-friction metal, and (*b*) alloys specially designed to withstand the corrosive effect of chemicals in food-stuffs, certain gases, and even acids.

An alloy which is largely employed is :

90 per cent. tin ; 4 per cent. copper ; 6 per cent. antimony.

Of the aluminium group the most commonly used alloy is that known as No. 12, consisting of :

92 per cent. aluminium ; 8 per cent. copper.

Sometimes a small quantity of magnesium is added and also frequently from 1 to 3 per cent. of nickel.

The finish or skin of the casting is smooth and in a large measure satisfactory, but polishing, plating, and bronzing are frequently employed, for protective as well as decorative purposes.

Shrinkage is an important consideration ; this varies greatly with the alloys used. Approximate allowances are :

White brass or zinc base alloys . . .	0·0035
Lead base alloys	0·002
Aluminium alloys	0·007

The alloys generally used for die-castings are of relatively low melting-point and are of such a constitution that they have a period of plasticity during the process of solidification. This adds much to the chance of the production of sound castings.

Temperature of working is an important factor ; if too high, splashing results, with production of long fins, oxidation, and discomfort in working. Too low temperature produces bad finish with little folds and creases on the surface, due to premature solidification.

The metals known as yellow metals—that is, high copper alloys—have hitherto been cast almost exclusively by the gravity process. In these cases great accuracy is not obtainable for long runs, owing to the wear of the dies at the high temperatures employed.

A great advance has taken place, however, quite recently in the introduction of several processes for the production of yellow metal die-castings of very high finish, having high mechanical properties, by pressure machines. Fry's Metal Foundries, Ltd., have secured the Empire rights of a process which uses the metal to be cast in a plastic or semi-fluid condition. The machine employed is known as the ' Polak,' and operates by hydro-pneumatic pressure.

The yellow metal alloys usually employed for gravity castings are: copper 90 to 95 per cent., aluminium 10 to 5 per cent. ; also copper 60 to 65 per cent., zinc 40 to 35 per cent. For pressure work the series of alloys of the latter class are mostly employed.

Mechanical Treatment of Metals and Alloys.

Method of Treatment.	Details of Method to be avoided.	Consequences of Details of Method of Work.
Hot working . .	1. Excess 'reduction' or rate of 'reduction.'	Cracking of metal.
	2. Deficient 'reduction' or rate of 'reduction.'	Coarse, weak metal.
	3. Excess temperature at commencement.	Weak, brittle metal and formation of fissures therein.
	4. Deficient temperature at conclusion.	Internal strain and consequent hardening. Possible 'season cracking.'
	5. Retention of scale on metal during operations.	Rough and pitted surface and possible cracking.
Cold working .	1. Excess 'reduction' or rate of 'reduction,' without annealing.	Surface cracking and hollow drawing. Strain-hardening.
	2 Deficient 'reduction' or rate of 'reduction.'	Insufficient hardening — where hardening is expected.
	3. Unequal 'reduction' . . .	'Season cracking' or cracking during annealing.

Heat Treatment of Metals.

Method of Treatment.	Details of Method to be avoided.	Consequences of Details of Method of Work.
Annealing	1. At too high a temperature	'Burning' and 'over-heating' in steel and brass. Brittleness and sponginess generally, owing to abnormal grain size.
	2. At too low a temperature	Insufficient structure change.
	3. For too long a period — at correct temperature.	Diffusion and aggregation of micro-constituents of metal or alloy.
	4. Careless charging in or firing of furnace.	Sudden access of heat may lead to cracking of cold metal.
	5. In an injurious atmosphere or in contact with injurious substances.	Copper is injured by reducing gases, steel by oxidising gases. Water and oil have injurious effects in many cases.
Quenching	1. From too high a temperature	Cracking, warping and brittleness.
	2. From too low a temperature	The object of the operation is defeated.
	3. Unequal	Warping.
Tempering	1. At too high a temperature	Removes the effects of quenching.
	2. At too low a temperature	Does not remove the effects of quenching sufficiently.
	3. Unequal	Lack of uniformity and consequent failure.

See also p. 2697.

SECTION XL

PART II

WELDING AND CUTTING.

(Contributed by D. Richardson, Wh. Exh., A.M.I.Mech.E., Capt. R.A.F.)

OXY-ACETYLENE WELDING.

The term ' autogenous welding ' is a cumbersome and loose description for the process, which has become synonymous with oxy-acetylene or gas welding, although its real meaning, ' self-producing,' does not fully apply. The term was chosen to describe a process in which two sections of metal are melted together by means of a blowpipe in contrast with the then existing methods, in which a joint was made by the addition of a different metal, as in brazing. The term ' fusion ' is considered to be much more appropriate than ' autogenous,' and is applied to a weld worked in the fluid or vapour state.

The easy control and intensity of the oxy-acetylene flame, the low cost of operation, the relatively inexpensive equipment required, the range of metals which can be welded, and the adequate supplies of carbide of calcium, oxygen, etc., available, are factors which have led to its many-and varied applications.

The industrial development which has taken place can also be attributed to the increased efficiency of the welder and the efforts that have been made to reduce the art of welding to a more scientific basis. The uncertainty of the results, upon which opposition and criticism are usually based, can, in general, be traced to one or more of the following causes : (*a*) inefficiency of the welder ; (*b*) use of defective or unsatisfactory apparatus ; (*c*) use of defective or unsuitable filling materials ; (*d*) lack of proper supervision and maintenance of the generating plant and equipment ; (*e*) unsuitability of the material welded.

It is not difficult to eliminate these causes, and a well-trained welder using good apparatus, correct filling materials, etc., can produce welded joints on mild steel, cast iron, copper, brasses, bronzes, aluminium and aluminium alloys, lead, etc., with mechanical and physical properties approaching those of the base metal, and in which the strength and service of the joint are absolutely reliable.

Equipment.

The apparatus required for an oxy-acetylene installation depends upon several factors, such as (*a*) the class of work to be welded; (*b*) whether a portable or stationary installation meets the case ; (*c*) whether production or repair work, or both, are possible applications ; (*d*) the facilities for installing equipment and for handling the work in prospect. In general, the apparatus and accessories will comprise a generator for generating the acetylene gas, which may be a fixed or portable plant—the most portable type of plant uses compressed or dissolved acetylene in portable cylinders instead of a generator ; chemical and mechanical purifying apparatus, when the gas is obtained from a generator ; oxygen apparatus for the production of oxygen, or oxygen in cylinders ; oxygen regulators for lowering and regulating the pressure of the gas after it leaves the cylinder, to the pressure required for the blowpipe, and, in the case of dissolved acetylene, an acetylene regulator for the same purpose ; hydraulic safety valve or other safety device for dealing with explosive mixtures of oxygen and acetylene or air and acetylene ; welding and cutting blowpipes of various capacities ; accessories, such as flexible tubing, goggles, welding benches, blowpipe support, preheating devices, tools, jigs, fluxes and filling materials, carbon paste and blocks, fire protection, protective equipment for the operator, etc., etc.

Fig. 1 shows in diagrammatic form an elevation and plan of the general layout of an acetylene welding and cutting installation. The letters refer to the same details in both views.

The acetylene generator, shown at A, is usually automatic—that is, the generation of gas is controlled entirely by the gasholder regulating the water supply or the carbide supply; B is

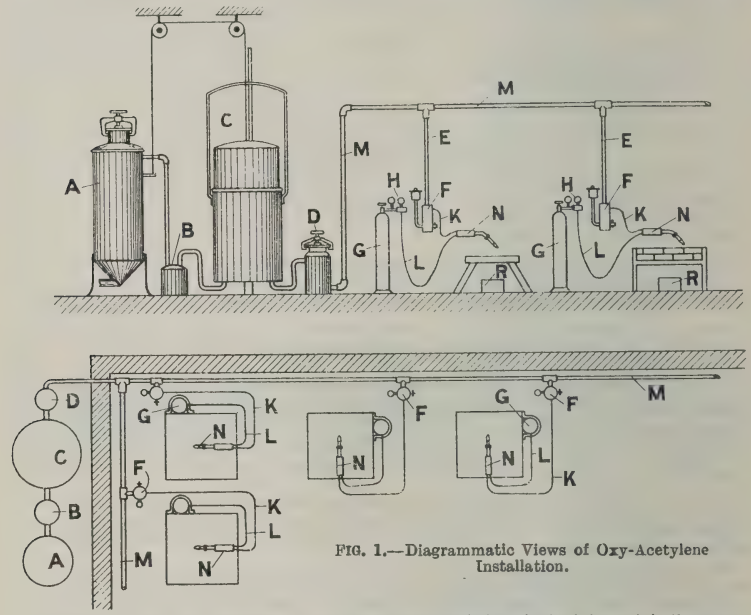


FIG. 1.—Diagrammatic Views of Oxy-Acetylene Installation.

a condenser, washer, and water-seal; C is a gasholder, consisting of a tank to contain the water-seal and a bell to store the gas; D is a purifying chamber and filter. The main piping, M, is connected to the purifying chamber and usually forms a complete circuit of the welding shop; branch pipes, E, are taken from the main for supplying acetylene to each welding station; F is the

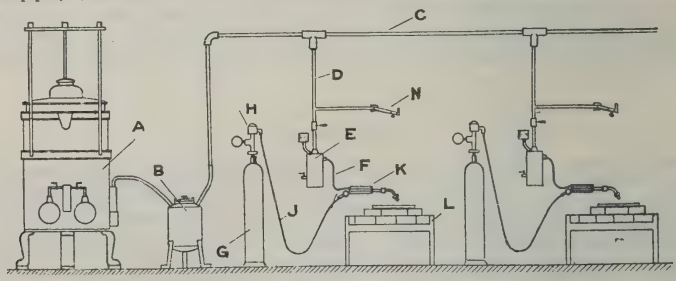


FIG. 2.—Diagrammatic View of Low-Pressure Oxy-Acetylene Installation.

hydraulic safety valve or other safety device; a flexible tube, K, connects the hydraulic safety valve to the welding or cutting blowpipe, N; G is the oxygen cylinder, on to which is screwed the oxygen regulator H; a flexible tube, L, connects the oxygen supply to the blowpipe N; two common types of welding benches, with blowpipe cooling tanks R, are shown.

For comparatively small installations, the generator, condenser, washer, water-seal, and gas-holder form a single piece of apparatus, as shown at A in fig. 2. The diagram shows at N a pilot light bracket attached to the branch tube D. An acetylene jet burner consuming about 5 litres of acetylene per hour, screwed into the bracket, forms an efficient and economical method of lighting the blowpipe. The burner should be at least 18 inches from the hydraulic safety valve and not more than 6 ft. above floor level. If arrangements are not made for an acetylene or coal-gas blowpipe lighting circuit a friction lighter should be used, not matches.

An installation of the type described is known as a low-pressure installation, to distinguish it from the type in which the acetylene is obtained from cylinders, known as the high-pressure system.

Fig. 3 shows the equipment for the high-pressure system. After passing through suitable acetylene and oxygen regulators the gases are conveyed to the blowpipe by means of flexible tubes.

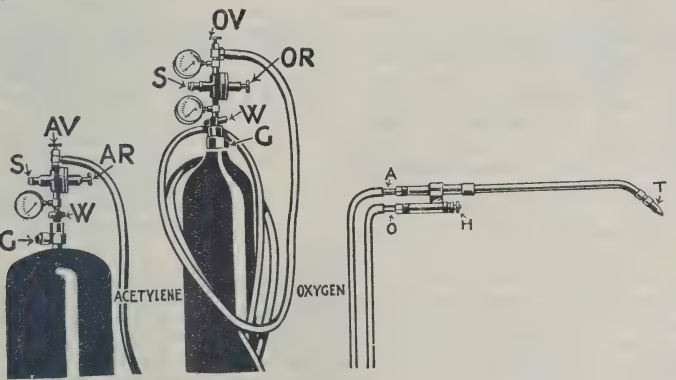


FIG. 3.—High Pressure or Dissolved Acetylene Installation.

OXYGEN SUPPLY.

The oxygen may be obtained in cylinders or may be produced on the premises. There are three methods of production, namely :—(1) chemical processes ; (2) the electrolytic process ; (3) the liquid air process. The older chemical processes are practically obsolete as regards oxygen for welding and cutting. Modern chemical processes are represented by the Oxygenite, Oxylithe, and Epurite processes. The main objections to chemical processes, which are now seldom used in welding and cutting processes, are the cost and impurity of the gas. The effect of impure oxygen in welding is to retard the welding operation, and thus increase the cost, and a tendency to produce defective welds. In cutting operations the purity of the oxygen is a most important factor, and tests have shown, for example, that with oxygen of 91 per cent. purity the cutting speed is halved when compared with oxygen of 98 per cent. purity. Simple and cheap apparatus is available for rapidly and accurately testing the purity of the gas.

The electrolytic and liquid-air processes may be used for producing oxygen on the premises. The gas can then be charged into cylinders or piped into the welding shop from large receivers into which it has been compressed, or it may be piped direct from the oxygen-producing plant. For information on ' Industrial Oxygen ' reference should be made to a paper read by T. Campbell Finlayson on the subject before the Institution of Chemical Engineers, in June 1923, and the Report of the Oxygen Research Committee, Department of Scientific and Industrial Research, Sept. 1923.

The problem of distributing oxygen in a liquid state instead of by the existing method of distributing the gas in steel cylinders at 120 to 150 atmospheres pressure is, at present, receiving considerable attention. Demonstrations showing the advantages of this system were given at the Annual Meeting of the German Acetylene Association. A liquid oxygen plant for oxy-acetylene welding has recently been installed at Middlesborough.

Oxygen produced by the electrolytic or liquid air processes has a purity of the order 99 per cent. Unless the consumption is very large the first cost of an oxygen installation is comparatively high, and the majority of users obtain the gas in cylinders from the manufacturers. The cylinders supplied by the manufacturers for welding and cutting usually contain 100 cu. ft. of gas at a pressure of 1,800 lbs. per sq. in. (120 atmospheres). The Gas Cylinders Research Committee of the Department of Scientific and Industrial Research have made a series of recommendations for ordinary commercial cylinders used for the storage and transport of gases including oxygen,

and are at present engaged on the standardisation of the fittings of gas cylinders.* They have also issued a Second Report dealing with the periodical heat-treatment of gas cylinders †; a Third Report dealing with alloy steel light cylinders ‡; and a Fourth Report, which completes the work for which the Committee was appointed, on cylinders for liquefiable gases.§ The Oxygen Research Committe have issued a Report mainly dealing with the storage and transport of liquefied gases, but also including a summary of the recommendations relating to compressed oxygen.‖ The British Engineering Standards Association have also formed a committee with a view to standardising cylinder valves and fittings.¶

In handling oxygen cylinders care must be taken against leakages, and particular care must be exercised not to oil or grease any part of the oxygen apparatus or supply line. Dust and moisture should be expelled by opening the cylinder valve before the cylinder is put into use. Repairs to cylinder valves should not be attempted by the user, and full cylinders should be handled carefully. All oxygen cylinders are fitted with valves having right-hand connections, and are generally painted black. No attempt should be made to transfer oxygen from one cylinder to another by rigging up some form of communicating tube. In the majority of cases a separate cylinder is used for each welding station or blowpipe in use. The arrangement of manifolding oxygen cylinders is being increasingly used as the handling of single cylinders, and wear and tear on the pressure regulators, is largely obviated. Oxygen manifolds should be able to withstand a pressure of at least 3,000 lbs. per sq. in., and each cylinder connection should be equipped with a high-pressure shut-off valve. It is extremely advisable that the necessary manifolds be purchased from a responsible manufacturer or made to specification. A cylinder valve should never be opened suddenly when the oxygen regulator is attached. In the event of the oxygen supply being restricted owing to the freezing of moisture in the cylinder valve, hot water only should be used to put it right. Leakages can be discovered by the hissing noise produced by the escaping oxygen, by the use of soap-suds spread over likely places, or by closing the valves on the blowpipe, cylinder, and regulator, and noting how long the pressure holds up on the gauges. Cylinders should be placed where they cannot be easily knocked over, and never near sources of heat. Spare cylinders should be safely and properly stored, care being taken not to store them near radiators, oil, grease, or combustible materials.

For estimating welding costs, the following table will be found useful. The table gives the number of cubic feet of oxygen remaining in a 100 cu. ft. cylinder corresponding to various readings of the high-pressure gauge on the oxygen regulator. The amount of oxygen indicated by the gauge reading is more or less approximate, as it depends upon the temperature of the oxygen. For practical work the deviations can be ignored, and would not affect the usefulness of the table. In general, the volume of oxygen in a cylinder is nearly proportional to its pressure, and thus, in the absence of the table, the volume of gas remaining in a cylinder can be determined by observing the gauge reading. A regulator provided with a pressure gauge which serves the double purpose of registering the pressure of the gas in the cylinder, and indicating at any period the quantity of gas in the cylinder, can be used.

Gauge Pressure in Atmospheres.	Cubic Feet of Oxygen in Cylinder.	Gauge Pressure in Atmospheres.	Cubic Feet of Oxygen in Cylinder.
10	8·3	70	58·4
20	16·7	80	66·7
30	25·0	90	75·0
40	33·4	100	83·4
50	41·7	110	91·7
60	50·0	120	100·0

At atmospheric pressure and temperature oxygen is rated at 12·08 cu. ft. per pound.

Oxygen is supplied to the trade at prices which vary according to the size of the cylinder, quantity purchased, and other conditions. Cylinders are loaned, or may be purchased outright. Demurrage charges are made on hired cylinders when retained above a certain period. The cylinder valve should always be closed when the cylinder is not in use and when returned for re-filling. Empty cylinders should be handled with the same care as for full cylinders. The proper care of cylinders and valves and their prompt return to the filling station are important factors in obtaining cheap oxygen. Oxygen cylinders are conveyed by the railway companies under strict regulations and conditions.

---

* *First Report of the Gas Cylinders Research Committee* (1921).
† *Second Report of the Gas Cylinders Research Committee* (1926).
‡ *Third Report of the Gas Cylinders Research Committee* (1929).
§ *Fourth Report of the Gas Cylinders Research Committee* (1929).
‖ *Report of the Oxygen Research Committee* (1923).
¶ Cylinder Fittings, British Engineering Standards Association, Publications Nos. 341 and 349 (1929).

B.O.C. SERVICE
for all Welding Processes

Oxy-Acetylene · Electric Arc and Electric Resistance

ENGINEERS and manufacturers are invited to make use of the Company in any matter relating to welding. A staff of experts and demonstrators in each of the three main systems is always available to discuss problems and to show what can be done under practical conditions. Speed of work, strength and economy are the objects of all welding processes. Their adoption is daily increasing in all sorts of manufacturing and repair work. For the fullest information and enthusiastic help, write or telephone to the Company's Head Office, or to the nearest Branch.

The BRITISH OXYGEN Co LTD
Inc. ALLEN-LIVERSIDGE Ltd.

Head Office and Engineering Works:
ANGEL ROAD, EDMONTON, N.18

Offices & Works: Barrow-in-Furness, Belfast, Birkenhead, Birmingham, Bristol, Cardiff, Coventry, Crewe, Derby, Dundee, Erith, Glasgow, Hull, Jarrow, Leeds, Leith, Liverpool, London (N.W. & S.E.), Manchester, Newcastle-on-Tyne, Plymouth, Sheffield, Southampton, Stockton-on-Tees, Swansea, Witham, Wolverhampton: and in Australia, India and South Africa.

B.O.C. EQUIPMENT
FOR WELDING, LIGHTING AND METAL-CUTTING

INCLUDES:

OXY-ACETYLENE WELDING AND METAL-CUTTING

Oxygen, Dissolved Acetylene, Automatic Oxygen Cutting Machines, Acetylene Generating Plant, Blowpipes, Regulators, Fluxes, Wires, Tubing, Gloves, Goggles, Welding Tables, Trolleys, Carbide of Calcium, and all Accessories.

ELECTRIC WELDING

Electric Arc Welding Plant, Electric Resistance Welding Machines (Spot, Butt and Seam), Electric Rivet Heaters, Flux-coated and Bare Wire Electrodes, Electrode-holders, Masks and all Accessories.

PORTABLE LIGHTING

Alda-Lights (Dissolved Acetylene Flare Lights), Imperial Lights, Portable Lamps for all purposes, Searchlights, Motor Vehicle Lighting Equipment, Carbide of Calcium.

The BRITISH OXYGEN Co LTD
Inc. ALLEN-LIVERSIDGE Ltd

Westminster Offices:
HORSEFERRY ROAD, S.W.1 and VICTORIA STATION HOUSE, S.W.1

Offices and Works: Barrow-in-Furness, Belfast, Birkenhead, Birmingham, Bristol, Cardiff, Coventry, Crewe, Derby, Dundee, Erith, Glasgow, Hull, Jarrow, Leeds, Leith, Liverpool, London (N.W. & S.E.), Manchester, Newcastle-on-Tyne, Plymouth, Sheffield, Southampton, Stockton-on-Tees, Swansea, Witham, Wolverhampton: and in Australia, India and South Africa.

56

F. 2763

Acetylene Supply.

The acetylene may be obtained compressed in cylinders filled with a porous material, the pores of which hold a film of acetone in suspension, it is then known as dissolved or compressed acetylene, or may be generated on the premises. The precautions in handling acetylene cylinders are much the same as for oxygen cylinders. The gas should not be drawn off at too high a rate—in other words, the greater the demand the larger the cylinder. An investigation has been made to determine the percentage of acetone vapour in dissolved acetylene at varying temperatures and pressures. The figures show that the output of the flame diminishes by 1·8 per cent. for 1 per cent. of acetone vapour in the flame. The temperature of the cylinder should be kept low and the cylinder pressure should not fall too low in order to keep the percentage of acetone vapour low. For heavy continuous work two or more cylinders should be manifolded together by means of special adaptors supplied by the manufacturers. For general work cylinders containing 60 to 275 cu. ft. of gas are advisable. For light work such as thin sheet metal, lead burning, etc., small cylinders can be used with advantage. All acetylene cylinders used for oxy-acetylene welding are provided with valves screwed with left-hand threads, and are painted a dark reddish colour. Cylinders for welding and cutting can be hired or purchased. Always close the valves of empty cylinders. Stocks of oxygen and acetylene cylinders, whether full or empty, should not be stored in the same place unless there is a fire-resisting partition between them. Cylinders should be kept far enough away from the actual welding or cutting operation so that sparks or flame will not reach them. Acetylene should not be supplied from a battery of cylinders to a system of shop piping without consulting the suppliers of equipment.

Dissolved Acetylene Cylinders.

(British Oxygen Co., Ltd.)

Approximate Capacity. Cu. Ft.	Approximate Weight, Fully Charged. Lbs.	Approximate Dimensions.	
		Diameter.	Height.
		Ins.	Ft. Ins.
3½	4	3	1 0
5	5½	3	1 2
10	8¼	4	1 4
20	23	6¼	1 3
40	28	6¼	2 0
60	70	8⅝	2 1
120	120	8⅝	3 4½
200	168	10	4 0

Before the gas is compressed into cylinders it is first purified and dried by several processes, and then compressed into solid drawn seamless steel cylinders.

The acetylene and oxygen being under high pressure, a perfectly uniform mixture of both gases can be maintained in the blowpipe and the pressure varied according to requirements. The elimination of the injector device, necessary in low-pressure blowpipes, and the hydraulic safety valve are advantages claimed for dissolved acetylene. Dissolved acetylene can be used with any type of oxy-acetylene blowpipe, but it should be recognised that the device for dealing with back-fires and flash-backs does not, as a rule, form part of the details of low-pressure blowpipes.

The amount of acetylene in a cylinder cannot be determined, as in the case of oxygen, from the pressure gauge reading. The only accurate method of calculating the gas used from, or left in, a cylinder is by weighing before and after use. The empty and full weights are shown on a label fixed to each cylinder after charging. For calculating the volume of acetylene used the following are the equivalents:

1·1 oz. of acetylene = 1 cu. ft. (approx.)
1 lb. ,, ,, = 14·5 cu. ft. ,,

By means of an acetylene generator acetylene is produced by the interaction of calcium carbide and water. There are three general types of acetylene generators which may be classed under the heads of water to carbide type, carbide to water type, and the dipping or displacement type. They can also be classified as low-pressure and pressure generators. Briefly, low-pressure generators deliver acetylene to the blowpipe at a pressure of about 8 ins. water pressure (0·036 lb. per sq. in.), whilst pressure generators deliver the gas up to a maximum fixed by the Home Office at 250 ins. water pressure (9 lbs. per sq. in.). The latter are not in general use.

The design and layout of a generating plant must be such that the acetylene reaches the blowpipe in a good condition, in sufficient quantity and at sufficient pressure, otherwise the welds may be porous, brittle, and burnt. The acetylene generator should be of good and solid

construction, and the gasholder capable of storing all gas generated under working conditions or after stoppage of work. There should be no signs of polymerisation or overheating of the gas even when working at the maximum delivery. The cross-section of tubes, pipes, and passages through valves should be such that the gas flows normally at the maximum delivery without loss of pressure. The pressure of gas should be as high as possible, and at least equal to 6 ins. of water pressure in the case of small plants, and 8 to 10 ins. in larger plants. The pressure should remain practically constant during working and not vary more than $\frac{1}{2}$ in. water pressure. The charging and cleaning of the plant should not result in any appreciable waste of gas, and be made without introducing sufficient air to interfere with the proper working of the blowpipe. The generator should comply with the Regulations of the British Acetylene and Welding Association. The size of carbide suitable for the type of generator should be used. For commercial purposes carbide is broken and packed in the following sizes: 80/120, 50/80, 25/50, 15/25, 7/15, 4/7, 2/4, and 1/2 millimetres. The figures refer to the approximate mean thickness of a lump in millimetres. The 50/80 size is in general use for welding and cutting generators. The choice of carbide-to-water, water-to-carbide, etc., generating plants depend on several factors, such as location, class of work, etc.

The Carbic system of generation is based upon the use of a carbic cake, which consists of specially prepared and treated carbide compressed into cylindrical cakes. It is claimed that the process renders every particle of carbide impervious to moisture, so that generation can only take place while water is in actual contact with the cake. Freedom from danger in handling, and from after-generation, convenience in handling, elimination of waste, and purity of gas are factors which have led to the success of this system.

Overloading of the plant is a common cause of defective welds. In carbide-to-water generators an allowance of not more than 1 cu. ft. of acetylene per hour for each pound of carbide in the rated charge is recommended. In the case of water-to-carbide generators the maximum rate of gas consumption should be half this value. In well-designed generators the yield of acetylene gas from the generator will be 85 to 90 per cent. of the carbide yield. Practically, for every 10 lbs. of carbide used in the generator a yield of 40 cu. ft. of gas at the blowpipe can be assumed. The proper care, cleaning, charging, and maintenance of generators is of first importance if reliable welds are required. Leakages of acetylene should be traced by means of soap-suds and never with naked lights.

The acetylene generating plant should be installed in a well-ventilated shed or building, or in a lean-to having no direct communication with a workshop or adjacent building, and no unauthorised person should be allowed to enter the building or enclosure. Where artificial light is required in a generator house or near a generator, electric incandescent bulbs, encased in outer gas-tight globes, should be used. A conspicuous notice prohibiting lights or smoking should be exhibited on the outside of the generator house. Generators should be cleaned and charged in daylight. Electric switches, fuses, etc., should be placed outside a generator house, or they should be so constructed that risks due to sparking or fusing are avoided. To comply with existing regulations, a licence under the Petroleum Acts must be obtained from the local authority for the storage of carbide. A sliding scale of licence fees, varying from 5*s.* to £5, according to the quantity stored, has been fixed, and severe penalties for contravening any condition printed or endorsed on the licence by the local authority are included. The London County Council have issued a circular setting forth the conditions which acetylene apparatus should fulfil before being approved for use on licensed premises.* A raised platform in the generator house should be provided for the storage of carbide. Drums containing carbide should not be placed in the welding shop or near a flame. Care must be exercised in opening drums to avoid striking sparks; a proper drum opener should be provided for this purpose. When the contents of an opened drum are not entirely used in charging the generator, the remainder should be transferred to a storage bin which can be sealed.

All users and prospective users of oxy-acetylene equipment should consult the memorandums issued by the Factory Department of the Home Office† and the Board of Trade.‡

Purification of Acetylene.

The chief impurities found in acetylene gas are (*a*) those which affect the quality of the weld, such as sulphur, phosphorus, silicon, finely subdivided lime; and (*b*) those which affect the cost of the weld, such as hydrogen and hydrogen compounds, nitrogen and nitrogen compounds, and water vapour. An efficient chemical purifer must be provided capable of removing sulphur and phosphorus compounds, the acetylene must pass through an efficient water-scrubber to remove or partially remove the impurities soluble in water and, finally, the gas should be filtered to remove solid impurities not eliminated by washing or chemical purification. A first-class purifying material will generally remove chemical impurities, moisture, and lime dust which are a common cause of defective welds.

Purification of the gas must be efficient, and the care and maintenance of the purifying chamber is very important. The cross-sectional area of the purifier should be ample, and the depth of purifying material should not exceed that specified by the makers, otherwise loss of

* 'Carbide of Calcium,' L.C.C. Public Control Department (Circular 80).
† Dangers from the use of acetylene gas and in oxy-acetylene welding in factories (Form 1704).
‡ Instructions to Surveyors. Repairs to Boilers (Circular 1596).

pressure and overheating of the gas will take place. The quality of the gas can be simply and quickly tested by the use of a white absorbent paper, saturated with a 10 per cent. solution of silver nitrate, exposed to the gas. Should the paper turn black after a few seconds' exposure, the gas is unsuitable for welding purposes.

Among the best known purifying materials are Heratol, Catalysol, Klenzol, and Suder. The three last are distinguished by their regenerative properties—that is, they recover their activity by simple exposure to the air, and thus can be used over and over again. In purifying acetylene the cost should be based on the volume of gas purified and not on the first cost of the material.

Preservation of Acetylene Apparatus.

The question of preventing the corrosion or deterioration of the material used in the manufacture of acetylene generators, purifying vessels, and other parts of the generating plant is important in view of the fact that according to the efficiency of the coating the life of the plant may be a question of months or years.

There are five cases to be considered, namely: (*a*) preservation of the external parts of the apparatus; (*b*) preservation of the immersed parts; (*c*) preservation of parts in contact with moist and impure gas; (*d*) preservation of the interior of purifying vessels; and (*e*) preservation of carbide containers. With regard to (*a*), good ironwork paints give good results when applied well to properly prepared surfaces. At least two coats should be applied. If the apparatus is exposed avoid paints used for woodwork, plaster, etc. For (*b*) it is necessary to use a waterproof paint which will resist the prolonged action of water. The best results are obtained when the base consists of tar and graphite. For (*c*) the paints used for immersed parts of acetylene apparatus give the best results, but the work must be done with great care. For (*d*) the majority of industrial preservative paints give satisfactory results, except those containing a zinc or similar base. Several coats are necessary, and great care is required in applying the coats on to well-prepared surfaces. For (*e*) tests have shown that no paint or coating will satisfactorily resist the action of the lime residue at the temperatures usually present, and that the lime residue appears to have a preservative action on the parts with which it is in contact. It is therefore unnecessary to protect these parts.

Oxygen and Acetylene Regulators.

Regulators in common use to reduce the pressure of gases are of the diaphragm type, and may be roughly divided into two classes—first, where the pressure of the gas closes the inlet, and, second, where the inlet is closed by a spring. They are designed to automatically deliver gas to the blowpipe at any uniform pressure to which the regulator is set. They should be designed for working over a wide range of pressures, and should be sensitive even at low pressures. Oxygen regulators are usually equipped with two gauges, a high-pressure gauge, which shows the pressure of gas in the cylinder, and in some types the quantity of gas in the cylinder, and a low-pressure gauge. Pressure gauges should be provided with a check to prevent a sudden inrush of gas. For successful welding the regulator should be provided with a low-pressure gauge, giving the pressure at which the gas is delivered to the blowpipe. The range of the low-pressure gauge should not be too great, and a maximum reading of 75 lbs. per sq. in. will be sufficient for the majority of welding and cutting operations. Regulators must be kept in good order, otherwise defective welds are inevitable. Regulators intended for use when welding should not be used when cutting unless the work is within the range of the regulator. It is dangerous to use oil, grease, or any organic material for lubrication. If it becomes necessary to lubricate the pressure-regulating screw, or to repack a needle valve, a little glycerine should be used. The pressure-adjusting screw should always be released after ceasing work and before changing from one cylinder to another. Creeping regulators or regulators with leaky valve seats should be overhauled, because they are likely to become dangerous and uneconomical. The gradual increase of pressure when the blowpipe is shut off indicates a creeping regulator. Devices in the form of electric heaters, which prevent freezing of regulators and thus interference with the supply of oxygen, are now available and are compact and convenient where electric current is available. They are interposed between the cylinder and the regulator. Regulator gauges should be tested at regular intervals but never tested with an oil-testing pump.

Acetylene regulators are usually painted red and have a left-hand screw thread corresponding to that on the standard valves of acetylene cylinders. Regulators for oxygen are usually painted black and have a right-hand screw thread. Any possible confusion is obviated by this distinction. Before connecting the oxygen regulator to the cylinder open the oxygen cylinder valve for an instant and then close it.

Cylinder valves should be opened very slowly so that pressure does not suddenly come on to the gauges or diaphragm. The proper working of regulators largely depends upon the care given them. They should not be left in places where dirt or dust may get into the interior and be deposited on the seat or inlet nozzle, and they should never be repaired by inexperienced workmen.

Hydraulic Safety Valves.

These are indispensable in low-pressure and medium-pressure installations, i.e. where the acetylene is generated on the premises. Their function is to provide a water-seal and to prevent explosive mixtures entering the acetylene piping or generating plant. The valve should be chosen so that the maximum supply of gas is obtained without burning the metal. Safety

devices based on mechanical principles, such as valves, etc., may more or less prevent the return of the flame, but are not infallible in preventing explosive mixtures forming in the generating plant.

Hydraulic safety valves should be charged and examined daily, and should be installed so as to facilitate filling to the proper level. The local authority responsible for issuing the carbide licence generally insists on the provision of a metal shield for the valve, to avoid accidents in the event of the valve being fractured. In general, the valve should be charged with water whilst the acetylene tap admitting gas to the valve is open. The design should be such that no water is carried over with the gas. An investigation has shown that the water vapour contained in the gas after passing through the hydraulic valve rises with the temperature, and can attain 7 per cent. when the gas is at 97° F. With 2·5 per cent. of water vapour, there is a reduction of over 8 per cent. in the speed of welding. No safety valve should have more than one outlet to which a blowpipe can be connected.

Pressure relief valves or safety valves used on pressure generators should be tested periodically; they do not replace the hydraulic safety valve, which is still required in a modified form.

Piping and Flexible Tubes.

The acetylene piping should be large enough to avoid loss of pressure. The size of piping should be dependent upon its length and the maximum supply of gas required. All piping should, when possible and convenient, be run overhead. When piping has to be run underground it should be placed in properly ventilated trenches. Piping and connections should be installed by workmen experienced in the installation of acetylene apparatus. Piping should be so arranged that any moisture will drain back to the generator, and where low points cannot be avoided these must be drained through drip cups closed with screw caps or plugs. No pet cocks should be used. A main gas cock should be arranged where it can be conveniently reached and near the point of entrance, so as to allow for shutting off the gas supply from any workshop. Gas cocks should be provided on the branch tube leading to the hydraulic valve so that a valve may be removed without discharging the line. All cocks, valves, unions, and joints should be full bore, and the installation should be thoroughly tested when the system is completed, and at regular intervals. It should not show a loss of pressure exceeding two inches of mercury within twelve hours when subjected to a pressure equal to a column of fifteen inches of mercury. Acetylene piping should be painted a distinctive colour. Galvanised iron pipe with welded or threaded connections should be used for piping acetylene. The pipes should be of uniform diameter, and bends—the number of which should be as small as possible—should be used in preference to square elbows. Compressed oxygen should never be used for testing or cleaning acetylene piping. If compressed air is used for this purpose the acetylene plant should be disconnected.

The following table will serve as a guide in installing acetylene piping for welding purposes. Manufacturers of welding equipment will supply full particulars as to pipe sizes suitable for any particular layout.

Acetylene Main Piping.

Approximate Diameters of Service Piping. For working pressure of 6 *in. to* 10 *in. water column.*

Consumption of Acetylene per hour.	Length of Pipe.	Internal Diam. of Piping.
Cubic feet.	Feet.	Inches.
15	100	$\frac{1}{2}$
40	100	$\frac{3}{4}$
80	100	1
140	100	$1\frac{1}{4}$
225	100	$1\frac{1}{2}$
400	200	2
1000	200	3
2500	200	4

Ordinary black steel piping may be used for low-pressure oxygen, and extra heavy steel pipe for high-pressure lines. The suitability of steel piping for this purpose is being investigated, and some manufacturers recommend copper piping from the point of view of safety and convenience. It is advisable for prospective users of oxygen pipe lines, headers, and manifolds to purchase them from, and have them installed by, reliable manufacturers familiar with the correct shop practice with reference to their construction and installation. No oil or grease must be used for the connections; litharge and glycerine should be used. The diameter of oxygen piping will depend on the length of the lines. A $\frac{3}{4}$-in. pipe is suitable for supplying twenty welding stations, provided the length of the line is not more than 300 ft. and the maximum consumption is not more than 500 cu. ft. per hour.

The flexible tubes conveying the gases to the blowpipe should not be too long, too flexible, or too rigid. The strength should be such that for welding purposes the tube should withstand a pressure of 100 lbs. per sq. in. without bursting, swelling, or leaking, and for use in cutting operations it should withstand 300 lbs. per sq. in. Armoured or wired hose is not necessary or recommended for ordinary welding and cutting operations. Three-ply tubing with a core of pure rubber is satisfactory for ordinary work, and for welding and cutting in confined spaces fire-resisting hose should be used. Avoid greasing the ends of the tube to facilitate fixing to the blowpipe or other connectors; use water. No white lead, oil or grease, or other pipe-fitting compounds, should be used for making tight joints. Tube clips should be used in preference to wire, etc., for securing the tube firmly to the connectors.

Flexible tubing should be examined regularly, and defective tubing should never be used. Avoid tangling or kinking the hose, long lengths—about 10 ft. lengths are suitable for stationary equipment and 15 ft. for portable—dragging the hose along the floor, and exposing it where it is likely to get damaged. Special care should be taken to avoid the interchange of the oxygen and acetylene tubes when connecting up to the blowpipe. Both acetylene and oxygen tubing should be blown through occasionally, so that dirt and dust will not get carried into the blowpipe, and tubing should be examined and tested periodically.

Welding Blowpipes.

Welding blowpipes are classified in relation to the pressure of the acetylene. Low-pressure blowpipes are those fitted with an injector to aspirate the acetylene, which is supplied from a generator under a pressure of 6 to 12 ins. of water. Medium-pressure blowpipes, sometimes called positive pressure, use acetylene generated from a pressure generator at a pressure up to 9 lbs. per sq. in. An injector or partial injector forms part of the blowpipe. High-pressure blowpipes use the acetylene supplied from acetylene cylinders, and are designed to mix acetylene and oxygen in equal volumes. No injector is required, and a change of nozzle with a corresponding adjustment of pressures is all that is required when changing over from welding one thickness of material to another.

In the low-pressure system thorough mixing of the gases in the blowpipe is very necessary. To obtain this, well-designed blowpipes carefully adjusted to the oxygen pressure recommended by the maker, are essential. A typical low-pressure blowpipe working on the injector principle is shown in fig. 4.

The size or power of blowpipe which should be used depends upon the thickness to be welded. By power is meant the amount of acetylene consumed per hour. Too small a power produces the defects of adhesion, inclusion of oxide, overheating, and burning of the welds. Too large a power produces a molten bath which is much too large, holes, and adhesion in the lower part of the weld.

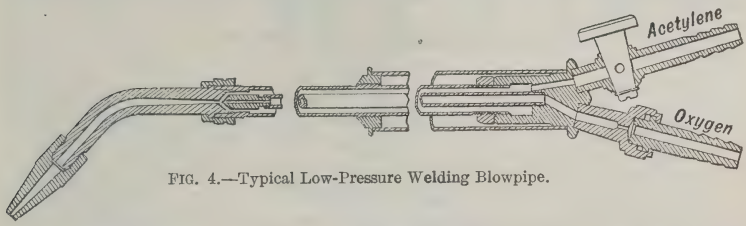


Fig. 4.—Typical Low-Pressure Welding Blowpipe.

The figure denoting the power of blowpipes is usually supplied by the maker, and is sometimes marked on the blowpipe. In welding materials other than wrought iron and steel the power of blowpipe to be used is dependent upon the material, its dimensions, and whether the job has been preheated; as a general rule the power of blowpipe for a given thickness of material is greater, as in melting, for example, cast iron, copper, or aluminium, more heat is required to melt a given weight of the metal, and, in addition, the high-temperature part of the flame is held farther from the molten metal.

All blowpipes should be carefully handled and treated as precision tools. The procedure to be followed in lighting and extinguishing the blowpipes varies with the design and type of blowpipe. The manufacturer's instructions should be closely followed. Lighting devices are convenient and safer than matches. A blowpipe should never be put down, unless a blowpipe stand is provided, until the gases are turned off, and should never be hung from regulators or other equipment. Gas economisers which serve the purpose of a stand and automatically cuts off the gases, except sufficient acetylene for a pilot jet, are valuable, especially in repetition work. Blowpipes in which the acetylene valve is entirely independent of a single valve which controls both gases, thus obviating the adjustment of the flame every time the blowpipe is lit, can be obtained. Carefully avoid the use of grease or oil, and never dismantle a blowpipe with a view to overhauling, as this may lead to irregular working after assembling. In the event of obstruction of the nozzle, a soft brass wire should be used for cleaning. Dust in the interior of the blowpipe can be

removed by blowing oxygen through from the nozzle end. Equipment should be inspected at frequent intervals by a competent operator, and only standard parts should be used in repairs to apparatus.

In the case of back-firing or flash-back due to overheating, the tip should be cooled in water, taking care to close the acetylene valve and to leave the oxygen valve slightly open. By back-fire is meant the momentary return of the flame into the blowpipe tip, which may relight immediately upon withdrawing the blowpipe away from the work or necessitate the reignition of the gases by means of a lighter. By flash-back is meant a sustained back-fire, the flame burning in the mixing chamber with a squealing sound accompanied by a smoky, sharp-pointed flame, necessitating the immediate cutting off of the gas supply, to prevent severe heating and possible destruction of the blowpipe head, and the subsequent cleaning to remove carbon deposits. If internal ignition occurs frequently the blowpipe should be sent for repair. An iron or steel wire should never be used for cleaning the nozzle.

Back-firing and flash-backs, which are a common cause of defective and expensive welds, may be due to one of the following causes:—

(*a*) Faulty design and manipulation of the blowpipe.
(*b*) Enlarging the nozzle orifice when removing obstructions.
(*c*) Working in confined spaces or with an accumulation of slag on the nozzle.
(*d*) Necessity for overhauling and internal cleaning.
(*e*) Restriction and fluctuation in the acetylene supply due to bad lay-out, exhausted purifier, etc.
(*f*) Defective hydraulic safety valves leading to air and moisture being carried with the gas.
(*g*) Defective oxygen regulators.
(*h*) Presence of solid matter in the gas either from the acetylene generator or inferior flexible tubing.

Oxy-Acetylene Flame Characteristics.

The combustion of acetylene in oxygen produces a flame which apparently consists of two parts —an inner cone, incandescent, clear, and well defined; and a non-luminous outer envelope which may be transparent or more or less opaque according to the purity of the acetylene. In a well-regulated and correctly designed blowpipe there is, between the inner white cone and outer envelope, a reducing zone which is distinctly visible when the acetylene is impure. The exact character of the flame will depend on the proportion of the oxygen and acetylene as they emerge from the nozzle. The oxygen and acetylene should mix in approximately equal proportions. Gas-ratio, or the ratio of the oxygen to the acetylene consumption, is an important factor in determining the quality and cost of welds. Varying proportions of the gases produce three characteristic types of flames, commonly known as carbonising, neutral or normal, and oxidising. These

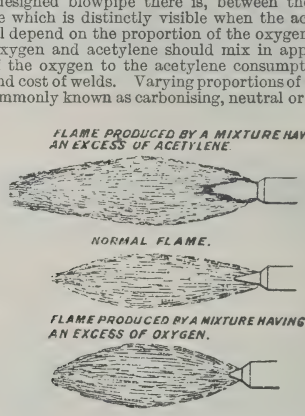


Fig. 5.—Flame Characteristics.

are shown in fig. 5. It is important that in practically all welding operations the flame should be neutral—that is, there should be no excess of oxygen or acetylene. Defective welds are produced with carbonising and oxidising flames. It is particularly important that there should be no excess of oxygen, which not only tends to produce weak welds, but also to blow the metal forward and produce adhesion. Regulation of the flame during important welding operations is essential, as very few blowpipes maintain a neutral flame under welding conditions, and further, the operator has a tendency to adjust the flame oxidising. It is advisable to adjust the flame whilst wearing dark glass goggles.

It is necessary to emphasise that the white cone of the flame should never be brought into contact with the molten metal, and that its distance from the molten metal depends, in general, upon the metal being welded and its thickness.

FERROX

WIRES *for* FINEST WELDS

MADE ONLY FOR WELDING

The raw material which forms the starting-point of FERROX manufacture is of at least as high quality as the best of other brands. EVERY COIL as it enters the factory is CHECKED BY ANALYSIS before release for treatment under the unique FERROX process, whereby the metal is purified, softened and impregnated with elements which impart to the welding process and the weld those qualities which have made FERROX WIRE stand for FINE WELDING. FEED RODS and FLUXES for welding of grey cast iron, aluminium and other non-ferrous metals and alloys.

THE FERROX COMPANY, LTD.

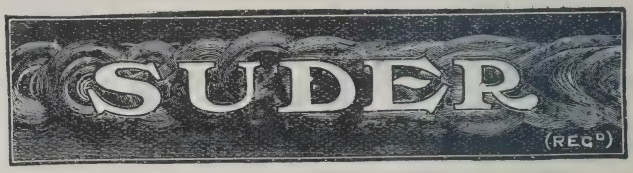
12 GROSVENOR GARDENS, LONDON, S.W.1.

'PHONE: Sloane 6101 (5 lines).

Samples and literature *gladly* sent free on request.

TELEGRAMS: Ferroxicon, Phone, London.

F. 2768



BRAND

Fluxes, Wires and Rods

for

OXY-ACETYLENE WELDING

Flux Coated Electrodes

for

ELECTRIC ARC WELDING

Specialities: "T" CAST IRON WELDING RODS (Patented)

ALUMINIUM FLUX (Patented)

Fluxes, Wires, Rods and Electrodes for all metals.

ACETYLENE PURIFYING MATERIALS.

"SUDER" Fluxes and other supplies are manufactured under the control of our Chemists and Metallurgical Engineers.

Price List and advice gratis on Welding Problems on application to:—

THE DISTRICT CHEMICAL CO., LTD.

106 FENCHURCH STREET, LONDON, E.C. 3

MONUMENT 3874. NATRON, LONDON.

F. 2769

Goggles and Protective Equipment.

For welding with acetylene it is necessary to protect the eyes with goggles fitted with suitable coloured glasses. There are a great many different kinds of special safety glasses on the market and many combinations of ordinary coloured glass are in common use. The eyes must be protected from injurious rays, the heat and glare from the work, and from particles of hot metal that fly up from the weld. To obtain the clearest definition or visibility with the least amount of glare, the selection of the colour tint in safety glasses should properly be decided by an expert; the depth of the tint may be determined best by the operator himself, owing to the individual difference which will permit one man to see clearly through a glass that would be too dark for another man. Amber, dark-amber, or dark-amber-green glasses give efficient protection. Glasses showing blue or violet tints should be avoided, except in certain combinations where they are used to obscure other colours. For the frames a tough heat-resisting material like vulcanised fibre is advantageous. Goggles should be light, comfortable, and well ventilated, and should be obtained from the manufacturer of welding supplies.

Welders employed on heavy work should be supplied with fireproof gauntlets and aprons, as a protection against radiated heat, and where welding is being performed in the interior of some structure or confined space all the outer clothing should be fireproof.

Filling Materials.

The necessity for employing the best materials for every welding operation cannot be too strongly emphasised. It is only by using the most suitable filling-in wires, rods, and fluxes, specially manufactured and controlled for each particular metal or alloy, that the most satisfactory work can be done.

Thin material is frequently welded by flanging, bringing the edges into contact and melting them together, but generally a filling material or ' welding rod ' is melted into the groove or crack.

Filling materials must, where possible, not only be of the correct chemical requirements to (*a*) replace loss in the molten zone, (*b*) counteract the tendency of the metal to oxidise, (*c*) assist in making the molten metal flow readily, and (*d*) to break down oxides when formed, but they should also act satisfactorily under the blowpipe. As a general rule, the welding wire or rod should have practically the same composition as the material to be welded.

The choice of the right size of welding wire or rod is important, and melting of both should take place in the molten bath and not in the flame. The following table gives the sizes of wire for ' forward ' welding sheet iron and steel. In the case of other metals the figures hold good, except for very thin material, when a size of not less than $\frac{1}{16}$ in. should be used.

Sizes of Welding Wire for Iron and Steel (' Forward ' Welding).

Thickness of Material. No.	Size of Welding Wire.		
	S.W.G.	Decimal Equivalent.	Nearest $\frac{1}{64}$.
20 B.G.	19	0·040	3
14 B.G.	14	0·081	5
10 B.G.			
8 B.G.	11	0·116	8
6 B.G.			
4 B.G.	8	0·160	10
3 B.G.	6 to 4	0·192	12
1 B.G.		to 0·232	15
1/0 B.G.	4 to 2	0·232	15
5/0 B.G.		0·276	18

The following table also gives the sizes of wire for ' backward ' welding; it should be pointed out that this method of welding is more reliable and economical than ' forward ' welding, and is rapidly displacing the latter for welding iron and steel of medium and heavy thicknesses.

SIZES OF WELDING WIRE FOR IRON AND STEEL ('BACKWARD' WELDING).

Thickness of Material.	Size of Welding Wire.		
	S.W.G.	Decimal Equivalent.	Nearest $\frac{1}{64}$.
Up to 6 B.G.	11	0·116	8
6 B.G. to 2 B.G.	8	0·160	10
1/0 B.G.	6	0·192	12
1/0 to 6/0 B.G.	4	0·232	15

Fluxes.

It is not always possible to include deoxidisers and special elements in the welding rod itself. By using a combination of chemicals in the form of a powder, liquid, or paste, known as a flux. the difficulties of producing sound and reliable welds are frequently overcome. A flux is used for (*a*) cleaning the parts to be welded; (*b*) eliminating and preventing the formation of oxide; (*c*) replacing elements burnt out by the flame; (*d*) facilitating the flow of the metal, and, when melted, (*e*) protecting the surface of the molten metal from the action of the flame and preventing blow-holes. The reducing zone of a well-regulated and well-manipulated flame protects, in a measure, the molten metal from oxidation. A flux is necessary in welding steel (except in cases where the steel welding wire contains deoxidising materials, as, for example, Ferrox), cast iron, copper, aluminium. etc.; for successful welding the proper flux should be used, and as the welding rod and flux are important factors in welding they should only be obtained from reliable manufacturers.

Preparation of the Work.

One of the first principles of sound welding is securing complete fusion and perfect union of the filling material and the edges of the parts to be welded. To ensure this, good preparation is necessary, which includes, as a rule, the cleaning of the edges and the welding wire, the bevelling or chamfering of the edges, the adjustment and maintenance of the parts in position, and allowance for expansion and contraction effects. In general, the preparation of the parts varies according to the thickness and nature of the metal and with the organisation and equipment of the workshop.

Cleaning the parts to be welded and the welding wire or rod is essential. Bevelling is necessary for materials $\frac{1}{8}$ in. and above in thickness. Bevelling ensures good penetration, increase in ductility and strength, economy, and reduction in the overheating effects. The angle of the bevel depends upon the thickness of material and whether 'forward' or 'backward' welding is used. All metals from $\frac{1}{8}$ in. up to $\frac{3}{8}$ in., and in some cases $\frac{1}{2}$ in., should be bevelled from one side only. Pieces exceeding $\frac{1}{2}$ in. may be bevelled from both sides. In forward welding the 'V' formed by the two bevelled plates should be about 90° for material exceeding $\frac{1}{4}$ in. in thickness, and 50° to 60° for material $\frac{1}{8}$ to $\frac{1}{4}$ in. In backward welding for material up to $\frac{1}{4}$ in. an angle of about 45°, and above $\frac{1}{4}$ in. an angle of 60°, can be used. The bevel should extend to the bottom of the plate, and thorough penetration and complete fusion should be obtained. In repair work preparation of the work is a very important feature and will save considerable time and keep the cost of welding low. Should it be necessary to re-weld a joint previously welded, it is essential that the weld metal be entirely removed, and on old work the same instruction applies, and corroded metal must be removed. In the case of aluminium repairs, for example, good judgment, as shown by the preparation, means success, and bad judgment a ruined casting.

Expansion and Contraction.

It is impossible to produce satisfactory work unless expansion and contraction effects are properly allowed for. Warpage and residual stresses, followed by failure, are the common result of welders failing to master this subject. The methods commonly employed are, in the case of thin sheets, tacking; in the case of heavier material and castings, setting the plates so as to form an angle, heating the entire work, partial heating, heating restraining members, breaking a member so that expansion and contraction can take place freely, bending or springing the material, etc.

If expansion can take place in all directions, it will give the welder no trouble, but he should operate so as to leave practically no residual stresses. If, however, welding takes place at a place that is confined by various parts or by the particular construction of a piece, it is necessary to give it due consideration.

The effects of contraction must be considered equally with those of expansion, as failures, cracks, etc., can often be traced to neglect of this property.

Preheating.

This may be partial or total, depending upon the work in hand, and is mainly employed with a view to (*a*) allowing for expansion and contraction, (*b*) decreasing the cost of welding, and (*c*) producing a more homogeneous joint. Preheating is usually carried out with the welding blowpipe, oil or gas burners, charcoal, or other suitable fuel. Where fuels such as charcoal are used for preheating, the combustion gases should, where possible, be carried away. In any case ventilation should be provided to avoid the harmful effects of the gases produced by combustion. The method used should be governed by the particular work in hand.

No general rules can be given for preheating temperatures, as the degree of heat necessary will, of course, depend upon the size and shape of the object to be welded. As a general rule, however, cast iron should be preheated to a dull red heat for small and simple castings, and to a blood-red heat for large or complex castings; aluminium and its alloys should be preheated very carefully, as too high a temperature will lead to the collapse or distortion of the article being welded.

Preheating machines burning paraffin and giving a flame of high calorific value, smokeless and transparent, give satisfactory results where portability and a powerful flame are required.

After-Treatment of Welds.

Improvement in the structure of welds with a consequent increase in strength and ductility is the result of proper after-treatment of welds. Thermal or mechanical treatment, or both, carried out in a correct manner is invariably beneficial, except in the case of defective welds.

Strength of Welds.

With proper equipment and suitable filling materials, the strength of a weld will depend mainly upon the care and skill of the operator. A good operator should be able to obtain welds 95 per cent. the strength of the original material on material of good weldability; for the average welder 85 per cent. should be taken as the lower limit of strength.

Recognition of Materials.

The identification of the metal, especially in repair work, is invariably indispensable. Once the metal or alloy is known the next step is to get familiar with all the facts relating to its weldability and, in case of doubt, proceed to the necessary tests for determining these. Some of the means by which the materials of the parts to be welded may be recognised are as follows: colouring, appearance of surface, contour, grinding spark test, chisel cut, fracture, blowpipe test.

Steel Welding.

The welding of wrought iron and mild steel is apparently simple; in reality it is somewhat difficult to produce strong and reliable welds. Oxidation and inclusions are the most serious defects, and must be reduced to a minimum. Welding wires containing a small percentage of vanadium or nickel are used with highly satisfactory results. To produce the minimum oxidation the blowpipe must be well designed and handled efficiently, and the parts to be welded must be thoroughly cleaned. Expansion and contraction must receive proper consideration. The use of a flux in the form of a powder has not proved satisfactory. It has been found that deoxidising materials in the form of a liquid gives excellent results. The product known as 'Suder' is specially prepared for use with a welding wire which does not contain deoxidising elements. At the high temperature of the flame it slags the oxide, keeps the surfaces clean, and prevents the inclusion of oxide. The weld can be carried out rapidly, thus reducing overheating effects. The liquid is applied by means of a brush to the edges of the plates and the welding wire previous to welding. The welding wire should be a material low in phosphorus and sulphur, and free from oxide. The use of ordinary iron or mild steel is unsatisfactory, and welding wire, such as 'Suder' or 'Ferrox,' made especially for the oxy-acetylene process, should be insisted upon. It will be found that economy is on the side of the controlled, but slightly more costly, and dependable product.

The following abstract from a specification for a welding wire for iron and mild steels is of interest and represents the American Welding Society's Specification for the welding wire for oxy-acetylene welding mild steel, structural work, plates, bars, low carbon steel forgings and castings. The surface of the wire is to be smooth, of uniform homogeneous structure, free from irregularities in surface hardness, segregation, oxides, pipe, seams, etc. Diameter shall not vary more than

+ 3 per cent. of the diameter specified. The chemical composition to be within the following limits :—

Carbon	not more than	0·06 per cent.
Silicon	,,	0·08 ,,
Manganese	,,	0·15 ,,
Sulphur	,,	0·04 ,,
Phosphorus	,,	0·04 ,,

The surface shall be smooth and free from scale, rust, oil, or grease. In the hands of an experienced welder the wire shall demonstrate good weldability, and shall flow smoothly and evenly without any unusual characteristics.

For welding mild steel and to produce a weld of greater strength but of lower ductility than that obtained with ordinary welding wire, the wire may contain 0·15 to 0·25 per cent. of carbon, 3·25 to 3·75 per cent. of nickel, and 0·5 to 0·8 per cent. of manganese.

A simple test to indicate the fitness of welding wire is made by laying the wire flat on a clean surface and applying a good neutral oxy-acetylene flame to the wire in such a manner as to heat it up for a distance of about 3 or 4 ins., playing the flame backwards and forwards until the wire is red and then slowly melting the wire, moving the flame so that the wire only melts halfway through its diameter. If the flame is withdrawn as soon as the metal begins to melt, the impurities can readily be seen being thrown off in the form of sparks, or a boiling action in the case of inferior metal ; and in some cases the metal will not flow together but ' runs away.' The metal from gas absorbing wires will, when cold, contain numerous spongy, volcano-like irregularities. In the case of clean metal free from impurities the wire will melt and flow evenly without any disturbing action.

The following table is useful when ordering welding wire :

Size. Inches	$\frac{1}{16}$	$\frac{3}{32}$	$\frac{1}{8}$	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$
Lbs. per 100 ft. .	1·04	2·3	4·1	6·5	9·3	16·7	26	37·5
Ft. per 100 lbs. .	9,615	4,273	2,403	1,537	1,067	600	383	266

The flame must be carefully regulated and held so that the tip of the white cone is about $\frac{1}{8}$ in. from the surface of the molten metal.

A good practical rule in welding steel is that the hourly consumption of acetylene should be $4\frac{1}{4}$ to $5\frac{1}{2}$ cu. ft. (120 to 160 litres) for every $\frac{1}{16}$ in. in thickness.

The following useful table is based upon these figures, in which the approximate bore of the nozzle orifice is also included :—

POWER OF BLOWPIPE FOR IRON AND STEEL.

Thickness of Material in Inches.	Consumption of Acetylene per Hour.		Bore of Nozzle in $\frac{1}{64}$ inch.
	In Litres.	In Cubic Feet.	
22 gauge	60–80	$2\frac{1}{2}$–$2\frac{3}{4}$	2
$\frac{1}{16}$	120–160	$4\frac{1}{2}$–$5\frac{1}{2}$	3
$\frac{1}{8}$	240–320	$8\frac{1}{2}$–11	3 full
$\frac{3}{16}$	360–480	$12\frac{3}{4}$–$16\frac{1}{2}$	4
$\frac{1}{4}$	480–640	17–22	5
$\frac{3}{8}$	720–960	$25\frac{1}{2}$–33	6
$\frac{1}{2}$	960–1280	34–44	7
$\frac{5}{8}$	1200–1600	$42\frac{1}{2}$–55	8

The blowpipe must be given a motion which will cause the flame to describe a series of overlapping circles in the case of light and medium sheets, and oscillated in a series of overlapping semicircles in the case of heavy sections. This is known as ' forward ' welding. A variation of these movements has been introduced which gives more economical and better welds. The welding wire, in this case, follows the flame instead of preceding it, and the flame is directed towards the starting point instead of towards the finishing point. The blowpipe is given a slight gyratory motion, whilst the welding rod is given a more or less elliptical or longitudinal motion. This method of welding is known as ' backward ' welding. Subsequent thermal treatment, which may consist of annealing or normalising, according to the particular requirements, will improve the structure of steel welds and materially increase the strength and toughness of the weld. Hammering may be resorted to with good effect, provided the hammering is not too vigorous or done at too low a temperature.

The following table gives the speed at which the welding of mild steel plates can be done under average conditions :—

Thickness of Plate in Inches.	Speed of Welding.	
	Feet per Hour.	Feet per Hour for Average Day's Work.
$\frac{3}{64}$	30 to 33	23 to 26
$\frac{5}{64}$	20 „ 21	16 „ 20
$\frac{1}{8}$	15 „ 16	13
$\frac{5}{32}$	13 „ 15	10
$\frac{13}{64}$	10 „ 12	9
$\frac{1}{4}$	9 „ 10	$7\frac{1}{2}$
$\frac{9}{32}$	8 „ 9	$6\frac{1}{2}$
$\frac{21}{64}$	7 „ $7\frac{1}{2}$	6
$\frac{23}{64}$	$5\frac{1}{2}$ „ 6	5
$\frac{15}{32}$	$4\frac{1}{2}$ „ $4\frac{1}{2}$	$3\frac{1}{4}$
$\frac{15}{32}$	$3\frac{1}{4}$ „ $3\frac{1}{2}$	$2\frac{1}{2}$ to 3
$\frac{19}{32}$	$2\frac{1}{4}$ „ $2\frac{1}{2}$	$1\frac{1}{2}$ „ 2

Owing to metallurgical changes that take place when melting medium and high carbon steel with the blowpipe, the welds obtained are invariably unsatisfactory. As a general proposition, the lower the carbon content of the steel, the more successful the welding operation. It is possible to get fair results on medium steels, but a flux, rapid welding, and thermal after-treatment, which requires skill to properly carry out, are necessary.

The data at hand show that for welding medium carbon steel, welding wires having a carbon content ranging from 0·06 to 0·60 per cent. are of no particular value.

For welding medium and high carbon steel and worn surfaces where great resistance to abrasive wear is desired and where machining is not necessary, a welding wire containing 0·60 to 0·75 per cent. of carbon and 0·30 to 0·90 per cent. of manganese, or a special welding wire such as chrome-vanadium, is desirable.

Alloy steels, such as chrome and manganese steel, are weldable, provided the percentage of carbon does not bring them into the range of medium and hard steels, which is frequently the case. The welding wire should be of the same character as the steel, and a flux must be used. In the case of nickel steel it is necessary to use a flux which will form a varnish over the molten metal and prevent the absorption of the gases from the flame of the blowpipe. It is important to recognise that alloy steels are used largely because of the excellent physical properties that are developed by suitable heat treatment, and that the uneven heat treatment produced by welding temperatures may produce considerable variations in quality. Hence they should be welded with a full knowledge of the alloy and the results likely to be obtained. This also applies to the welding of stainless steel, which has recently been thoroughly investigated. The principal difficulty is the formation of the oxide of chromium which is dealt with by the use of a first-class flux. The method of applying the flux is an essential factor for success, and consists in coating the welding wire or rod and the underside edges of the sheets to be joined. To obtain an adhesive coating the wire and edges are first painted with a solution of silicate of soda and then sprinkled with the flux.

Cast-Iron Welding.

The two principal defects in cast-iron welding are the production of hard spots and blowholes in the weld and subsequent cracks or fractures in the weld or adjacent metal. The oxide formed must be removed or destroyed by the use of a flux. The welding rod must be of good material and specially made for cast-iron welding. One great trouble with cast-iron rods in general is that the quality is by no means uniform. This is especially true with the cheaper and uncontrolled products. Rods made from ordinary cast iron produce blowholes and hard spots. Special rods containing titanium, producing sound, soft, and reliable welds, are obtainable. Expansion and contraction should be considered as of great importance. Preheating should be used to some extent in all cast-iron welding. It is advisable to weld in position after preheating. The welder is protected by making use of asbestos sheet or simply plate. Specially designed blowpipes of considerable length are available for this class of work, and two or three welding rods are welded together. The parts must be thoroughly cleaned, and in the case of cracks their extension must be prevented by drilling holes a short distance from the ends in the direction the crack would follow. The weld should be arranged horizontally, and where this cannot be arranged carbon blocks should be used to prevent the metal flowing away. Carbon products, in the form of blocks, powder, or paste, play an important part in modern welding operations. They are far superior to fireclay or fire-bricks, which are liable to melt under the blowpipe and invariably cause blowholes. Carbon compound, in powder form, which can be made into a plastic mass with water, is valuable for making moulds, dams, etc., and for lining up repairs and holding parts in position. To protect machined surfaces during preheating operations, they should be coated with flaked graphite and oil. This can be made into a paste and painted on, or the surfaces can be oiled and the

graphite dusted on. The graphite must be coarse, as fine flake is not satisfactory. The flame must be held so that the tip of the white cone is between ¼ and ⅜ in. from the molten metal, the distance depending on the thickness of the metal. The after-treatment consists in annealing or allowing the metal to cool very slowly. In well-executed welds on cast iron the quality of the weld is generally superior to that of the metal of which the casting is made. The grain is finer and the welded zone is sound, homogeneous, and can be easily worked.

Certain cast irons of great strength and in which the carbon is in the combined state are difficult to weld. Cast iron which has been submitted to the heat of a fire for a considerable time is frequently unweldable. The metal in both cases melts under the blowpipe in a distinctly different manner from ordinary cast iron, and the welder should have no difficulty in recognising that the blowpipe should not be applied to make a joint. The bronze 'welding' of cast iron, iron, steel, etc., is a process in which the oxy-acetylene flame is used in conjunction with a copper-zinc alloy, such as 'Suder Bronze,' 'Sifbronze,' or manganese-bronze, to produce joints which would be difficult or expensive to obtain by the usual method of melting the metal of the casting. The procedure in making joints is very much the same as in actual welding. The 'V' opening need only be about 15°. The heat must be just right—too much will burn the bronze, too little will prevent any alloying—and the surfaces must be clean. It has been found in actual operations that the welding time is greatly reduced when the parts are prepared by sand-blasting. Extensive researches have been made on the bronze welding of cast-iron pipes.

Malleable Cast Iron.

It is difficult to produce satisfactory joints on malleable cast iron by welding. The welds are usually hard, brittle, and porous. The best results are obtained by brazing with the oxy-acetylene blowpipe, using the same methods of preparation as for cast iron, and using 'Suder' or 'Sifbronze' as a filling material. The two pieces to be joined are brought to a point just below fusion, great care being taken that they do not become fused. When the edges are at a right temperature, the rod of manganese bronze is fused into the groove with the aid of a good flux.

The introduction of acetylene for brazing purposes provides a fuel which has the highest known flame temperature and high calorific value. By using dissolved acetylene with a special brazing blowpipe no air-bellows or oxygen is necessary, as the blowpipe is atmospheric and a clean and relatively small flame of suitable power is obtained.

Aluminium Welding.

Sheet aluminium welding can be handled in very much the same way as steel as regards preparation and allowance for expansion and contraction. The principal difficulty lies in the formation of the oxide, which must be dealt with by an energetic and reliable flux. The filling material is pure aluminium wire; before starting to weld, the wire and parts to be welded must be thoroughly cleaned by scraping. A delicate touch and considerable practice are necessary, especially on thin material. Material under 22 gauge is best welded after flanging the edges. The flanged edges should practically have no radius, so as to leave the minimum of space between the underside of the sheets. The depth of the flange should be about two and a half times the thickness of the sheet. The white cone of the flame should be level with the surface of the plates and the blowpipe held so that the flame is inclined to the horizontal at about thirty-five degrees. If the methods are correct the upper and lower surfaces of the weld appear identical. The power of the blowpipe to be used is not in direct ratio to the thickness of the metal as in steel welding. For example, on 16 to 19 gauge material a blowpipe consuming 2 cu. ft. of acetylene per hour would be satisfactory, whilst for material ⅛ in. thick 10 cu. ft. is required, and for ¼-in. material 40 cu. ft. per hour. The flame must be carefully regulated, and the tip of the white cone should be from ¼ to ⅜ in. from the molten metal. After welding, the welds should be thoroughly washed with warm water and, where possible, internal strains should be removed by annealing. The structure and strength of the metal in the welded zone are improved by hammering when cold. Subsequent corrosion of the welded seams may be due to the physical condition of the aluminium sheet and the welds, and in the case of important work preliminary corrosion tests should be made on welded strips cut from the sheets.

Pure aluminium is not usually employed for castings subject to stress. In order to raise the tensile strength and increase the hardness varying percentages of zinc, copper, or nickel are added. There is an endless variety of aluminium alloys, but from a welding point of view they fall into two classes, viz., those containing zinc as their principal hardening medium and those containing copper. The material expands greatly and is very weak at high temperatures, consequently contraction strains are likely to produce cracks and distortion unless the work is properly handled. In welding castings—crank cases, for example—it is often necessary to preheat them either completely or partially. The preheating must be done carefully, otherwise the casting may collapse. For supporting the casting evenly and lining up, carbon putty is useful. For missing parts, and where the welder is not well skilled in this class of work, it is advisable to back up with sheet copper or galvanised iron. In rigging up jobs for repairs, care must be taken that expansion and contraction effects are not interfered with. The parts should be carefully cleaned and free from oil and grease. The repair of oil saturated castings is frequently one of considerable difficulty. Cleaning, by dipping in caustic soda, followed by washing in scalding hot water,

THREE
CELEBRATED
PAPERS

"THE ENGINEER"
(*Established 1856*)

"THE IRONMONGER"
(*Established 1859*)

"THE CHEMIST & DRUGGIST"
(*Established 1859*)

Proprietors:—

MORGAN BROTHERS (PUBLISHERS) LTD.
28 ESSEX STREET STRAND
LONDON W.C.2

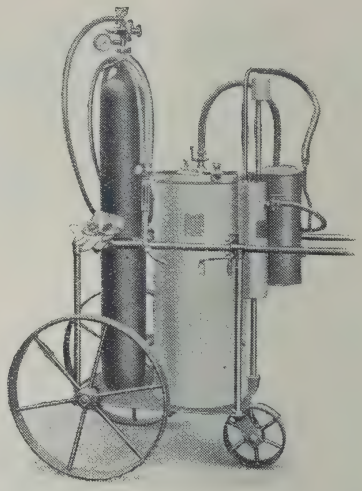
Cutting and Welding at Low Cost

THE cheapest way of obtaining Acetylene is to use a Carbic Low Pressure Plant and Produce your own from Carbic Cakes, a specially prepared and compressed form of Carbide.

CARBIC Cakes yield a guaranteed volume of gas of exceptional purity, and entirely without waste.

CARBIC Generators are portable, low in first cost, economical in working, and highly efficient at the weld. Being automatic in action, they need no attention while in operation, and maintain a regular supply of cool, pure gas at unvarying pressure.

WRITE for your complimentary copy of the NEW Carbic Cutting and Welding Book. Particulars of Carbic Flare Lights and Hand Lamps will also gladly be sent upon request.



Carbic Limited,

Specialists in Acetylene

(Dept. E.Y.),

51 Holborn Viaduct,

London, E.C.1.

Branch Offices and Stores at:— Cardiff, Leeds, Liverpool, Manchester, Newcastle-on-Tyne, Southampton, Glasgow and Dublin

F. 2775

and drying, bevelling to a larger angle than usual and sweating out the oil and scraping it away, indicate the methods which lead to successful repairs. A good grade of filling material of average composition is necessary; pure aluminium wire should not be used. The acetylene should be pure and the blowpipe should give a soft, non-oxidising flame. Two methods of welding are adopted, one known as the flux and the other as the puddle method. In the former a flux is used in the ordinary way and in the latter no flux is used, a steel rod and paddle being used to remove or break down the oxide and to smooth the weld respectively. All traces of flux must be washed off. The casting should be reheated slightly to remove any local strains and then allowed to cool very slowly. For finishing off a single cut-file or a 'dreadnought' file is most suitable.

The welding of the well-known aluminium alloy known as 'Duralumin,' which contains copper, manganese, magnesium, etc., is satisfactory or not according to the mechanical properties required in the joint. The alloy has important properties which are reduced, and in some cases almost eliminated, by welding operations. The metal should not be welded when the joints are subject to bending action, shocks, or when the full properties of the metal are desired.

The welding of aluminium-silicon alloys, such as Alpax, in the form of castings presents no particular difficulty. In sheet form, however, the difficulties are greater than with pure aluminium sheet, and the correct welding procedure must be followed. Bevelling for material slightly exceeding $\frac{1}{16}$ in. is advisable to obtain good penetration. A welding wire of Alpax and a good flux are essential.

Copper Welding.

Copper may be welded, but only satisfactorily where the operator is familiar with the technique. Many failures can be attributed to the commercial metal being in an unsatisfactory condition for welding. Material which may be satisfactory for brazing may prove quite unsuitable for welding. Where possible, a sample strip 2 or 3 ins. long should be tested by making a weld and applying a bending test. It is necessary to clean and bevel the edges and to support the underside of welds made on thin material, copper tubes, etc., by means of a mandrel consisting of an iron tube on which is wound or fixed asbestos, which must be perfectly free from moisture. Owing to its lack of strength when hot, the tacking of copper welds should be done carefully. For the same reason welds should not be started at the edges of the sheets, but at a point about one-third the length of the seam. The power of the flame should be practically the same as that used for welding steel of the same thickness. The flame must be carefully regulated. The melting should not be too rapid or too slow. Without a correct welding rod, which contains a small quantity of phosphorus, it is impossible to satisfactorily weld copper. A flux is useful. Preheating to avoid adhesion is necessary. The tip of the white cone should be about $\frac{1}{4}$ in. from the surface of the molten metal. The welds should be hammered at a dull-red heat to improve their ductility and strength, and afterwards annealed. The technique for the repair of copper fireboxes and tube plates has been worked out and applied with great success. A good sense of fusion, the addition of the metal in thin layers, and satisfactory equipment are essential for success. Welds can be obtained with mechanical properties almost identical with those of the metal welded. Such welds can be planished, stamped, spun and, in general, submitted to the ordinary workshop operations carried out on sheet copper.

Brass and Bronze Welding.

Brasses and bronzes are only welded indifferently without the correct welding wire or rods. The filling material should be practically of the same composition as the metal to be welded, which, in the case of brass, should contain a little aluminium, and in the case of bronze a little phosphorus and traces of aluminium. Bronze filling materials are cast. Fluxes should be used in both cases, and the power of the flame is less than that used for the same thickness of copper—that is, distinctly less than for steel plates of the same thickness. The regulation of the flame is very important: a slightly oxidising flame will give a surface free from wrinkles. The position of the flame is the same as for copper. The after-treatment of the welds consists in annealing at definite temperatures. Identification of the particular brass or bronze to be welded is important in view of the importance of choosing the correct welding wire or rod, and the subsequent heat-treatment which is necessary. Reclaiming castings and repairing worn or broken parts, such as gear wheels, bearings, etc., constitute the general run of the work on bronzes. In the case of castings similiar methods to those outlined for the repair of cast iron are necessary.

Lead Welding.

The oxy-acetylene blowpipe is rapidly supplanting the older methods of welding lead, or 'lead burning,' as it is generally called. It has proved superior in convenience, quality of joints, and economy. A small type of blowpipe is required, consuming from $\frac{1}{2}$ to 6 cu. ft. per hour. When it is necessary to fill up a hole or to build up any part of a lead object, the work is facilitated by using a mould or a backing of steel plate. When welding patches in repair work avoid rectangular patches on vertical faces, which make two edges of the patch horizontal. The patch should be made lozenge shape, so that the welds are done on the incline. The welding wire or rod should be of the same metal as the sheet. Special outfits have been designed for lead burning. using dissolved acetylene; they are very compact and portable.

Nickel Welding.

Nickel when melted has the property of readily absorbing gases, especially oxygen. Various methods have been tried to protect the molten metal from the gases of the blowpipe and to prevent their absorption. The welds obtained are very porous, and it is extremely difficult to obtain satisfactory welds. A method which has given fair results consists in placing the two parts to be welded on a heated anvil and then heating the parts to a white heat with the blowpipe, afterwards carefully hammering the parts together. The key to the production of sound welds lies in the deposition of metal which is metallurgically sound. It has recently been proved that the welding of nickel and its commercial alloys—such as Monel metal—can be satisfactorily accomplished by the use of a flux, in which manganese acts as a desulphurising agent, or a borated acetylene flame.

Machines for Welding.

Acetylene welding machines for the manufacture of tubes have been in use for many years. The development of automatic welding machines for the manufacture of tanks, etc., is more recent. For repetition work machines have been designed which accomplish the work in less time with comparatively unskilled men. The machines consist of two parts, (*a*) holding device for the work, and (*b*) apparatus for moving the blowpipes, with or without oscillatory motion, over the seam.

Recent developments in welding machines have been mainly in the direction of the use of multi-jet blowpipes. In order to obtain speed it was recognised that more than a single jet of flame should be employed, and from two jets in tandem it has become customary to employ a large number of jets extending for a considerable distance along the line of the weld. The defects of this method have been overcome by distributing the heat on both sides of the centre line of the seam. The welded section produced in this way is wider, stronger, and more certain to penetrate than is the case with the narrow weld made by a central line of flame.

Training Welders.

An oxy-acetylene welder has been defined as a person who has the requisite manipulative skill and technical knowledge successfully to operate welding and cutting apparatus, and, at the same time, has skill in the operation of hand tools used in the preparation and finishing of the work in connection with welding, and also has a reasonable amount of ingenuity. He must also have a knowledge of accuracy commensurate with the type of work to be welded. Naturally, a trade in which so much depends on the personal equation has drawn attention to the need of training men to produce sound work. No doubt the best method is that which expounds principles followed by practice under expert supervision. Welding skill grows rapidly under a well-worked-out system of exercises which aim at developing skill in making typical welds on various metals. Testing the welds is also an important requisite of a training course. Classes for training welders are held in the principal centres, and manufacturing concerns who train their workmen must follow methods depending upon the size of shop and class of work.

A common error is to attempt work, especially repair work, in a shop where the tools and apparatus for successful all-round welding are not provided, or to endeavour to repair jobs which are beyond the capacity of the welder. It is much more economical, in such cases, to hand over work of this character to firms, such as *Barimar Ltd.*, who have a world-wide reputation for repairs of all kinds, especially large castings which frequently must be welded *in situ* and require specialised knowledge and a choice of the most suitable welding process. A firm such as this, specialising in general repairs, have the necessary equipment for carrying out and testing the work, and welders who are not mere blowpipe operators.

Testing Welds.

The importance of testing welds has, in recent years, become very great. It is well known that the quality of a weld cannot be assumed from the general appearance, finish, etc., although considerable information can be obtained from the visual inspection of welds. At the present time the usual methods of testing include tensile and impact tests on specimens welded under identical conditions, microscopic examinations of specimens, bending and hammering tests and X-ray examination. In the latter the actual welds can be examined, and flaws, blowholes, and other defects discovered. Recent investigations indicate that electrical and magnetic test for the non-destructive examination of welds will soon find commercial application. In fact, simple and portable apparatus based on the magnetic properties of iron and steel is now available for the non-destructive examination of welds on iron and steel of any thickness.

The American Bureau of Welding have drawn up Standards for Testing Welds which are intended primarily to serve as a standard uniform basis for testing and comparing sample welds as distinguished from welds on structures. The testing of the latter must necessarily be by non-destructive methods, which would therefore be, in general, quite different from those employed in examining sample welds made for the purpose of testing a welder's proficiency, comparing welding wires, or comparing welding methods. For the shop standard a bending test is recommended; for the commercial standard, a bending and tensile test on approved lines.

Miscellaneous Applications.

There are numerous applications of the oxy-acetylene process not included in the term welding, but which have a direct or indirect connection with it and are worth consideration. The oxy-acetylene flame, for example, is a convenient and simple method of applying local heat for numerous workshop operations, such as forging, straightening, riveting, expanding for shrinkage fits, heat-treatments, etc.

Brazing operations are more efficiently carried out with the oxy-acetylene flame than by the older methods. Case-hardening and surface hardening can also be efficiently and economically done with the flame.

OXY-ACETYLENE CUTTING.

Wrought iron, steel, and steel castings can be cut with the ordinary cutting blowpipe, and with a special cutting blowpipe or the ordinary cutting blowpipe operated in a special manner, cast iron can be cut.

The cutting blowpipe usually consists of a compound jet made up of one or more heating flames with a pure oxygen jet in the centre. The purity of the oxygen is of much greater importance when used for cutting than when used for welding. As the purity decreases the oxygen consumption and time increases, and the speed of cutting decreases as the following figures, which represent tests on steel plates 5 feet long by 1 inch thick, show :—

No. of Test.	Purity of Gas. Per cent.	Time in Seconds.	Oxygen Consumption in Cu. Ft.	Remarks on Cuts.
1	99·7	280	8·0	Very smooth and clean.
2	99·4	291	8·8	Smooth.
3	99·1	315	9·8	Satisfactory.
4	98·0	334	11·3	Good but rough
5	96·0	365	13·2	Not smooth.
6	95·0	412	15·4	Very irregular.
7	90·0	644	20·0	Rough, surface bad.
8	87·0	675	23·0	Very rough, surface bad.
9	85·0	696	25·8	Very rough, unsatisfactory.
10	82·5	—	28·4	Very rough & not through.

A spot on the metal is first heated to a bright red heat with the heating flame, and then the oxygen jet is turned on. It has been proved that only partial combustion takes place, the phenomenon of cutting being really a disintegration by the current of gases from the blowpipes. The oxy-hydrogen or oxy-coal-gas flame is often used for cutting in place of acetylene. Material up to 36 ins. has been cut. The same acetylene equipment is used for cutting as for welding. For heavy cuts special oxygen regulators are required. There are two kinds of cutting blowpipes, known as the 'central' and 'following jet' types. Mechanical contrivances have been devised for special forms and shapes and for heavy cutting. Automatic cutting machines using the oxy-acetylene or oxy-coal-gas blowpipe, with or without preheated cutting oxygen, have reached a high degree of perfection. These machines usually have the full range of movements found on a universal machine tool, and have a degree of accuracy such that accurate gear teeth can be cut, and any desired number of thousandths can be left for finishing.

APPROXIMATE RESULTS FOR STEEL CUTTING.

(*British Oxygen Co., Ltd.*)

Thickness of Material.	Nozzle.		Speed of Cutting.	Pressure of Oxygen at Regulator.	Gas Consumption. Cu. ft./hr.	
	Size.	Distance from Work.			Oxygen.	Acetylene.
Ins.	Ins.	Ins.	Ft./hr.	Lbs./sq. in.		
$\frac{1}{4}$	$\frac{3}{3\cdot2}$	$\frac{3}{16}$	66	20	42	16
$\frac{1}{4}$	$\frac{1}{16}$	$\frac{3}{16}$	*80*	*20*	*110*	*16½*
$\frac{1}{2}$	$\frac{1}{3\cdot2}$	$\frac{3}{16}$	58	25	50	20
$\frac{1}{2}$	$\frac{1}{16}$	$\frac{3}{16}$	*65*	*25*	*135*	*21½*
$\frac{3}{4}$	$\frac{1}{16}$	$\frac{1}{4}$	55	32	142	23
$\frac{3}{4}$	$\frac{1}{3\cdot2}$	$\frac{3}{16}$	*35*	*35*	*62*	*23*
1	$\frac{1}{16}$	$\frac{5}{16}$	48	35	150	24
$1\frac{1}{4}$	$\frac{1}{16}$	$\frac{3}{8}$	44	35	150	$24\frac{1}{2}$
$1\frac{1}{2}$	$\frac{1}{16}$	$\frac{3}{8}$	40	40	175	25
2	$\frac{1}{16}$	$\frac{1}{2}$	30	45	190	28
3	$\frac{1}{16}$	$\frac{1}{2}$	24	55	210	31
4	$\frac{1}{16}$	$\frac{1}{2}$	20	60	230	36
5	$\frac{1}{16}$	$\frac{5}{8}$	17	65	250	40
6	$\frac{1}{16}$	$\frac{5}{8}$	15	70	270	43

The above figures are based on actual cumulative results and the chief variation to be expected is the cutting speed, which will vary according to the nature of the work, experience of the operator, etc.

Figures shown in full are recommended ; figures in italics may be used if desired.

HEAVY STEEL CUTTING.

Material Thickness.	Nozzle Size.	Distance of Nozzle from Material.	Oxygen Pressure.	Approximate Speed of Cutting.
Ins.	Ins.	Ins.	Lbs./sq. in.	Ft./hr.
7	$\frac{3}{3\cdot2}$	$\frac{3}{8}$	70	13
8	$\frac{3}{3\cdot2}$	$\frac{3}{8}$	75	12
9	$\frac{3}{3\cdot2}$ or $\frac{7}{0\cdot4}$	$\frac{3}{8}$	85	11
10	$\frac{7}{0\cdot4}$	$\frac{3}{8}$	95	10
12	$\frac{7}{0\cdot4}$ or $\frac{1}{8}$	$\frac{7}{16}$	120	8
14	$\frac{1}{8}$	$\frac{7}{16}$	150	6
16	$\frac{1}{8}$	$\frac{7}{16}$	180	5

Blowpipes can be obtained for cutting steel under water. Under-water cutting is based on the fact that the water must be forced away from the heating flame and cutting jet.

Compressed air has been used for this purpose, and one type of under-water cutting blowpipe makes use of the products of combustion. The cooling action of the water necessitates using a heating flame five to seven times as powerful as that used for cutting the same thickness of material in the ordinary way. With an efficient arrangement an ordinary hand-cutter can be adapted for cutting under water. In salvage work steel plates have been cut at a depth of 40 ft. Various devices for lighting the flame under water when it becomes extinguished have been successfully introduced, being in the form of a pilot light burning continuously and forming part of the blowpipe. The British Oxygen Co. have found, with their under-water cutting blowpipe, that it is possible to cut as fast, or even faster, under water than in the dry.

Cast iron can be cut by means of the oxygen lance or by a special oxy-acetylene blowpipe. In the former case an iron tube from 6 to 12 feet long, $\frac{3}{8}$ in. external and $\frac{3}{16}$ in. internal diameter, containing three wires of 12 gauge, is supplied with oxygen at a pressure of from 70 to 150 pounds

per square inch, according to the thickness to be cut. The point at which the cut is to be started and the end of the tube are raised to a red heat by means of an oxy-acetylene blowpipe, the oxygen turned on and the heating blowpipe removed. Combustion of the cast iron and the tube proceeds rapidly; it takes 20 seconds to penetrate a piece of cast iron 4 inches thick and less than 2 minutes for material 24 inches thick. In the second method a cutting blowpipe with a flame having a distinct excess of acetylene is used, or a blowpipe in which the cutting jet is modified and not the heating flame. In the latter type a certain quantity of acetylene is introduced into the cutting jet and leads to an increase in the temperature of the oxygen.

The following is an abstract from a manufacturers' table of cutting data:—

APPROXIMATE RESULTS FOR CAST IRON CUTTING.

(British Oxygen Co., Ltd.)

Metal Thickness.	Nozzle.		Oxygen Pressure.	Length. of White Cone.	Approx. Speed of Cutting.	Gas Consumption. Cu. ft./ft. run.	
	Size.	Distance from Work.				Oxygen.	Acetylene.
Ins.	Ins.	Ins.	Lbs./sq. in.	Ins.	Ft./hr.		
Up to 1½	3/64	3/8	110	2	7	20	7·2
1½— 3	1/16	1/2	120	3	6	40	12·5
3— 5	3/32	1/2	130	3½	5	55	17·6
5— 8	3/32	5/8	160	4	4	80	23
8—11	7/64	5/8	150	4	2½	200	48
11—14	1/8	3/4	150	4	2	300	75
14—16	1/8	3/4	160	4	1½	400	105

The above pressures, speeds, and consumptions are merely given as a guide to indicate the results obtained on cast-iron cutting and they will vary according to the composition of the iron, etc.

ELECTRIC WELDING.

Electric welding processes can be placed in two distinct groups, one producing heat by the resistance of the metal being welded to the passage of the current, the other using the heat of the electric arc. They are known as Resistance Welding and Arc Welding, respectively. These two main groups have formed the basis for many processes of welding. The differences rest entirely on methods and special applications. The subdivisions are as follows:—

Arc Welding.

(1) Carbon or graphite electrode.
(2) Metallic electrode.
  (*a*) Bare electrode.
  (*b*) Coated electrode.
  (*c*) Covered electrode.
(3) Electric blowpipe.
(4) Atomic hydrogen.
(5) Shrouded arc.
(6) Gas shielded.

Resistance Welding.

(1) Butt welding.
  (*a*) Flash welding.
  (*b*) Upset welding.
  (*c*) Automatic butt or contact flash.
(2) Mash welding.
(3) Seam welding.
  (*a*) Butt seam welding.
  (*b*) Lap seam welding.
  (*c*) Bridge seam welding.
  (*d*) Flash-seam welding.
(4) Percussive welding.
  (*a*) Electro-static.
  (*b*) Electro-magnetic.
(5) Spot welding.
  (*a*) Lap spot welding.
  (*b*) Bridge welding.
  (*c*) Button welding.
  (*d*) Disc depression welding.
  (*e*) Point welding.
  (*f*) Multiple point welding.
  (*g*) Multiple electrode welding.
  (*h*) Ridge welding.
  (*i*) T-spot welding.
  (*j*) Duplex spot welding.

The many systems of welding which have developed from these main groups are generally associated with the inventor's name or company exploiting the process, such as:—

Benardos System. Thomson System. Daysohms System. A.C. Welding System.
Slavianoff System. Alloy Welding Processes. Cyc-Arc Process. Fluxode System.
Kjellberg System. Plastic-Arc System. Atomic-hydrogen Process. Electronic Tornado Process.
Zerener System. Quasi-Arc Process. Ferro-Arc Process.

ARC WELDING.

Commercial efficiency of arc welding processes is estimated by the relation that the quality of the work bears to the cost. The principal factors which decide the quality of the work are: (*a*) electrodes; (*b*) current (D.C. or A.C.); (*c*) welding equipment; and (*d*) efficiency of the welder. In assessing the cost the principal factors are: (*a*) capital expenditure; (*b*) depreciation and maintenance charges; (*c*) wages; (*d*) cost of electrodes and, (*e*) cost of electrical energy. The speed of welding enters into the question of cost provided increased speed is not obtained at the expense of welding efficiency. The absolute requisites for good work are: (*a*) low voltage; (*b*) short arc; and (*c*) constant heat per unit area in the weld. An arc welding circuit should possess the following characteristics: (1) ease of arc establishment when the work and electrode are cold; (2) freedom from undue tendency to freezing of electrode or extinguishing of arc with maintenance of short arc streams; (3) stable arc with maintenance of short arc streams; (4) limited current increase with growth of liquid globule; (5) limited increase of current at the instant and during the period the globule short circuits the arc stream; (6) the arc voltage should increase rapidly at the instant of globular detachment or on breaking a momentary short circuit in order to facilitate re-establishment of a stable arc. The subject of welding nomenclature, definitions, and symbols is receiving consideration.

Carbon Arc Welding.

The characteristic principle of this process, commonly known as the Benardos process, is that an electric arc is held between the metals to be joined, which forms one electrode of the circuit, and a carbon rod or pencil manipulated by the welder which forms the other electrode. By placing the carbon electrode in contact with the metal, thus closing the circuit, and instantly withdrawing it again, an electric arc is formed between the two electrodes and, owing to the high temperature of the arc, the metal is melted. Steel up to $\frac{3}{32}$ in. is satisfactorily melted together without using a filling material, but for greater thicknesses a 'melt bar' is melted into place by the heat of the arc. Experience has shown that the metal to be welded should be the positive electrode, and the carbon the negative electrode. If this order is reversed, a shorter arc is obtained and particles of carbon can enter the molten metal and cause brittleness.

The current used is direct, the voltage being from 60 to 90, and the amperes vary from 250 to 600, depending upon the thickness of the metal.

The use of graphite electrodes in place of hard carbon electrodes is increasing as the merits of the graphite electrode are more widely known. The chief advantages are: (*a*) breakage in handling and in operation is appreciably reduced; (*b*) breakage and cracking due to sudden changes of temperature is eliminated; (*c*) less carbon is carried into the weld; (*d*) the arc is much steadier; (*e*) the life is from two to three times as long as for an amorphous carbon electrode of the same diameter; (*f*) electrical conductivity is four times that of the carbon electrode of the same cross-section, consequently the heating caused by resistance is negligible and much higher current densities can be carried. Considerable development is being made in the art of carbon arc welding where graphite electrodes are used. The single or double graphite arc has proved useful in the welding of non-ferrous metals. Automatic carbon electrode welding machines are now extensively used owing to the increased speed and lower costs obtained, as compared with hand work.

A recent development, the results of which it is claimed will be far-reaching, is the Electronic Tornado process of automatic carbon arc welding, in which control of the arc is established by super-imposing a strong magnetic field on the arc flame. This control permits the arc to travel through the various fields without disturbance, and the arc is given a gyratory motion of great velocity which also tends to hold it in a straight line. Increased strength, ductility, and high welding speeds are obtained. The process can be applied to light sheets or heavy plate.

Metallic Arc Welding.

The use of metallic electrodes for arc welding has proved more satisfactory than the use of carbon or graphite electrodes, which necessitate feeding the new metal into the arc by means of a rod or wire. A very wide variety of metal electrodes are used and can be classified as follows: (*a*) bare wire; (*b*) coated electrodes manufactured by dipping, painting, or spraying with a liquid; (*c*) covered electrodes, which are wrapped with a refractory material or otherwise encased in a much heavier covering than that used for coatings. They can also be classified as: (*a*) heavy slag depositing electrodes; (*b*) light slag depositing electrodes; (*c*) flux-coated electrodes which deposit no slag; and (*d*) bare wire. Sheathed electrodes have a flux between a metal core and sheath, whilst composite electrodes have one or more conducting materials combined mechanically with the flux. The effect of surface materials on electrodes from the standpoint of some of the scientific laws governing the metallic arc has been investigated and it has been shown that the presence of non-metallic materials on the surface is required. Generally speaking, there is a sphere of utility for each type, and provided care is exercised in their selection, a weld of maximum efficiency should be obtained. Information on automatic and semi-automatic metallic arc welding is given on p. 2788.

Bare Electrode.

The Slavianoff System, using a bare metal electrode, has not been developed in this country to any extent. There is considerable discussion as to the efficiency of bare electrodes, and although it may be possible to successfully weld mild steel with such electrodes, where toughness is desirable or necessary, or for the welding of steel alloys, a coated or covered electrode will give better and, in many cases, the only successful results. The decision of Lloyd's in approving the use of covered electrodes has not caused the adherents of the bare-metal electrode to abandon their position. The bare electrode imposes a minimum thermal load, but requires the presence in the equipment of a high stabilising factor. The following is an abstract from a specification for bare iron and steel electrodes for welding mild steel, steel forgings and castings, plates, bars, etc.:

Carbon	0·06	per cent.	maximum.
Manganese	0·15	,,	,,
Phosphorus	0·04	,,	,,
Sulphur	0·04	,,	,,
Silicon	0·08	,,	,,

In the hands of an experienced welder electrodes should demonstrate good weldability in 'flat' and 'overhead' positions, and should pass through the arc smoothly and evenly without any unusual characteristics.

The data at hand show that bare steel electrodes having a carbon content between 0·18 and 0·85 per cent. are of no particular value.

Coated and Covered Electrodes.

The use of coated and covered electrodes aims at improving the metal in the weld, permitting the use of alloy steels, non-ferrous metals, etc., as electrodes, facilitating electrode manipulation and the use of alternating current. Covered electrodes may be placed in two general classes, the first class being the electrode which has a heavy covering of scientifically prepared material containing a deoxidiser. The second class of covered electrode is of the type which has a very thin coating on the wire and aims at obtaining the same results as the heavy covered electrodes without manipulating the heavy coating of slag deposited during the welding operation. It is possible to obtain as many as twenty different types of flux-covered electrodes, each specially designed for depositing a particular material. As a result of research work, electrodes are now available for the purpose of welding and depositing non-ferrous metals.

The percentage, by weight, of an electrode which passes over into a weld, sometimes called the deposit efficiency of the electrode, is affected by the following factors: (*a*) current, but not current density; (*b*) polarity; (*c*) thickness of the work; (*d*) chemical composition of the electrode; (*e*) surface materials of electrode; (*f*) occluded gases, slag inclusions, etc.; (*g*) arc length. Experiments so far recorded indicate that a good commercial electrode will show from 80 to 90 per cent. deposit efficiency, whilst a poor rod may show as little as 50 to 60 per cent. The bearing of this on welding costs merits attention.

Kjellberg System.

The effect of refractory coverings on the metal electrode was first discovered by Kjellberg. In the Kjellberg process, controlled by the *Anglo-Swedish Electric Welding Co., Ltd.*, the steel rod is covered with a coating or sleeve of normally non-conducting materials, the function of the cover being, amongst other things, to direct the metal where it is required, to protect the fused metal from oxidation, and to make up the loss in the arc of certain constituents of the iron or steel forming the object.

In carrying out the process the covered rod is held in a clip or holder connected to a continuous current supply, and forms the positive pole, the object forming the negative, though in some cases the poles are reversed.

The covering is removed from the end of the rod to enable the current to pass, and an arc is struck between the rod and the object. As soon as the arc is struck a portion of the metal of the object, having an area approximately equal to the section of the rod, fuses together with metal from the end of the rod, and at the same time the covering, which is of a friable nature, partly fuses and breaks away at the same rate as the metal of the rod is deposited. In this way metal can be gradually deposited on an object until a sufficient quantity has been obtained for the purpose required. From time to time the deposited metal is hammered so as to consolidate it in exactly the same way as a weld made in the fire.

A further advantage of the covering on the rod is that the tensile strength of the weld can be varied within certain limits to suit the purpose required. For example, by this process a tensile strength of from 20 to 35 tons per sq. in. can be obtained. It is possible also to obtain various degrees of hardness merely by altering the composition of the covering.

'Quasi-Arc' Electrodes.

These electrodes are the special product of *The Quasi-Arc Company, Limited*, and are patented throughout the world. The company also supply special plants which enable electric arc welding to be carried out with both A.C. and D.C. supplies of any voltage in the most economical and efficient manner.

'Quasi-Arc' Electrodes consist of a special steel wire coated with a mineral flux having distinct physical and chemical properties whereby a weld of the highest mechanical strength and purity is obtained.

In making a weld or reinforcing a worn surface the electrode is held in an insulated holder connected by a flexible lead to the positive pole of an electric generator, and an arc is struck between the electrode and the work which carries the negative connection. The welding operation is continued by holding the point of the electrode just under the surface of the molten slag and feeding down uniformly the electrode as the metal is deposited on the work.

The flux covering of the 'Quasi-Arc' Electrode has special properties to enable a low current to be used in fusing the electrode. The fused covering is acid in reaction, automatically removing any mill scale or iron oxide from the work; it is also reducing in its chemical character, so that it absorbs oxygen, and prevents atmospheric gases from causing defects in the weld metal and at the weld junction. The electrode therefore deposits a maximum possible amount of metal free from defects with a low current consumption, so that welding is not only reliable but very economical. The aluminium wire incorporated with the electrode further prevents oxidation of and removes gases from the weld metal. The low current used not only gives economy but prevents local overheating and embrittleness in the adjacent steel of the work, so that the weak brittle zone formed in the work when using many other types of electrodes is absent.

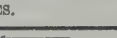
The metallurgical principles of the 'Quasi-Arc' Electrode follow closely those used in modern steel-melting practice.

Apart from Mild Steel Electrodes, the company manufacture special electrodes for welding Cast Iron, Carbon Steel, Manganese Steel, Stainless Steel; also 'Mangecross' Electrodes used for welding and reinforcing worn or fractured Manganese Steel, and 'Uranium' Electrodes which are suitable for welding work to be subjected to stresses at high temperatures, such as hot forging, flanging, etc.

The following table gives the comparative strength of riveted joints and joints made with 'Quasi-Arc' Electrodes. These figures are the results of official tests carried out by Lloyd's Register of Shipping, and it was on the excellent results obtained that 'Quasi-Arc' Electrodes were the first to be approved for welded ship construction.

TABLE SHOWING COMPARATIVE TENSILE TESTS OF LAP JOINTS OBTAINED BY RIVETING AND THOSE OBTAINED WITH 'QUASI-ARC' MILD STEEL ELECTRODES.

WELDED WITH 'QUASI-ARC' ELECTRODES.



Specimens of Mild Steel.	Dimensions. Overlap.	Breadth.	Thickness.	Sectional Area. Sq. in.	Tons. Max. Stress. Total.	Per sq. in.	Remarks.
Plain bar .	—	5·5 in.	0·25 in.	1·375	39·3	28·55	
Welded bar .	1·5 in.	5 in.	0·25 in.	1·25	36·15	28·9	Broke 4½ ins. clear of weld.
Plain bar .	—	2 in.	0·50 in.	1·0	28·4	28·4	
Welded bar .	2 in.	5 in.	0·50 in.	2·5	64·75	25·9	Broke 6 ins. clear of weld.
Plain bar .	—	2 in.	0·75 in.	1·5	42·0	28·0	
Welded bar .	2·25 in.	5 in.	0·75 in.	3·75	89·5	23·85	Broke through plate at weld due to bending.

TREBLE RIVETED.

Specimens of Mild Steel.	Dimensions. Overlap.	Breadth.	Thickness.	Rivets. Diameter.	Spacing.	Sectional Area. Sq. in.	Tons. Max. Stress. Total.	Per sq. in.	Remarks.
Plain bar .		2 in.	0·50 in.			1·0	27·4	27·4	
Riveted bar .	7·5 in.	18·25 in.	0·526 in.	¾ in. dia.	2·625 in.	9·6	165·5	17·21	Broke through outer line of rivet holes.
Riveted bar .	9 in.	18·625 in.	0·489 in.	⅞ in dia.	3·125 in.	9·1	172·0	18·91	

The use of 'Quasi-Arc' Electrodes has been authorised by the British and Foreign Admiralties, Lloyd's Register of Shipping, the British Corporation for the Survey and Registry of Shipping, Det Norske Veritas, and Nederlandsche Scheepvaart-Inspectie Bureau, for use in the more important parts in ship construction.

'Quasi-Arc' Electrodes were exclusively employed in the construction of the all-welded motor ship *Caria* (ex *Fullagar*), of 400 gross tonnage. This vessel, built in 1920, has each year successfully passed Lloyd's Annual Survey, and is at present in service at British Columbia, having been recently renamed *Shean*.

Among later achievements by the use of 'Quasi-Arc' Electrodes is the application to a completely welded 3,000,000 cu. ft. gasholder, 200 ft. in diameter, 100 ft. high, with three lifts. Since this gasholder was satisfactorily completed, others have been constructed by the same method, together with the whole of the necessary framework. In addition, gasworks plant, including retort houses, condensers, gas mains, etc., have also been welded with 'Quasi-Arc' Electrodes. Hydro-Electric Pipe Lines have also been completelv welded with 'Quasi-Arc' Electrodes and a number of oil storage tanks, the largest being 140 ft. in diameter, with a capacity of 3,000,000 galls.

'Quasi-Arc' Electrodes are used in all the Royal Dockyards and by the principal railway and tramway companies, collieries, mines, engineering firms, and shipyards in Great Britain and abroad.

An impact or alternating impact and bending test gives a much better measure of the quality of a weld, especially the toughness, than a tensile test. Moreover, such a test conforms more to service conditions. A 'Quasi-Arc' weld being of sound and pure metal gives a very high result under such a test, whereas a weld made with an electrode not having a special reducing flux gives a value, due to the oxide defects present in the metal, sometimes a third of that given by a 'Quasi-Arc' weld.

In view of the importance which the subject of corrosion bears to that of welding, the following notes on this matter may be added :—

Fundamentally, corrosion is electrolytic in character, and depends on the difference in potential of contiguous areas.

Commercial iron and steel contain areas which in the presence of an electrolyte (which may be merely atmospheric moisture) are capable of forming voltaic circuits, the necessary conditions being afforded by the presence of small defects of iron oxide, etc., or by local conditions of strain in the metal itself. These impurities or strains give rise to areas of higher electrolytic potential, and form electro-positive points which are first attacked by the corroding agent. Thus, in a caulked joint, corrosion will commence at the caulking, and similarly a riveted joint will be attacked round the rivet head, owing to the local strains in the metal at these points.

In a welded joint, if the metal at the weld be less pure than the surrounding metal, the weld will be first attacked. Similarly, if the added metal be not homogeneous, local differences in potential in the weld itself will cause corrosion at the weld.

A microscopic examination of the metal deposited by 'Quasi-Arc' Electrodes shows it to be of great purity, and the usual laboratory tests show it to be always electro-negative to the surrounding metal. Hence it follows that this weld metal will have a high factor of resistance to corrosive influences.

In order that the best results may be obtained with 'Quasi-Arc' Electrodes it is essential that the electrical conditions be satisfactory. The Quasi-Arc Company have carried out a considerable amount of research work in order to determine the best designs for various types of plant. Although primarily the electrode governs the quality of work carried out by the arc welding process, it is of great importance that the plant be able to provide the correct amperage for the electrode in use, and that the arc remain stable at this current. The Quasi-Arc Company supply special D.C. generators of the drooping voltage type for single operator installations, and also constant voltage machines for multiple welder sets which fulfil these conditions in every way. These can be supplied as motor generators for A.C. or D.C. supplies or self-generating units driven by internal combustion engines.

The company have also paid particular attention to static transformer equipment for A.C. supplies, and can offer sets with which it is possible to carry out with 'Quasi-Arc' Electrodes welding of quality equal to that effected with a D.C. current.

They supply a welding unit for A.C. welding of particular interest, which is known as a Compensated Rotary Transformer. This machine supplies an alternating current to the arc, and generally has the economic advantages of static transforming plant. It has, however, several additional features which render it suitable for use on supply systems where static plant is not advisable. These may be summarised as follows : (1) The load taken from a polyphase supply is perfectly balanced ; (2) the power factor is of a value equal to that taken by well-designed induction motors ; (3) the ease of welding operation is increased by supplying the arc with current at a frequency three times that of the frequency of supply.

All 'Quasi-Arc' plants have been designed to obtain a maximum electrical efficiency with the simplest methods of control, so that the welder can adjust his welding current without difficulty and without electrical knowledge.

Alloy Welding Processes System.

'A.W.P.' electrodes are flux-covered, and have, in general, a spiral winding of asbestos yarn as a basis of the covering. Asbestos in itself is a flux, as it fuses and combines with iron oxide, forming a compound silicate, which reaction assists the operation of the flux coating. This spiral asbestos winding makes an excellent vehicle and support for the covering, which contains various valuable constituents, and is the chief feature of 'A.W.P.' electrodes. A suitable binder is used, making a tough-coated and concentric sheath to the metal core. These electrodes, therefore, travel and handle without damage to the covering. They are approved by Lloyd's Register of Shipping, the British Corporation for the Survey and Registry of Shipping, and Bureau Veritas, and give equally good results on either direct or alternating current.

The chief problem in electric arc welding is to prevent as far as possible the partial combination of the molten metal with oxygen and nitrogen—in other words, to avoid the formation of metallic oxides and nitrides, as these compounds are, in general, very detrimental to the success of welding.

By the aid of the flux covering, the molten metal is shielded from contact with the air, and if properly chosen it also serves the purpose of disposing of whatever oxides or nitrides may have unavoidably formed.

These fluxes, however, when carefully chosen not only tend to eliminate undesirable elements in the welded metal, but can be made effective in producing any desirable chemical composition in the deposited metal.

The 'A.W.P.' process has made it possible to lay down deposited metal of any predetermined composition, and to duplicate in the weld such metals as manganese steel, high-carbon steel, high-speed steel, stainless steel, etc., and other steels of very special qualities, which it would be impossible to secure with an electrode having a core of alloy steel. This subject is fully dealt with in the company's bulletins, which can be obtained on application.

'A.W.P.' electrodes are made in over twenty different types to suit all classes of work.

The growing importance of arc welding in engineering operations has caused *Alloy Welding Processes, Ltd.*, to give special attention to the design and construction of welding generators, which besides being light in weight, and therefore easily portable, will be capable of meeting every requirement of the welder in regard to its ability to handle electrodes of any make and any size within the range at present manufactured.

The 'A.W.P.' patent welding generator is a plain shunt-wound machine without separate exciter, or reverse compound winding. It is of the drooping characteristic type, having an open circuit voltage of approximately 70, which, however, drops to the pressure required at the arc immediately welding commences.

The machine can be adjusted, by means of a shunt regulator, to suit any size of electrode at present made.

On top of the dynamo frame is fixed a casing containing an oil-immersed transformer, the primary of which is connected in series with the main circuit and the secondary connected in the shunt field circuit.

The object of this transformer is to inject voltages into the shunt winding during sudden changes of load in the main circuit. This it does so effectively that the tendency for the main current to vary is checked immediately, in consequence of which a very stable arc is obtained. With a machine of this description there is a complete absence of spluttering at the arc, and the metal deposited is homogeneous and free from holes or sponginess.

The 'A.W.P.' dynamo can be supplied direct coupled to engine-driven plants for portable use, as motor generator sets fitted with either an A.C. or D.C. motor, or arranged for belt drive.

Alloy Welding Processes have also specialised in multi-operator machines for three welders and welding with alternating current, and are the proprietors of the Daysohms patent A.C. welder system, which is as follows:—

Daysohms' System.

This system is designed to work on ordinary alternating current supply circuits, and is portable or stationary as desired. One of the advantages claimed is the special adaptability of the device for the efficient welding of very thin metal; strips of mild steel, No. 20 S.W.G., have been welded in angle form at the rate of 1 ft. in 2 mins., the current being 27 amperes at 65 volts on open circuit with a power factor of 0·45. The supply current is transformed to the necessary voltage and then passes through a choking coil. This has a straight core, with pivoted keeper which can swing, the speed of its movement being controlled by a dashpot. Limits can be set to the movement on each side by means of cam-form stops. With passage of current the keeper of course tends to be attracted towards the core, and the nearer it approaches, the lower the flow of the current, owing to the increased choking effect. The limit of the approach is set by the adjustable cam, and in this way any welding current within the limits can be obtained. The other cam, which controls the striking current, similarly regulates the distance by which the keeper can move away from the core by gravity or spring. As has been stated, the speed of the movement is controlled by the dashpot, and this can be regulated by means of a perforated disc in the piston. Consequently both striking and welding currents can be adjusted independently, and the keeper is free to move between the stops under the control of an adjustable dashpot, so that the characteristic curve connecting volts and amperes is the resultant of the two curves which would have been obtained if the keeper had been fixed in either position. The consequence is that it is possible

to weld with a lower open-circuit voltage and, in addition, to weld thinner material without burning through, as the striking current can be set so as to be very little in excess of that required for the actual welding operation. In working the firm recommend that the current for welding should be adjusted first and then the striking current reduced until there is just sufficient excess to enable the arc to be struck easily.

These welding machines are available in two standard sizes, one for currents between 25 and 100 amperes with infinite gradation, and the other similarly for currents up to 200 amperes. The machine is supplied as a complete unit ready for connection to the supply main, and any ordinary flux-covered electrode can be employed.

Premier System.

The Premier system of arc welding aims at distributing the current at the required welding voltage without any mechanical control whatsoever. This is obtained by the use of a dynamo of a special type connected with the arc through a special reactance coil in the main circuit. This coil absorbs no appreciable energy, but effectively damps out any sudden rise in current, and at the same time helps momentarily to build up the current when any sudden reduction takes place.

The company manufacture plant of the multiple operator type which does away with large and wasteful series resistances.

A special type of plant for welding thin sheet metal has been developed. The machine consists of a very small transformer and choke coil, and sheets down to 18 gauge can be welded satisfactorily. Flexin welding tools and electrodes by means of which a flexible electrode is fed to the arc by means of a pneumatic or electrically driven hand tool represents a continuous welding system which ensures high welding speed and elimination of electrode waste.

Ferro-Arc System.

The Ferro-Arc system employs electrodes with a coating of flux and slag-forming materials. The core metal of Ferro-Arc electrodes is of the highest quality and made to an exact specification for arc welding. The principal component of the flux is powdered blue asbestos fibre specially treated and combined with other substances which control the melting point of the asbestos and enhance its power of dissolving oxides of iron and other impurities. Blue asbestos is peculiarly suitable for use as a flux, and it is doubtful whether any other material or combination can give such successful results in arc welding.

Ferro-Arc flux fuses evenly and level with the melting of the metal core, and when molten is fluid, free, and mobile, entirely covering and protecting the deposited metal. The flux has definite deoxidising properties which, together with its powerful solvent action for oxides, ensures the deposition of clean metal free from impurities. The scavenging action of the flux removes scale and rust from the surfaces to be welded, and so results in perfect fusion and interpenetration of weld and parent metals. The necessity for a substantial coating on the electrode for the achievement of best results should be emphasised. It is not possible to deposit sound metal free from oxides and impurities and which is thoroughly united to the parts to be welded without an adequate amount of flux and slag. A substantial coating in no way hinders the operator. On the contrary it assists him greatly, improving both the speed of deposition and ease of manipulation, while ensuring the soundness of the deposit and the neatness of finish. Valuable ingredients may be imparted to the weld from the coating.

The company manufactures three grades of electrodes for the welding of mild steel, *i.e.* Types A, B, and C, which vary in quality and price. The coating in each case is based on blue asbestos powder. There is also Type AA for the welding of mild steel, where exceptionally high strengths (about 30 tons per sq. in.) are required, or where it is necessary to hot forge the part after welding. As a recent development the *Ferro-Arc Welding Company* have undertaken the manufacture on an extensive scale of electrodes coated with blue asbestos yarn as distinct from powder. The uniform dependability and the consistent high quality of rods of this class are unrivalled for withstanding damage and deterioration both from rough usage in transit and from changing atmospheric and climatic conditions.

All the above types may be used on any standard plant using either direct or alternating current. Remarkably good results are obtained with these electrodes, the outstanding features of which are smoothness of working, penetration of the metal, neatness of finish, and high strength of the weld.

For those who require only a very lightly coated rod at a low price, the company provides its well-known ' Kathodes,' which give best results when connected to the negative pole of a direct current supply.

For the welding of cast iron two types of electrode are made. The standard type is specially designed to deposit a true cast iron in the weld, while Type D deposits a nickel-copper alloy which has the great advantage of forming a soft machinable deposit without preheating ; with this electrode cast iron may be welded ' in the cold ' without the formation of hard spots, particularly in the line of junction.

Electrodes to deposit 14 per cent. manganese steel and 0·5 per cent. and 0·8 per cent. carbon are also supplied.

A new development is an extremely successful electrode for welding stainless steel, particularly 'Staybrite.' This electrode deposits a metal containing approximately 8 per cent. nickel and

18 per cent. chromium. The composition of the electrode could be suitably altered so as to give other desired combinations of nickel and chromium. In working it is particularly smooth, fusion is perfect, and neatness of finish remarkable. The deposit is very tough and entirely free from blowholes. Its resistance to corrosion is as high as that of the metal on which it is deposited. It may be used for the welding of mild steel to stainless steel.

The Ferro-Arc Welding Company's system of electric arc welding employing Ferro-Arc electrodes has been tested and approved by Lloyds Register of Shipping.

A.C. System.

The A.C. welding system is one which embodies novel features. The nature of a welding arc is such that the use of bare wire with alternating current for welding purposes is impossible unless special provision is made to counteract the effect of the periodic cessation of power.

Before the introduction of the A.C. welding system, the use of alternating current for welding purposes was condemned as inferior to D.C., and was only used in conjunction with heavy slag-forming electrodes.

A paper which was read before the Institution of Welding Engineers* explained the theory of arc stability, and showed the difference between the A.C. welding system and the transformer and fixed reactance stabiliser, which had hitherto been in vogue. It also shows that the tendency of the arc to die out is counteracted by the method of arranging the choking effect in the circuit as a variable quantity instead of being a fixed value. The 'Alternarc' transformer of the A.C. system can be adjusted to have a certain average value for its reactance, and this governs the average value of the power supply to the arc, in the same way as a fixed reactance choke coil is designed to determine the value of the power supply to the arc. The reactance of the 'Alternarc' transformer, however, is not a fixed quantity, but varies with the current over the whole load cycle, in such a way that when the current is at minimum the reactance value is a minimum, and *vice versa*.

The effect of this method of arc control is to give very high arc stability; so high, in fact, that it is claimed possible to do equal, and even more efficient, welding work with alternating current than with direct current.

Alternating current has been condemned for welding on many occasions, owing to the fact that the arc stability factor supplied by the transformer and choke combination was so extremely low. This bad feature of poor arc stability has now been overcome by the A.C. system, and consequently, with maximum welding efficiency possible with alternating current, coupled with the desirable features of high electrical efficiency, portability, and low first cost, the A.C. welding system offers a most efficient and economical welding system.

The way in which the variable reactance control is arranged is as follows:—The 'Alternarc' transformer consists of a primary winding wound on one limb of a closed core, having two secondary windings, one of which is wound to have high mutual inductance and low leakage inductance with the primary coil, and giving a heavy leakage flux on full load. In some cases an alternative path is arranged for the leakage flux by extending the core yokes between which extensions a movable flux-diverter is arranged. Both sections of the primary windings are tapped at intervals to give adjustment to the voltage and also to the current, the movable flux-diverter giving a fine adjustment for accurate work.

The material used in the construction of the transformer is fireproof, the windings being composed of asbestos-covered wire, insulated with pure mica. Each of the larger type machines is provided with a cooling fan, and is enclosed in an oak case arranged with Sedan handles, in such a way that a whole apparatus is a portable unit.

To summarise, the claims made by the A.C. Welding Company for their system are—maximum welding efficiency, maximum arc stability (giving the choice of any electrode on the market); maximum electrical efficiency (giving minimum running cost); minimum upkeep (there being no adjustments of brushes, attention to commutators, etc.), and no maintenance charges. The machine is guaranteed against burning-out, and can only break down under wilful bad treatment.

The machines are normally supplied with an open circuit voltage ranging from 60 to 100, according to the settings of the secondary connections, but if required they are supplied with a standard 'Alternac' safety device, which limits the open circuit voltage to a maximum of 25 volts.

Electric arc cutting has been developed on the A.C. system. The system of cutting consists in the use of special cutting electrodes, which liberate free oxygen at the arc, steel being the basis of the electrodes on currents of 300 amperes, and carbon on currents from 600 to 1000 amperes.

Wilson Plastic-Arc System.

A wide and varied range of arc welding machines are manufactured on this system by *Messrs. G. D. Peters & Co., Ltd.* A special feature of all Wilson Plastic-Arc Welders is the Control Panel, which gives the machine welding qualities available in no other way. These panels, which have a range of adjustment to suit the particular welder, give close current regulation without the use of a separate exciter or brush moving. The whole equipment, which includes all accessories,

* *Theories and Facts to Aid in the Choice of an Efficient Arc Welding System*, by D. T. Smout, A.M.I.E.E.

The Hall Mark

of a FIRST-CLASS ELECTRODE lies in the DEPOSITED METAL

FERRO-ARC

ELECTRODES

are manufactured in several grades to suit varying requirements, but each excels in its own type.

Blue Asbestos yarn coated electrodes of the highest quality, fully fluxed paste coated electrodes and lightly coated rods are all included in the Company's products.

Special electrodes for developing mechanical and hot forging properties in the weld are produced, as are also electrodes for welding Cast Iron, Manganese, Steel, Nickel-Chromium alloys, etc. etc

The Ferro-Arc System of Welding is approved by Lloyd's Register of Shipping.

The BRITISH OXYGEN Co LTD
Inc. ALLEN-LIVERSIDGE Ltd

Head Office and Engineering Works:
Angel Road, Edmonton, N.18

Sole Selling Agents in Great Britain and Ireland for

FERRO-ARC WELDING CO., LTD.

F. 2786

The Engineer.

ESTABLISHED 1856.

1/- WEEKLY

The Metallurgist

IS A FREE MONTHLY SUPPLEMENT

to The Engineer.

EDITORIAL and PUBLICATION OFFICES

28 ESSEX STREET STRAND LONDON W.C.2

ANNUAL SUBSCRIPTION RATE £3 5 0

F. 2787

is compact and built for service and can be supplied with a motor of practically any characteristics to suit the local source of power supply, or the generator can be supplied without a motor, to be driven by any means locally available, such as, for example, a belt. Alternately the unit can be petrol driven, and portable petrol sets are a speciality of the firm.

The electrodes supplied by the firm have a world-wide reputation and are known as the Wilson 'Colour-tipt' Welding Metals. The composition of each grade of metal, which covers all welding requirements, is the result of exhaustive experiments to provide an electrode having proper flowing qualities, penetration, and characteristics as deposited metal, to produce a sound, mechanical weld. The packing and marking of these electrodes has received considerable ingenuity and attention which ensures economy and their ready identification under practically all conditions of use. The welding plants and electrodes have the full approval of Lloyd's. The close current control, stability, and value is such that special alloys, such as the non-corrosive chrome alloys, can be welded without losing the effect of their essential elements in the weld.

Fluxode System.

The Fluxode system, Engineering Developments, Ltd., uses flux-covered and bare-wire electrodes which can be used with any arc welding plant. Mild steel, high carbon and manganese electrodes of both types suitable for high-class welding with high tensile strength or where a hard deposit, such as for reinforcing or building up tramway and railway rails, is required.

Arc welding plant to meet all conditions and for use on alternating or direct current has been designed from the point of view of economy and efficiency. The single operator set, for example, is suitable for any standard voltage on direct or alternating current supply varying from 100 to 500 volts. The set consists of an electric motor coupled to a specially wound welding generator which possesses a variable voltage characteristic similar to that obtained by the combination of a level voltage supply with a stabilising resistance in series with the arc.

Electric Blowpipe Welding.

In this process the electric blowpipe is essentially a horizontal-flame arc lamp using two carbons mounted like the letter V. The process was introduced by Dr. Zerener in 1890, and is usually known as the Zerener system. The flame of the arc was directed away or deflected downwards by means of an electro-magnet in series with the arc. The apparatus is bulky and complicated, and for even moderate-sized work it must be suspended from above. This method of welding eliminates the passage of the current through the work. It has not proved as successful as the methods already mentioned, and has been superseded by them. The Voltex Process is a variation of the Zerener method, in which the carbon electrodes are impregnated with metallic oxide which is vaporised by the flame of the arc.

Arc Welding Equipment.

Electric-arc welding may be done with a direct-current or alternating-current arc, having a reasonable degree of stability and sufficient amount of heat to accomplish fusion. The effect of the 'type' of current on the efficiency of metal deposition by the arc is threefold. It affects the speed of deposition, the amount of penetration or fusing of the metal, and arc stability. The design of the welding generator or transformer affects the cost of welding. The necessity for a special machine rests upon economic factors, and the differences between the various types used are those of power, cost of operation, and difference in speed of operation due to the relative stability of the arc.

The question of direct or alternating current raises many interesting points which have never been clearly determined. Good welding can be done with both direct and alternating current. Flux-covered electrodes are essential for certain types of alternating current machines, apparently because the lagging effect of the flux and slag maintains the necessary temperature during the periodical current reversals. Comparative data by independent authorities is missing, and the question of whether direct-current welding is faster or slower, whether flux-covered electrodes enable direct current to take the lead in welding speed, whether the practical losses in motor generator and resistance compare favourably or otherwise with the alternating-current system, etc., etc., are questions which are answered by contradictory claims by both parties to this controversy. However, there are advocates for both systems, and satisfactory equipment can be obtained for arc welding with direct or alternating current.

The successful application of arc welding is largely dependent upon the selection of the proper equipment. There are two systems in general use—(*a*) single operator system, where a separate machine is provided for each operator, each machine receiving its current supply from the supply circuit; (*b*) multiple operator system, where more than one operator receives current direct from the same machine, but each operator is provided with a control panel which enables him to vary the current without interfering with the other operators.

The equipment consists of a motor generator, control panel, and accessories. When using alternating current use is made of a transformer to reduce the distribution voltage to that suitable for the weld.

Amongst the accessories are protective equipment and electrode holders.

Protective Equipment.

The intense glare emitted in the process of arc welding consists of a combination of invisible infra-red rays, visible light rays, and invisible ultra-violet rays, and special safety devices are required to protect the operator from their harmful effects. For light electric welding it is necessary only to protect the eyes with goggles fitted with suitable coloured glasses. Hand shields are made for medium-weight electric welding work which can be done with one hand. They serve the double purpose of protecting the eyes of the operator and also shielding the face from the heat rays and ultra-violet radiations. For heavy electric welding it is common practice to use a helmet or hood which provides protection for the eyes, head, face, and neck, and leaves the hands free. With regard to the tint of the protective glasses, a reference to the note on goggles in the section on oxy-acetylene welding will be useful. The filtering glass used should cut off both ends of the spectrum, the ultra-violet and infra-red, and transmit, in reduced quantity, only rays of high visibility, such as the yellow and green radiations, thus giving the maximum visibility with the minimum reception of energy. For the protection of the hands and wrists leather gloves are used. These devices are subject to modifications to meet the requirements of special work. Neighbouring workmen and operators should be protected from the dangerous rays of the electric arc.

Precautions in Arc Welding.

The precautions that should be observed in arc welding have been set forth in a memorandum issued by the Factory Department of the Home Office (Form 329, June 1921). These deal with: (*a*) hand-screen; (*b*) helmet or mask; (*c*) gloves; (*d*) aprons; (*e*) protection from hot metal; (*f*) chipping of slag. The memorandum also deals with the comparative dangers of direct and alternating currents, use of direct current, injurious effects of radiations, etc. The memorandum on the Electricity Regulations (Form 928, 1925) should also be consulted.

Automatic Metallic Arc Welding.

Depositing metal using an automatic feed for the electrode results in a materially increased rate of deposition. Machines have been developed which are capable of welding steel 11 B.G. thick at the rate of 22 ins. per minute. This method of depositing metal is particularly advantageous on sections of uniform shape upon which a considerable amount of metal is to be deposited. Automatic welding does not eliminate the human element in welding, but does in certain cases greatly increase the speed of the process, and in other cases obtains results which are beyond the limitations of manual operation. Automatic welding with the bare metallic electrode is in more general use at present than any other method. In one type of machine the electrode is fed to the arc from a reel by means of feed rolls actuated by a 30-volt motor which is shunted across the arc when the arc is struck. Regulation of feed is maintained by two contractors which change the direction of rotation of the feed motor when the arc voltage increases or decreases above or below a certain amount. Research has proved conclusively that a faster, stronger, and more homogeneous weld can be made on material $\frac{1}{8}$ in. or more in thickness by use of a multiple arc automatic welder, depositing the metal in two or more layers, using about 350 amperes in each arc, rather than by depositing in one layer, using an excessively high amperage. A number of welding applications involve all the requirements of automatic welding except that the size or weight of the pieces to be welded prohibits convenient handling, clamping, and traversing. In such cases a semi-automatic welding machine has proved to be of great advantage. One type of machine feeds the welding wire through a flexible conduit to a welding nozzle in which the control switch for the feed motor is located. The control mechanism operates to start and stop the feed motor. The maintenance of a uniform and steady arc depends on the operator.

Speed of Welding in Feet per Hour.

Thickness of Metal.	Hand Welding.	Semi-Automatic Welding.	Automatic Welding.
$\frac{1}{16}$	—	25 to 50	50 to 100
$\frac{1}{8}$	7 to 16	15 to 30	30 to 50
$\frac{3}{16}$	6 to 12	12 to 25	20 to 35
$\frac{1}{4}$	5 to 10	8 to 15	10 to 25
$\frac{3}{8}$	3 to 5	5 to 8	5 to 8

Arc Welding Applications.

The application of arc welding to repair and manufacturing, particularly for the replacement of castings, is very extensive, but its application should be made with a full knowledge of the possibilities and limitations of the process. In the welding of iron and steel, with their great variety of physical properties, a familiarity with the wide range of conditions is necessary to meet and solve the problems presented. In welding cast iron, copper, brass, aluminium, etc., the application of arc welding should be looked upon as in the development stage, where welds with physical, chemical, and mechanical properties similar to the base metal are required.

A complete investigation into the welding of cast iron is being made, and the results, as far as they have gone, demonstrate that it is possible to weld some grades of cast iron if the proper method and conditions of welding are obtained. The following points have been considered :—(1) preparation of the parts; (2) prevention of slag inclusions; (3) suitable values of current; (4) method of deposition and its effect on the strength and hardness of the weld; (5) reversed polarity versus straight polarity; (6) effect of long versus short arc; (7) effect of speed on the softness of the weld; (8) the hard zone and its removal; (9) relative weldability of various grades of cast iron; (10) tests of electrodes.

Cost of Arc Welding.

The cost of electric arc welding with the metal electrode process depends upon several variable factors, such as: (1) the price of labour; (2) the characteristics of the arc, (*a*) arc easy to maintain, (*b*) arc difficult to maintain, (*c*) amount of heat available; (3) the cost of power; (4) the cost of electrodes; (5) supervision required, (*a*) production work, (*b*) work of a varied character; (6) interest charge on equipment.

Welding speed is fundamentally dependent upon the rate of metal deposition. This rate varies with such factors as, energy required for liquefaction of electrode material, arc stability, current density, and electrical resistance of electrode material. It has been shown that high current welding results in an increase of 100 per cent. in the speed of welding a butt joint. Butt joints in ¼-inch plate may be welded at a maximum rate of 5·3 ft. per hour when employing 175 amperes; at 10 ft. (or more) per hour when using 400 amperes.

RESISTANCE WELDING.

There is a marked distinction between the joining of metals by the two electric processes, viz., arc welding and resistance welding. In the former the heat is localised through the action of the electric arc and no pressure is applied; in the latter the heat is localised by the resistance to the flow of the current and pressure is used.

The process of resistance welding is based upon the phenomenon that a poor conductor of electric current will heat if current is forced through it, or that a good conductor will also heat if enough current is passed through it. Since an imperfect joint, such as the abutting ends of two pieces of metal, or the lapping of two pieces of metal which are to be welded, is a poor conductor and offers great resistance to the passage of the current, it will naturally heat and finally cause the metal to fuse and flow sufficiently to weld. In practice the operation is very rapid because comparatively large amounts of current are used.

For this type of welding a single-phase alternating current of any commercial frequency is used. The transformer in the machine will take the current from an ordinary shop circuit and reduce it to a suitable welding current.

Resistance welding is only suitable for duplicate or repetition work. In general one machine is designed for handling pieces of the same metal and for making joints of similar design. Repair jobs have been done in special cases, but it is only in rare cases that such an adaptation is attempted.

A resistance welding machine is simply a single-phase transformer having a secondary of one or two turns, the voltage required between the terminals being only five to ten volts, and the current being limited only by the size of the pieces to be handled. The transformer is built into a heavy frame which supports the clamping jaws to handle the work and mechanical devices to regulate the heat and pressure. The machine has reached a high degree of perfection, and large savings of time and labour have resulted in production work. The principle involved has been applied in the construction of a variety of machines.

By some one of the resistance welding processes, nearly all metals and alloys can now be welded. Tin is the notable exception. The cross-section of stock that it is possible to weld by the resistance process now seems to be limited only by the electric power available. Special adaptations for heating rods, bars, billets, rivets, etc., and electric brazing and soldering machines are available. Special machines for zinc tube welding, automatic spot welding, rivet heating, forging hearths, welding hollow articles, etc., etc., have been designed, and their main features have been dealt with in a paper read before the Institution of Welding Engineers.*

* 'Resistance Welding,' by F. E. Saunders, M.I.E.E.

Butt Welding.

Electric butt welding consists in joining pieces end to end by securing complete fusion of the entire cross-section of the pieces. The pieces to be welded are placed in the clamping jaws, one of which is stationary and the other movable; enough pressure is exerted to bring the ends into contact and the current is then turned on. As soon as the welding temperature is reached the operator increases the pressure, thus forcing the pieces together, and turns off the current. Nearly all metals used commercially have been welded by this process.

Butt welding may be subdivided into three classes, namely, (1) upset welding, (2) flash welding, and (3) contact flash welding or automatic butt welding.

As regards the first subdivision, in which the weld is made by butting the cold ends of the stock to be welded and then applying the electric pressure, it is apparently being superseded for most purposes by flash welding.

Flash welding is made by applying the current before the ends are in actual contact and utilising the flash or minute arcs that play between the ends of the stock. Some advantages of flash over upset welding are (*a*) less power required, (*b*) less extrusion or upsetting, (*c*) burnt metal is thrown off, (*d*) increased production, (*e*) stronger welds. The best technique for the flash welding of tungsten steel tips to medium carbon steel shanks in the manufacture of machine cutting tools, and the serviceability of such tools as indicated by bending and service tests, has been determined.

The third subdivision, contact flash welding, is in some respects similar to percussive welding. The weld is made almost instantly by using high continuous contact pressure and heavy current, the current being cut off by automatic methods of control.

Spot Welding.

Electric spot welding consists in pressing the pieces together while passing the current through them, and localising the flow of the current so that the weld is confined to 'spots' between the electrodes, which are used to secure the pressure. The term automatic spot weld is given to a spot weld made on a welder which applies the pressure, makes and breaks the circuit all automatically.

The material to be welded is placed between copper die points supported on the ends of two arms, and the points are brought firmly together on the metal by the use of either foot or motive power. The current is then turned on. The metal, being a poor conductor, becomes heated, and when heated to a sufficient degree, pressure is applied. In the case of automatic control, the current is turned on and off and the necessary welding pressure applied automatically by controlling mechanism usually driven by a variable-speed motor. The majority of commercial metals can be spot welded, but it is essential that the metal to be welded should offer a considerable resistance to the electric current, otherwise sufficient heat cannot be generated. The technique for spot welding galvanised iron has been determined and gives very satisfactory results. The requirements for successful welds are rounded die points and twice the current and half the pressure required for ungalvanised metal of the same thickness. The spot welding of light materials guarantees a successful product and increased production. A commercial spot welder for ¼-inch plates is in use.

During the past few years considerable investigation has been carried out with regard to heavy spot welding for ships and constructional work. Large machines capable of supplying a welding current up to 100,000 amperes, and equipped to exert a pressure up to 30,000 lbs. per square inch on each spot, have been constructed. The main difficulties which have been encountered are, briefly, electrode-tip protection and the preparation of the materials to be welded. Two secondary considerations have been the separation of the electrodes and the shape of the head of the spot welder.

The following indicate the variations of spot welding:—

Button and Disc-Depression Welding.

Where very thick work is to be spot welded it is often desirable to find means to localise the current and cut down the amount of pressure required. Small metal discs or buttons can be used for this purpose placed between the head of the welder and the metal to be welded. Disc-depression welding differs from button welding in that in the disc-depression method an annular depression is provided in one of the sheets, around the disc. The method makes possible the use of a comparatively light machine for welding heavy steel plate, and also the use of less power and time for making the welds.

Bridge Welding.

This method is used to join thin sheets edge to edge without overlapping and without reducing the dimensions of the pieces. The sheets are brought together in their correct position, and then thin strips or discs are spot welded across the joint.

Multiple Electrode Welding.

Time and power can frequently be saved by making several welds at once, in which case the spot welder is equipped with two or more electrodes. It is advisable to have projections on one of the pieces to be welded to assist in localising the current. Successful multiple welding requires a separate transformer for each weld.

T Spot Welding.

When pieces are welded in the form of a T by holding one piece in a hollow electrode and welding it to the other the resulting weld is called a T spot weld.

Point or Projection Welding.

In this method the metal to be welded is provided with points or projections. The points are brought in line with each other, and the welding effect is to provide a smaller contact point which increases the resistance and allows the weld to be made with the consumption of less time and power. Ridge welding is the same as point welding, except that, instead of a single point, a ridge is raised in the metals in such a way that the ridges on the two pieces come together in the form of a cross.

Duplex Spot Welding.

In this type of spot welding two transformers are used simultaneously on opposite sides of the sheets to be welded and their currents pass through two secondaries, welding two spots simultaneously.

Combinations of all the above processes have been made for special jobs. Sometimes it is only necessary to make a change in the construction of the electrodes to make a machine do work of a varied character.

Seam or Line Welding.

This process practically combines the features of butt welding and spot welding, and is used for joining pieces along a continuous line or seam, usually by moving the joint in the path of the current, so that fusion is secured continuously. Machinery has been developed, but is usually limited to work on light stock and short seams. A recent development is the use of interrupted current for making longer seams on heavier stock. Excellent welds have been obtained by the step-by-step method in which the roll moves and stops alternately. During the rest period the rolls are energised and the welds made. The speed of welding is rather slow. The construction of all-steel motor bodies has given rise to a development which can be classified as flash seam welding, and consists of flash welding a long seam in a machine which really consists of two butt welding machines mounted on a single base. A 56-in. seam is flash welded in approximately 5 seconds. Standard seam welders are usually equipped with two electrodes, one connected to the work and the other a roller running over the seam, or both electrodes may be converging rollers, with the converging edges resting on opposite parts of the work near the edge to be welded. The work is passed beneath the rollers, which carry the current into the metal, and makes a continuous line of welding. The work must be carefully cleaned and be securely held by the machine to get good results. The width and strength of the weld depend upon the current, pressure, and speed adjustments as well as upon the size and character of the electrodes and the shape, size, and character of the metal being welded.

Considerable development work has been done during the past two years. It has been applied to the manufacture of steel barrels, casks, tanks, ventilators, heating and ventilating pipes, etc. Seams can be made at a rate of 40 ft. per minute on $\frac{1}{16}$ in. sheets, and 20 ft. per minute on $\frac{1}{4}$ in. sheets. The welds may be butt, lap, or bridge, and be made continuously or progressively.

Mash Welding.

Mash welding is a process of welding rods, wires, strips, etc., by crossing or overlapping the pieces and welding between relatively large electrode surfaces under pressure. The process is used in the manufacture of lamp-shade frames, wire waste baskets, etc.

Electro-percussive Welding.

This is a process which is chiefly intended for manufacturing applications in which large quantities of duplicate parts are to be produced. The underlying principles utilised in percussive welding were discovered in 1905 by Mr. J. Chubb, of the Westinghouse Company. Following his original experiments, machines were developed for welding metals in wire form of rather small cross-section. Widely different physical properties offered no obstacle, and

ductile welds can be made with copper and aluminium, platinum and lead. Electrical devices are used whereby a rapid small separation of the parts to be welded causes an intense arc to melt the surfaces. At the proper moment a forging hammer forges the pieces together. The advantages claimed for the processes are (*a*) power saved ; (*b*) time of welding negligible ; (*c*) welds of unequal sections are possible ; (*d*) welds of unlike metals can be made ; (*e*) uniform welds; (*f*) cost of finishing welds reduced. Recent results have shown that percussive welding has many valuable applications, especially in non-ferrous and steel-alloy welding.

Experimental work has been directed toward extension of the process to larger cross-sections. A large impulse transformer has been built capable of storing 20,000 joules. Sections up to ¾ in. diameter have been successfully welded and no apparent limitations are anticipated from a process point of view.

ELECTRIC CUTTING.

Although cutting with a blowpipe has many advantages over electric arc cutting, yet there are many places where the electric arc can be used advantageously. The groove is not so narrow as that cut with a blowpipe, nor is the surface cut as clean. One advantage of cutting with the arc is that scale, either on the surface or in pockets, cold shuts, sand pockets, and blowholes have no appreciable affect on the cut and will not extinguish the flame or arc as is the case of oxy-acetylene cutting.

The electric arc will cut any metal, ferrous or non-ferrous, and has been found particularly useful for the cutting of cast iron. The thickness that can be economically cut by the arc varies from 4 to 6 inches, for straight cutting. The current necessary for cutting operations on general and heavy cutting varies from 800 to 1,200 amperes. The voltage of the cutting arc varies between 25 and 45, a good average being 37. The best arc for cutting is a reasonably long one or a reasonably short one, whichever is preferred. The chief point is to have the liberation of energy near the work to be cut.

It will be seen that heavy values of current are required, and special apparatus has been designed so that the arc could be controlled with these heavy currents. An important part of these developments is the design of a handle to protect the operator, hold the electrode, and to carry 1,000 amperes. A small item like the use of pure graphite instead of carbon for electrodes means the difference between success and failure.

Increase of current means larger electrodes and hence more metal to be cut out. The latest development for this work is a low resistance graphite electrode and the approximate sizes for various current densities are as follows :—200 amperes, ¼ to ⅜ in. electrodes ; 300 to 400 amperes, ⅜ to ½ in. electrodes ; 400 to 600 amperes, ½ to ⅝ in. ; 600 to 800 amperes, ⅝ to ¾ in ; 800 to 1,200 amperes, ⅞ to 1 in. electrodes.

Cutting on heavy sections is generally best accomplished from the underneath side or from a vertical face tipped slightly greater than 90 degrees so that the molten metal can flow away, although the action is not entirely simply melting metal. In cutting with the arc a reasonably smooth surface can be obtained by paying attention to one side only, or the cut may be made rough and trimmed up smooth by a second application of the arc.

Recent developments consist of a special electrode for cutting with the arc, and for such work as piercing holes. This special electrode, covered with a coating giving off free oxygen under the temperature of the arc, gasifies the entire metal melted, so that the boring action is uninterrupted after once started. This electrode is used by starting the arc and immediately pushing the electrode into the puddle. After the hole is through the piece it can be widened to any size or shape by drawing the arc around its edges. This type of electrode only takes from 250 to 350 amperes. A development in under-water cutting is an adaptation of the carbon electrode arc cutting process, in which the carbon electrode is pierced by two small brass tubes, through which a special cutting gas is supplied from the surface. The gas oxidises and blows away the metal melted by the arc, and also converts the surrounding water into steam, thus making a gaseous envelope for the arc. The combined action serves to greatly speed up the operation of the cutting electrode, as the metal is blown aside before it has a chance to solidify. This device has been used successfully for cutting metal between three and four inches thick at a depth of about 70 ft.

ATOMIC HYDROGEN WELDING.

Equipment is now available for atomic hydrogen welding. The process consists in passing a stream of hydrogen gas through the arc formed between two electrodes. The heat of the arc breaks up the hydrogen molecules into atoms, and these combine again a short distance beyond the arc, and in so doing liberate a large amount of heat. Since atomic hydrogen is a powerful reducing agent, it reduces any oxides which may form on the surface of the metal, and metals can be melted without the slightest trace of oxidation.

The equipment represents the results of continuous experiment, research, and trials over a considerable period. The outfit consists of a single-phase transformer for converting the voltage of a 60-cycle supply to one suitable for the welding equipment, a specially designed variable reactor for providing the proper welding current and voltage, and a welding blowpipe by means of which the actual work of welding is performed.

The blowpipe consists of a holder supporting two tungsten wire electrodes, the electric conductors connecting these to the reactor, and the tubing for the hydrogen gas. Each electrode is supported inside a nozzle through which the hydrogen is forced out around the electrode. The combination of electrodes and nozzles is set at an angle, and the distance between the electrodes, or arc length, and also the flow of gas, are readily adjustable. In contrast to the usual method of arc welding, there is no current flowing from the electrodes to the work to be welded ; the circuit is completed from one electrode to the other. When welding ceases, the welding circuit is automatically interrupted until the operator is ready to weld again, when the circuit is automatically restored

In making welds the metal is melted under the heat of the arc, but to add additional metal a welding wire or rod must be used and fed into the arc in much the same way as that followed in oxy-acetylene welding.

The reactor consists of a U-shaped core provided between with pole faces, between which a pivoted armature can move. The legs of the core are provided with coils connected in series with the electrodes of the blowpipe. The amount of welding current and voltage is governed by the movable armature which changes the air gap of the magnetic circuit and therefore the reactance.

The equipment is made for operating from 60-cycle, single-phase circuits, and for use on ordinary metals less than ½ in. in thickness, or on hitherto unweldable metals of greater thickness. Extremely thin material can be welded. The process is particularly adaptable to the welding of metals and alloys so far not weldable by other methods.

THERMIT WELDING.

Thermit is a trade name for a mixture of finely divided aluminium and iron oxide, which when ignited reacts to produce a superheated liquid steel at 5000° F. The underlying principle of the process is the high chemical affinity of aluminium for oxygen. Up to a temperature of 2800° F. thermit is an inert mixture. At that temperature the aluminium unites with the oxygen of the iron oxide and comes down as liquid steel at about twice the temperature of ordinary molten steel. If steel at this high temperature is poured around the sections to be united, especially if the sections have previously been preheated, it will melt them and amalgamate with them so as to form a fusion weld.

Four varieties of thermit are produced for welding ferrous material, *i.e.* plain, cast iron, railroad, and wabbler thermit.

In making a thermit weld, the parts to be united are first lined up with a space between the ends, the extent of which depends upon the size of the sections to be welded. The ends are thoroughly cleaned and a wax pattern is formed around them, the exact shape of the reinforcement of thermit steel which is to be cast at that point to make the weld. A sand box is next rammed around the pattern and inside a sheet iron box, provision being made for pouring gates, heating gates, and risers. The parts to be united are brought to a good red heat by suitable means, such as compressed air liquid fuel. The charge of thermit is placed in a conical shaped crucible supported over the pouring gate of the mould and ignited by means of an ignition powder. In 25 to 35 seconds the reaction is completed and the crucible tapped. The slag does not enter the mould.

The process is extensively employed for locomotive repairs, marine repairs, crankshaft repairs, steel mill repairs, pipe welding, rail welding, and miscellaneous heavy welding. Detailed instructions are available for carrying out thermit welds in the fields mentioned. The process is adapted to all sizes and weights of pipe up to 6 in. diam., and an important feature possessed by the thermit pipe welding process is the ease with which it can be applied to the welding of coils. The joints can be welded either before or after the pipes are bent.

ADDENDUM.

The following is a brief review of welding progress in 1930:

In spite of the stagnation in practically every branch of engineering the demand for welding equipment has increased. Arc-welding has made marked progress, particularly for the welding of steel structures and 'redesign engineering,' in which castings, such as frames of machines, are replaced by welded steel construction. Considerable improvements have been made in resistance welding machinery, and projection welders have been built that require over 3000 KVA power to operate them. The bronze welding of cast iron, etc., is now extensively employed.

The use of automatic arc and oxy-acetylene welding of several different forms has increased. An improved portable semi-automatic arc-welding equipment suitable for use with either the open or hydrogen-shielded arc has been perfected. The atomic hydrogen process was further extended in usefulness by its adaptation to automatic welding. This permits of greater speeds with higher currents than were feasible with the hand operation of the process. Several new types of arc-welding machines have been designed to give the welding circuit of single-operator machines greater stability.

SECTION XLI

EXPLOSIVES

(Contributed by Professor W. R. Hodgkinson, F.R.S. (Edin.), Ph.D., F.I.C., C.B.E.)

SECTION XLI

EXPLOSIVES.

(Contributed by Professor W. R. Hodgkinson, F.R.S. (Edin.), Ph.D., F.I.C., C.B.E.)

Explosives may be classified into 'Propellants,' which are required to produce a rapid but steady rise in pressure in some kind of firearm; others may be termed 'Detonators,' or the less definite term 'High Explosive.' The latter are practically instantaneous in action and produce a wave or vibration of high velocity. Under certain conditions some substances may behave as belonging to either class.

The first propellant, 'Black Powder,' has since 1770 (about) consisted of nitre 75 per cent., charcoal 15 per cent., and sulphur 10 per cent. Practically all the gases, which may be 700 times the volume of the powder, arise from the nitre, the volume varying according to the density of the powder and to the conditions at burning.

Gunpowders do not detonate, but may give a very rapid rise of pressure if burnt in a closed space. Being porous, though dense, the first formed and very hot gases are driven through the remainder of the charge, igniting it *en masse*.

Cordite.

Cordites and the like are practically the most satisfactory propellants invented; owing to their gelatine-like structure they are non-pervious to gases. All true colloids show this property in some degree; the rate of burning can therefore be well regulated by the area of surface, volume of gas, and temperature. Owing to their composition, these explosives are practically smokeless. The cold-gas volume (i.e. volume at ordinary temperature) is in the region of 1,000 times that of the powder, and the temperature seldom less than 2,000° C., and may be above 3,000° C., according to composition. 'Cordite' proper consists of gelatinised high grade gun-cotton with a certain amount of nitroglycerine and a mineral jelly incorporated while the gelatinising agent is still present, the agent eventually drying off. A number of varieties of these powders are made with lower grades of gun-cotton, their physical properties being much the same. These powders are not easily detonated and are useless for blasting purposes.

Gun-cotton.

Gun-cotton' and 'Cellulose Nitrates' are made by the action of a cool mixture of nitric and sulphuric acids on cleaned cotton waste or the like. Strength of acids, temperature, and time of contact determine the extent and nature of nitration. Real gun-cotton contains about 13·8 per cent. of nitrogen, nearly corresponding to $C_6H_7O_2(NO_3)_3$.

Varieties of nitrated cellulose containing 10 to 11 per cent., or less, nitric-nitrogen than real gun-cotton are also used in some classes of propellants.

Gun-cotton is unaffected by water and does not burn if containing about 10 per cent. of moisture. When dry it ignites at between 160° and 180° C. It burns without smoke and is easily detonated by a sharp blow on a hard surface. When wet it can only be detonated by a primer of dry gun-cotton. When pulped and pressed into slabs, gun-cotton (13·8 per cent. N.) is used for blasting; its action is very local.

Nitroglycerine.

Nitroglycerine is formed by the action of mixed nitric and sulphuric acids on glycerine at a low temperature. It is a colourless oily liquid of sp. gr. 1·6. It freezes easily, and melts at 4° C. It is practically insoluble in, and unaffected by, water. Solid or liquid, it is easily detonated by percussion or by heating to about 150° C., hence the danger of thawing the solid nitroglycerine. As a liquid it is unpractical as an explosive, but when mechanically absorbed into a porous inert material, or incorporated in the substance of a jelly, useful preparations result.

Dynamite.

'Dynamites' which fulfil most of the requirements for blasting, and are reasonably safe, consist of nitroglycerine incorporated in kieselguhr, cork charcoal, and the like non-gritty powders. Generally some metallic nitrate, colouring or binding dope is added. Dynamites can be detonated by fulminate detonators. They are comparatively safe to handle and store for a moderate time in a place of medium temperature. Water may cause the extrusion of nitroglycerine from most forms of dynamite.

Gelignite.

'Gelignites' are mainly a thin jelly of a low grade gun-cotton in nitroglycerine; warm nitroglycerine is usually the gelatinising agent, but others may be employed. They may contain about 90 per cent. nitroglycerine. They are plastic, waterproof, and more powerful and convenient than ordinary dynamites. They should be stored in a place of cool even temperature. Most of the gelignites are doped, for some real or imaginary object.

Permitted Explosives.

'Permitted Explosives' for use in mines are such that comply more or less perfectly to certain tests, such as non-ignition of fire damp or coal dust, not producing a dangerous amount of poisonous fumes, considerable range of safety in storage and use. In many of these explosives, ammonia nitrate is the main oxygen carrier, the combustibles being benzene derivatives such as nitrobenzene, nitronaphthalene, and the like, with or without wood meal or inert powders to cool the flash.

Perchlorate of ammonia is the chief active material in some blasting compositions for civil or military use (cheddites). They are useful in fiery mines, as a quantity of hydrochloric acid gas is produced on explosion which acts as a damper of flame. Almost any kind of organic matters may be used in this class. Paraffin wax is often used, as, amongst other properties, it is waterproof.

Perchlorate compositions are stable and keep well, but all are liable to explode on moderately strong percussion. Generally they simply burn violently on ignition by a flame.

Ordinary chlorate compositions should be completely barred, as they are never safe and are erratic in use.

Detonators.

Detonators are substances employed to start the disruptive explosion or detonation of other compounds which alone may not explode by a flame or a moderately hard blow. Owing to the extreme rapidity of their explosion, detonators initiate a molecular decomposition of the main explosive, or it responds to the wave blow of the detonator.

Mercury fulminate, $HgN_2C_2O_2$, is the oldest and most commonly used detonator. It may be used alone or diluted with nitre, antimony sulphide, potassium chlorate, or even inert substances in some proportion, in order to regulate its sensitiveness, or reduce or modify the manner of exploding.

Cap and igniting compositions contain same proportion of $KNO_3Sb_2S_3$ and $KClO_3$.

Small quantities of fulminate, say 2 grams, may flash off if fired open, but the flimsiest cover will cause it to detonate on firing by a flame contact. A very slight suddenly applied pressure causes detonation. Its function in cap compositions is to respond to the blow of the hammer and ignite the other ingredients.

Lead azide, PbN_6, acts similarly to fulminate, is stronger and quicker in action, but probably not so amenable to regulation as fulminate.

Fuses.

These are, generally, some arrangement for firing an explosive at a desired moment. They consist, as a rule, of finely ground black powder contained in a paper tube protected by tape, machine-wound over it and waterproofed by some resinous or pitchy composition. Rate of burning may be controlled by fineness of powder, and in some cases by its composition, diameter of tube, etc.

In military fuses the powder has to burn along a circuitous route of determinable length in a metal container.

Fuses for blasting are packed in barrels and cases containing 300 or 500 coils of 24 ft. length. The average rate of burning of safety fuses is 75 to 90 secs. per yard. (See also 'Trinitrotoluene.') The fuse end is inserted in the detonator cap, which is then secured in proper contact with the explosive cartridge or charge, which may be tamped in the bore hole or not according to nature of explosive. (See also 'Blasting,' page 2572.)

Lyddite, or Picric Acid.

Lyddite, or picric acid, $C_6H_2OH(NO_2)_3$, is mainly a military explosive. It is somewhat difficult to detonate, otherwise there is no reason against its use for rock blasting. When some quantity is burnt there is a liability for it to pass into the detonation stage. The chief danger in connection with it is the liability to form metallic compounds (picrates), which are extremely sensitive to heat, friction, or percussion. There are several related substances with similar properties and uses, such as dinitrophenol, nitrocresols, etc.

Trinitrotoluene.

Trinitrotoluene (T.N.T.) and trinitroxylene (and other stages of nitration of toluene, xylene, etc.) are mainly military explosives, but T.N.T. has been used for some time in instantaneous fuses for civil work. Under proper conditions all these substances could be used for blasting.

Amatols.

Amatols (of various catagories) are essentially ammonia nitrate compositions with T.N.T. and similar substances. The violence of their explosion is increased by the addition of small quantities of powdered aluminium, magnesium, and other substances.

Amatols have a great future as blasting agents, but some of them are very sensitive and are not quite waterproof.

Blasting Machines.

Dynamo Electrical Blasting Machines are specially built to explode all blasting charges with absolute safety, either in stone quarries, coal mines, or oil shale workings. Being water and damp proof they are always ready for action.

SECTION XLII

WORKSHOP RECIPES, PROCESSES, ETC.

PAINTING — VARNISHES — METAL LACQUERS AND BRONZES — LUBRICANTS AND OILS — CLEANING-COMPOSITIONS — MISCELLANEOUS

SECTION XLII

PAINTING — VARNISHES — METAL LACQUERS AND BRONZES — LUBRICANTS AND OILS — CLEANING-COMPOSITIONS — MISCELLANEOUS.

(*All Parts by Weight unless otherwise Specified.*)

PAINTING.

Painting Woodwork.

A mixture of linseed oil and white lead is the best preservative of wood. The linseed is the preservative; the white lead gives body and opacity. Spirits of turpentine is added as a solvent, to enable the paint to penetrate into the pores of the material; the solvent should be light and easily evaporated.

Boiled linseed oil dries quicker than raw, but is darker and thicker than the latter.

To Remove Oil Paint.

2 lbs. washing soda, 2 lbs. quicklime, mixed to consistence of cream, spread on, and kept moist for an hour. The paint can then be washed off.

Glazier's Putty.

Whiting and raw linseed oil, well beaten up.

To Remove Old Putty.

Paint it with nitric or hydrochloric acid, and after about an hour it will be soft enough to remove; or a hot iron will soften it at once.

Painting.

In painting woodwork for the first time, kill knots with coating of fresh slaked lime; then in 24 hours scrape off, and paint knots with mixture of red and white lead and linseed oil; or use red-lead powder mixed with strong glue-size, and apply warm—the work can then be gone on with in a few minutes; when dry, smooth with pumice stone. Next put on priming coat. All holes are then stopped with glazier's putty, and the required coats added; each coat in good work is rubbed smooth with glass-paper before adding another.

Painting required to have no gloss has an additional coat, made with turpentine only, and no oil, called a *flatting coat*.

For old work, if much soiled, wash with soap and water and scour; if foul with smoke and grease, wash with lime and water; then commence with *second* coat.

New woodwork should have 4 coats; for periodical repainting 2 are sufficient.

Priming.

This generally consists of a mixture of white lead and red lead, irrespective of the colour to be used in finishing. The lead mixture is diluted with boiled linseed oil and a small quantity of turpentine. A special asbestos priming for painting fibro cement and all classes of asbestos sheeting is supplied by *Messrs. Walter Carson & Sons*, Grove Works, Battersea, London, S.W. 11.

'Muraprime.'

'Muraprime' (*Walter Carson & Sons*) fills a long-felt want for a decorative and reliable weatherproofing paint for rough cast, cement and stucco work. It is specially recommended for porous plaster, brick and cement surfaces, rendering them damp-proof and ensuring dry inside walls. As an undercoating where it is proposed to use paint, enamel or paper, 'Muraprime' is invaluable owing to its property of holding back stains and resisting damp. It is made in white, cream, yellow, green, and pink; other colours are supplied to order.

'COVERINE.'

This paint (*Walter Carson & Sons*, Grove Works, Battersea, London, S.W. 11) is a white undercoating, which saves time in the preparation of undercoats. It is ready for use in five minutes, and requires to be thinned with turpentine only. One coat is equal to two coats of white lead paint, and is consequently a saving of time and material. It dries hard, and can be coated in a few hours. The finishing coat can be applied and the work completed in one day. It forms a good undercoating for gilding, and will stand washing.

'MURALINE.'

This is a washable water paint (*Walter Carson & Sons*), and is sold in dry powder, in forty artistic shades. Water added to the powder in an equal proportion makes a sanitary washable paint, which forms a highly perfect and durable surface. Compared with paste distempers (which contain about one-third of water), it is claimed to effect a considerable saving, owing to its great covering power. 1 cwt. covers 600 to 700 square yards on non-porous surfaces. It will never deteriorate if stocked in a dry place, unexposed to the air. 1 cwt. of powder with 1 cwt. of water makes 2 cwt. of paint. It can be applied to new lime plaster, if dry, without injury to the colours. It works and flows evenly under the brush, and leaves a hard, smooth surface. Intermediate shades may be produced by mixing colours or by reducing with white. It is non-poisonous and an excellent disinfectant. The colours are fast to light, and do not discolour. It does not rub off, scale, or flake. The white is an excellent covering for ceilings. Unlike whitewash, removal is unnecessary before repainting if in sound condition.

Painting Ironwork.

For new ironwork, 3 coats Torbay or other oxide of iron paint answers well; 1 lb. mixed with oil will cover from 8 to 12 yards. For periodical painting, 2 coats. For the first, or first and second coats on iron, red lead is very generally used; 1 lb. mixed with oil will cover about 5½ yards.

'FERRI-SILICON.'

This paint (*Walter Carson & Sons*), which can be made in any colour, is a cheap and durable preservative for ironwork, woodwork, and all exposed surfaces. It is a combination of iron and other oxides with silicates, which, when incorporated with a special enamel oil, produces a paint closely resembling enamel, but with a very tough and elastic surface, absolutely impenetrable. The manufacturing process which the enamel oil undergoes is claimed to render it at least twice as durable as the finest linseed oil. The paint is specially suitable for use on gasometers, as the colours are not affected by contact with sulphuretted hydrogen gas, sulphide of ammonia, etc.

'ANTI-CORROSION.'

This paint (*Walter Carson & Sons*) is for out-door work as a preservative for iron, wood, stone, brick, compo, etc. It is employed by many of the leading railways both at home and abroad, and also on life-boat stations. It is well adapted for coating the roofs of corrugated iron buildings, etc., and is especially recommended for buildings exposed to the corrosive action of sea air as it does not blister and wash off under these trying conditions.

SYRONITE ANTI-RUST PAINT.

Syronite is a thin transparent syrup which can be painted upon new ironwork before, during, or after erection, or upon existing structures that are already rusting. If the structure is in a bad state before the Syronite treatment is undertaken, the surface is first well brushed with a stiff wire brush to remove all loose rust. A coat of Syronite is then applied, its action being to penetrate through any remaining rust and cause this to shed off. When shedding has taken place, another coat is applied, and then the treatment is complete.

Pigments can be combined with Syronite when required.

A peculiar feature of the material is its capacity of 'creeping' or spreading. Ordinary paint stays exactly where the brush leaves it, and as the brush cannot always get under and along joints, round rivets, etc., these parts do not get equal protection with the general surface.

Preservative for Ironwork.

(1.)

Coal-tar boiled till the watery vapour and naphtha are expelled, and mixed when cold with ½ pint of naphtha and the same amount of boiled oil to each gallon of tar.

(2.)

Experiments carried out by the Military Engineering Department at Brest have shown that as an effective preservative for metal work coal tar is rendered far more efficient by the addition of a small quantity of quicklime, in the proportion of one part of lime to two parts of tar. The mixture, after being heated to the boiling point, is brushed upon the ironwork at a temperature of about 120° F. The tar should be used over a coating of red lead.

(3.)

A valuable rust-inhibitive coating for general priming or finishing work may be prepared from sublimed blue lead. Two parts of blue lead and one part of linseed oil containing about 5 per cent. of turpentine drier make a paint of the right consistency. When this paint is used for top-coat work in marine exposures—battleship gray—the addition of 1 per cent. of carbon black to the blue lead aids in the maintenance of the colour.

Rust-proof Coating for Iron or Steel.

Linseed oil, 25 parts; calcium resinate, 36 parts; manganese borate, ½ part; lead acetate, 1 part; naphtha, 37½ parts; artificial graphite, 25 parts. These are mixed and applied to the metal by brushing, or dipping, or any other method. The article is then baked at 300° F. for 1 hour and 40 minutes. The coating is highly lustrous and resistant to corrosion.—*American Machinist.*

Note on Ironwork Painting.

According to experiments published in the *Bulletin de la Société des Ingenieurs Civils de France*, a number of coats of paint form a much less effective rust preventive for metal than a single layer does. Iron plates well cleaned and polished were coated, some with one coat and some with two, three, or more coats of paint, and then exposed to the action of steam for a whole day, after which the paint was removed and the plates were examined. The first was as bright as it was before exposure to steam, the second showed a few rust spots, the third was still more attacked, and the fourth entirely so. This result is stated to be due to the fact that the deeper layers dry badly and tend to dissolve the uppermost coating, increasing its porosity to air and moistare.

Painting Zinc.

Dissolve 1 part chloride of copper, 1 nitrate of copper, and 1 sal ammoniac, in 64 of water, and add 1 part commercial hydrochloric acid. Paint the zinc with this, and leave to dry for 24 hours. Any oil colour will then firmly adhere to it.

To Paint in Imitation of Ground Glass.

White lead, 4 parts; boiled oil, 3 parts; spirits of turpentine, 1 part; add a small quantity of dryers. The colour must be exceedingly thin, and put on the panes of glass with a large brush in as even a manner as possible. Dab the paint over lightly with a pad of cotton-wool covered with an old linen rag.

Hydraulic Cement Paint.

Hydraulic cement mixed with oil forms an incombustible and waterproof paint for roofs of buildings, outhouses, etc.

Outside Whitewash.

Slaked lime, ½ bushel. Add common salt, 1 lb.; sulphate of zinc, ½ lb.; sweet milk, 1 gallon.

Flexible Paint.

Used for painting on canvas. 2½ lbs. good yellow soap dissolved in 1½ gallon boiling water; grind the hot solution with 3½ gallons of good oil paint.

Red Paint, quick drying (8 to 20 Minutes).

(Used for Pillar-boxes.)

Japanner's gold size	40 per cent.
Red lead	60 „

When dry, varnish with outside oak varnish.

Waterproof Paints.*

For General Purposes.

(1.)

2 gallons linseed oil, boiled with 11 ozs. each of rosin and litharge, 1½ oz. each of minium and umber; add gradually 8 ozs. of sulphate of zinc and a solution of 12 ozs. of potassium hydrate, and a like quantity of alum in 4 gallons of water. For preparing the ground colour, compound 10 lbs. of chalk and 2 lbs. of zinc white with ¾ gallon of water in which 1½ oz. of alum has been previously dissolved, and mix with this a solution of 1½ lb. of glue in ½ gallon of water. This ground paint is mixed with 4 to 5 lbs. of the above composition, and the mixture diluted with petroleum.

(2.)

12 ozs. of melted rosin, thoroughly mixed with 6 gallons of fish oil and 1 lb. of melted sulphur, and some ochre or any other colouring substance rubbed up with linseed oil. Apply several coats of the hot composition with a brush. The first coat should be very thin. Suitable for either wood or stone.

For Casks and Waterspouts.

Boiled oil, 1 part; charcoal, finely powdered, 1 part. Mix to consistence of paint.

For Vessels and Submarine Works.

Sulphate of copper, 400 lbs.; grape sugar, 100 lbs.; concentrated solution of potash, 200 lbs. The precipitate of hydrated oxide of copper formed by heating this mixture is filtered, dried, and mixed with 8 lbs. of 75 per cent. carbolic acid. Heat the mass moderately, and add about 12 gallons of crude linseed oil. When the paint is to be used it is reduced with linseed oil, and then applied. This composition is claimed to prevent the deposition of marine animals on the surface to which it is applied.

Paints for Waterproofing Outside Walls.

(1.)

Rosin, 4½ ozs.; litharge, 3 ozs., boiled in 1 pint of clear linseed oil until the wooden spatula used for stirring becomes brown. Give the walls 2 or 3 coats of this, preferably on a hot and dry summer's day.

(2.)

A solution of paraffin wax in heavy coal-tar oil is excellent for painting houses, and especially walls exposed to the action of the weather. Several experiments in painting damp walls with this solution have given very satisfactory results. Wall-paper which formerly became moist and detached from the walls during rainy weather remained perfectly dry after the wall had been painted with this solution, which is prepared by dissolving 1 part of paraffin wax in 2 to 3 parts of coal-tar oil at a moderate heat. A sufficient quantity of oil must be used so that the solution does not entirely congeal on cooling. To heat the paint while applying it, place the vessel containing it in hot water. It is best to apply the solution on a warm day, when the bricks are dry. Generally 1 coat is sufficient, but even if 2 coats are given the cost is considerably less than that of oil paint.

Paint for Wooden Floors.

Soak 2 ozs. good light-coloured glue in cold water for 12 hours, and dissolve it with constant stirring in thick milk of lime (prepared from 1 lb. of caustic lime) heated to the boiling-point. Stir in 8¾ ozs. linseed oil. Mix with any colour not affected by lime, and dilute with water if necessary. For yellow-brown or brown-red colours, boil in the ground colour ⅓th of its volume of a solution of shellac and borax. A coat of varnish produces a bright surface.

Paint for Red Brick Floors.

Wash the bricks with soap-water containing $\frac{1}{20}$th part of carbonate of soda. Dissolve 1 part of glue in 16 of boiling water and 4 parts of red ochre, and stir the mass thoroughly together. Apply 2 coats of this to the bricks, and then give a coat of linseed oil varnish.

---

* It has been proved that no paint film, of which linseed oil is one of the chief constituents, can be made *thoroughly* impervious to moisture.

Preservative Solution for Stonework.

1 part white shellac, dissolved in 8 parts wood spirit. Apply to surface by means of a watering pot provided with a rose.

Paint for Roofs.

Pulverised slate (argillaceous schist), 35 parts; pulverised mica slate (mica schist), 30 parts; pulverised rosin, 35 parts; mix with ½ its volume of pure coal-tar, and boil to a fluid mass. This paint gives a very durable and pliant coating, which does not melt in the greatest heat of summer nor crack or break in the greatest cold. It resists moisture, and retains its lustre and smooth surface. It will last for 4 or 5 years.

Paints for Preserving Wooden Posts.

(1.)

50 parts rosin, 40 parts powdered chalk, 4 parts linseed oil; melt together in an iron boiler, add 1 part of natural cupric oxide, and stir very carefully into the mixture 1 part of sulphuric acid. Apply with a stiff brush. When dry the mass forms a coating as hard as stone.

(2.)

1½ part rosin (melted), 48 parts fish oil, 1½ part sulphur. When the mixture is thoroughly combined and liquid, add sufficient ochre rubbed up with linseed oil to give the desired shade of colour. Apply the mass while still warm in as thin a coat as possible, and in a few days, when the first coat is dry, repeat the operation.

Paint for Felt Roofing.

Washed graphite, 2 parts; red ochre, 2 parts; cement, 16 parts; barium sulphate, 16 parts; plumbic oxide, 6 parts. These materials to be ground in an oil varnish prepared as follows: 100 parts of good linseed oil are boiled for 8 hours in a copper boiler with 5 per cent. of pyrolusite, 10 parts of flowers of sulphur and 20 parts of French pitch are then dissolved in the mixture, and the mass filtered before it becomes cold. 25 lbs. of this composition and 1½ gallons of linseed oil varnish or linseed oil, for reducing the paint, are sufficient to give 2 coats to 1,000 square feet of roof surface. The first coat, while still wet, is uniformly covered with fine dry sand by means of a sieve. The sanding should be done during the progress of the painting, so that the workman is not obliged to step on to the wet paint. The free sand is renewed with a broom before the second coat is given, and it is best to apply this 8 days after the first. The second coat need not be sanded, its purpose being to combine with the first to form an insoluble mass hard as stone, and to give to the roof a neat, finished appearance.

Quickly-drying Oil Paint.

Boil for 15 minutes in an earthenware pot 1 part of soft curd soap in 3 parts of water. Pour the mass through a colander, wash it with cold water, and press out the water in a linen cloth. To 1 part of the curd soap add ¼ part of unslaked lime and ¾ part of water. Triturate the fat slime thus formed in oil or water with the necessary pigments. Walls, ceilings, stairs—in short, anything of stone, plaster of Paris, or zinc—can be painted with this. If the paint is to be used on wood, add $\frac{1}{10}$ part of linseed oil. Ochre, chrome yellow, Berlin blue, indigo, lead, and zinc, are best adapted for colouring substances. The mixture dries so quickly that 3 coats can be applied in a day. It is entirely without odour, and is about one-third the cost of ordinary oil paint.

Heat-Proof Paint.

Paint with good heat-proof properties, for smoke boxes, etc., can be made from 3 lbs. lamp black, 2 lbs. black lead, 1 lb. of black manganese oxide; these are mixed together; add ½ gal. of gold size, 1 pint of turpentine, and ½ pint of terebene; this, when thoroughly mixed in, will bring the mixture to the consistency of paint, and if about two coats of this are applied to surfaces exposed to heat, they will be found to resist same very well.

Fire-resisting Paint.

(1.)

20 lbs. finely pulverised glass, 20 lbs. finely pulverised porcelain, 20 lbs. any sort of stone in powder, 10 lbs. calcined lime, and 30 lbs. water glass (silicate of soda). The solid elements, having been powdered as finely as possible and sifted, are moistened and then intimately mixed with the water glass. This yields a syrupy mass that may be employed for painting, either alone or mixed with colour. The coating is applied with a brush, as other paints are, as uniformly as possible over the surface to be protected. The first coat hardens immediately, and a second one may be applied 6 hours or more afterwards; two coats are sufficient. The paint must be used immediately after it is made up.

(2.)

The most practical means so far discovered by the United States Forest Products Laboratory, Madison, Wis., for economically rendering small quantities of wood fire-resistant consists in the

use of fire-retardant paints. Ordinary whitewash or calcimine has proved in tests to be as fire-resistant as any paint covering tried, and it also possesses the advantages of cheapness and ease of application. While it will not prevent the burning of wood continuously exposed to a high temperature, a good coat of calcimine will decrease the likelihood of a blaze spreading from small sources of fire, such as flying sparks and matches. Calcimine is naturally more effective for interior than for exterior use. For work of the latter class the Forest Products Laboratory has developed an effective paint composed of linseed oil, zinc borate, and chrome green, which has maintained its fire-resisting properties through more than three years of exposure to the weather.

Heat-indicating Paint.

A paint for indicating an excessive rise of temperature in machinery bearings is composed essentially of a mixture of the double iodide of silver and mercury and copper mercuric iodide. The former is a pale yellow powder at normal temperatures, but turns to a vivid carmine at 90° to 100° C., while the copper salt, which is normally of a strong scarlet vermillion, changes to a chocolate brown at 60° to 70° C. A mixture of the two in the proportion of 85 per cent. of the copper salt and 15 per cent. of the silver changes promptly from vermillion to nearly black at 60° to 70° C. The paint is made with the aid of white shellac spirit varnish (100 grms. of the mixture to 70 cub. cm. of the spirit), and is applied in the form of a band or bull's-eye to the bearing. After the paint has become dry it is to receive a coat of colourless oil-proof varnish.

See also p. 962.

Painting Portland Cement.

Portland cement work which is to be painted must be thoroughly hardened and dry. It is advisable to let the work stand for a year before oil paints are applied. The durability of the paint will be assured by, first, brushing over the surface with dilute sulphuric acid—one part of acid to 100 parts water—and allowed to dry before painting. A preparatory coating for oil paint is a solution of common water glass in three or four parts of water. Two applications of this, followed by a washing with water, and then another application of water glass, will be found effective.

Coal-Tar Paint.

(1) Coal-tar, 8 parts; Portland cement, 1 part; kerosene oil, 1 part. (2) Coal-tar 16, Portland cement 4, and kerosene 3 parts. The Portland cement neutralises the action of the tar on iron and steel, while the kerosene acts as dryer. It is suitable for painting ironwork of all kinds, and is a satisfactory coating, it is said, for ships' sides, and may also be used on galvanised iron.

Graphite Paint.

The best and longest records of satisfactory protection against corrosion are said to be held by graphitepaints containing silica in intimate admixture with tough crystalline foliated graphite.

An example of this is the 'Foliac' Silica-Graphite Paint manufactured by *Graphite Products, Ltd., Battersea*, which is British manufactured throughout, and has been specified for a large number of steel structures, including Admiralty airship sheds, oil fuel tanks, gas holders, docks, bridges, station roofs, railway wagons, &c.

The relative covering power of foliated silica graphite paint is very high, and owing to the lubricating properties of the pigment the speed of application to the surface is very much more rapid than with other linseed oil paints. Owing to the large adhesion surface between the vehicle and the foliated graphite a very flexible tough paint is obtained which stands wearing action due to dust, for example, much better than paints which become brittle on drying. The resistance of the pigment to the action of the atmosphere found in industrial districts is due, of course, to the fact that it is chemically inert.

Bituminous Paints.

Bituminous paint is a good preventive to rust as it adheres closely to the material to be protected, it is strongly resistant to sulphurous vapours prevalent in the air of industrial centres, and very few of the mineral acids attack it, with the exception of nitric acid. Nitrous fumes have little or no action on the better bituminous paints. A good bituminous paint should dry within four or five hours, and the chances of injury by rain or similar eventualities during the setting are, therefore, correspondingly small. A high-class bituminous paint may be applied at a cost of something like a farthing per square foot with two coats of the paint applied. The life of a well-compounded bituminous paint extends up to twenty years' efficient protection.

Luminous Paint.

Mix 50 parts of lime with 20 parts of flowers of sulphur, heat them together in a closed crucible until all fumes cease to be evolved. When quite cold grind the product up with varnish or paint which is quite free from lead. Luminous paint produced from oyster shells is the most efficient in retaining its luminosity after dark. Take clean shells and, after washing them in warm water, place them in the fire for half an hour, and when cool, after removal, pound them up very small and fine, taking away any grey portions as being of no use, then put this powder in a crucible with alternate layers of flowers of sulphur, place on the lid and cement it round with sand made into a stiff paste with beer; as soon as this is dry, place the crucible in the fire and bake for one hour. Wait until quite cold before opening the lid. The product should be white; any grey portions must be separated as non-luminous. Sift it through fine muslin, and mix with either varnish or gum water. For use with buoys or in places exposed to the weather, varnish of the best quality only must be used.

VARNISHES.

(Incombustible Varnish for Wood.

An application of a solution of equal parts of alum and isinglass to the place exposed to the flame prevents ignition, but not the transmission of heat. By coating wooden vessels with this varnish fluids may be boiled in them over an ordinary fire.

Varnish for Wood not acted upon by Boiling Water.

Boil in an untinned copper boiler 750 parts of linseed oil. Suspend in this, in a bag which must not touch the bottom, 150 parts of litharge and 90 parts of pulverised minium. Let the oil boil until it has assumed a dark brown colour; then remove the bag, and replace it by one containing 7 to 8 bulbs of garlic. Now melt 500 parts of pulverised amber in 60 parts of linseed oil over a strong fire; add it while boiling to the prepared linseed oil, and let it boil for 2 to 3 minutes longer, stirring it vigorously. Then take it from the fire, allow it to settle, pour off the clear liquid, and when cold put it in bottles, which should be hermetically closed.

Quick-drying Varnish.

A quick-drying varnish that is less liable to crack than that made from kauri gum is made in the United States from the nuts of the Chinese wood oil tree (*Aleurites fordii*). The tung (or wood) oil has also been found of special value for waterproofing cement.

Varnish to prevent India-rubber Gas-tubing from Leaking.

Treacle, 1½ part; gum arabic, 2 parts; white wine, 7 parts; strong alcohol, 3½ parts.

Waterproofing Material.

A new French material for waterproofing leather, cloth, paper, and other materials, consists of a mixture of eight parts of amyl acetate and four of castor oil, with one part of sulphur chloride stirred into it. This forms a jelly, giving off hydrochloric acid; it liquefies after being kept tightly covered a few days, and, the acid being then neutralised with barium carbonate, gives a colourless solution on filtering. To make a varnish, nitrocellulose is dissolved in the liquid after the addition of alcohol or benzine

Cellulose Lacquers and Varnishes.

Cellulose lacquers and varnishes are extensively used for motor-cars and other purposes where a brilliant finish coupled with great resistance to weather is desired. They are made by dissolving nitrocellulose or cellulose acetate in amyl nitrate or acetate, castor-oil, or other suitable solvent, and may be applied with a spray or a brush, according to the nature of the lacquer and the surface to be decorated.

Cellulose lacquers have been used for twenty-five years or more, but their great popularity is of recent growth. In many instances they have replaced oil enamels, but in numerous other cases the reverse has occurred. For motor-cars, the American cellulose lacquer finish is superior to the American oil-varnish finish, but inferior to the English oil-varnish. English cellulose lacquer finish is superior to the American product and approaches, but is not quite equal to, English oil-varnish. In appearance the cellulose lacquer finish and the oil-varnish finish are almost equal, and it is a question of workmanship rather than material which gives the advantage, sometimes to one and sometimes to the other. The relative durability remains to be proved, short time tests being unreliable. Cellulose lacquer is more liable to crack under vibration, but is more resistant to changes in temperature, and whilst it is harder than oil-varnish it is more brittle and more liable to peel.

The cost of cellulose lacquer is greater than that of oil-varnish, but in large quantities it is cheaper to apply, so that the cost of covering a given area is less with cellulose than with oil-varnish. Although the use of cellulose lacquers is increasing rapidly, these materials have not superseded oil-paints, water-paints, and stove-enamels. They are limited in scope, cannot be repaired so readily when damaged, and many of them are inflammable and produce vapours injurious to the health of those applying them. These are temporary disadvantages which will probably disappear as still further improvements are made in cellulose lacquers. In selecting such lacquers care should be taken to avoid those of poor quality, of which there are too many available.

METAL LACQUERS AND BRONZES.

Lacquer for Iron and Steel.

Mastic, 10 parts; camphor, 5 parts; sandarach, 15 parts; elemi, 5 parts. Dissolve in alcohol and apply without heat.

A Rustless Coating for Steel.

A solution of white wax in benzine applied with a camel-hair brush. When the film of wax hardens, the bulk of it is wiped off with a chamois leather, just sufficient remaining to prevent rust, but not to be noticeable.

Angus Smith's Coating for Pipes.

Coal Tar 1 cwt., Tallow 7 lbs., Quick Lime (slaked) 10 lbs., Pine Resin 4 lbs., Coal Naphtha, sufficient to thin the composition to a degree suitable for proper coating. Heat mixture, and immerse pipes, and allow to remain until the same temperature is diffused through the mass; then withdraw pipes. This treatment leaves a very firm and hard coating of pitch. It is very essential that the mixture be heated to a proper degree, viz., 350° to 450° F. If heated too much the pitch will not adhere properly, but will scale off.

Bronzing Liquids.

(1.)

Sulphate of copper, 1 oz.; sweet spirits of nitre, 1 oz.; water, 1 pint.

(2.)

Solution of chloride of platinum, laid on surface with a feather. When dry, blacklead over with an ordinary blacklead brush, and then lacquer.

(3.)

Boil 1 oz. of ammonium carbonate and a like quantity of blue vitriol in 1½ pint of vinegar until the latter is nearly evaporated. Then add 1½ pint of vinegar in which has been dissolved ½ drachm of oxalic acid and a like quantity of sal ammoniac. Place the mixture (known as Walker's Chemical Bronze) over the fire until it commences to boil, then allow it to cool; filter, and put by in well-closed bottles.

If a medal, etc., is to be bronzed, it is first thoroughly cleansed, then heated, and the liquid applied by means of a badger-hair brush. In a short time boiling water is poured over the medal, and when dry it is rubbed with a cotton rag dipped in oil, and then with dry cotton.

Browning for Gun Barrels.

Tincture of muriate of iron, 1 oz.; nitric ether, 1 oz.; sulphate of copper, 4 scruples; rain-water, 1 pint. When the barrel is finished let it remain in lime-water for a short time; then scratchbrush and apply bronzing.

Lacquer for Gun Barrels after Browning.

Shellac, 1 oz.; dragon's blood, ½ oz.; rectified spirit, 1 quart. Dissolve and filter.

To Chemically Blacken Brass.

Dissolve copper in nitric acid (out of doors) until all action ceases; when cold decant into another jar.

Sand-blast work, or clean in hot lye and soda; dry in sawdust; heat to 212° F.; immerse in the (cold) nitrate solution; remove and heat again sufficiently to dry off solution and burn off green colour; repeat if necessary; cool and free from oxide by brushing; dip in ammonia; rub dry with soft cloth or in sawdust—work is then brown. To make it dead black, heat just enough not to burn hands, and rub with soft leather and fine olive oil; heat, to dry oil.

Highly-polished work, if well cleaned and dipped in commercial ammonia before applying nitrate, will take a permanent black.

Sand-blasted work takes a black that cannot be removed by ordinary wear.

Black-bronzing Aluminium.

(1) Immerse the articles, after being thoroughly cleaned, in a hot solution containing 8 oz. of white arsenic, 8 oz. of iron sulphate, ¾ gallon of hydrochloric acid, and ¾ gallon of water. When sufficiently blackened the articles are removed, washed and lacquered. A more adhesive deposit may be obtained by adding to the solution a small quantity of ammonium molybdate and yellow prussiate of potash. (2) Paint the articles with a solution made by dissolving 60 grammes of copper chloride in 300 c.c. of water and add ammonia in excess to re-dissolve the precipitate first formed. Then heat the articles, care being taken not to melt the aluminium. A deeper black may be got by repeating the process, or by dipping the articles in a solution of sodium sulphide.

Black Finish on Steel Shears, etc.

Prepare a saturated solution of caustic soda and add about ¼ lb. of saltpetre to a five-gallon solution. Boil the solution for a short time, and allow it to cool. The clear liquid should only be used, and this should be brought to the boiling point in an iron kettle. The articles should be wired as for plating and immersed in the solution. The articles will take on a grey colour at once with black underneath, and should be left in the solution until the grey finish disappears and a beautiful black remains. Rinse the articles in cold water, dry in sawdust, oil with linseed oil, and wipe clean. The resulting finish will be a fine blue-black that wears well.

LUBRICANTS AND OILS.*

Greases for Railway Axles.

(1.)

Water, 1 gallon; palm oil, 6 lbs.; clean tallow, 3 lbs.; common soda, ½ lb.

(2.)

Tallow, 8 lbs.; palm oil, 10 lbs. Heat to 212° F., and well stir.

Antifriction Grease.

(1.)

Tallow, 100 lbs.; palm oil, 70 lbs. Boil together, and when cooled to 80° F. strain through a sieve and mix with 28 lbs. common soda and 1½ gallon of water.

(2.)

Blacklead, 1 part; lard, 4 parts.

Rosin-Oil Soap.

90 lbs. powdered slaked lime, stirred into 100 lbs. rosin oil. Heat the mixture, constantly stirring it well until a uniform paste of the consistence of syrup is obtained. This rosin oil is a component of all the patent wagon greases.

Blue Patent Grease.

2 lbs. powdered slaked lime, heated in 550 lbs. of crude rosin oil for 1 hour, and then allowed to cool. Skim the oil, and stir in 10 to 12 lbs. of rosin-oil soap, until the compound is of a butter consistency and of a blue colour.

Yellow Patent Grease.

1 part turpentine, 20 parts caustic lye. Add to 350 parts of blue grease.

Black Patent Grease.

2 parts lampblack, 110 parts blue grease.

Patent Palm-Oil Wagon Grease.

Melt 10 lbs. rosin-oil soap in 10 lbs. palm oil: add 550 lbs. rosin oil, with as much rosin-oil soap as will give the whole the consistence of butter; finally, add 7½ to 10 lbs. of caustic soda lye.

Adhesive Grease for Machine Belts.

Lubricate the belts with fish oil to which 10 per cent. of tallow has been added. This will make them flexible, and augment their friction on the pulleys.

Grease for Waterproofing Leather.

Oleic acid, 24 parts; crude stearic acid, 6 parts; ammoniacal soap, 18 parts; extract of tannin, 6 parts; water, 24 parts. Melt the oleic acid together with the stearic acid; then add gradually the ammoniacal soap, the extract of tannin, and finally the water. The ammoniacal soap is obtained by adding to heated oleic acid caustic ammonia until, after continued stirring, the odour of ammonia remains apparent and the whole congeals to a jelly-like mass.

Cooling Compound for Hot Bearings.

(1.)

Mercurial ointment mixed with black cylinder oil and applied every quarter of an hour, or as often as expedient. The following is also recommended as a good cooling compound for heavy bearings:—Tallow, 2 lbs.; plumbago, 6 ozs.; sugar of lead, 4 ozs. Melt the tallow with gentle heat, and add the other ingredients, stirring until cold. For lubricating gearing, wooden cogs, etc., nothing better need be used than a thin mixture of soft soap and black lead.

(2.)

For cooling a hot bearing Professor Lecocq, of the Charleroi Higher Industrial School, recommends that the oil usually employed for lubrication be dissolved in an equal quantity of petroleum spirit, the evaporation of which latter will, he says, afford the cooling desired.

---

* See also pages 977-992.

Watchmaker's Oil.

Expose the best olive oil to a temperature a few degrees below the freezing point, which will cause all foreign substances to separate. Pour off carefully the supernatant clear oil, and filter through a cup of linden wood or pith of elder wood. By this process an oil is obtained which will remain liquid for several years, and does not attack the delicate machinery.

Neat's-foot oil treated in the above manner furnishes a less useful oil, since it loses much fatty matter by exposure to cold.

A very useful oil is obtained by dissolving 1 part of pure neat's-foot oil in 3 of pure benzine. Allow the compound to remain for several days in a closed vessel; then filter, and expose the solution to a temperature of 40° F., at which it is again filtered, and the benzine distilled off. The oil should be kept in small, dark vials protected from the air.

A very fine lubricant for clocks and watches is, according to Artemus, obtained by mixing 2 parts of solar oil with 1 of rape-seed oil.

To test the fitness of Oils for Lubricating Watches and Clocks.

Pour a drop of the oil to be tested upon different metal plates, as iron, brass, tin, lead, etc.; keep them in a place free from dust, and examine the drops during 8 to 14 days in regard to their liquidity. Oil remaining liquid after the lapse of this time can be safely used.

Viscosity of Various Standard Oils.

Description.	Viscosity.	
	At 70° F.	At 120° F.
Sperm oil	100	45
Whale oil	190	70
Olive oil	213	75
Lard oil	225	78
Neatsfoot oil	247	80
Rape-seed oil	250	88
Axle oil	600	120
Valve oil	—	550
Cylinder oil	—	700

In the above table sperm oil is taken as the standard for viscosity at 70° F.

The viscosity of thick oils can be greatly reduced by adding 8 per cent. of naphthalene to the oil.

An objectionable characteristic in an oil is its tendency to become resinous with use, due to the oil undergoing the process of oxidation. This change in the oil indicates, to some extent, the degree of refining process the oil has undergone. A simple test can be made in the following way:—Place a small quantity of oil in a glass beaker and put in a nearly equal amount of nitric acid. Should the oil have no tendency to gum, the acid and oil will combine and get very thick, while should there be any objectionable characteristic the oil will remain thin.

CLEANING AND PRESERVATIVE COMPOSITIONS.

To clean Brass.

Oxalic acid, 1 oz. Rotten stone, 6 oz.; powdered gum arabic, ½ oz.; sweet oil, 1 oz. Rub on with piece of rag.

Soap for cleaning Metal-work.

The soaps used for cleaning metal-work usually consist of mixtures of vaseline, oleic acid, and fat, mixed with a small quantity of rouge. When freshly prepared they leave nothing to be desired; but, unfortunately, such mixtures soon turn rancid and become unfit for use. A newer soap for metal-work, which is stated to be free from this objection, is made from cocoanut butter in the following way:—2·5 kilogrammes of the butter are melted in an iron vessel, together with a little water, and to the mixture is added, with constant stirring, 180 grammes of chalk, 87·5 grammes of alum, 87·5 grammes of cream of tartar, and 87·5 grammes of white lead. This mixture is then poured into moulds and allowed to solidify. The soap so obtained is made into a paste with water, rubbed over the metal to be cleaned, and finally removed by a dry rag or chamois leather.

To clean Grease from Paper.

(1.)

Warm the spotted part, and press blotting-paper upon it until all the grease is extracted that is practicable by this method; then with a brush apply very hot turpentine, and wash off with alcohol.

(2.)

Apply benzine collas freely to the spot.

To clean Marble.

Powdered chalk, 1 part; powdered pumice stone, 1 part; soda, 2 parts. Mix with water. Paint over marble; then clean, and wash off with soap and water.

To clean Grease from Stone or Marble.

Soft soap, 1 part; fuller's earth, 2 parts; potash, 1 part. Mix with boiling water. Lay it on the spots, and let it remain for a few hours.

A Solvent for Rust.

It is often very difficult, and sometimes impossible, to remove rust from articles made of iron. Those which are very thickly coated are most easily cleaned by being immersed in a solution (nearly saturated) of chloride of tin. The length of time they remain in this bath is determined by the thickness of the coating of rust. Generally twelve to twenty-four hours is long enough.

To cleanse Paint Brushes from Dried-in Paint.

Suspend the brush in a tumbler containing a solution of 1 part of crystallised sodium carbonate in 3 of water, in such a manner that it hangs several inches from the bottom of the tumbler. Let it stand from 12 to 24 hours in a moderately warm place (140° to 150° F.). The dried paint will be softened so much that it can be easily washed out with soap and water. Brushes which have become as hard as stone can be restored by this process.

To remove Rust from Iron.

(1.)

Freely apply turpentine or paraffin oil, leave for twelve hours, and then rub with the finest emery cloth.

(2.)

Dip in strong hot potash bath, for about ½ hour. Then immerse in a cold muriatic acid pickling solution—2 parts water, 1 acid. This removes rust in a few minutes, the metal being attacked but very little. Without the potash it will take about 65 minutes against 4 when both baths are used. The appearance of the rust when treated is black and soft and it can almost be entirely rubbed off with the hands. The metal so treated has a smooth, glossy finish.

(3.)

A dip of 40 parts of muriatic acid and 60 parts of water is effective for removing rust. After the rust has been removed, it is necessary to dip the steel in a strong alkaline bath to remove every trace of acid. The alkaline solution usually employed for this purpose is composed of 2 oz. each of soda ash, sodium cyanide, and caustic lime.

To Remove Rust from Iron Electrically.

A very simple and effective way of cleaning rusted iron articles, no matter how badly they are rusted, consists in attaching a piece of ordinary zinc to the articles, and then letting them lie in water to which a little sulphuric acid is added. They should be left immersed for several days, or a week, until the rust has entirely disappeared, the time depending on how deeply they were rusted. If there is much rust, a little sulphuric acid should be added occasionally. The essential part of the process is that the zinc must be in good electrical contact with the iron; a good way is to twist an iron wire tightly around the object, and connect this with the zinc, for which a remnant of a battery zinc is suitable, as it has a binding post. Besides the simplicity of this process, it has the great advantage that the iron itself is not attacked in the least as long as the zinc is in good electrical contact with it. When there is only a little rust, a galvanised iron wire wrapped around the object will take the place of the zinc, provided the acid is not too strong. The articles will come out a dark grey or black colour, and should then be washed thoroughly and oiled. The method is specially applicable to objects with sharp corners or edges, or to files and other articles on which buffing wheels ought not to be used. The rusted iron and the zinc make a short-circuited battery, the action of which reduces the rust back to iron, this action continuing as long as any rust is left.

To keep Machinery from Rusting.

Take 1 oz. of camphor, dissolve it in 1 lb. of melted lard; mix with it (after removing the scum) as much fine blacklead as will give it an iron colour. Clean the machinery, and smear it with this mixture. After 24 hours rub clean with soft linen cloth. It is stated that this mixture will keep machinery clean for months under ordinary circumstances.

To protect Iron and Steel from Rust.

The following method is but little known, although it deserves preference over many others: Add $1\frac{3}{4}$ pints of cold water to 7 ozs. of quicklime. Let the mixture stand until the supernatant fluid is entirely clear. Then pour this off, and mix it with enough olive oil to form a thick cream, or rather to the consistency of melted and recongealed butter. Grease the articles of iron or steel with this compound, and then wrap them up in paper; or, if this cannot be done, apply the mixture somewhat more thickly.

To keep Tools from Rusting.

Half-ounce camphor dissolved in 1 lb. melted lard; skim and mix in as much black-lead as will give it an iron colour. Smear tools, and after 24 hours rub clean with soft linen cloth.

Or, 6 oz. lard to 1 oz. rosin. When semi-fluid it is ready for use.

Or, use vaseline, to which, when warm, a little powdered camphor has been added.

To remove Rust from Tools.

Tools which have become very rusty should be treated with a chemical solution, instead of trying to scour the rust off by means of emery cloth. A good solution for removing rust may be made as follows. Into one quart of distilled water dissolve, little by little, sufficient chloride of tin to obtain a saturated solution, that is, until the water will not dissolve any more of the salt. Put the tool into a receptacle containing the solution and let it stand over night. In the morning rinse the solution off in running water and dry thoroughly with a piece of chamois or cloth.

Blueing Steel.

Outfit.

The outfit consists of a cast-iron kettle, one tank for boiling water, and one tank for oil (aquiline). The mixture consists of nine parts of saltpetre, by weight, and one part pyrolusite. The articles are hung in perforated baskets. A scraper is necessary for the melting kettle to stir the solution and to scrape the sediment from the bottom.

Method.

The desired temperature for treatment is obtained when all the mixture has melted and begun to boil, and when a pinch of fine sawdust thrown in bursts into flame the moment it reaches the surface. The temperature must be kept constant to give good results. In operation, the full basket may be hung on the chain at one end of a lever and lowered into the solution. The articles are left in for three minutes, then are slowly lifted out to allow the surplus solution to drain off: they are then dipped into hot water to clean off the solution; after the water has drained off, the articles are dipped into the oil and taken out as quickly as possible and allowed to drain. It is essential to note that there be no rust on the work, and that red-hot cast iron is attacked by these chemicals.

Blueing Small Screws.

Use a boiling bath composed of:

Water	1 gallon.
Hyposulphite of soda	2 ozs.
Acetate of lead	2 „

The hyposulphite of soda and the acetate of lead are first dissolved in the water and the whole is then brought up to boiling point. The work, which must be absolutely clean and free of grease, and preferably polished, is immersed in the bath for a few seconds, and then withdrawn. At first the work will be a kind of yellow, changing to red and then to blue almost immediately. The work is rinsed in water and quickly dried. Changing the length of time during which the work is immersed in the bath will give varying depths of colour. When the desired tint is obtained, note the time immersed, and give subsequent dippings the same period in the bath.

The brilliancy of the blues obtained by the above method is greatly enhanced if the work is copper plated before immersion in the colouring bath. The plating may be accomplished by the dipping process. To a quantity of distilled water add all the copper sulphate it will dissolve, then for every four ounces of distilled water add from 10 to 14 drops of sulphuric acid. Start with 10 drops in the first instance and add more sulphuric acid if required. Before coppering it is absolutely essential that the work be free from grease. It should be noted that in all colouring processes the higher the initial polish on the work the greater the brilliance of the finished work.
(*'Machinery.'*)

Blueing Small Hardened Steel Springs Without Heating.

Thoroughly clean the springs in boiling lime water and afterwards remove any remaining lime by means of a brush. Dip the springs in a nitric acid bath; wipe clean and oil; linseed oil is generally used for this purpose. (*W. R., 'Machinery.'*)

Waterproof and Fire-Resisting Boards.

The boards are first laid in a waterglass solution of 5° to 10° Bé., where they are left 10 to 12 hours, when they are taken out and allowed to drip off. After drying, they are placed in a solution (gravity 2° to 3° Bé.) of calcium chloride, magnesium chloride, and ammonium chloride. In this they are left four to six hours, and after dripping off and drying again, they are ready for use.

MISCELLANEOUS.

To Prevent Rusting-in of Screws.

In machines exposed to heat or moist air the screws soon rust in, even with the use of oil, which later on renders it difficult to take the machines apart. By dipping the screws before use in a thin paste of graphite and oil they can be readily taken out, even after years.

Drilling Glass.

To drill holes in glass a common steel drill, well made and well tempered, is the best tool. Whilst drilling, the hole should be moistened with turpentine in which some camphor has been dissolved.

To drill a large hole in thin glass, make in a piece of wood a hole of the size required, and fasten the wood with beeswax upon the glass so as to act as a guide. A piece of brass or copper tubing, of the size of the hole required, is placed in the hole in the wood, supplied with emery powder and water, and is twirled round between the fingers or by means of a bow-string. This will cut a hole in a few minutes.

To Divide Glass Bottles.

Breaking glass tubes or vessels with a clean edge by the aid of a string dipped in an inflammable oil is a well-known method, but the latter is not suitable for large vessels, such as a carboy; to break such vessels, fill to point required with cold water, then pour boiling oil over surface of water: before oil has time to cool, dash cold water on outside of vessel, when a clean break is said to result.

A still better method is to make a small mark, with a triangular file, at the point where the break or division is required. A tiny gas flame, not larger than $\frac{1}{8}$ in. diameter, is then applied to the file-mark. If a crack does not start quickly the hot glass is wetted with a drop of water. When a crack has started the flame is held in contact with the glass, just in front of the crack; the crack will then extend to the flame and by causing the flame to travel, step by step around the vessel, complete severence will be effected with accuracy and certainty.

To Cut Glass Tubing of Large Diameter.

Nick the glass tube with a file or a cutter at the point where the fracture is desired. Wrap three or four thicknesses of blotting paper around the tube about $\frac{1}{2}$ in. from each side of the mark, the blotting paper being held in place by copper wires. Soak the blotters with water, and expose the narrow section of tube between the two wet blotters to the heat of a bunsen burner. The flame will cause a fracture starting at the file nick and extending in almost a straight line around the tube. The method describes for glass bottles (above) may also be used.

Cutting Glass under Water.

Cutting a glass sheet with a pair of scissors may seem difficult, but it is not, although the job must be done under water and both the glass and scissors must be submerged. A good way is to first cut a paper the size or shape of the glass desired; place it on the under side of the glass for a pattern. It is not expected that it is possible to cut through the centre of a large piece of glass, but by trimming a little at a time a good job can be done. The cutting sensation is a good deal the same as that of cutting cardboard.

To make Ivory and Bone Permanently White.

Hydrated oxide of zinc is dissolved in ammonia and the articles laid in it. To remove the yellow tint, a few drops of an aqueous solution of blue vitriol is added; after impregnation the articles are repeatedly washed with cold water, dried in the air and polished. The zinc-oxide of ammonia solution is prepared as follows:—40 parts by weight water mixed with 25 parts of pure zinc, with (after a quarter of an hour) 50 parts concentrated hydrochloric acid added. To the zinc solution 150 parts hot water is added, then (gradually stirring the while) caustic ammonia until the precipitated white hydrated oxide of zinc is dissolved.

Hard Rubber.

The finest grade of hard rubber, such as is found in a good comb or fountain pen, may contain 75 per cent. rubber and 25 per cent. sulphur, and may have been vulcanised ten hours. The specific gravity of pure hard rubber ranges from 1·12 to 1·25. The tensile strength in the finest and purest grades may be over three tons, and the dielectric strength is specified by the United States Navy for sheets at 10,000 volts per $\frac{1}{32}$ in. of thickness.

Fibre.

There are three kinds of fibre in general use—viz., two kinds of vulcanised fibre, known as hard and flexible, and horn fibre.

The flexible is restricted to such uses as pump valves, washers and gaskets. The tensile strength of vulcanised fibre varies from 10,000 lbs. to 20,000 lbs. per sq. in. (as compared with 3,000 lbs. to 6,000 lbs. for leather). Its resistance to compression varies from 40,000 lbs. to 60,000 lbs. per sq. in. (for wood it varies from 3,000 lbs. to 11,000 lbs.). Its resistance to shearing varies from 9,000 lbs. to 13,000 lbs. per sq. in. (compared to wood, with the grain, 225 lbs. to 906 lbs.; to aluminium casting, 12,000 lbs.; and to aluminium forgings, 16,000 lbs.). There is no clearly defined elastic limit, but it approximates 4,000 lbs., and there is no permanent set even at rupture point. The specific gravity is about 1·3, varying from 1·2 to 1·5 (sp. gr. of pure aluminium in a cast state is 2·58; of cork, 0·24; of wood, 0·35 to 1·33). It possesses the strength and density of a metal, yet is elastic and nearly as light as hardwood. A square foot of $\frac{1}{8}$ in. fibre weighs about $\frac{1}{4}$ lb., and the same area of 1 in. stock about 7 lbs.—i.e., about 20 cub. ins. to the pound.

Vulcanised fibre is insoluble in all the ordinary solvents, and is not affected by immersion in alcohol, ammonia, benzine, or any of the animal, vegetable, or mineral oils. It absorbs water freely, and swells when wet, but is not otherwise injured. When heated this fibre does not melt or soften, but at a high temperature it chars and loses its elasticity. Vulcanised fibre is not brittle, and will stand a great amount of pounding and rough usage. It can be machined, and will take a fine polish, but cannot be moulded.

One of the purposes to which vulcanised fibre is being applied is the thrust bearings in light, sensitive, drilling machines. Ball bearings cause endless trouble in this position if one of the balls fractures or chips, and it has been found that a fibre disc answers the purpose admirably. The permeability of the fibre to oil renders it self-lubricating, and its life under these conditions is remarkable.

To Face Oilstones.

On a sheet or plate of iron with an accurately even or straight face (it ought to be planed), scatter a little emery or fine sand, add a little water, and rub the face of the stone on the sheet, renewing the emery or sand and water as requisite, finishing with an addition of water without emery or sand.

Etching and Etching Fluids.

The nearest approach to a perfect protective for general purposes is beeswax of the proper consistency, which can be easily applied in any desired thickness. Before applying the wax it is important that the surface be thoroughly clean and absolutely dry, and the difference in temperature between the wax and the article to be etched should be slight. If it is necessary to dip the piece into melted wax, the article should be kept immersed for a few moments until it acquires the same temperature as the molten wax. The design is cut in the wax and through to the metal, after which the etching liquid is applied.

The following fluids have been found to work satisfactorily:

Iron and Steel.—Hydrochloric acid (full strength) should be used. Nitric acid, however, must not be used.

Brass.—Nitric acid.

Copper.—A mixture containing 2 parts of nitric acid and 1 part of sulphuric acid.

Silver.—Nitric acid, 3 parts; water, 1 part.

Gold.—A mixture containing 1 part of nitric acid and 3 parts of hydrochloric acid. This mixture should be prepared just before being applied, and should be used warm, under a hood or fume closet.

Platinum.—The same mixture as that used for gold.

Lead.—Nitric acid.

Aluminium.—A 10 per cent. solution of caustic soda or potash.

Zinc.—A mixture containing equal parts of hydrochloric acid and water, used warm.

Glass.—Hydrofluoric acid. The article may be immersed in the liquid acid for a few minutes or it may be exposed to the fumes for five to fifteen minutes. Extreme caution should be used in handling this acid, using rubber gloves for the hands and a lead or hard rubber container for the acid. Contact with the skin will cause severe burns, and the action seems to be continuous, producing burns far more severe than those caused by the mineral acids.

All the acids may be used full strength, but if the etching is to be of considerable depth most of them may be diluted with water before applying. This will require more time, but will produce a cleaner cut. The exception to this is the mixture of gold and platinum, which must always be used full strength and applied as warm as the melting point of the wax will permit.

Bending Small Metal Tubes.

Put a cork or plug into one end and fill with slightly damp sand well rammed in, then close with another cork; heat the tube until it is just uncomfortably hot to the hand, and then bend round a bit of board sawn to the required curve.

In making coils and spirals of small tubing, the Bureau of Standards at Washington has adopted the plan of filling the tubes with ice. This has the advantage of being very easily got rid of after the completion of the operation, and has proved very effective. To produce the ice filling, the tube, after charging with water, is immersed in a bath of ice and salt.

To Drill Hard Steel.

Use turpentine instead of oil; it will drill readily even when the drill lubricated with oil will not touch the metal.

Turning and Cutting Aluminium.

One of the best lubricants for cutting aluminium in the lathe is paraffin; this prevents tearing, and enables a better finish to be obtained.

To Remove a Stubborn Nut.

Heat the spanner end and place on nut for a moment or two or apply a red hot bar to the nut; this will expand the nut only, whereas using a blow-lamp, etc., heats both nut and bolt. If the above method fails, the nut should be split.

Silvering a Mirror.

(Up to 14 square inches area.)

Well scrub glass with brush, soap, and hot water; rinse repeatedly in cold water; soak in nitric acid for few minutes; rinse in distilled water. Do not touch surface after soap stage.

Solutions.

(*a*)		(*b*)		(*c*)	
Silver nitrate .	50 grs.	Grape sugar .	50 grs.	Caustic potash .	50 grs.
Distilled water .	1 oz.	Distilled water .	1 oz.	Distilled water .	1 oz.

Add strong ammonia drop by drop to (*a*), when a yellowish precipitate will form, and later liquid will clear. No more ammonia must then be added; swirl after each drop. Add (*c*), when black precipitate will form; add ammonia drop by drop until solution clears; add 2½ ozs. distilled water and mix in sugar solution (*b*). *Immediately* put into dish and carefully insert glass with face (downwards) just below the surface. When sugar is added solution becomes cloudy, so it should only be added when quite ready for use. Leave glass undisturbed for 30 minutes. Remove glass and again wash in distilled water. Drain for few minutes and swish off any drops of water. If cloudy, polish with a pad of cotton wool covered with soft chamois leather and jewellers' rouge, by light circular strokes. Lacquer with No. 2 cold colourless lacquer from Messrs. Gedge, 90 St. John's Street, Clerkenwell. Back with Brunswick black. Absolute cleanliness in vessels and liquids is essential to success. It is usually cheaper and more satisfactory to send the glass to a mirror-maker.

Size of Clock Dials.

Allow 1 ft. diameter for every 10 ft. from the ground, and then add 6 ins. to this; thus, if the dial is to be 40 ft. from the ground, the diameter should be 4 ft. 6 ins.

Wire Drawing.

Wire may be reduced in diameter or *drawn* by forcing it through holes in a piece of steel or other substance. A steel die is used where practicable, but for some purposes diamond dies are preferable. The diamonds used are either natural stones or chips from large stones used in jewellery. For a hole ·055 in. in diameter a 5-carat stone is needed, whilst stones weighing half-a-carat suffice for dies with holes from ·014 in. down to ·009 in. in diameter and under. In copper wire the smallest size made commercially is 0·002 in. in diameter. Thirteen diamonds are needed to reduce the wire from ·010 in. in diameter to 0·003 in. in diameter. Wires thinner than ·002 in. in diameter are exceedingly expensive to make. A lot drawn for exhibition purposes from ·0022 in. in diameter down to ·001 in. in diameter took 7 drafts. The life of the diamonds when drawing copper wire is six months to a year, provided the stones are of good size, hard, smooth, and well drilled. The weight produced per operator per day falls very rapidly as the size of the wire is reduced. Thus, whilst one man will produce 1,100 lbs. per day of No. 22 (B. and S. gauge), he can only draw 45 lbs. of No. 40 (B. and S.). Diamonds are also used for drawing steel, though they are not economical, nor do they produce a better quality of wire than steel die-plates; but if fine steel wire is wanted very round and true to size, the diamond dies seem necessary. A steel die will not keep to size for more than one coil of No. 33 wire or finer. The life of a diamond is much shorter on steel than on copper, averaging only about three days, when it must be polished out for the next larger size. The surface produced on steel wire is not so smooth when diamonds are used as it is with steel. The speed at which the wire is drawn appears to have little effect on the life of the diamond die, which depends solely on the length passed through. About 200 lbs. weight of steel wire can be drawn through a No. 32 (B. and S.) diamond die-plate before it is off size, 15 lbs. through holes ·005 in. to ·003 in. in diameter, and less than 1 lb. through holes smaller than ·002 in. in diameter. The smallest commercial size is ·0025 in. steel brush wire, drawn from low or medium carbon steel. Difficulty was at one time experienced in drawing hard music wire smaller than ·012 in. in diameter without breaking the diamond; but nowadays it can be drawn as small as ·005 in. in diameter, using 2 to 2½-carat diamonds for the dies. Below about ·009 in. or ·008 in. diameter, however, music wire becomes very curly, and no method of keeping it straight has yet been discovered.

Cracks in Metal.

A crack in a piece of metal is prevented from extending further by the well-known means of drilling a hole where the rent ends; but when the hole is not bored on just that spot the crack is apt to continue beyond the hole. To facilitate the search for the exact point, moisten the cracked surface with petroleum, then wipe it, and immediately rub it with chalk. The oil that has penetrated into the crack exudes, and thus indicates with precision where the crack stops. The best method of healing cracks is by electric or acetylene welding.

Rendering Celluloid Incombustible.

In order to overcome the undesirable quality of celluloid to ignite, it may be dissolved in a mixture of ether and alcohol and added to an ether-alcohol solution of ferric perchloride. The clear syrupy liquid is poured into a suitable vessel and is left for spontaneous evaporation. The resulting substance of shell-colour is, after washing and drying, a celluloid of great pliability and transparency, and is incombustible.

See also 'Celluloid,' by Dr. Bückmann· Fr. London, Scott, Greenwood & Son, 8 Broadway, Ludgate Hill, London, E.C. 4. Price 8*s.* 6*d.*

Wood-Worms in Oak

It is stated that the most effective means of treating oak work when attacked with wood-worm is to give it several coats of paraffin, well soaked into the wood. This should be done by a careful workman, making quite sure that every hole has been well soaked with the paraffin; the paraffin does not damage the oak, but has a tendency to improve the colour.

The extermination of the *Xestobium tessellatum* in the oak roofing of Westminster Hall has been effected by spraying the timber with a mixture of 50 per cent. tetrachlorethane, 6 per cent. cedar wood oil, 2 per cent. solvent soap, 2 per cent. paraffin wax, and 40 per cent. trichlorethylene. The tetrachlorethane is a perfect insecticide, but it is so dangerous that those who handle it must use gas masks.

Thermostatic Metal.

Thermostatic metal, manufactured by the *British Thomson-Houston Company, Ltd.*, in appearance a homogeneous metal, is formed of two metal strips having widely different coefficients of expansion. The two metals are welded together along their entire length, and are quite inseparable under normal working conditions. When the duplex strip is subjected to change of temperature, one half will expand more quickly than the other, causing the strip to bend. A change of one degree of temperature is said to affect its curvature to a measurable extent, and practically instantaneously. In bending, it exerts a force sufficient for the direct operation of levers, the opening and closing of ventilators, etc. The metal can be cut, stamped, or machined. it is supplied in various thicknesses from 0·015 in. to 0·25 in.

Setting a Spirit-Level Glass.

A simple method of setting a spirit level glass is to use a small piece of putty to support each end temporarily. The plastic putty can be made to hold the glass in the required position, it being an easy matter to depress either end so that it will be parallel or level with the working surface of the levelling bar. When properly levelled up in this way plaster of Paris is poured around the supporting ends of the glass and allowed to set. The glass will then be held permanently in place. (J. H. B., *Machinery*.)

Supporting Power of Ice.

Ice 1½ inches thick will support a man.
„ 4 „ „ „ cavalry and light guns.
„ 5 „ „ „ an 84-pound cannon.
Ice 10 inches thick will support a crowd.
„ 18 „ „ „ a railway train.

Removing Enamel.

The ordinary paint-removers, designed to remove air-dried paint and varnishes, are not adapted, or intended to be used, for the removal of baked enamel from metallic surfaces such as those of automobiles. It is customary in the removal of baked enamel, particularly enamel which is baked under high temperatures, to laboriously chip it off. D. F. Clark, of Harrisburg, Pa., has compounded a body (U.S. Patent No. 1,495,547) which may be applied with a brush, and which will adhere to the enamelled surface, and on allowing it to remain for from 15 mins. to 2 hours, depending upon the temperature at which the enamel was baked, the enamel may be easily brushed off. The best results are obtained from a mixture of the following ingredients in the proportions mentioned; 20 parts of methyl acetone; 20 parts of denatured alcohol; 10 parts of aqua ammonia (of 26° to 32°); 10 parts ammonia carbonate; 15 parts benzol; 10 parts xylol; 5 parts paraffin wax; 5 parts celluloid chips; 5 parts soap. The benzol, xylol, and methyl acetone are boiled, and the wax and celluloid are stirred in until thoroughly incorporated. The mixture thus far made is then permitted to cool. After cooling, the denatured alcohol is added. Finally, the ammonia is added. The purpose of the celluloid and the wax is to impart a viscous consistency to the mass and prevent evaporation of the volatile constituents. Some enamelled metals have so adhesive an enamel that this treatment will not remove it.

Flow of Sand.

The velocity of flow of sand and other fine materials through openings of various shapes and in different circumstances, is determined mainly by the size of the particles and of the opening, but is slightly diminished by angularity of the particles or by moisture on them. The specific gravity of the particles, the height of the material above the opening, and the shape of the upper contour of the material have little effect. If D is the diameter of the opening, d that of the particles, both in inches, the mean velocity of flow V in feet per second is given with sufficient accuracy for most practical purposes by the equation, $V = (D - 2d)/\sqrt{D}$.

(*Hersam: Journal of the Franklin Institute.*)

SECTION XLIII

RADIO COMMUNICATION

(Pp. 2821-2830.)

SECTION XLIII

RADIO COMMUNICATION.

(Contributed by the Editor of 'The Wireless World.')

Glossary of Technical Terms.

The following list of definitions contains terms that are usually unfamiliar to engineers who have not made any special study of radio technique. Expressions that have more or less passed into everyday language since the inception of broadcasting in this country have been omitted.

Amplification Factor (of a valve).—The ratio of the change of anode voltage necessary to bring about a given change of plate current, to the change in grid potential required to produce the same change in anode current. More simply, the number of volts by which the anode potential must be lowered to maintain the anode current constant when the grid potential is raised by one volt. It represents the maximum voltage amplification that the valve can theoretically provide.

Anode.—The electrode of a thermionic valve, which, when maintained at a higher potential than the cathode, attracts negative electrons from it, thus bringing about a current flow between the two.

Anode Bend Detection.—A method of rectification in which the grid circuit of a valve is so adjusted that the application of a given increase in oscillatory grid voltage will bring about an increase of anode current much greater than the decrease of this current due to a reduction in grid voltage by the same amount. The valve is operated on the lower bend of its characteristic curve by applying a suitable negative polarising voltage from a local source.

Astatic Coil.—An inductance wound in such a way that its external field is strictly limited, thus preventing magnetic interaction with other coils in close proximity.

Band-Pass Filter.—A network of coupled circuits arranged to pass a definite band of frequencies.

Beats.—When two wave motions of slightly dissimilar frequencies are superimposed, a pulsation is produced by the alternate coincidence and opposition of the waves. These pulsations or 'beats' have a frequency equal to the difference between the frequencies of the two superimposed wave motions. In wireless reception, the resultant frequency can be rectified and the principle applied in various ways.

Damping.—A term relating to the rate at which an oscillation dies away or decays, and also to the forces responsible for this progressive decrease in amplitude.

Detector.—In a wireless receiver, this term is applied to a device intended for converting incoming oscillations into unidirectional pulses of current. (See *Anode Detection* and *Grid Detection.*)

Diaphragm.—The sound-producing vibrating member of a loud-speaker or telephone receiver.

Eliminator.—An appliance comprising voltage-regulating, smoothing, and, if necessary, rectifying devices, and designed to provide the necessary anode, filament, or plate potentials for a wireless receiver from electric supply mains, whether A.C. or D.C.

Fading.—Should be applied exclusively to variations in signal strength due to changes in the reflecting properties of the Heaviside Layer, and not to changes brought about by alterations in transmitting power, etc.

Field Strength.—The intensity of the electric and magnetic field due to the radiation of a wireless transmitter at a given distant point. It is conveniently measured in terms of millivolts induced in an aerial having an effective height of a given number of metres, and is then expressed as ' millivolts per metre.'

Full-Wave Rectification.—A system of rectification in which both positive and negative half-cycles of an alternating current are used.

Ganged Controls.—An arrangement in which two or more controls of a wireless receiver (generally variable condensers) are linked together mechanically in such a way that they may be operated from a single control knob.

Grid.—The wire gauze or spiral wire electrode interposed between anode and filament of a valve, which controls the electron stream emitted by the filament.

Grid Detection.—A method of rectifying high-frequency oscillations, based on the unilateral conductivity of the grid-filament space inside a valve, and depending for its efficacy on use being made of the curvature of the grid volts/grid current characteristic. A small condenser is connected in series with the grid, and when the latter receives a negative charge, the value of anode current is reduced. This charge is allowed to leak away through a high resistance, in order that the grid may assume its mean operating potential before dealing with the next impulse.

Heaviside Layer.—It is generally assumed that the upper rarefied stratum of the earth's atmosphere is ionised ; it is in consequence a partial conductor of electricity, and therefore a reflector of wireless waves. This theory accounts for the transmission of wireless signals round the curvature of the earth, and also for such phenomena as ' fading ' and ' skip distances.'

High Frequency.—Generally applied to electrical oscillations of a frequency above the limits of audibility.

Indirect Ray.—Radiated waves deflected downwards from the Heaviside Layer (*q.v.*). The term ' direct ray ' is applied to radiation which reaches its destination *via* the earth's surface.

Metal Rectifier.—A power rectifier consisting of a contact junction between two dissimilar metals, as, for instance, copper with a film of copper oxide and lead. A number of such elements may be connected in series to deal with high voltages.

Motor-Boating.—A term of American origin applied to undesirable oscillations in a low-frequency amplifier, brought about by unwanted reaction effects between the various circuits. When an amplifier affected with this trouble is connected to a loud-speaker, a sound similar to the exhaust of an internal combustion engine is produced.

Neutralised Amplification.—A system of high-frequency amplification in which a tendency towards unwanted self-oscillation, due to interelectrode capacity of the valve, is balanced out by providing means whereby equal potentials of opposite phase may be passed back from anode to grid circuits.

Pentode.—A thermionic valve essentially designed for low-frequency amplification, and comprising five electrodes, including three grids. It has the advantage of a high amplification factor, and gives a very considerable power output with relatively low anode potentials.

Power Valve.—A type of low-frequency amplifying valve intended to provide a large power output from its anode circuit rather than to afford high voltage magnification.

Reaction.—An arrangement applied to a valve whereby part of the amplified oscillatory energy in its plate circuit is fed back to the grid circuit, thus allowing incidental losses in this latter circuit to be nullified, with the practical result that signal voltages are greatly increased. The extent to which this retroactive or feed-back effect may be applied is usually under control, so that it may be adjusted to meet the conditions under which the receiver is being operated.

Rectification.—Conversion of an alternating current into a unidirectional pulsating current. The term is applied both to low-frequency currents and to high radio-frequencies, but there is a tendency for the expression 'detection' to be allotted to the latter.

Rejector Circuit.—A tuned oscillatory circuit with its constants so arranged that it offers a very high impedance to oscillations of certain definite frequencies, while allowing others to pass readily.

Screen Grid Valve.—A valve designed exclusively for high-frequency amplification : a fourth electrode, generally in the form of a closely meshed grid, is interposed between the normal control grid and the anode, and acts as an electrostatic screen, reducing incidental capacity between these two electrodes. A positive polarising potential is applied to the extra grid.

Screening.—The provision of metal partitions so arranged that circuits carrying oscillatory currents cannot affect adjacent circuits.

Selectivity (of a wireless receiver).—The relative extent to which a receiver is capable of selecting one particular transmission to the exclusion of other frequencies corresponding to interfering radiation.

Side-Band.—The effect of modulation as applied to the carrier wave of a telephony transmitter is to produce a number of high-frequency oscillations with values above and below the actual frequency of the carrier wave, and in considering the width of the frequency spectrum occupied by a station, these additional frequencies must be taken into account.

Thermionic Valve.—Embraces the various valves with from two to five electrodes generally used in radio transmission and reception, and excludes gas-filled rectifying tubes. A modern valve comprises a cathode or filament (sometimes both, the filament being in the form of a heater element), at least one grid, and an anode, all being contained in a glass envelope, exhausted to the highest degree possible.

Triode.—A three-electrode thermionic valve.

Tuned Anode.—A simple form of intervalve coupling used in high-frequency amplification, in which an oscillatory circuit connected in series with the anode of the valve is tuned to act as a rejector of the frequency that is to be received.

Wavelength.—The distance at any given instant between the maximum positive points of two successive waves. Irrespective of their length, electro-magnetic waves travel with a velocity of three hundred million metres per second.

Useful Formulæ.

WAVELENGTH.—The wavelength of an oscillatory circuit is ascertained by applying the formula :

$$\lambda = 1884\sqrt{LC}.$$

In terms of frequency :

$$f = \frac{10^6}{2\pi\sqrt{LC}}$$

where

λ = wavelength in metres,
L = inductance in microhenries,
C = capacity in microfarads,
f = frequency in cycles per second.

CAPACITY.—Capacity of fixed condensers :

$$C = \frac{0{\cdot}0885\ A\epsilon\ (n-1)}{d}.$$

Reactance of a condenser :

$$\frac{1}{2\pi fC}$$

C = capacity in farads,
A = area of overlap of plates in sq. centimetres,
ϵ = dielectric constant of separating material (see below),
n = number of metal plates,
d = distance separating plates (in cms.),
f = frequency in cycles per second.

Dielectric constants of insulating materials in common use, as compared with air (air = 1)

Ebonite	2·56 to 3·48	Fibre	5·1 to 5·9
Glass	6·6 to 10	Mica	5·0 to 7·0
Shellac	2·95 to 3·73	Paraffin Wax	1·92 to 2·47
India Rubber (Para)	2·34	India Rubber (vulcanised)	2·94

Condensers in parallel:

$$C = C_1 + C_2 + C_3 + \text{etc.}$$

Condensers in series:

$$\frac{1}{C} = \frac{1}{C_1} + \frac{1}{C_2} + \frac{1}{C_3} + \text{etc.}$$

INDUCTANCE.—The following formula gives the inductance of a single-layer coil with closely-wound turns:

$$L = \pi^2 d^2 n^2 l K,$$

where

L = inductance in microhenries,

d = diameter of coil (in cms.),

n = number of turns per cm.

l = length of coil (in cms.),

K = correction factor depending on ratio of diameter to length of coil (see following Table).

$\frac{d}{l}$	K	$\frac{d}{l}$	K	$\frac{d}{l}$	K
0·00	1·000	0·80	0·735	4·0	0·365
0·10	0·959	0·90	0·711	5·0	0·320
0·20	0·920	1·00	0·688	6·0	0·285
0·30	0·884	1·5	0·595	7·0	0·258
0·40	0·850	2·0	0·526	8·0	0·237
0·50	0·818	2·5	0·472	9·0	0·218
0·60	0·788	3·0	0·429	10·0	0·203
0·70	0·761				

Reactance of an inductance coil (in ohms):—

$$2\pi f L$$

where

f = frequency in cycles per second,

L = inductance in henrys.

H.F. AMPLIFICATION.—The maximum stage H.F. amplification attainable from a screen grid valve, without such adventitious aids to stability as neutralisation, is given by the following formula, which refers to a wavelength of 200 metres. Stability at higher wavelengths can, of course, be assumed.

$$A = \frac{-1 + \sqrt{1 + 2\mu R_o \omega C}}{\omega C_o R_o}.$$

Stage amplification obtainable with a transformer with ratio other than the optimum is given by:

$$A = \frac{N\mu R}{R + N^2 R_o}$$

A = stage amplification,

C_o = anode-grid capacity in farads,

μ = amplification factor of valve,

R_o = A.C. resistance of valve (in ohms),

R = Dynamic resistance of tuned circuit (in ohms),

N = Step-up turns ratio of transformer,

$\omega = 2\pi \times$ frequency (at 200 metres, $\omega = 10^6 \times 2\pi \times 1{\cdot}5$).

L.F. AMPLIFICATION.—Voltage stage amplification (approx.) of a transformer-coupled amplifier :—

$$\mu \frac{N_2}{N_1} \times \frac{X}{\sqrt{X^2 \times R_0^2}}$$

where

μ = voltage factor of valve,
N_2 = number of secondary turns,
N_1 = number of primary turns,
X = reactance of primary coil (in ohms),
R_0 = A.C. resistance of valve.

Copper Wire Tables.

SINGLE SILK COVERED.

Diam.	S.W.G.	Diam. Overall.	Turns per inch run.	Turns in one sq. in.	Wt. per 1,000 yds.	Yds. per lb. or oz.
in.		in.			lbs.	per lb.
·064	16	·067	15·0	222	37·9	26·4
·048	18	·050	20·0	400	21·3	46·8
·036	20	·038	26·3	692	12·0	83·3
·028	22	·030	33·3	1 090	7·30	137
·022	24	·023 5	42·5	1 810	4·50	222
·018	26	·019 3	51·8	2 680	3·02	332
·014 8	28	·016 1	62·1	3 860	2·05	488
·012 4	30	·013 7	73·0	5 330	1·44	695
·010 8	32	·012 1	82·6	6 820	1·10	912
·009 2	34	·010 5	95·2	9 060	·800	1 250
·007 6	36	·008 9	112	12 540	·551	1 815
·006 0	38	·007 3	137	18 770	·348	2 871
					oz.	*per oz.*
·004 8	40	·006 1	164	26 900	*3·62*	*276*
·004 0	42	·005 2	192	36 860	*2·58*	*387*
·003 2	44	·004 4	227	51 530	*1·67*	*599*
·002 8	45	·004 0	250	62 500	*1·33*	*752*
·002 4	46	·003 6	278	77 300	*1·00*	*1 000*
·002 0	47	·003 2	312	97 300	*·727*	*1 375*

DOUBLE SILK COVERED.

Diam.	S.W.G.	Diam. Overall.	Turns per inch run.	Turns in one sq. in.	Wt. per 1,000 yds.	Yds. per lb. or oz.
in.		in.			lbs.	per lb.
·064	16	·068	14·7	216	38·3	26·1
·048	18	·051	19·6	384	21·6	46·3
·036	20	·039	25·6	655	12·1	82·5
·028	22	·031	32·2	1 040	7·44	134
·022	24	·025	40·0	1 600	4·59	218
·018	26	·020 5	48·8	2 380	3·08	325
·014 8	28	·017 3	57·8	3 340	2·09	478
·012 4	30	·014 9	67·1	4 500	1·48	675
·010 8	32	·013 3	75·2	5 650	1·13	887
·009 2	34	·011 7	85·5	7 310	·82	1 220
·077 6	36	·010 1	90·1	8 120	·57	1 750
·006 0	38	·008 5	118	13 900	·27	3 760
					oz.	*per oz.*
·004 8	40	·007 3	137	18 800	*3·88*	*258*
·004 0	42	·006 2	161	25 900	*2·80*	*358*
·003 2	44	·005 4	185	34 200	*1·86*	*536*
·002 8	45	·005 0	200	40 000	*1·48*	*675*
·002 4	46	·004 6	217	47 100	*1·15*	*871*
·002 0	47	·004 2	238	56 600	*·845*	*1 190*

SINGLE COTTON COVERED.

Diam.	S.W.G.	Diam. Overall.	Turns per inch run.	Turns in one sq. in.	Wt. per 1,000 yds.	Yds. per lb.
in.		in.			lbs.	
·128	10	·135 5	7·4	54	150·8	6·63
·104	12	·111 5	9·0	81	99·7	10·3
·080	14	·087 5	11·4	130	59·3	16·9
·064	16	·071 0	14·1	198	38·3	26·1
·048	18	·054 5	18·3	335	21·6	46·3
·036	20	·041 5	24·1	581	12·24	81·7
·028	22	·033 5	29·8	888	7·48	134
·022	24	·027 0	37·0	1 369	4·57	219
·018	26	·023 0	43·5	1 892	3·21	311
·014 8	28	·019 8	50·5	2 550	2·21	452
·012 4	30	·017 4	57·5	3 300	1·58	634
·010 8	32	·015 8	63·3	4 010	1·20	835
·009 2	34	·014 2	70·5	4 970	·781	1 280
·007 6	36	·011 6	86·2	7 430	·619	1 610
·006 0	38	·010 0	100·0	10 000	·392	2 550
·004 8	40	·008 8	112·5	26 600	·255	3 910

DOUBLE COTTON COVERED.

Diam.	S.W.G.	Diam. Overall.	Turns per inch run.	Turns in one sq. in.	Wt. per 1,000 yds.	Yds. per lb.
in.		in.			lbs.	
·128	10	·141	7·1	50·3	152	6·58
·104	12	·117	8·5	72	101	9·09
·080	14	·093	10·75	115	60·2	16·6
·064	16	·075	13·3	177	38·9	25·6
·048	18	·058	17·3	299	22·0	45·4
·036	20	·046	21·7	473	12·6	79·4
·028	22	·038	26·3	692	7·76	129
·022	24	·031	32·3	1 043	4·92	203
·018	26	·027	37	1 400	3·40	294
·014 8	28	·023 8	42	1 790	2·37	422
·012 4	30	·021 4	47	2 210	1·70	587
·010 8	32	·019 8	50·5	2 550	1·32	755
·009 2	34	·018 2	55	3 020	·977	1 024
·007 6	36	·015 6	64	4 010	·677	1 477
·006 0	38	·014 0	71·5	5 110	·437	2 287
·004 8	40	·012 8	78	6 080	·290	3 456

These tables give information necessary in designing high- and low-frequency inductance coils, etc. In the columns headed 'turns per inch run' and 'turns per sq. inch,' an average thickness of covering is assumed; there is not complete uniformity in this matter.

Empirical Formulæ.

FRAME AERIALS.—Provided a frame aerial is wound in a conventional manner, is of ordinary dimensions, and is shunted by a 0·0005 mfd. condenser, an inductance suitable for covering the 200–550 metres waveband is given if a total length of 80 ft. of wire is used.

Similarly, the use of 250 ft. of wire will provide an inductance suitable for covering the waveband 900–2,000 metres.

OPEN AERIALS.—The natural wavelength of a normal open aerial is from 4 to 5 times its length.

VALVE GRID BIAS.—For amplifying conditions, an approximation to the correct grid bias (in volts) is given by

$$\frac{\text{Anode voltage applied}}{\text{Amplification factor of the valve} \times 2}$$

For anode bend detection, the value ascertained by the above means is doubled.

BARE COPPER WIRE.

Diam.	S.W.G.	Section Area.	Ohms per 1,000 yds.	Length per Ohm.	Wt. per 1,000 yds.	Approx. safe Current.
in.		sq. in.		yds.	lbs.	amps.
·128	10	·013	1·87	535	148·8	35
·104	12	·008 5	2·83	353	98·2	28
·080	14	·005 03	4·78	208	58·1	19
·064	16	·003 22	7·46	134·6	37·2	13
·048	18	·001 81	13·27	53·4	20·9	7
·036	20	·001 02	23·6	42·4	11·8	4
·028	22	·000 616	39·0	25·6	7·12	2·5
·022	24	·000 380	63·2	15·8	4·40	1·5
·018	26	·000 254	94·3	10·60	2·94	1·0
·014 8	28	·000 172	139·5	7·18	1·99	·7
·012 4	30	·000 121	199	5·03	1·40	·5
·010 8	32	·000 091 6	262	3·82	1·06	·4
·009 2	34	·000 066 5	361	2·77	·769	·25
·007 6	36	·000 045 4	529	1·89	·525	·15
·006 0	38	·000 028 3	849	1·18	·327	·1
				in.	*oz.*	*milliamps.*
·004 8	40	·000 018 1	1 326	*27·15*	*3·35*	*70*
·004 0	42	·000 012 6	1 910	*18·87*	*2·32*	*50*
·003 2	44	·000 008 04	2 985	*10·77*	*1·49*	*30*
·002 8	45	·000 006 16	3 899	*9·24*	*1·14*	*25*
·002 4	46	·000 004 52	5 307	*6·78*	*·834*	*20*
·002 0	47	·000 003 14	7 642	*4·71*	*·581*	*12*
·001 6	48	·000 002 01	11 941	*3·02*	*·372*	*8*
·001 2	49	·000 001 13	21 230	*1·70*	*·209*	*5*
·001 0	50	·000 000 78	30 570	*1·18*	*·145*	*3*

ENAMELLED.

Diam.	S.W.G.	Diam. Overall.	Turns per inch run.	Turns in one sq. in.	Wt. per 1,000 yds.	Yds. per lb. or oz.
in.		in.			lbs.	per lb.
·064	16	·067 5	14·8	219	37·8	26·4
·048	18	·050 7	19·7	388	21·3	46·9
·036	20	·038 7	25·8	666	12·0	83·3
·028	22	·030 5	32·8	1 080	7·30	137
·022	24	·024 3	41·1	1 690	4·52	221
·018	26	·019 8	50·5	2 550	3·03	330
·014 8	28	·016 4	60·1	3 610	2·05	488
·012 4	30	·013 6	73·5	5 400	1·44	694
·010 8	32	·012 0	83·3	6 940	1·09	915
·009 2	34	·010 2	98·0	9 600	·832	1 202
·007 6	36	·008 6	116	13 450	.543	1 840
·006 0	38	·007 0	143	20 450	·340	2 810
					oz.	*per oz.*
·004 8	40	·005 5	182	33 100	*3·49*	*286*
·004 0	42	·004 6	217	47 100	*2·43*	*411*
·003 2	44	·003 7	270	72 900	*1·56*	*642*
·002 8	45	·003 3	303	91 800	*1·195*	*835*
·002 4	46	·002 8	357	127 500	*·885*	*1 128*
·002 0	47	·002 3	435	189 000	*·613*	*1 630*
·001 6	48	·001 9	526	277 000	*·392*	*2 540*
·001 2	49	·001 4	714	510 000	*·221*	*4 510*
·001 0	50	·001 2	833	694 000	*·154*	*6 480*

Wavelength and Frequency Tables.

The use of these Tables, which give inductance/capacity values for radio frequencies, will be evident from the following examples:—

(*a*) Given a tuned circuit whose total capacity is 0·0003 microfarads and inductance 150 microhenries, what is its natural wavelength and its natural frequency?

The L.C. constant of the circuit is $0{\cdot}0003 \times 150 = 0{\cdot}045$, and the corresponding wavelength is found from the tables to be 400 metres, while the frequency is 750 kilocycles.

(*b*) What inductance will be required to tune a condenser of 0·0002 mfds. to 350 metres?

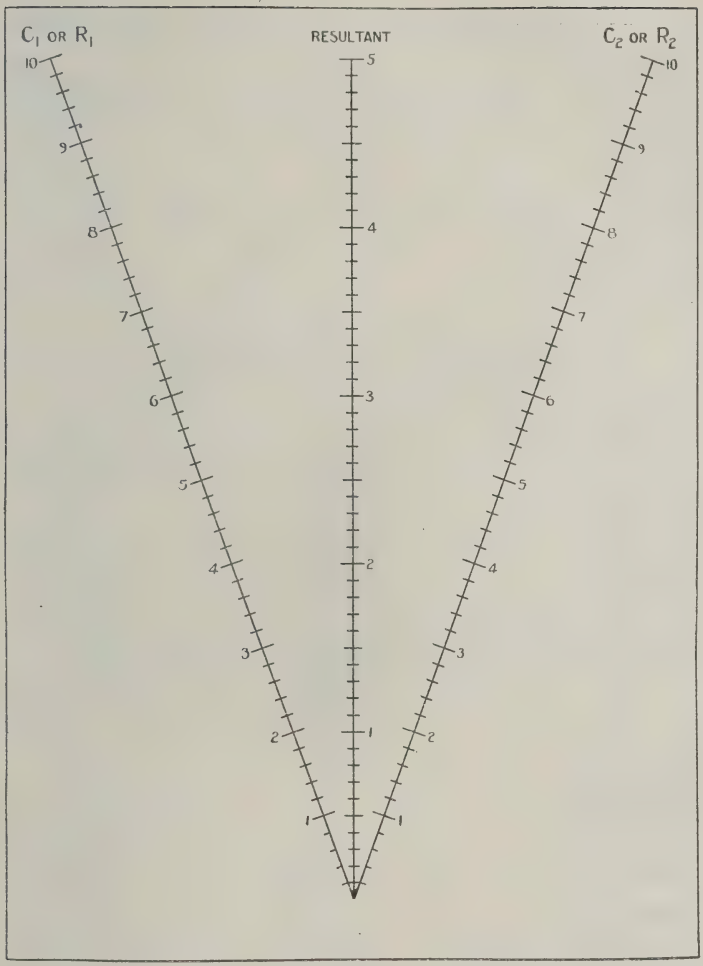
From the Table, L.C. for 350 metres is 0·03448; hence the inductance is 0·03448 divided by 0·0002, that is 172 microhenrys.

(*c*) A circuit with a natural frequency of 1,000 kilocycles is required, the inductance of the tuning coil being 127 microhenrys; what capacity should be connected across the coil?

L.C. for 1,000 kilocycles is 0·02533; hence the capacity is 0·02533 divided by 127, that is 0·0002 microfarads.

(L. is in microhenrys and C. in microfarads.)

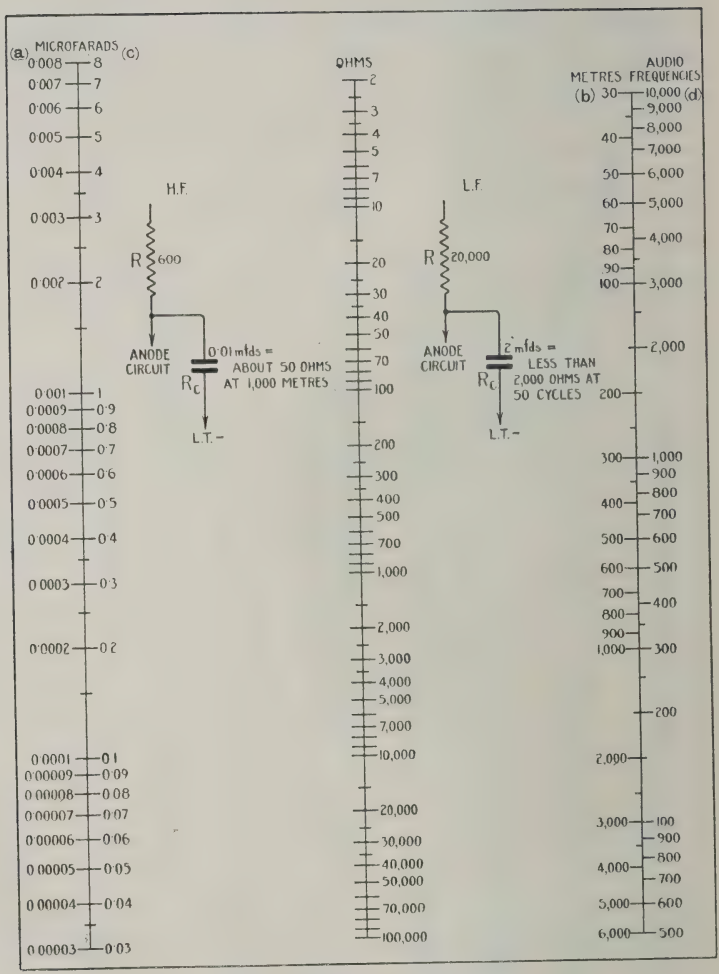
Wave-length. Metres.	Frequency Kilocycles.	L.C.	Wave-length. Metres.	Frequency Kilocycles.	L.C.	Wave-length. Metres.	Frequency Kilocycles.	L.C.
10	30,000·00	·0000282	800	375·00	·1801	1,860	161·30	·9741
15	20,000·00	·0000634	820	365·90	·1893	1,880	159·60	·9948
20	15,000·00	·0001126	840	357·10	·1985	1,900	157·90	1·016
25	12,000·00	·0001755	860	348·80	·2082	1,920	156·30	1·038
30	10,000·00	·0002533	880	340·90	·2179	1,940	154·60	1·060
40	7,500·00	·0004503	900	333·30	·2280	1,960	153·10	1·081
50	6,000·00	·000704	920	326·10	·2381	1,980	151·50	1·104
75	4,000·00	·001583	940	319·10	·2487	2,000	150·00	1·126
100	3,000·00	·002816	960	312·50	·2595	2,100	142·90	1·241
110	2,727·00	·003404	980	306·10	·2704	2,200	136·40	1·362
120	2,500·00	·004052	1,000	300·00	·2816	2,300	130·40	1·489
130	2,308·00	·004757	1,020	294·10	·2927	2,400	125·00	1·622
140	2,144·00	·005518	1,040	288·40	·3045	2,500	119·00	1·760
150	2,000·00	·006335	1,060	283·60	·3161	2,600	115·40	1·903
160	1,875·00	·007204	1,080	277·80	·3283	2,700	111·10	2·052
170	1,765·00	·008134	1,100	272·70	·3404	2,800	107·10	2·207
180	1,667·00	·009120	1,120	267·90	·3531	2,900	103·50	2·366
190	1,579·00	·01016	1,140	263·10	·3660	3,000	100·00	2·533
200	1,500·00	·01126	1,160	258·60	·3787	3,200	93·75	2·883
210	1,429·00	·01241	1,180	254·20	·3921	3,400	88·24	3·255
220	1,364·00	·01362	1,200	250·00	·4052	3,600	83·33	3·648
230	1,304·00	·01489	1,220	245·90	·4190	3,800	78·95	4·065
240	1,250·00	·01622	1,240	241·90	·4326	4,000	75·00	4·503
250	1,200·00	·01760	1,260	238·10	·4469	4,200	71·43	4·966
260	1,154·00	·01903	1,280	234·40	·4610	4,400	68·18	5·446
270	1,111·00	·02052	1,300	230·80	·4757	4,600	65·22	5·960
280	1,071·00	·02207	1,320	227·30	·4906	4,800	62·50	6·485
290	1,034·50	·02366	1,340	223·90	·5053	5,000	60·00	7·039
300	1,000·00	·02533	1,360	220·60	·5208	5,200	57·69	7·606
320	937·50	·02883	1,380	217·40	·5359	5,400	55·56	8·208
340	882·40	·03255	1,400	214·30	·5518	5,600	53·57	8·836
360	833·30	·03648	1,420	211·30	·5674	5,800	51·72	9·467
380	789·50	·04065	1,440	208·30	·5837	6,000	50·00	10·14
400	750·00	·04503	1,460	205·50	·5998	6,200	48·55	10·82
420	714·30	·04966	1,480	202·70	·6165	6,400	46·87	11·54
440	681·80	·05446	1,500	200·00	·6335	6,600	45·45	12·25
460	652·20	·05960	1,520	197·40	·6502	6,800	44·12	13·02
480	625·00	·06485	1,540	194·80	·6677	7,000	42·86	13·78
500	600·00	·07039	1,560	192·30	·6849	7,200	41·67	14·59
520	576·90	·07606	1,580	189·90	·7028	7,400	40·54	15·40
540	555·60	·08208	1,600	187·50	·7204	7,600	39·47	16·26
560	535·70	·08836	1,620	185·20	·7387	7,800	38·46	17·14
580	517·20	·09467	1,640	182·90	·7573	8,000	37·50	18·01
600	500·00	·1014	1,660	180·70	·7756	8,200	36·59	18·93
620	483·90	·1082	1,680	178·60	·7946	8,400	35·71	19·85
640	468·70	·1154	1,700	176·50	·8134	8,600	34·88	20·82
660	454·50	·1225	1,720	174·40	·8329	8,800	34·09	21·79
680	441·20	·1302	1,740	172·40	·8520	9,000	33·33	22·80
700	428·60	·1378	1,760	170·50	·8720	9,200	32·61	23·81
720	416·70	·1459	1,780	168·50	·8916	9,400	31·91	24·87
740	405·40	·1540	1,800	166·70	·9120	9,600	31·25	25·95
760	394·80	·1626	1,820	164·80	·9327	9,800	30·61	27·04
780	384·60	·1712	1,840	163·00	·9530	10,000	30·00	28·16



RESISTANCE AND CAPACITY CHART.

This ABAC gives resultant values of resistances in parallel or capacities in series.

The outer lines can be taken as representing Resistance values in units, tens, etc. of ohms, or Capacities in microfarads or fractions of microfarads, etc. By joining together the appropriate points on the outer lines, the resultant value can be read at the point of intersection with the centre line.



REACTANCE OF CONDENSERS AT RADIO- AND AUDIO-FREQUENCIES.

This chart is intended essentially for determining the relation between the Reactance of a Condenser and its associated Resistance as used in 'de-coupled' circuits. The low-capacity scale (*a*) is read in conjunction with the metres scale (*b*) for H.F. circuits and the high-capacity scale (*c*) with that giving Audio-frequencies (*d*). To read (*c*) in conjunction with (*b*) divide the ohms scale by 1,000. R should exceed ten times Rc.

ADDENDA.

Voltage amplification of a resistance-coupled L.F. magnifying stage is:

$$\mu \times \frac{R}{R + R_a}$$

where
R = external coupling resistance in ohms.
R_a = A.C. resistance (impedance) of valve.
μ = amplification factor of valve.

Formula for peak separation (in kilocycles) in a capacity-coupled filter.

$$\frac{\sqrt{\frac{1}{C_m^2\omega^2} - r^2}}{2\pi L}$$

where
ω = $2\pi \times$ frequency,
C_m = common coupling capacity in farads,
r = equivalent series resistance of the whole tuned circuit,
L = inductance of one filter coil in henrys.

To find equivalent diameter of ribbed coil formers, the overall diameter over ribs should be multiplied by the following constants:

No. of ribs	4	5	6	7	8	9	10
Multiply by	0·806	0·867	0·910	0·936	0·95	0·962	0·970

Power output (in milliwatts) of a valve is given by the formula:

$$KH\mu \left(\frac{V_a}{\mu}\right)^{\frac{5}{2}}$$

where H = a coefficient depending upon the ratio of external to internal impedance in the output valve. If there is no watts dissipation limit, H = 0·041, but with large power valves the load impedance may amount to 4 times the A.C. resistance of the valve, and H becomes a minimum of 0·032.

K = a constant obtained from the maker's valve curves:

$$\frac{I}{\frac{V_a}{\mu} - E_{if}}$$

V_a = anode voltage.
μ = amplification factor.

SECTION XLIV

PATENTS, TRADE MARKS, AND DESIGNS

(Contributed by a Chartered Patent Agent.)

SECTION XLIV

PATENTS, TRADE MARKS, AND DESIGNS.

(Contributed by a Chartered Patent Agent.)

PATENTS.

Application for a patent must be made by the inventor or inventors either alone or in conjunction with one or more co-applicants or a limited company. By inventor is understood either the actual inventor or a person who first introduces an invention from abroad or to whom an invention is communicated from abroad.

When an application is made it is accompanied by either a 'provisional' or a 'complete' specification. In the former case the specification must describe the nature of the invention, but need not particularly describe the same, and the applicant has the right to use and publish the invention during nine months from the date of application, which time may be employed in perfecting the invention and in applying for foreign patents. This protection against the consequences of use and publication is known as Provisional Protection. Should a complete specification not to be filed within the prescribed period of nine months, the application is deemed to be abandoned, and no official publication of the provisional specification takes place. When the application is accompanied by a complete specification, full protection is obtained from the date of the acceptance, and between this date and that of the sealing of the patent the applicant has all the privileges and rights accorded by the patent, except that he will not be entitled to institute any proceeding for infringement, unless and until a patent for the invention has been granted.

When a provisional specification has been filed and it is not desired to abandon the application, a complete specification must be filed within nine months from the date of filing the provisional specification; one month's extension of this time, however, can be obtained on payment of a fee. In such a case the complete specification must describe substantially the same invention as that the nature of which is disclosed in the provisional specification; if not, the Comptroller may require the complete specification to be amended to avoid any disconformity between the two specifications.

A complete specification—whether filed with the application or after a provisional specification—must fully describe (if necessary with the assistance of drawings) the manner of carrying out the invention in practice, and at the same time clearly and correctly point out the features of novelty by a distinct statement of the invention claimed. The complete specification on being filed is submitted to an examiner who makes an investigation to ascertain whether the invention described therein has been wholly or in part claimed or described in any specification (other than a provisional specification not followed by a complete specification) published before the date of the application, and deposited pursuant to any application for a patent made in the United Kingdom within fifty years next before the date of the application, or in the complete specification of any prior application, although such complete specification may not have been filed when the application was made. If any anticipating specifications are found, the applicant will have the opportunity of amending the specification, and, if the amendment is not effected to the satisfaction of the Comptroller, the Comptroller may insert a reference to any of the prior specifications in the specification by way of notice to the public. Where the Comptroller is satisfied that the invention has been wholly and specifically claimed, he may refuse to grant a patent.

When an applicant has two or more provisional applications running concurrently and the subject-matter of such applications is of such a nature as to constitute a single invention which may properly be covered by one patent, the Comptroller may accept one complete specification in respect of the whole of such applications and grant a single patent thereon bearing the date of the earliest of such applications; in case of legal proceedings, however, the respective dates of the several applications may be considered.

Patents of addition for improvements in, or modifications of, an invention already patented may be applied for for any unexpired term of the original patent, and no renewal fees, other than those payable in respect of such original patent, will be payable.

Great Britain is a party to the International Convention for the protection of industrial property, under which an applicant for a patent in any one of the States being parties to the Convention may obtain priority in any of the other States—that is to say, an application may be made in any such States, notwithstanding that publication of the invention may have been made therein, providing that such application be filed within twelve months from the date of the original application.

When a complete specification has been accepted such acceptance is advertised, and the specification will be open to public inspection. Notice of opposition to the grant of a patent may be given at any time within two months from the date of such advertisement, but on the following grounds only, viz.:—

(*a*) That the invention was obtained from the opponent or from a person of whom he is the legal representative.

(*b*) That the invention has been published in the complete specification of a British patent filed in the United Kingdom within fifty years before the date of application for the patent, the grant of which is being opposed, or has been made available to the public by publication in any document (other than a British specification) published in the United Kingdom prior to the application.

(*bb*) That the invention has been claimed in the complete specification of a British patent which, though not published at the date of the application for a patent, the grant of which is opposed, was deposited pursuant to an application for a patent which is, or will be, of prior date to such patent.

(*c*) That the nature of the invention or the manner in which it is to be performed is not sufficiently or fairly described in the complete specification.

(*d*) That the complete specification includes matter not in the provisional specification, and forms the subject of an application made by the opponent in the interval between the leaving of the provisional and complete specifications.

(*e*) That, in case of an application under the International Convention, the specification describes or claims an invention other than that for which protection has been applied for in a foreign state or British possession, and that such other invention forms the subject of an application made by the opponent in the interval between the leaving of the application in the foreign state or British possession and the leaving of the application in the United Kingdom.

If there is no opposition, or, in case of opposition, if the determination is in favour of the grant of a patent, the Comptroller causes a patent to be sealed.

The decision of the Comptroller in most cases is subject to appeal to the Law Officer.

Letters patent extend to Great Britain, Ireland, and the Isle of Man, and are granted for sixteen years, subject to the payment of renewal fees commencing at 5*l.* in respect of the fifth year, and increasing 1*l.* for each succeeding year during the term of the patent, the last payment being 16*l.* These renewal fees are payable before the commencement of the year in respect of which the payment is made, but extensions of time for making these payments up to three months can be obtained, with the addition of a fine, and in the event of a patent lapsing through the non-payment of a fee, the Comptroller is empowered to revive the patent on certain conditions.

At any time after the sealing of a patent the Comptroller may, if requested by the patentee, cause the patent to be endorsed with the words 'licences of right' and any person shall then be entitled, as of right, to a licence under the patent upon such terms as, in default of agreement, may be settled by the Comptroller.

Furthermore, any person interested may apply to the Comptroller alleging that there has been an abuse of the monopoly rights under a patent and asking for relief. Such rights will be deemed to have been abused, if, after four years from the date of the patent, the invention—being capable of being worked in the United Kingdom—is not being worked on a commercial scale; or if such working is being prevented or hindered by the importation from abroad of the patented article; or if the demand for the patented article is not being met to an adequate extent and on reasonable terms; or if trade or industry is being prejudiced by refusal to grant licences, or by conditions attached by the patentee to the purchase, or use of the patented article or to the working of the patented process. If the Comptroller is satisfied that an abuse of the monopoly rights has been proved, he may order the patent to be indorsed with the words 'licences of right' and grant a licence to the applicant on such terms as he may think expedient; or he may order the patent to be revoked.

For 'PATENT SPECIFICATION DRAWINGS,' see page 90.

TRADE MARKS.

Trade marks are registered under the Trade Marks Act of 1905 as amended by the Act of 1919, which provides that a registrable Trade Mark must contain, or consist of, at least one of the following essential particulars:—

1. The name of a company, individual, or firm represented in a special or particular manner.

2. The signature of the applicant for registration or some predecessor in his business.

3. An invented word or invented words.

4. A word or words having no direct reference to the character or quality of the goods, and not being, according to its ordinary signification, a geographical name or a surname.

5. Any other distinctive mark; but a name, signature or word or words, other than such as fall within the descriptions in the above paragraphs 1, 2, 3, and 4, shall not be registrable under the provisions of this paragraph except upon evidence of its distinctiveness.

Provision is also made for the registration of any special or distinctive word or words, letter, numeral, or combination of letters or numerals used as a Trade Mark by the applicant or his predecessors in business before the 13th August, 1875, and which has continued to be used down to the date of the application for registration.

The Trade Marks Act of 1905 as amended by the Act of 1919 provides for a new kind of registration of Trade Marks whereby certain marks, not registrable under the principal Act, can now be registered, provided they have been in *bona-fide* use in the United Kingdom for not less than two years. For this purpose the Register is divided into two parts called Part A and Part B, the new kind of marks being registered in Part B.

An application for registration is made on a form bearing the necessary stamp and should be accompanied by four additional representations of the mark properly mounted. The application is then examined and the mark compared with other marks on record in the Trade Marks Office (this provision may be dispensed with in the case of application to register in Part B), and if, in the opinion of the registrar, any mark or marks be found having such resemblance to the mark sought to be registered as to be calculated to deceive, the applicant will be notified of the conflicting mark or marks, and if, within one month from the date of issuing such notice, no action is taken by the applicant the application will be deemed to be abandoned. Should, however, the applicant so desire, the registrar will appoint a hearing in order that the applicant may attend before him, when the registrar will decide whether the application can proceed. Appeal can be taken from the decision of the registrar to the Board of Trade.

When an application is allowed the mark is advertised in the Official Journal of the Trade Marks Office. The application is then open to opposition for one month from the date of the Journal in which the advertisement appears, and should no opposition be entered the registrar will, upon payment of the registration fee, issue his certificate of registration.

The term of registration of a Trade Mark is fourteen years and may be renewed.

DESIGNS.

For articles which do not form the subject of a patent, protection may in some cases be obtained by design registration where the novelty consists of shape, pattern or ornamentation. Copyright is granted for five years and is renewable for a further similar term, and it is within the discretion of the Comptroller of Patents and Designs to extend the term of a copyright for a third period of five years.

SECTION XLV

COST OF ENGINEERING WORKS

(Contributed by John T. Rea, Author of 'How to Estimate.')

SECTION XLV

COST OF ENGINEERING WORKS.*

(Contributed by John T. Rea, Author of 'How to Estimate.')

AERIAL ROPEWAYS.

Two broad types—Single-rope or Monocable, and Double-rope or Bi-cable. In the former the load carriers hang from and move with an endless hauling wire rope, for gradients up to 1 in 4; in the latter the load carriers hang from a fixed wire rope (double line), and are drawn along with an endless hauling rope, for any gradient.

Aerial ropeways fixed complete, including wire ropes, pulleys, carriers, wooden supports, engine, driving gear, machinery, labour, etc., to carry 10 or 20 tons per hour, in loads not exceeding 5 cwts. per carrier, and gradients as above, costs (1930):—

	Per Mile.
	£ £
Running single-rope system, to carry 10 tons per hour . . .	1,800 to 2,200
" " " 20 " . . .	2,700 " 3,300
Fixed-rope system (double line), to carry 10 tons per hour . .	3,000 ., 4,000
" " " 20 " . .	4,500 " 6,000

Prime Cost of Plant with Steel Structures.—The following table, by Mr. John H. Huntley, M.I.E.S., shows the pre-war standard amounts for aerial ropeways with no peculiar complications, using spiral construction ropes up to 20 tons capacity and locked coil ropes for 40 to 80 tons capacity, and represents the total cost of the plant erected. For lengths over 5 miles costs in proportion.

Capacity in Tons per Hour.	Length of Line in Miles.			
	½ Mile.	1 Mile.	2 Miles.	5 Miles.
Tons.	£	£	£	£
5 per hour . .	1,900	2,000	3,300	6,900
10 " . .	2,100	2,300	3,650	7,400
20 " . .	2,350	2,600	3,900	8,000
40 " . .	2,500	2,870	5,000	10,500
60 " . .	2,850	3,200	5,700	11,900
80 " . .	3,100	3,500	6,300	13,100

Working Costs per Day.—The following pre-war costs, by Mr. John H. Huntley, M.I.E.S., include depreciation, but exclude cost of driving, which varies extremely owing to the difference of height between stations, etc., and cannot be standardised.

Length in Miles.	Capacity per Day in Tons.					
	50 Tons.	100 Tons.	200 Tons.	400 Tons.	800 Tons.	1,000 Tons.
	s. d.	s. d.	s. d.	s. d.	s. d.	s. d.
½ mile . .	17 6	21 0	25 0	32 0	42 4	48 0
1 " . .	19 5	24 0	28 10	37 7	52 0	58 5
2 " . .	26 6	31 6	36 4	49 7	68 10	76 10
5 " . .	45 7	51 0	61 0	83 3	126 0	156 10

* All prices mentioned are pre-war figures unless otherwise stated; present rates (1930) nearly double.

Aerial Ropeway at Loch Leven, Argyllshire (1906).—Erected for the construction of a dam for water-power purposes. Length of line about 4¼ miles. Arranged on the single-rope system, and has a mean gradient of 1 in 22·8, or a difference in altitude of 1,100 ft. against the loads. Three bends in the line, with a station at each bend, and including the power station and both terminals, there are six stations for loading and unloading. Eighty-six trestles, averaging 24 ft. high. Trestles, terminals, angle, and power stations of pitch pine. Driving power is from a Pelton wheel of 250 brake horse-power. Speed of line, 300 ft. per minute. Capacity, 700 to 1,000 tons per day, with 580 steel buckets, each having an individual load of 6 cwt., and spaced about 90 ft. apart. Twenty men are engaged in working the line. Total cost, including carriage, erection, and foundations, etc., complete, about 12,500*l.* or 2,600*l.* per mile.

Cost of operating per ton-mile, allowing for redemption in 3 years, labour and 10 per cent. on the labour account for supervision, was estimated at 2*d.*

Plant for Single-rope Fixed-grip System.—Prices vary greatly according to the ground passed over and the material transported, so the following figures are merely approximate. They are the rates for articles at manufacturers' works to which must be added the cost of packing and marking for export, about 4 per cent. overall, carriage, loading and unloading, and erection with concrete foundations, etc., as well as of motive power, which always differs. The value of timber structures would be about 15 per cent. less than that of corresponding structures in steel. From quotations received in August, 1922.

Description of Parts.	10 tons per 10 hrs.' line. £	25 tons per 10 hrs.' line. £	50 tons per 10 hrs.' line. £
1. Steel wire rope, trestle mountings, grips and carriers for a length not exceeding 1 mile—per mile	400	600	825
Ditto over 1 mile but not exceeding 3 miles in length—per mile	440	660	910
2. Driving and tension gears for a line not exceeding 1 mile	140	160	175
Ditto over 1 mile but not exceeding 3 miles in length	250	300	330
3. Mountings for each angle station giving any degree of deviation	70	80	90
4. Steel trestle structures, average about 16 per mile, height 20 ft., per trestle	18	22	20
5. Steel structures for terminal gears for a line not exceeding 1 mile	100	120	135
Ditto, ditto, for a line over 1 mile, but not exceeding 3 miles in length	160	180	200
6. Steel structures for angle stations, each	50	65	75

Example of Approximate Estimate.—To illustrate the method of estimating from the foregoing particulars the following example is given. The cost of a ropeway ¾ mile long to carry 25 tons per 10 hours with one angle station: steel trestles, and timber structures for terminal and angle stations. From quotations received in August, 1922:

	£	*s.*	*d.*
Steel wire rope, trestle mountings, grips, carriers, as No. 1—¾ of 600*l.* per mile	450	0	0
Driving and tension gears as per No. 2	160	0	0
Mountings for angle station as per No. 3	80	0	0
Steel structures for 12 trestles as per No. 4—22*l.* × 12	264	0	0
Timber structures for terminal gears as per No. 5—120*l.*, less 15 per cent. (18*l.*)	102	0	0
Timber structures for angle stations as per No. 6—65*l.*, less 15 per cent. (9*l.* 15*s.*)	55	5	0
	1,111	5	0
Packing and marking for shipment, 4 per cent.	44	9	0
Total cost of ropeway materials for ¾ mile	1,155	14	0

Add to the foregoing carriage, unloading, erection, foundations, and motive power.

ARTESIAN WELLS.

Artesian bored tube wells, including boring by the percussion, percussion hydraulic, or rotary methods, according to the sub-strata, supplying steel-socketed lining tubes with butt ends, flush-screwed, and special solid steel cutting shoe, and providing all materials, plant and labour, up to 100 ft. in depth; add 10 per cent. for every additional 10 ft. Telescoping the lining tubes

is obsolete, as they should be one continuous tier. Special sizes run up to 4 ft. and 6 ft diameter. See also 'Tube Wells,' p. 2887. Estimated costs (1930) :—

Soil.	Tubes.	Per ft. deep.
In easy soils, such as earth and sand	3 in. to 4 in. external diameter	15*s.* to 20*s.*
	6 in. „ 9 in. „ „	25*s.* „ 34*s.*
In soft rocks, such as chalk or shale	3 in. „ 4 in. „ „	20*s.* „ 25*s.*
	6 in. „ 9 in. „ „	32*s.* „ 42*s.*
In hard rocks, such as basalt or sandstone	3 in. „ 4 in. „ „	30*s.* „ 35*s.*
	6 in. „ 9 in. „ „	46*s.* „ 62*s.*

London.—The charges of the Metropolitan Water Board for domestic supply are based upon 5 per cent. of the rateable value of property, and in the case of a large building, such as business premises or a bank, are out of all proportion to the water used. Charges of 200*l.* to 600*l.* per annum are common, and even 4,000*l.* has been paid. Roughly speaking, any building owner in London paying more than £120 a year for water will find it better to sink an artesian well. Boreholes over 1,000 ft. deep have been sunk in the London basin, and yield 100,000 gallons per hour. The flow is reliable, the water is of a low temperature, pure and soft (generally about 5° of hardness compared with 16° of the Water Board supply), leaves no deposit on pipes and boilers, and the cost is very small.

The capital outlay for many important buildings in London for an independent water supply, including sinking an artesian well and providing the pumping plant, where the quantity runs from 1,000 to 4,000 gallons per hour, varies from 1,500*l.* to 4,000*l.*

The cost of pumping artesian wells depends on the lift and power required, and may be as low as ¼*d.* per 1,000 gallons on a lift up to 400 ft., and is commonly 1½*d.* per 1,000 gallons for manufacturing purposes. Pumping by electric motors, with current at 1½*d.* per unit, will work out to about 2½*d.* per 1,000 gallons when the water is raised some 300 ft. from the borehole to the storage tanks above. The capital invested in a separate installation would thus be repaid in three to six years, leaving only the pumping costs to be considered as an annual expense, so that the value of artesian wells is apparent.

BORING.

Percussion boring by the ordinary 'English' method to depths of 500 ft. costs 10*s.* to 20*s.* per foot, advancing 2 to 6 ft. per 24 hours.

Rope boring for oil, salt, etc., holes 6 to 15 ins. diameter, 20 to 60 ft. per day, costs 2*s.* to 8*s.* per foot. Plant for a hole 2,000 ft. deep weighs 80 tons, and costs about 2,000*l.*

Diamond boring in hard rock, 5 ins. diameter, costs in this country 8*s.* per foot down to 500 ft., and 12*s.* to 18*s.* down to 1,500 ft. Advance 5 to 10 ft. per day. Bort 'diamonds' are 2*l.* to 3*l.* per carat for the softer rocks, and carbon 6*l.* to 12*l.* per carat for the harder ones. Boreholes above 6 ins. diameter are thus almost prohibitive for expense.

Chilled shot method for relatively large boreholes, 6 in. diameter, in hard ground, costs 10*s.* to 12*s.* per foot. It is slower but cheaper than diamond boring, and consists of chilled steel shot fed into a thick steel pipe notched at the lower end and rotated upon the rock to be pierced, with water forced down.

Actual Costs—The following are from practical pre-war experience :—	Per ft. *s.*	*d.*
Rope boring at Middlesbrough, 8 ins. diameter, to depths of 1,000 ft., at rate of 60 ft. per day	8	0
Ditto. in Pennsylvania and Ohio, 6 to 15 ins. diameter, at rate of 20 ft. per day, average contract price	6	0
Diamond boring in Scotland, 5 ins. diameter, to depth of 1,800 ft., advancing about 6 ft. per day	6	4
Ditto. in British Columbia, 2,200 ft. deep, at rate of 10 ft. per day	8	6
Ditto. in Nova Scotia, 2 ins. diameter, sunk by the Government in fairly hard ground to depths of 200 to 400 ft. in 1913, averaged	9	0
Ditto. in Victoria, also sunk by the Government, averaged in the coal measures in 1912	9	3
Ditto. ditto. in the harder rocks	14	0
Chilled shot 6 in. Government boreholes in Nova Scotia in 1913, averaged	10	6
Some important boreholes of the world—		
Bezuidenville, Transvaal, 5,800 ft.	42	0
Czechow, Rybnik, Upper Silesia, 7,350 ft.	43	4
Paruschowitz, Silesia, 6,572 ft. (12 in. bore decreasing to 2½ in.)	11	6
Schladebach, Prussia, 5,737 ft.	21	2

BREAKWATERS.

Breakwaters formed of large rough blocks of stone as a foundation with heavy masonry walls above, forming a harbour. Average width 300 ft., and average depth 50 ft. below low-water mark. 150*l.* to 300*l.* per foot run complete.

Alderney.—Composite type of rubble mound and upright wall above. Mound terminates 12 to 15 ft. below low water. 338 ft. wide at base and 36 ft. at top. 90 ft. mean depth of water. 259*l.* per foot run.

Algiers.—Mound type, with large random concrete blocks surrounding the rubble. 50 ft. mean depth of water. 122*l.* per foot run.

Bizerta.—Composite type, with wall built on caisson system. 58 ft. mean depth of water. 60*l.* per foot run.

Colombo.—South-west breakwater, 1874–85, is 4,212 ft. long, built of concrete blocks weighing 18 to 30 tons, on the sloping bond system, and cost 705,200*l.*, or 167*l.* per foot run. Extension to ditto as a further protective arm, 1907–12, is 1,800 ft. long, also of concrete blocks, and cost 380,000*l.*, or 211*l.* per foot run.

North-east breakwater, 1894–7, is 1,100 ft. long, and constructed as a rubble embankment; the north-west or island breakwater, 1900–6, is 2,657 ft. long, and made of concrete; the two together cost 500,000*l.*, or 133*l.* per foot run.

Total length of the four breakwaters about 9,770 ft., with five lighthouses on them, and total cost 1,585,200*l.*, or an average of 162*l.* per foot run. Depth of sea, 30 to 36 ft. at low water. Rise of tide at Ceylon is only 2 ft. Engineers, Sir John Coode, Son, and Mathews.

Dover.—Wall type, composed of concrete blocks weighing 26 to 40 tons, 82 ft. wide at base and 40 ft. at top. Depth of water, 61 ft. high; 42 ft. low. 415*l.* per foot run.

Holyhead.—Composite type, with very predominant mound of rough rubble stone, 400 ft. average width at base and 40 ft. at top. Sea and harbour walls built upon the mound. 54 ft. mean depth of water. 163*l.* per foot run.

Marseilles.—Mound type. Outer casing of concrete blocks, 25 tons each, and average thickness 14 to 20 ft. Inner casing of rubble stone, 2 to 5 tons weight, about 12 ft. thick. Rubble hearting, ½ to 2 tons stones, and 6 ft. thick. 35 ft. mean depth of water. Joliette section, 72*l.* per foot run, and new section, 109*l.* per foot run.

Naples.—Composite type. 93 ft. mean depth of water. 186*l.* per foot run.

Plymouth (1811–48).—Rubble mound with walls on top, nearly a mile long. Slope of 5 to 1 seawards. 360 ft. wide at base and 43 ft. at top. Average depth, 45 ft. below low-water mark. Absorbed 3,670,440 tons of rubble and 22,150 cubic yards of masonry. Engineer, Sir John Rennie. 250*l.* per foot run.

Portland.—Mound type with large rubble blocks and heavy masonry wall above, 270 ft. wide at base. The largest of its kind in England and in two portions—inner about 2,000 ft. long and outer over 6,000 ft., with 400 ft. opening between them. Depth of water, 58 ft. high; 51 ft. low. 6 million tons of stone used. Convict labour. Total cost, 1,250,000*l.* 140*l.* per foot run.

Maintenance.—This is less for walls than for rubble mounds, as the latter are greatly affected by wave action, and need constant repair. Cost varies according to situation and exposure, and rarely exceeded 1*l.* per foot run per annum, pre-war. For some time it averaged 1*s.* 3*d.* at Holyhead, 7*s.* at Geneva, and 13*s.* at Naples.

(See Section XXVIII.)

BRIDGES.

Types.—(1) Arched, (2) Cantilever, (3) Girder or lattice, (4) Movable (bascule or lifting, rolling, swing, and transporter), (5) Pontoon (with barrels, boats, planks, etc.), (6) Suspension, (7) Trestle (with very high steel or wood framework piers), and (8) Viaducts or "land" bridges.

Costs include foundations, piers, abutments, approaches, footpaths, machinery, etc., complete.

Description.	Span.	Number of Spans.	Cost per ft. sup. (1930).
	ft. ft.		
Brick or masonry	50 to 100	Single	£2 to £4
	50 „ 100	Multiple	£3 „ £6
Ordinary concrete	30 „ 50	Single	£2 „ £3
	30 „ 50	Multiple	£3 „ £4
Reinforced concrete	50 „ 100	Single	£1 „ £2
	50 „ 100	Multiple	£2 „ £3
Steel bascule or lifting	50 „ 100	Single	£5 „ £8
	100 „ 200	Multiple	£8 „ £12
Steel trussed girder or lattice	50 „ 100	Single	£2 „ £5
	50 „ 100	Multiple	£3 „ £6
Steel suspension, with towers	50 „ 100	Single	£2 „ £4
	100 „ 200	Multiple	£4 „ £7
Steel swing, with pivot pier	50 „ 100	Single	£3 „ £5
	100 „ 200	Multiple	£5 „ £8

Brooklyn Bridge (1870–83).—Wire cable suspension bridge over the East River, connecting New York with Brooklyn. 5,990 ft. total length, including 1,596 ft. central span from tower to tower, two spans of 930 ft. from the towers to anchorage on either side, and the approaches of ironwork and masonry, 1,563 ft. on New York side and 971 ft. on Brooklyn side. Width 85 ft., divided into five passage ways for vehicles, trams, and foot passengers. Clear height above high water 135 ft. in centre and 118 ft. at towers, giving a free passage to shipping. Towers 277 ft. above high water, upon rock foundation 80 ft. below surface of river and 40 ft. beneath its bed. Supporting the spans are four cables nearly 16 ins. diameter, and 3,758 ft. long, with anchorages of solid stone masonry weighing 60,000 tons each, and rising 90 ft. above river's edge. Each cable contains 5,282 galvanised steel wires in 19 separate strands, of 278 lengths, and each strand has over 200 miles of continuous wire. Wires laid parallel, not twisted, with spiral wire wrapping. Deflection of cables between towers is 128 ft. Engineer, John A. Roebling. Total cost, including approaches, 3,100,000*l.*, or 518*l.* per ft. run, or 6*l.* per ft. sup.

Forth Bridge, Scotland (1883–90).—At South Queensferry, near Edinburgh. Steel latticework, with two main cantilever spans 1,710 ft. each, and 24 land spans, 42 to 688 ft. each. Each cantilever 690 ft. long, united by a girder 250 ft. span. The steel towers supporting the cantilevers are 361 ft. high, and the steel tubes of both are 12 ft. diameter at base, tapering to extremities. 50,960 tons of steel employed and 6½ millions of rivets. Complete length, 8,100 ft., and with approaches, 8,300 ft.; width, 30 ft.; 145 acres of surface to be painted. Double line railway. Height of railroad, 152 ft. above high water. Main piers of masonry, and concrete foundations in rock, 70 ft. below high water. Central pier of granite standing on the islet of Inchgarvie. Engineers, Sir John Fowler and Sir Benjamin Baker. Contractors, Sir Wm. Arrol & Co. Total cost, including approaches, 1,700,000*l.*, or 25*l.* per ft. run or 7*l.* per ft. sup.

Gôkteik Viaduct (1901).—In the Shan States, Upper Burmah, the viaduct forming part of the railway where it crosses a deep wide valley about 80 miles from Mandalay, and runs on to Kunlon at the Chinese frontier. The second most lofty steel trestle bridge in the world, but for combination of height, length, and weight of material easily the first. It is straight for 1,638 ft. at the centre, with terminal curves of 800 ft. radius, 341 ft. and 281 ft. long respectively. Total length between abutments, 2,260 ft. There are 15 trestles or skeleton towers, varying from 250 ft. to 320 ft. in height, 10 spans of 120 ft., and 7 spans of 60 ft., or 17 spans altogether, formed with lattice or plate girders. 5,000 tons of metal, and 200,000 rivets. All the metal parts, some 230,000 pieces, were prepared in the United States, shipped 10,600 miles, and transported 460 miles up country by the Burmah railway, where they were re-assembled and put up without a hitch. A single track was first laid, but space was left for a second when the traffic required it. Total cost, 61,000*l.*, or 27*l.* per ft. run, or 3*l.* 10*s.* per ft. sup.

Newcastle-Gateshead High-level Road Bridge (1924–28).—A combined single steel arch and suspension-bridge over the Tyne, forming a link in the Great North Road between Newcastle and Gateshead. It spans the river from bank to bank, 531 ft. length of actual crossing, being the longest arched bridge in Great Britain. The total distance of the roadway carried by channel steel hangers from the arch, together with the approach spans on either side, is 1,270 ft., and the complete weight 8,000 tons (including arch, 2,400 tons; deck of roadway, 1,600 tons; and approach spans, 4,000 tons). Masonry approaches, 581 ft. in addition. This great arch is of the two-hinged type, constructed of two mild steel trusses 45 ft. apart, with parabolic arched ribs and web members, 20 ft. maximum depth at centre. The arch was built out from both sides of the river simultaneously, and erected without any staging or supports from below, so as not to interfere with shipping. The foundations necessitated sinking heavy caissons, 84 ft. × 28 ft., to solid rock 70 ft. below ground level, each caisson containing 1,000 tons of concrete. The towers at either end of the arch are steel-framed buildings faced with granite, and may be used as warehouses. The approach spans were built up at the shore ends of the bridge and rolled forward into position, and are supported by octagonal steel columns 70 ft. maximum height, each standing upon a cast-iron cylinder, 21 ft. diam., filled with concrete and sunk to a firm bearing. The bridge road-way is 38 ft. wide, carries a double line of tramways and ordinary traffic, and two footpaths cantilevered to a width of 9 ft. on either side. It is designed to take exceptional loads of 100 tons on four wheels, and two 2 ft. diam. water mains and two 2 ft. gas mains are carried under the paths. The roadway is 93 ft. above high water, with 84 ft. clear headroom for vessels below, and the rise of arch is 170 ft. Engineers, Messrs. Mott, Hay and Anderson, with whom were associated Messrs. Coode, Fitzmaurice, Wilson and Mitchell. Architect, Mr. R. Burns Dick, F.R.I.B.A. Contractors, Messrs. Dorman, Long & Co., for whom Mr. Ralph Freeman, M.I.C.E., acted as consulting engineer, and Mr. Charles Mitchell as civil engineer. The contract was 571,225*l.*, but street improvements, acquisition of land and property, clearance of slums, etc., brought the total up to 1,150,000*l.*, of which the Ministry of Transport gave 60 per cent. Unit cost of bridge on contract price, with arch span and two approach spans or an overall length of 1,270 ft., came to 450*l.* per ft. run, or 8*l.* per ft. sup., including width of footpaths.

Newport Bridge, Mon. (1924–7).—Over the river Usk. Replaces an older stone and iron bridge of five arches and only 35 ft. total width, built in 1800, and widened in 1864. The new bridge, opened in June, 1927, is partly of reinforced concrete and stone, with five arched steel spans between four river granite piers with concrete hearting and two end abutments of stone backed with concrete. The piers are in cross pairs, built up on two rectangular caissons sunk by compressed air and filled with concrete, the space between each pier being bridged by a semi-

circular cross arch, 41 ft. 6 ins. diameter, leaving the old pier foundation undisturbed but eventually protected with concrete. Length of bridge from face to face of abutments is 357 ft., with a 36-ft. carriage-way and two footways each 12 ft. wide, making an overall width of 60 ft. between the cast-iron parapets. The concrete roadway and stone-flagged footpaths are carried in each of the five spans by decking plates on seven steel arched ribs, 8 ft. 6 ins. apart, the paths being cantilevered outwards from the face rib on each side. The bridge carries gas and water mains, electric and post-office cables, and twelve bronze lamp standards. Engineers, Messrs. Mott, Hay, and Anderson, conjointly with the borough engineer, Major W. E. Lloyd, M.I.C.E., and the resident engineer, Mr. G. L. Groves. Contractors, Sir Wm. Arrol & Co. Contract price, including demolishing the old bridge, was 140,275*l.*, or 393*l.* per ft. run, or 6*l.* 11*s.* per ft. sup.

Niagara Suspension Bridge (1852–5).—Connects the United States with Canada. A wire-rope suspension bridge for road and railway, with a single span of 821 ft., centre to centre of piers and 245 ft. above the water. Superstructure is a hollow rectangular box, 24 ft. wide × 18 ft. deep, on the top of which the railway is laid, while the bottom forms a roadway for public traffic, both floors being constructed of timber beams, etc. Railway floor is suspended from two wire cables, 10 ins. diameter, at intervals of 5 ft., and the lower floor by two similar cables. 120 stays. Two masonry towers at each end of the bridge, 60 ft. high, 15 ft. square at base, tapering to 8 ft. square at top, and pierced by an arch. Anchor chains embedded in masonry 7 ft. × 3 ft., and 25 to 35 ft. deep. Engineer, John A. Roebling. Total cost, 80,000*l.*, or 97*l.* per ft. run, or 4*l.* per ft. sup. This suspension bridge was replaced by a steel arch bridge in 1897.

Tower Bridge, London (1890–4).—Steel, with two fixed shore suspension spans 270 ft. each, and a lifting central span of 200 ft. Two large steel towers encased with stone, with two bascules or drawbridges in the centre and connected at top by a high-level footbridge, 140 ft. above high water. Bascules weigh 1,000 tons each, and are raised and lowered by hydraulic machinery which is contained in the chambers of the piers supporting the towers, each pier being 180 ft. long × 70 ft. wide. Staircases and hydraulic lifts are provided in the towers for foot passengers crossing the high-level bridge when the bascules are raised for the passage of shipping. Complete length, with approaches, 2,980 ft. Width of fixed spans between parapets, 60 ft., and breadth across centre span 50 ft. Engineer, Sir John Wolfe Barry. Sir Horace Jones, the City Architect, designed the architectural details. Contractors: Sir John Jackson for foundations; Sir Wm. Arrol & Co. for steelwork; and Sir W. G. Armstrong & Co. for hydraulic machinery. Total cost, including machinery and approaches, 903,000*l.*, or 303*l.* per ft. run, or 5*l.* per ft. sup.

Suspension Bridges.

Estimated costs by a well-known firm of steel manufacturers of strong steel wire rope suspension bridges of large power, designed to carry 15-ton traction engines with wagons in line on the bridge floor, the prices being for the whole of the iron and steel work, including lattice piers surmounted by cast-iron saddles for the ropes to run over, all bolts and nuts, holding down bolts, steel rod suspenders, open-steel parapets, high tension steel cables, steel cross girders to support decking, steel wind ties under floor, etc., ungalvanised but painted two coats before despatch, delivered f.o.b., London or Liverpool, or to railway stations in England; supplied only, and exclusive of timber floor, foundation work, and erection. Pre-war rates. If a concrete or asphalt floor is required the prices will be subject to an increase, owing to the additional strength necessary and the greater weight put on the structure. The span is centre to centre of piers, and the feet super. for area unit price, have been reckoned accordingly.

Span.	Width.	Area.	Total Price.	Price per Foot Super.	Weight of Steelwork.
Ft.	Ft.	Ft. Sup.	£	*s.* *d.*	Tons.
50	19	950	565	11 11	31
75	19	1,425	785	11 0	44
100	19	1,900	945	9 11	53
125	19	2,375	1,115	9 5	60
150	19	2,850	1,275	8 11	66
175	19	3,325	1,430	8 7	75
200	19	3,800	1,575	8 3	81
225	19	4,275	1,740	8 2	91
250	19	4,750	1,880	7 11	100
275	19	5,225	2,065	7 10	110
300	19	5,700	2,230	7 10	118

The prices may therefore vary from about 8*s.* to 12*s.* per ft. sup. If erected complete, including timber floor and concrete, stone, or brick foundations or abutments, and anchorages, the costs would be at least double the foregoing. Narrower bridges, down to 3 ft. wide, for lighter loads, such as foot traffic, cattle, mules, horses and carts, would be of reduced construction and correspondingly cheaper.

Actual costs of steel wire-rope suspension bridges, delivered and erected, including steel piers, wire-ropes, suspension bands, and timber for floor, but exclusive of any foundation work or anchorages. Pre-war rates. The span is centre to centre of piers, and the feet super. for area unit cost have been reckoned accordingly as before.

Locality.	Load.	Span.	Width.	Area.	Total Cost.	Cost per F.S.
		Ft.	Ft.	Ft. Sup.	£	*s. d.*
Bournemouth, across Alum Chine	Donkey carts and bath chairs	230	6	1,380	590	8 7
Derby sewage works	Foot traffic	86	3	258	165	12 9
Dryburgh, across the Tweed, near Melrose Abbey	Foot traffic	260	6	1,560	405	6 11
Linley, over the Severn. Two approach spans in addition, making a total span of 390 ft.	A row of 10-ton carts	213	12	2,556	1,700	13 3
Llandilo	Foot traffic	144	5	720	220	6 1
Llangollen	Foot traffic	146	3	438	190	8 8
Rhayader, for the Birmingham Corporation Waterworks	Cart traffic	110	10	1,100	450	8 2
Shrewsbury. Two approach spans of 30 ft. each in addition, making a total span of 210 ft.	Light vehicular traffic	150	8	1,200	950	15 10
Swalwell, over the Derwent	Foot traffic	125	4½	562	285	10 2
Warrington	Foot traffic	183	6	1,098	430	7 10
Wrexham, across the Dee.	Row of loaded haycarts	130	10	1,300	670	10 4

The average rate was therefore about 10*s.* per ft. sup. If erected complete, including foundation work and anchorages, the costs would be at least double foregoing.

Bridges of Concrete.

Coulouvrenière Bridge, Geneva (1895–6).—Across the Rhone, 351 ft. wide, and total length of bridge over the abutments 492 ft. Width between parapets 61 ft. 4 ins., two footways 12 ft. 6 ins. each, with a carriageway 36 ft. 4 ins. between the kerbs. Two large arches with spans 131 ft. 3 ins., and rise of 17 ft. 6 ins. Arches 3 ft. 3 ins. thick at both springings and centre, about 4 ft. 6 ins. at the haunches, and steel pivoted. Two smaller arches, 35 ft. 5 ins., and 39 ft. 4 ins., but are not pivoted, being elliptical. Very handsome bridge. Arches, spandrels, 9 spandrel walls, 2 ft. thick, 12 in. top table, and filling, all of concrete, 1 to 4½, 6, 7, and 8. Piled abutments. Facing of spandrel walls, columns, cornice, and parapet of stone. Wooden centres were used, with sand boxes for adjusting and striking. Total cost, 68,000*l.*, or 138*l.* per ft. run, or 2*l.* 5*s.* per ft. sup. of roadway and footpaths. Cost per cubic yard, including centering and scaffolding, was 3*l.* 2*s.* 3*d.*

Eyach Bridge, Hohenzollern (1895).—Designed by Herr Max Leibbrand, of Sigmaringen. A single span of 98 ft. 5 ins., with a rise of 9 ft. 10 ins. Concrete arch and table, 1 to 7½, and foundations to abutments, 1 to 9, which are piled. Pivots or joints made of granite instead of iron or lead. Bridge is 143 ft. long, 12 ft. 6 ins. total width, with roadway 7 ft. 6 ins. wide, and each footpath 2 ft. 6 ins. The footpaths overhang the arch on either side and were formed of old iron rails. Plain iron railing as parapet. Cost, 900*l.*, or 6*l.* per ft. run, or 10*s.* per ft. sup.

Kirchheim Bridge, Württemburg (1897).—Over the Neckar, which is 361 ft. wide at this point, with 230 ft. of foreshore, or a total of 591 ft. Designed by President von Leibbrand, chief engineer to the State of Württemburg. Length of bridge from outside to outside of abutments, 592 ft., and space between parapets 17 ft. 8 ins., including road, 12 ft. 8 ins., and two footways, 2 ft. 6 ins. each ; open iron parapets. Four spans of concrete arches, 124 ft. 8 ins. each, with a rise at the centre of 18 ft. Thickness of arch at crown, 2 ft. 8 ins., increasing to 3 ft. at springing. Concrete piers 7 ft. 6 ins. thick at springing level, faced with stone at water level ; cutwaters also faced with stone. Concrete in arches and abutments, 1 to 7½, spandrel walls, tables, etc., 1 to 9, and foundations for piers, 1 to 4. Concrete on face was tooled and coloured, and concrete in piers rendered 2 ins. thick with 1 Portland cement and 2 sand. Lead joints in the centres and springings of arches 6 to 7 ins. wide, bedded and recessed in stone. Screw jacks were used for striking the centres in place of folding wedges or sand boxes. Cost, 7,000*l.*, or 12*l.* per ft. run, or 13*s.* 5*d.* per ft. sup. of bridge table. Approaches, 2,250*l.* in addition.

Munderkingen Bridge, Württemburg (1893).—Over the Danube. Designed by President von Leibbrand. One span only of 164 ft., and rise 16 ft. 5 ins. Thickness of arch at centre 3·28 ft., at springing 3·61 ft., and at haunches 4·26 ft. The first concrete bridge where steel pivots on bearing boxes were used, one set at each abutment and one set at the crown. Internal and external spandrel walls, abutments, horizontal and intermediate tables, cornice, and arch itself, were of concrete, 1 to 7½, and tinted. Spandrel face of limestone, and ornamental open iron parapet. Cost of bridge, including piled abutments, 3,600*l.*, or 21*l.* per ft. run, or 13*s.* 6*d.* per ft. sup. of road and footway surface. Approaches, 1,000*l.* in addition.

Neckarhausen Bridge, Hohenzollern (1900).—Over the Neckar. Designed by Herr Max Leibbrand, of Sigmaringen. The handsomest concrete bridge in Germany, the concrete being coloured, and no stone used. A single concrete arch of 164 ft. clear span, with a rise of 15 ft. Arch ring 2 ft. 10 ins. thick at the centre, 3 ft. at the springing, and 4 ft. at the haunches. The arch is tapered on plan, being 1 ft. 8 ins. wider at the springing than at the centre. Concrete spandrel columns and arches over, instead of solid spandrel walls and haunches, thus reducing both material and dead load. Arches pivoted with steel pins and cast-iron brackets or bearings, and those carrying roadway were strengthened with iron rails built into the top and abutments. Concrete in arch 1 to 7½, faced with ⅝-in. spalls of limestone. Voussoirs, cornice, etc., were coloured. Width of macadam roadway 13 ft., and overhanging footpaths 2 ft. 6 ins., which is a cheap method. Centering was supported on temporary piles, with screw jacks for striking. Cost, 3,000*l.*, or 17*l.* per ft. run, or 19*s.* 5*d.* per ft. sup. Long approaches, 5,000*l.* in addition.

BRIDGES OF REINFORCED CONCRETE.

Spans, 50, 75, or 100 ft. External width, including parapets, 25 ft. Height, from mean water level to underside of eliptical arch, 13 ft. 1*l.* to 3*l.* per ft. sup. (1930).

Items.	Estimated Cost (1930).		
	50 ft. Span.	75 ft. Span.	100 ft. Span.
	£	£	£
Excavations	140	220	370
Concrete	500	930	1,336
Centering	260	480	680
Reinforcing Steel	210	390	556
Preliminaries 10 per cent.	110	200	294
Contractors' Profit, etc., 15 per cent.	180	330	484
Total Cost	1,400	2,550	3,720

Alexandra Bridge, Pietermaritzburg (1914–15).—Reinforced concrete on the Kahn system, 170 ft. long × 30 ft. wide between the parapets, with two 5-ft. footpaths cantilevered over a 20-ft. roadway. Central arch 75 ft. span, and a half arch 38 ft. span at each end. Cost, including 5 per cent. for designs and supervision, 5,906*l.* for the bridge and 4,008*l.* for the approaches = 9,914*l.* total. 52*l.* per ft. run, or 2*l.* per ft. sup.

Brivio Bridge, Italy (1913–17).—Over the river Adda, and gives access to the railway station of Cisano Bergamasco. Entirely constructed of reinforced concrete, 444 ft. long and 30 ft. wide, with a 24 ft. roadway flanked by two raised footpaths, each 2 ft. 7 ins. wide, with edges of L irons. The roadway is about 26 ft. above the average level of the river, and suspended from three reinforced concrete arches overhead. Span of centre arch 177 ft., and of side arches 144 ft., all being supported on two massive concrete pillars and shoulders, each 43 ft. × 13 ft., built on a group of 48 steel-shod concrete piles made on the Considère system, 14 ins. square, and driven 43 to 53 ft. through the bed of the river by a steam hammer weighing 4 tons. Normal load on each pile averages 35 tons, uniformly distributed by a superimposed concrete slab 3 ft. thick. Foundations protected by pitch pine shuttering 5 ins. thick. Sole of roadway 5 ins. thick, covered with asphalt and a layer of concrete and gravel on top, with 12 in. × 30 in. transverse beams underneath attached to 11 in. × 24 in. vertical bars and to 18 in. × 40 in. longitudinal external beams. Cast steel rollers for expansion. Parapet is of cast ironwork. Bridge designed to carry a double-track electric tramway, a steam roller of 22 tons, and a dense crowd of people calculated at 10 cwt. per square yard. Total cost, 20,000*l.*, or 45*l.* per ft. run, or 30*s.* per ft. sup.

Chatellerault Bridge, France (1900).—Across the river Vienne, and is formed entirely of ferro-concrete on the Hennebique system. Centre span 164 ft. and two side spans 135 ft. each, with rises of respectively 16 ft. and 13 ft., and a width between the parapets of 26 ft. 3 ins., excluding two cantilevered footways which overhang 3 ft. 5 ins. Spans of four arched concrete ribs, 35 ins. deep × 19 ins. wide at springings, tapering to 9 ins. square at the soffit, and extrados connected together by an arched table. Roadway and footpaths supported by spandrel pillars, 8 ins. square, and longitudinal concrete beams, the top table being 10 ins. thick at the centre, tapering to 5 ins. at kerb. The bridge only weighs 250 lbs. per ft. sup. Cost, 8,000*l.*, or 18*l.* per ft. run, or 12*s.* 6*d.* per ft. sup.

Loughor Bridge, Glamorganshire (1920–23).—Spans the tidal estuary of the river Loughor, and replaces the old timber trestle structure. The new bridge is of reinforced concrete, from a design by Messrs. L. G. Mouchel and Partners, carried out by the Yorkshire Hennebique Contracting Co., Ltd. Length, 720 ft. between the abutments, divided into 18 spans of 40 ft. each. Width, 36 ft. between the parapets, providing two footways, 5 ft. wide, and a carriage-way of 26 ft. between the kerbs. Piers rest on reinforced concrete piles, 15 ins. square, driven into the bed of the river to various depths of 5 ft. to 37 ft. Four piles to each pier, each pile being encased in a reinforced concrete cylinder, 1 ft. 9 ins. diameter, cross-braced with reinforced concrete tie-beams, 10 ins. × 14 ins. Main longitudinal beams, 12 ins. wide × 2 ft. 8 ins. deep in the centre; secondary beams, 7 ins. × 18 ins. Four main or longitudinal beams and 12 secondary or cross beams to each span. 5½ ins. reinforced concrete decking, covered with 2 ins. cushion of sand and finished with 4 ins. of 'Tarmac' roadway. Ducts, 18 ins. × 9 ins., left under footpath concrete flagging to accommodate electric cables. Parapet of pierced concrete panels, with moulded handrail. Abutments have reinforced concrete wall supporting the ground behind. Steel reinforcement was of plain round steel bars, $\frac{5}{16}$ in. to 1⅛ in. diameter, and the bridge was built in half-widths, so as not to interrupt the traffic. It was designed to carry a normal working super. load of 120 lbs. per square ft., and a rolling load of 48 tons. Total cost, excluding approaches, 56,700*l.*, or 79*l.* per ft. run, or 2*l.* 2*s.* 7*d.* per ft. super.

Pontypridd Bridge (1908).—Over the river Taff. Reinforced concrete on the Hennebique system, with the two piers on reinforced concrete piles. Total length 170 ft., and width 30 ft. Central river span 117 ft. Total, 3,600*l.* 21*l.* per ft. run, or 14*s.* per ft. sup.

Terralba Bridge, Italy (1916).—Part of a scheme for enlarging the Genova Brignole Station. It replaces about 600 ft. of the Terralba highway connecting St. Martino with St. Fruttuoso. Total length of bridge, including approaches, is 750 ft.; width 45 ft., including two footways each 5 ft. wide. Constructed of reinforced concrete and has six spans, four of 61 ft. and two of 58 ft., each span being built in three bays across. Part of the inclined approach was also formed in six spans, each of 35 ft. and three bays across, the remainder of the plane being constructed of massed work between supporting walls. Subsoil was a deep bed of soft, marshy clay, which prohibited piling and required a specially light foundation formed by 1 per cent. reinforced concrete rafts, the reinforcement being carried up into the pillars of the bridge. The rafts were supported on two pinewood piles per square yard, each 20 ft. long by 10 ins. wide, and tops coated with cement. Volume-ratio of reinforcement in superstructure was 1·8 per cent. to the gravel concrete. Total cost, 18,000*l.*, or 24*l.* per ft. run, including approaches, or 12*s.* 5*d.* per ft. sup. of concrete, including the foundation, and 6*s.* 8*d.* per ft. sup. of road and footpath.

Warrington Bridge (1913).—Over the river Mersey. Largest single-span reinforced concrete bridge in England, on the Considère system. River walls and abutments also of reinforced concrete. Clear span 134 ft., width 80 ft. between parapet walls, and ten parabolic ribs supporting upright and cross beams on which rest 6½ in. floor slabs. Roadway has wood-block paving, and footpaths have concrete slabs. Total cost, including approaches, 24,000*l.* or 34*s.* per ft. sup. ditto.

CANALS.

The total length of canals for inland navigation in the United Kingdom is 4,673 miles (3,641 in England and Wales, 184 in Scotland, and 848 in Ireland). 1,144 miles are owned by the railway companies, and 219 controlled by them. An ordinary inland canal in this country is 25–30 ft. wide at bottom, 40–45 ft. wide at water level, and 4–5 ft. deep. Canal boats are 70–75 ft. long and 7–8 ft. wide; barges are about the same length, but double the width, *i.e.* 14–15 ft. For maps and towns, etc., see the *Handbook on Canals*, with map, issued by the Canal Control Committee (Board of Trade), and obtainable from H.M. Stationery Office, Kingsway, London, W.C. 2, price 6*d.* net, or 8*d.* post paid. (November, 1918.) Total paid-up capital was 47,550,000*l.*, and the cost varied from about 5,000*l.* to 20,000*l.* per mile.

In 1905 the constructional cost of new canals in Great Britain, 40–45 ft. wide at bottom, 80–85 ft. wide at water level, and 8 ft. deep in centre, with locks 230 ft. × 22 ft. and 6½ ft. draught of water over sills, suitable for two lighters of 250 tons carrying capacity, was estimated at 30,000*l.* per mile. The cost of improving existing routes to reach the same standard was similarly reckoned at 15,000*l.* per mile. New barge canals would now cost (1930) 50,000*l.* per mile.

In France, in 1870, there were 3,000 miles of canals and 2,000 miles of canalised rivers. The average cost of the French canals up till then was 11,600*l.* per mile, and of the canalised rivers, 4,800*l.* per mile. Present length, 7,500 miles.

BARGE OR BOAT CANALS.

Basingstoke Canal (1794–96).—From Basingstoke, Hampshire, to the river Wey Junction Canal, Surrey. Length, 37 miles, including 29 locks. Total cost, constructional works only, 170,000*l.* or 4,600*l.* per mile.

Bridgewater Canal (1759–61).—In Lancashire; originated by the Duke of Bridgewater to bring coal from his mines on the Worsley estate to Manchester, but it was later extended to the Mersey. First length was only 10 miles, but complete length 38 miles. 50 ft. wide and 5½ ft. deep, with vertical stone walls instead of banks. The Barton aqueduct carried the canal over the river Irwell, at a height of 39 ft. Engineer, James Brindley. (The Bridgewater Canal was purchased by the Manchester Ship Canal Co. in 1887.) Total cost, constructional works only, 235,000*l.*, or 6,180*l.* per mile.

Dortmund and Ems Canal (1890–1900).—From Dortmund to the North Sea at Emden. Part of the 157 miles total length includes canalisation of the river Ems. Bottom width, 59 ft., 98 ft. at water level, and depth, 8 ft. Works include 1 canal lift, 3 masonry aqueducts, 185 bridges, and 20 masonry locks, 220 ft. long each. Artificial canal portion cost 29,000*l.* per mile, and river canalisation portion, 14,000*l.* per mile. The locks were 15,000*l.* each, and the weirs 8,500*l.* each. Total cost, 3,970,000*l.*, including 400,000*l.* for land.

Cost per mile of total length, including land	. . . .	25,286*l.*
,, ,, ,, ,,	constructional works only . .	22,740*l.*

Grand Junction (begun, 1790).—From the Oxford Canal at Bramston, Northamptonshire, to the Thames at Brentford, Middlesex. Width at bottom, 24 ft., and 36 ft. at water level. The main line is 93 miles long, but the total length, with branches, is 140½ miles (189 by Whitaker). Cost per mile, including land, etc., 19,500*l.*

Ulster Canal (1826–41).—Unites Lough Neagh with the upper Lough Erne, and extends from Charlemont, Co. Armagh, on the river Blackwater (which is navigable from Charlemont to Lough Neagh), passes Caledon, Monaghan, and Clones, and joins Upper Lough Erne, Co. Fermanagh, near Belturbet. Length, 47 miles. Total cost, constructional works only, 200,000*l.*, or 4,255*l.* per mile.

Worcester and Birmingham Canal (1791).—Connects these towns, and joins the Severn at the former. Runs through a tunnel 500 yards long, and three shorter tunnels. All are 18½ ft. wide, 7 ft. deep, and 18 ft. high. Length, 30 miles. Original cost per mile, 20,000*l.*

Irrigation Canals.

For irrigating large tracts of land for growing cotton, rice, sugar, wheat, etc., especially in America, Australia, Egypt, and India. The cost varies according to geographical features, nature of ground, available water supply, size of main canal, number of branches, distributing channels, and subsidiary works. In India the costs are approximately—

5,000*l.* to 10,000*l.* per mile of canal and main branches.
2*l.* to 3*l.* per acre of land irrigated.

Agra Canal (opened 1874).—In India, and draws its water supply from the Jumna at Okhla, ten miles below Delhi, irrigating 380,000 acres. The main and branch canals total 110 miles, and the distributing channels 600 miles. Total cost, including all works, 800,000*l.*

7,270*l.* per mile of canal and main branches
2*l.* 2*s.* per acre of land irrigated.

Cavour Canal (begun 1862).—In Italy, and starts from the river Po, not far from Turin, and terminates in a junction with the river Tessin above Novara. Length, 51 miles, with a total fall of 71 ft. The summer discharge of the Po being found insufficient, a branch canal was constructed for bringing a supplementary supply from the Dora Baltea, which rises from the glaciers of the Alps and possesses a large discharge in the summer when the Po is low. Total cost, 4,104,000*l.*, or 80,470*l.* per mile of main canal.

Ganges Canal (opened 1854).—The largest irrigation work in India, and waters the district between the Ganges and the Jumna. Designed by Sir Proby Cautley. The canal with its main branches is about 650 miles long, and can irrigate an area of 1,205,000 acres. It was remodelled about twelve years later. Total cost up to 1878 was 3,055,000*l.*

4,700*l.* per mile of canal and main branches.
2*l.* 11*s.* per acre of land irrigated.

Lower Ganges Canal (1870–90 approximately).—An extension to supplement the original foregoing canal. Designed for a total length of 582 miles. Irrigates 2,500,000 acres. Total cost about 4,500,000*l.*

7,730*l.* per mile of canal and main branches.
1*l.* 16*s.* per acre of land irrigated.

Upper Swat Canal.—In India, along mountainous country at the Malakand hills. Length, 150 miles, and irrigates 450,000 acres. Works include a tunnel, 11,235 ft. long by 18 ft. wide and 13 ft. high, seven shorter tunnels totalling one mile, 167 aqueducts across ravines, 20 to 100 ft. deep, a weir 700 ft. long, stop-dam, pitched channel, etc. Total cost about 1,300,000*l.*

8,670*l.* per mile of canal and main branches.

2*l.* 18*s.* per acre of land irrigated.

Ship Canals.

Large waterways as formed for ocean-going shipping, complete with channels, locks, docks, machinery, entrance breakwaters, etc.; construction works only, and excluding promotion, purchase of land, compensation, or legal charges:

100,000*l.* to 500,000*l.* per mile of actual canal.

Amsterdam Canal (1865–76).—Connects Amsterdam and the Zuyder Zee with the North Sea *via* Lake Y and Wyker Meer, and across a narrow neck of sand-hills in a westerly direction. Works comprise a dam 4,462 ft. long, with maximum bottom width 147 ft. and top width 13 ft., shutting off Lake Y from the Zuyder Zee. There are five large locks, six sluiceways, three swing-bridges (two railway and one roadway), and a harbour of 260 acres at the North Sea entrance, protected by two rubble stone breakwaters, 5,070 ft. long each, with pier on top formed of concrete blocks, 6 to 12 tons each. Canal is 16½ miles long, 88½ ft wide at bottom, 186 ft. at water level, and 23 ft. deep (now increased to 30 ft.). 21,000,000 cubic yards of soil and mud were excavated and dredged and 13,000 acres of land reclaimed. The sale of the latter realised about 70*l.* an acre, or a total of some 910,000*l.*, which mitigated the engineering cost. The largest type of dredger used could deposit 1,300 cubic yards of material on the banks in twelve hours, at 2*d.* per cubic yard. Permanent pumping machinery was installed for draining the adjacent lands into the canal and out again into the sea. Engineers, Sir John C. Hawkshaw and Mr. Dirke. Contractors for the whole of the works, Messrs. Henry Lea and Son. For full details see paper by Mr. Harrison Hayter read at the Institution of Civil Engineers in 1880, and printed in the *Proceedings*, vol. lxii. p. 1.

	Total.	Per Mile.
Constructional works only . . .	2,600,000*l.*	157,580*l.*
Other expenditure and charges . .	400,000*l.*	24,240*l.*
Whole cost	3,000,000*l.*	181,820*l.*

Caledonian Canal (1803–23).—From Inverness to Fort William, along the Great Glen of Scotland, and connects Inverness Firth with Loch Eil, having Lochs Ness (22 miles), Oich (6 miles), and Lochy (10 miles) in between. The summit level of 102 ft. is overcome by 28 locks, 14 on either side, each about 170 ft. long × 40 ft. wide; 23 miles of artificial or canal navigation in four stages, and 38 miles of natural lake navigation through the foregoing three lochs, the total distance between extremities being thus 61 miles. Canal is 50 ft. wide at bottom, 120 ft. at water level, and 17 to 20 ft. deep, for vessels up to 600 tons burden. Engineer, Thomas Telford. Total cost, including land, 1,311,270*l.*, or 57,010*l.* per mile of actual canal length of 23 miles.

Corinth Canal (1882–93).—Across the isthmus of Corinth, between Gulf of Corinth and Gulf of Aegina, in Greece, and mostly through solid rock with very deep cutting. The neck of land rises in the centre to 286 ft. above the bottom of the canal. Bottom width, 72 ft., and average depth, 26 ft. No locks. Length, 4 miles. Entrances at each end protected by solid jetties built out into the sea. Greek engineer, Matsos. Total cost, 1,400,000*l.*, or 350,000*l.* per mile.

Cronstadt and St. Petersburg Canal (1877–90).—Starts from the Neva and goes south-west for about 2 miles, when it curves and proceeds in a straight line to Cronstadt. Average width, 240 ft., and depth, 20 ft. No locks. Total length, 16 miles, including 10 miles bay channel and 5 miles canal proper. Costs per mile are based on the total length of 16 miles.

	Total.	Per Mile.
Constructional works only . . .	1,240,000*l.*	77,500*l.*
Other expenditure and charges . .	760,000*l.*	47,500*l.*
Whole cost	2,000,000*l.*	125,000*l.*

Elbe and Trave Canal (opened 1900).—In Germany. Width at bottom, 72 ft.; at top, 105 to 126 ft.; and depth, 10 ft. Length, 42 miles. 7 locks. Total cost, 1,177,700*l.*, or 28,040*l.* per mile.

Kiel Canal (1888–95).—Also known as the Kaiser Wilhelm Canal, or the Baltic and North Sea Canal, connecting Kiel in the Baltic to Elbe in the North Sea. Mostly through marshes or shallow lakes and river valleys, with tidal waterway throughout. Original dimensions: bottom width, 72–85 ft., surface width, 190–226 ft., and depth, 28–30 ft. (since increased to 150 ft., 335 ft.,

and 45 ft., respectively). Total excavation, 106,000,000 cubic yards. Two entrance locks, originally 492 ft. long × 82 ft. wide × 32 ft. deep over sill (now enlarged to 1,083 ft. × 148 ft. × 45 ft., and constructed in concrete, each lock having three pairs of caissons dividing the chambers into lengths of 725 ft. and 330 ft.). The original canal was completed in 1895, and the works for broadening and deepening in 1914. Breakwaters at each end. Length, 61 miles.

	Total.	Per Mile.
Original cost up to 1895, including land purchase	7,800,000*l.*	127,870*l.*
Enlargements and reconstructions up to 1914	11,200,000*l.*	183,605*l.*
Whole cost	19,000,000*l.*	311,475*l.*

Manchester Canal (1887–94).—Connects Manchester with Eastham, the Mersey port of Liverpool. Fifteen miles of canal proper, and 21 miles tidal waterway, or 36 miles total. Canal averages 120 ft. wide at bottom, 170 ft. at water level, and 26 ft. minimum depth (now increased to 28 ft.). Four sets of locks, with an average lift of 15 ft. each, thus raising the water level of the canal at Manchester to 60 ft. above mean sea-level at Liverpool. Capacity for 9,000 ton vessels, and it takes seven hours to pass through the whole distance and locks. The latter vary from 600 ft. × 80 ft. to 150 ft. × 30 ft., and sills 16–28 ft. deep below normal water-level of canal, the gates being moved by hydraulic machinery. The works include 160 acres of docks, with warehouses, machinery, and 95 miles of railway sidings, 6 high-level bridges, 7 swing-bridges, and the Barton swing aqueduct, 234 ft. long, for the Bridgewater Canal over the Irwell (to replace Brindley's fixed aqueduct), which latter canal had to be bought. 52,000,000 cubic yards of excavation and removal. Plant cost 980,000*l.*, and comprised engines, locomotives, steam pumps, cranes, piling machines, horses, wagons, temporary railway, etc. Principal engineer, Sir E. Leader Williams, assisted by W. Henry Hunter and others. Contractors, T. A. Walker (who died on November 25, 1889, during the progress of the work), and his executors, T. J. Reeves, L. P. Nott, and C. H. Walker, who continued. For full particulars see *History of the Manchester Ship Canal*, 2 vols., published by Sherratt & Hughes, 34 Cross Street, Manchester. Price 25*s.* Details of cost below are based on the total length of 36 miles :—

	Total.	Per Mile.
Purchase of Bridgewater Canal	1,783,000*l.*	111,438*l.*
Land and compensation	1,167,000*l.*	72,937*l.*
Constructing the canal and docks	9,000,000*l.*	562,500*l.*
Interest on capital during construction	1,050,000*l.*	65,625*l.*
Law charges, Parliamentary expenses, etc.	500,000*l.*	31,250*l.*
Whole cost	13,500,000*l.*	843,750*l.*
Average annual maintenance per mile of canal		2,000*l.*
Do. do. per acre of docks		400*l.*

Panama Canal (1904–15).—From Colon, on the Atlantic, to Panama on the Pacific. A concession was granted in 1876 to a French syndicate, which made surveys for two years and reported to an international congress at Paris in May, 1879. The Panama Canal Company was then floated, with Ferdinand de Lesseps as president, and preparatory work undertaken in 1881, the first excavation being made in January, 1882. The French scheme, however, was finally abandoned in 1904 owing to fever and bad administration, after the failure of both the first and a second company, and after only 12 miles had been cut out of the 54 originally proposed, and 50,000,000*l.* spent. On May 4, 1904, the United States Government bought the effects of the French company in Panama for 8,000,000*l.*, and started the American project as now completed. A strip of territory extending five miles on each side of the route of the canal was also purchased from Colombia for 2,000,000*l.* Colonel (now General) George W. Goethals, U.S.A. Engineers, was chairman of the Canal Commission which worked out the type and salient features.

Total length along axis, 50½ miles, but only 41½ miles within the coastlines. Rise of tide at Limon Bay in the Atlantic varies from 7 to 19 in., and from 8 to 21 ft. at Panama. The opening channel at sea-level, from Limon Bay to Gatun, is 7 miles long by 500 ft. wide and 41 ft. deep. At Gatun are three twin locks to raise vessels to the summit canal level, 85 ft. above sea, also an earthen dam, 8,000 ft. long, 1,500 to 2,000 ft. thick at the base, 100 ft. wide at top, and 115 ft. high, damming the Chagres River and forming the artificial Gatun Lake, 164 square miles area, with surface 85 ft. above sea-level, through which ships pass in a channel 500 to 1,000 ft. wide and 45 ft. deep. After 23½ miles of Gatun Lake the Gaillard Cut (formerly Culebra Cut) extends for 8 miles and 280 ft. maximum depth, with a channel 300 ft. wide and 45 ft. deep. Then a single twin lock, with 30 ft. drop, at Pedro Miguel to the small Miraflores Lake, where a channel, 1½ miles long by 500 ft. wide and 45 ft. deep, leads to Miraflores itself. Here two twin locks lower vessels, 31 ft. each lock, to the Pacific sea-level, and a channel 8 miles long, likewise 500 ft. wide and 45 ft. deep, conducts them to deep water in Panama Bay. In addition there are nearly 3 miles in locks and alongside approach walls. About 26 miles are artificial canal work, out of the total distance of 50½ miles.

Width of channel varies from 300 to 1,000 ft. at bottom, and depth 40 to 45 ft. The six pairs of locks are 1,000 ft. long × 110 ft. wide, with intermediate gates to divide them into two sections of 600 ft. and 400 ft., and 6 pairs of lock gates with steel frames and plates. There are 46 pairs of lock gates altogether, 46 to 82 ft. high, and the largest weighs 730 tons. Least depth of water over sills of locks when lake is at normal level is 41 ft.

The total excavation for the whole canal amounted to 220,000,000 cubic yards (130,000,000 in the dry, and 90,000,000 by dredges), of which about 112,000,000 yards was in the Culebra Cut, the most extensive excavation ever made. Concrete in the locks came to 4,500,000 cubic yards. Maximum number of employees at one time was 50,000 (6,000 Americans, 5,000 Europeans, and 39,000 negroes).

In addition to General Goethals, other engineers were Messrs. S. B. Williamson, chief of the Pacific Division, Gaillard, Sibert, Hodges, Goldwork, Schildhauer, Cornish, Zinn, Nichols, Alfred Noble, and others.

The canal was opened for traffic on August 15, 1914, the steamer passage being 10 hours each way. Ships are towed by electric cars or 'mules' running on rails alongside the waterway. Several landslides subsequently occurred necessitating temporary suspension of traffic till well into 1915.

Total cost was approximately as follows, the cost per mile being based on the total length of 50½ miles:—

	Total.	Per Mile.
Spent by the original French Canal Co. up to 1904 before abandonment . .	50,000,000*l.*	990,100*l.*
Purchase money paid to the French Canal Company	8,000,000*l.*	158,416*l.*
Sum paid to the Republic of Colombia or Panama for concession . . .	2,000,000*l.*	39,604*l.*
Constructional cost of American scheme, including terminal facilities . .	65,000,000*l.*	1,287,130*l.*
Whole cost	125,000,000*l.*	2,475,250*l.*

Saulte Ste. Marie Canal (opened 1895).—In Canada, connecting Lakes Superior and Huron, Known as the 'Soo,' and passes 20,000 vessels a year round the Saulte Ste. Marie Falls. Length, 1¼ miles. Width at bottom 142 ft., and depth 20 ft. It has the longest lock in the world, with gates operated by electric power. Total cost, 560,000*l.*, or 509,090*l.* per mile.

Suez Canal (1859–69).—Between Port Said on the Mediterranean and Suez on the Red Sea. M. Ferdinand de Lesseps, engineer and promoter. Total length all but 100 miles, including 60 miles through the shallow lakes of Menzaleh, Ballah, Timsah, and the Bitter Lakes, with and without dredging, and the rest actual canal formed by cutting. Sandy soil and a little rock. Sixty dredging machines were employed, and about 100,000,000 cubic yards of material excavated and dredged. Over 25,000 labourers were employed during the earlier period. Canal was originally 72 ft. wide at bottom, 196 to 328 ft. at water-level, and 16 ft. average depth (now increased to 147 ft., 420 ft., and 39 ft., respectively). No locks, as the canal is level throughout. Rise of tide is only about 1 ft. in the Mediterranean, and 6 ft. in the Red Sea. A slight current of 3 ft. per second keeps the canal clear. Passage takes 16 hours.

An auxiliary freshwater canal was constructed at the same time from the Nile, near Cairo, to Lake Timsah, and thence nearly parallel to the main canal as far as Suez, to supply fresh water and to convey provisions and materials, and afterwards to irrigate. It was 26 ft. wide at bottom, 56 ft. at surface, and 6½ ft. deep.

At Port Said the harbour encloses 450 acres, with an eastern breakwater 6,233 ft. long and a western breakwater 9,800 ft., composed of random concrete blocks weighing 20 tons. At Suez the outlet of the canal is protected on the south-west by a mole 2,900 ft. long, and on the north-west by a sold earthwork embankment.

Costs per mile are based on the total length of 100 miles.

	Total.	Per Mile.
Cost of promotion and preliminaries .	5,300,000*l.*	53,000*l.*
Constructional works only . . .	10,700,000*l.*	107,000*l.*
Enlargement, widening, and deepening since 1869.	4,000,000*l.*	40,000*l.*
Whole cost	20,000,000*l.*	200,000*l.*
Less indemnity granted in 1864 for loss of forced labour, about	4,000,000*l.*	40,000*l.*
Net cost	16,000,000*l.*	160,000*l.*

Welland Canal (opened 1833).—Constructed to avoid the break in communication between Lakes Erie and Ontario, Canada, caused by the falls in the Niagara River, and runs parallel to the latter on the west. Length, 26¾ miles. Twenty-seven locks, with total lift of 330 ft. Enlarged in 1867 to accommodate vessels of 400 tons. Again enlarged, 1874–87, to take vessels of 1,000 to 1,500 tons, so that the bottom width of canal would be 100 ft., and depth 14 ft. Locks increased to 270 ft. long by 45 ft. wide, and 14 ft. depth of water on the sills.

	Total.	Per Mile.
Cost of original canal opened in 1833	2,700,000*l.*	100,935*l.*
Cost of enlargement from 1874–87	2,300,000*l.*	86,019*l.*
Whole cost	5,000,000*l.*	186,954*l.*

Concrete.

(*London Prices, July*, 1929.)

The following prices are based on best British Portland cement at 55*s.* per ton to contractor delivered on site, including use of bags and returning empties. Clean pit sand 12*s.* per yard cube, Thames ballast 10*s.*, gravel 12*s.*, and 1½ in. broken stone 14*s.*, ditto. Wages, labourer, 1*s.* 4*d.* per hour.

The rates for measured work include materials, measuring and mixing by hand, wheeling up to 20 yds., laying and ramming, and 17 per cent. establishment charges and profit. Machine-mixed concrete would be about 3*s.* to 6*s.* per cubic yard cheaper.

Plain Concrete.

		s.	*d.*
Portland cement concrete, 1 to 6, with unscreened Thames ballast in foundations, walls, and masses exceeding 12 ins. thick	per yd. cube	36	0
Ditto, 1 : 3 : 6, with 1 cement, 3 sand, and 6 clean pit gravel, ditto	,,	37	0
Ditto, 1 : 3 : 6, with 1 cement, 3 sand, and 6 local broken stone, 1½ in. gauge graded down to ¼ in., in foundations, walls, and masses	,,	40	0
Add if in foundations and walls not exceeding 12 ins. thick	,,	2	0
Add if in underpinning in short lengths up to 5 ft.	,,	6	0
Add if executed between high and low water mark, including covering with canvas to protect against tides	,,	8	0
Deduct from the foregoing 1 : 3 : 6 proportions if the proportion is 1 : 4 : 8	,,	3	0
Portland cement concrete, 1 : 3 : 6, with 1 cement, 3 sand, and 6 clean pit gravel, spade finish. spread and levelled in beds, pavings, or ground floors, 6 ins. thick	per yd. sup.	6	0
Add for floated or trowelled surface with 1 cement and 2 sand, ¾ in. thick, as a screed or bed	,,	1	9
Granitic surface layer of 1 cement to 2 granite chippings, 1 in. thick, floated and trowelled to a smooth face whilst concrete is unset	,,	4	0
Hard dry filling under concrete, of clean broken brick, hard core, mason's spalls, etc., spread and levelled 6 ins. thick	,,	2	6
Portland cement concrete, 1 : 2 : 4, with clean pit gravel, ballast, or crushed aggregate, graded ¾ in. to ¼ in. gauge, cast *in situ* complete, with exposed faces left clean and perfect, including use of casings, for kerbs and channels to footpaths, roads, pavements, etc.	per ft. cube	2	4
Ditto pre-cast solid building blocks, exceeding 4 ins. thick, and set in 1 to 3 cement mortar	,,	2	10
Ditto pre-cast hollow blocks (measured as solid), ditto, ditto	,,	2	6
Ditto pre-cast lintels, door and window heads, window sills, steps, copings, cornices, quoins, and setting and jointing in 1 to 3 cement mortar	,,	4	9

Reinforced Concrete.

Reinforcement varies from ½ to 5 per cent. sectional area of the steel compared with the sectional area of the concrete, according to position.

1 per cent. volume of steel = 132 lbs. reinforcement per yard cube of concrete.

General economy of reinforced concrete construction for large structures, and over steel-framed buildings of equal strength, is 10–20 per cent., and over brick buildings and in heavy foundations 20–30 per cent.

Reinforced concrete factories, offices, stores, warehouses, etc., cost 10*d.* to 1*s.* 2*d.* per ft. cube.

Proportionate costs vary : ⅛ concrete, ⅛ steel, ⅛ formwork, to ½, ¼, ¼, respectively.

General proportions—1 : 1½ : 3 and 1 : 2 : 4. Cement should equal British Standard Specification (B.S.S. No. 12). Ferrocrete, Super-Cement, and Ciment Fondu harden much more quickly and are stronger, but more expensive.

Sand graded $\frac{3}{16}$ in. to $\frac{1}{10}$ in. (75 per cent. to pass $\frac{1}{8}$ in.), and aggregate $\frac{3}{4}$ in. to $\frac{3}{16}$ in. Concrete to be quickly deposited in layers not exceeding 6 ins. thick and 3 ins. next reinforcing bars, and in quantities not more than 2 ft. cube, closely punned round the steel.

Each class of work and story to be given separately, and heights stated. The space occupied by the steel is not to be deducted. Prices for mass concrete include forming chamfers, splays, rounded angles, etc., but the use of moulds for these should be taken under formwork.

The concrete, the steel, and the formwork are first priced separately, then combined into a condensed table of complete work.

Concrete Piles—

		s.	d.
Pre-cast reinforced Portland cement concrete, 1 : 1½ : 3, with 1 cement, 1½ washed sand, and 3 local broken stone, ¾ in. or ⅜ in. gauge graded down to $\frac{3}{16}$ in., in bearing or sheet piles, including all moulds, and planted in position ready for driving (exclusive of steel reinforcement and pile shoes)	per ft. cube	5	0
Steel reinforcement and shoes for ditto, and fixing	,,	4	0
Placing and driving ditto not less than 144 sq. ins. section, including all working apparatus, such as pile engine, etc., in ordinary ground (portion in soil only to be measured)	,,	4	0
Ditto ditto ditto in tide work	,,	5	6
Ditto ditto if driven from barges or floating stages, including use of same, extra over tide work	,,	1	6
Simplex reinforced concrete piles, 16 ins. diameter, cast in place and driven complete (8 or 10 piles, of 25 ft. average length and reinforcement provided, can be driven per 8-hr. day)	per ft. run	12	0
Allowance for bringing, erecting, and removing pile engine and tackle for driving, etc.	per job	15*l.*	to 25*l.*

Concrete Foundations, Walls, etc.—

		s.	d.
Reinforced Portland cement concrete (only), 1 : 2 : 4, with 1 cement, 2 washed sand, and 4 local broken stone, ¾ in. gauge graded down to $\frac{3}{16}$ in., in foundations and masses	per yd. cube	50	0
Ditto in walls exceeding 12 ins. total thickness	,,	54	0
Ditto beams, not exceeding 72 ins. sectional area, to floors or roofs	per ft. cube	2	2
Ditto large beams, exceeding 72 ins. but not exceeding 144 ins., ditto	,,	2	0
Ditto independent piers, columns, struts, or pilasters	,,	2	4
Ditto steps, stairs, winders, strings, carriages, etc., any section	,,	2	8
Ditto floor slabs or flat roofs, 4 ins. thick, and hoisting 30 ft.	per yd. sup.	6	4
Ditto ditto ditto 6 ins. thick, ditto	,,	9	6
Ditto walls, 6 ins. thick, measured between piers or projections	,,	10	0

Steel Reinforcement—

		s.	d.
Mild high-tension steel in accordance with B.S.S. No. 15, British Standard Specification for Structural Steel, Class A, in plain reinforcement (round, square, or flat bars), including cutting, bending, hoisting, 30 ft., assembling or putting into position, stout binding wire, and fixing complete in foundations, walls, columns, floors, etc., in			
¾ in. to 1½ in. bars	per cwt.	21	6
½ in. to ⅝ in. ,,	,,	22	6
¼ in. to ⅜ in. ,,	,,	23	0
Steel in proprietory systems of trussed or stirrup bars, and fixed	,,	26	0
Ditto ditto indented, twisted, or studded bars, ditto	,,	30	0
Fixing only sheet reinforcement, such as expanded metal, fabric, and mesh systems, 4–16 lbs. per yd. sup., including cutting, bending, or wiring (measured net, with no allowance for laps)	per yd. sup.	0	6

Formwork—

		s.	d.
Use and waste of 1½ in. simple flat centering, machine-planed one side, edges shot, to flat concrete floors and roofs, including about 14 ft. cube rough timber supports, ties and braces, and wedges, cleats, clamps, dog-irons, straps, etc., erection, cutting, maintenance, striking, and removal, average 12 ft. high	per square	52	0
Ditto ditto if to curved ceilings or roofs	,,	62	0

		s.	d.
Use and waste of 1 in. to 1½ in. strong shuttering to concrete walls, vertical or battering, including all supports, struts, etc., erection, cutting, maintenance, and removal (both sides measured)	per square	45	0
Ditto ditto if curved	,,	55	0
Add if boarding is machine-planed one side, thicknessed, and edges shot	,,	5	0
Limewashing internal surfaces of formwork, or treating with soft soap or mould oil, to prevent the concrete adhering	per yd. sup.	0	3
Use and waste of fir in rough timber supports, shoring, etc.	per ft. cube	3	0
Use and waste of casing, planed one side and edges shot, to concrete beams, average 12 ins. × 6 ins., including sides and soffits	per ft. sup.	1	0
Ditto to concrete pillars or columns, square, including supports	,,	0	10
Ditto ditto ditto circular, ditto	,,	1	3
Use and waste of casing in narrow widths, under 12 ins., to openings in concrete walls, floors, roofs, etc	per ft. run	0	6
Extra for fillets to internal angles of casing or shuttering to form chamfers, splays, rebates, rounded angles, etc., not exceeding 9 ins. girth	,,	0	3
Labour forming small splays, grooves, rounded edges, etc., not exceeding 2 ins. girth	,,	0	2
Raking, cutting, and waste to shuttering, not exceeding 1½ ins. thick	,,	0	3

Complete Work.—The following may give a general idea of the whole cost, including concrete, steel reinforcement, and formwork, but there are so many systems that it is impossible to be more definite, and the range of values must be great :—

			s.	s.
Reinforced concrete	12 ins. × 12 ins. piles, including driving, etc., complete	per ft. run	12 to	18
,, ,,	12 ins. pillars or columns, square or circular	,,	6 ,,	12
,, ,,	steps, stairs, and sills, average 12 ins. × 6 ins.	,,	4 ,,	8
,, ,,	floors, 4 ins. to 6 ins. thick, including beams	per yd. sup.	20 ,,	30
,, ,,	flat roofs, 4 ins. to 6 ins. thick, including beams	,,	20 ,,	30
,, ,,	walls, 4 ins. to 6 ins. thick, vertical or battering	,,	12 ,,	24
,, ,,	partitions, 2 ins. to 4 ins. thick, smooth on both sides	,,	10 ,,	16
,, ,,	foundations, thick walls, and masses	per yd. cube	70 ,,	100

Reinforced Granite Concrete Tubes.

Manufactured by John Ellis & Sons, Ltd., 16 Pocklington's Walk, Leicester, of best Portland cement and Mountsorrel granite, 1 to 3, crushed, washed, and passed through ¼-in. mesh, and with sufficient sand to fill up all interstices. The tubes are made both with and without steel ribs, the latter being ¼ in. welded steel rings placed in the centre of the material, 12 ins. apart. The concrete is cast in vertical steel moulds, filled in thin layers, and rammed with a heavy machine-driven hammer. The pipes are left in the moulds for three days, after which they are put in the open air to mature for six months.

Prices of reinforced circular sewer tubes free on rail at works, Barrow-on-Soar, L.M.S. Railway (July, 1928) :—

Internal Diameter.	Thickness in inches.	Stock Lengths.	Weight per ft. run.	Feet per ton.	Price of Tube per ft. run.
ins.	ins.	ft.	lbs.	ft.	s. d.
12	1¼	3	56	40	2 6
15	1½	2 and 3	86	26	3 0
18	1⅝	2 ,, 3	97	23	3 6
21	1⅝	2½ ,, 3	132	17	4 6
24	2	2 ,, 3	176	12¾	6 0
27	2	2 ,, 3	199	11¼	7 0
27	2¼	2 ,, 3	219	10¼	8 0
30	2½	2 ,, 3	276	8⅛	9 3
33	2½	3	289	7⅞	10 0
36	2½	3	326	6¾	11 0
36	2⅝	2 and 3	345	6½	12 0
39	3	3	415	5⅝	14 0
42	3	2 and	472	4¾	16 0
45	3¼	3	541	4½	18 0
48	3½	2 and 3	640	3¾	20 0
57	4½	3	943	2⅝	31 6
60	5	3	1120	2	37 6

Laying.—Cost of laying and jointing in addition. The tubes are usually laid on a square concrete bed raised half-way up the outside, and 6 ins. thick. The ogee joints are sealed with cement mortar, 1 to 1.

Connections.—Bends, junctions, circular manholes, well tubes, etc., are also supplied of granite concrete.

Other Tubes.—Socket and spigot sewer tubes, 15 ins. to 24 ins. diameter, and egg-shaped tubes, with internal measurements 18 ins. × 12 ins. to 60 ins. × 40 ins., are similarly made.

Stanton-Hume Reinforced Concrete Pipes.

The Stanton-Hume Reinforced Concrete Pipe is manufactured by the Stanton Ironworks Co., Ltd., near Nottingham, under sole licence from the Hume Pipe and Concrete Construction Co., Ltd., London. A hard, dense pipe is produced, owing to the centrifugal action of the moulding machinery, and the concrete mixture spreads itself evenly over the inner surface, while the circular motion expels excess air and water, so that there are no voids. All materials are mechanically tested in laboratories, then carefully graded and mixed by special apparatus, and the sand automatically washed. The pipe is particularly suitable for drainage and sewage work, as it is very strong, the interna. surface is abnormally smooth, owing to the longer lengths fewer joints are required, and cost is low. The reinforcement is a spirally wound steel wire cage of great tensile strength, kept in a central position by centrifugal tendency.

The pipes are made 4 ins. to 60 ins. diameter, and in classes A, B, C, D, E, for working pressures of 10, 25, 50, 75, 100 lbs. per sq. in. respectively. Proportions, 1 : 2 and 1 : 3 graded granite, finely crushed.

Prices delivered 50 miles from Stanton Gate at Ilkeston, near Nottingham (July, 1928) :—

Internal Diameter.	Thickness in inches.	External Diameter.	Stock Lengths.	Weight of Pipe for Class A.	Weight of Collar A.	Price of Pipe per ft. run.
ins.	ins.	ins.	ft.	lbs.	lbs.	*s.* *d.*
4	1	6	6	104	13½	0 9
6	1	8	6	146	17	1 1
9	1⅛	11¼	6	237	26	1 9
12	1⅜	14¾	6	380	40	2 9
15	1½	18	6	513	56	3 8
18	1⅝	21¼	8	887	75	4 8
21	1⅞	24½	8	1,111	92	5 9
24	1⅞	27¾	8	1,350	118	7 6
27	2	31	8	1,612	132	8 9
30	2⅛	34¼	8	1,906	175	10 6
33	2¼	37½	8	2,220	202	12 3
36	2⅜	40¾	8	2,540	240	15 0
42	2½	47	8	3,110	302	20 0
48	2¾	53½	8	3,900	415	25 0
54	3	60	8	4,790	560	32 6
60	3	66	8	5,310	620	38 6

Joints.—The above prices include one collar for each length of pipe. Pipes with ogee joints 5 per cent. less. Jointing requires skilled labour, but experienced men can be sent ; 1 cement to 2 sand.

Laying.— Cost of laying and jointing extra to rates in table. Reinforced bends, tees, angle branches, etc., are also made.

Deep-water Quays.

The structural conditions of deep-water quays are so variable that the assumption of a unit of cost is misleading. For ocean-going shipping in the Thames, however, the rates between 1919–22 (post-war) varied from 50*l.* to 200*l.* per ft. run.

Deep-water Quay, Greenhithe (1919–21).—On the south bank of the river Thames, and was an extension of the existing wharves of the Empire Paper Mills. Tidal range over 18 ft., with 17 ft. depth at low-water ordinary spring tides. Length of new section of the approach 455 ft., with minimum width of 8 ft., and new deep-water berthage at end 303 ft. by 52 ft. width throughout, and three splays at corners. Total deck area of new works approximately 24,600 sq. ft. Quay carried out in creosoted Oregon and pitch pine, with steel superstructure. Engineers, Messrs. Carey and Latham.

Total cost, 70,783*l.*, or 2*l.* 17*s.* 7*d.* per ft. sup. of new wharf accommodation obtained. About 93*l.* per ft. run, including approaches.

Deep-water Quay, Shellhaven (1920–21).—On the Essex side of the river Thames, and consisted of the extension of a deep-water berthage for an oil storage depot. Tidal range 18 ft., with 29 ft. depth at low-water ordinary spring tides. Length of new quay approach 560 ft. by 18 ft. wide, and new deep-water berthage at end 420 ft. long by 40 ft. deck width, the two forming a **T** shape. Total deck area approximately 31,000 sq. ft. All of reinforced concrete on the Coignet system. Engineers, Messrs. Carey and Latham, and contractors Messrs. Mowlem & Co.

Total cost 80,000*l.*, or 2*l.* 11*s.* 7*d.* per ft. sup. of new wharf accommodation obtained. About 82*l.* per ft. run, including approaches.

Docks.

(*Estimated Costs*, 1930.)

Wet docks, including wharves, gates, locks, machinery, etc., but excluding cost of land :— 50,000*l.* to 100,000*l.* per acre of water area.

Dry or graving docks for repairs, including equipment and pumping machinery :— 4*l.* to 8*l.* per yard cube of internal capacity.

Steel floating docks for repairs, in terms of lifting power 15*l.* to 25*l.* per ton.

Dock gates of greenheart, 70 to 90 ft. wide and 40 to 50 ft. high, including hydraulic machinery . . . 4*l.* to 5*l.* per ft. sup.

Ditto of steel, ditto, ditto . . . . . 3*l.* to 5*l.* ,,

Caissons, traversing or floating, 60 to 90 ft. wide, and 30 to 40 ft. high, including hydraulic machinery . 5*l.* to 10*l.* per ft. sup. of waterway.

Dock walls, of concrete, stone or granite, with concrete backing, including digging . . . . . 40*s.* to 90*s.* per yd. cube.

Actual Costs of wet docks, including dock walls, gates, locks, etc., per acre of water area :—

	£
Avonmouth Docks, near Bristol (1868–77)	22,600
Barry Docks, near Cardiff (1889). Water area, 114 acres. Total cost, 3,072,000*l.*	26,947
Cardiff. Water area, 50 acres. Total cost, 1,500,000*l.*	30,000
,, Bute Dock Extension. Water area, 35 acres	22,000
,, West Bute Dock (1839). Water area, 22 acres. Total cost, 400,000*l.*	18,180
,, Roath Dock (1887). Water area, 33 acres. Total cost, 500,000*l.* .	15,150
Glasgow, Queen's Dock	24,500
Hull, Alexandra Dock	28,900
Immingham, near Grimsby (1912). Water area, 56 acres. Total cost, including land, railway sidings, dock sheds, coal hoists, etc., 2,600,000*l.* . . .	46,430
Liverpool, Alexandra Dock	23,300
,, North Extension. Water area, 83 acres. Total cost, 2,640,000*l.* .	31,810
London, King George V Dock (opened 1921). Concrete walls, 7 reinforced concrete jetties, entrance lock, &c. Water area, 64 acres. Total cost, with equipment, 4,130,000*l.*	64,531
London, St. Katherine's Docks	36,000
Newport, S. Alexandra Docks (1893). Water area, 20 acres. Total cost, 500,000*l.*	25,000
Swansea, Prince of Wales' Docks (1879). Water area, 28 acres . . .	18,600

Gladstone Docks, Liverpool (1910–27).—The whole scheme embraces :—a new River or Entrance Lock (1,070 ft. long and 130 ft. wide, 22 ft. depth on the sills at low water ordinary spring tides, and 48 ft. at high-water ditto, allowing the largest ship to enter or leave on every single tide of the year); a Connecting Lock (645 ft. long and 90 ft. wide, between the existing dock system and the new docks); a Turning Dock, called the 'Gladstone Dock' (1,594 ft. long and 942 ft. extreme width, water area 25 acres, with a shed on the north quay, 925 ft. × 100 ft. and a shed on the west quay, 526 ft. × 100 ft.); a Branch Dock, called the 'Gladstone Dock No. 1' (1,420 ft. long and 400 ft. wide, opening out of the turning dock, with north shed 1,252 ft. × 100 ft., and south shed, 1,502 ft. × 150 ft., both of reinforced concrete, flat roofs and three storey); another Branch Dock, called the 'Gladstone Dock, No. 2' (1,285 ft. long and 400 ft. wide, also opening out of the turning dock, with north and south sheds, 1,202 ft. × 100 ft. and 1,277 ft. × 100 ft., of reinforced concrete as before); and the 'Gladstone Graving

Dock,' 1910–13 (1,050 ft. long and 141 ft. wide at bottom and 155 ft. at top, 43 ft. depth on sill at ordinary spring tides, being the largest dry dock in Europe, and can also be used as a wet dock, with a single-storey shed the full length of the north quay); also large crane and jigger equipment at sheds, tracks, rail facilities, electric power, engines, pumps, etc.

Engineers, Mr. T. M. Newell, M.I.C.E., engineer-in-chief of the Mersey Docks and Harbour Board, and Mr. J. A. Newell, A.M.I.C.E., and Mr. G. E. C. Callow, A.M.I.C.E., resident engineers. The work was carried out by direct labour, but the lock gates were made by Sir W. G. Armstrong Whitworth & Co., cranes by Messrs. Stothert and Pitt, Ltd.,of Bath, and details by other firms.

The whole extensions comprise—58¼ acres of water area, 3 miles of quayage, 19 acres of ground covered by sheds, and 60½ acres gross floor area of sheds. Total cost over 7,500,000*l.*, or 128,755*l.* per water acre.

Actual Costs of dry or graving docks for repairs, including equipment :—

	Per yard cube of internal capacity. £	*s.*	*d.*
Aberdeen. Internal size, 524 ft. × 73 ft. at surface level. Total 46,000*l.*	1	15	0
Belfast (1905–12). Length over all, 901 ft. Total cost, 350,000*l.*	2	1	0
Glasgow. Internal capacity, 81,555 yds. cube. Stone and concrete	2	9	0
Hartlepool. Brick and concrete, 570 ft. × 75 ft. × 17 ft. deep	1	8	0
Liverpool. Gladstone Dock (1910–13). Total cost, 500,000*l.*	1	13	0
Otago, New Zealand. Internal capacity, 37,000 yds. cube. Total 75,000*l.*	2	0	0
Portsmouth. Granite with brick and concrete backing; 42 ft. deep	2	0	0

Drainage.

4-in. and 6-in. stoneware drains, jointed in cement, including trenches 3 ft. deep and concrete bed, 6 ins. thick . . . . . 2*s.* to 3*s.* per ft. run

Ditto ditto with manholes, ventilating pipes, etc. . . 5*s.* to 6*s.* per ft. run.

Manholes and inspection chambers, 3 ft. × 2 ft. 6 ins. × 3 ft. 6 ins. deep, complete, including digging, concrete, brick sides, iron cover, etc. . 8*l.* to 10*l.* each.

Add extra for every additional foot in depth . . . . . 16*s.* to 18*s.* per ft. deep.

Drains and sewers for estate development and town planning schemes . 200*l.* to 400*l.* per acre.

Drainage and sewage disposal works for cities and towns . . . 3*l.* to 6*l.* per inhabitant, 20*l.* to 30*l.* per house.

Reconstruction of Drainage, Stanhope Lines, Aldershot (1893).—Combined water carriage system with 4 in., 6 in., 9 in., and 12 in. glazed stoneware pipes (total, 54,670 yds. run), 2 to 6 ft. deep, 500 Jennings' latrines, 500 urinal stalls, 80 w.c.'s, with 3 gal. flushing cisterns, 200 gal. galvanised iron automatic cisterns, and 16 100-gal. brick automatic flushing tanks, with disposal of sewage by irrigation on War Department land. Occupation of barracks, 8,000 adults, 200 children, and 200 horses. Total cost, exclusive of sewage disposal works, 20,500*l.*, or 2*l.* 10*s.* per head of occupation exclusive of horses; 7*s.* 6*d.* per yd. run of excavation.

Drainage of Land.

Agricultural, or subsoil, unglazed earthenware drain pipes laid dry in 12 in. or 15 in. lengths, without jointing material and ends simply abutting. Depths and distances apart vary according to nature of soil. Cost includes digging, filling, pipes, and laying.

	Cost per acre.
In compact or heavy soils, drains 5 to 8 yds. apart, and 2 ft. 6 ins. to 3 ft. deep	6*l.* to 10*l.*
In medium soils, drains 9 to 10 yds. apart, and 3 ft. to 3 ft. 6 ins. deep	5*l.* to 6*l.*
In porous or light soils, drains 11 to 22 yds. apart, and 3 ft. 6 ins. to 4 ft. 6 ins. deep	4*l.* to 5*l.*

Dredging.

Dredging Plant.—Modern dredging appliances are of two main classes, with subdivisions and pre-war costs as follows :—

Class I.—Digging dredgers, comprising :

(*a*) Bucket dredger, or ladder (single or double), or elevator dredger, with a continuous band of buckets.

	£
With 5 ft. cube bucket, single ladder, raising 40 ft.	7,000
,, 15 ,, ,, ,, 45 ,,	27,000
,, 27 ,, ,, ,, ,, 55 ,,	43,000

(*b*) Dipper dredger, with a single bucket at the extremity of a long arm or lever. Almost purely an American type.
With 1 to 2 yds. cube dipper, hull of timber, 1,600*l.* to 2,400*l.*
With 3½ to 4 yds. cube dipper, dredging to depth of 35 ft., 16,000*l.* to 18,000*l.*
With 10 yds. cube dipper, dredging to depth of 35 ft., 26,000*l.* to 28,000*l.*

(*c*) Grab or grapple dredger, with a drop bucket suspended at the end of a chain.
A twin-screw hopper dredger, with hopper capacity of 12,000 tons, fitted with five hydraulic grab cranes each of 5 tons bucket capacity, working to a depth of 47 ft., has cost 35,000*l.*

(*d*) Hand or bag and spoon dredger, which consists of a bag attached to a steel rim at the end of a long pole, and is suitable only for mud in ponds and canals.

Class II.—Suction or hydraulic dredgers, comprising:

(*a*) Sand pump dredger, with centrifugal pump.
A 2,000-ton suction hopper dredger would cost 50,000*l.*

(*b*) Suction cutter dredger, with water jets and rotary cutters.
A suction hopper dredger with hopper capacity of 70,000 cub. ft., capable of raising 4,000 tons of sand per hour from a depth of 70 ft., has cost 75,000*l.*

(*c*) Compound dredger, fitted with both the suction or suction cutter and a bucket ladder.
A sea-going hopper dredger, fitted with both ladder buckets (18 cub. ft.) and suction pumps, capable of dredging 400 to 650 yds. cube per hour from a depth of 50 ft., has cost 53,000*l.*
Rock cutters are not dredgers, but an auxiliary class.

Cost of Work.—Costs include use of plant, dredging, and removing and depositing excavated material at sea. Steam dredgers cost 10,000*l.* to 70,000*l.* each; steam hopper barges in attendance, of 800 tons or 800 yds. cube capacity, with a speed of 11 knots per hour, cost 18,000*l.* to 20,000*l.*; ordinary hopper and dredger barges, 300*l.* to 1,000*l.* Pre-war.

Description.	Material.	Per yd. cube.	Per ton.
Bucket dredging, or ladder dredging	Mud, sand, or soft clay	1*s.* to 2*s.*	9*d.* to 1*s.*
	Gravel or stiff clay	2*s.* ,, 4*s.*	1*s.* ,, 2*s.*
Grab dredging, or grapple dredging	Mud, sand, or soft clay	2*s.* ,, 4*s.*	1*s.* ,, 2*s.*
	Gravel or stiff clay	3*s.* ,, 5*s.*	2*s.* ,, 3*s.*
Hand dredging, or 'bag and spoon dredging'	Mud, sand, or soft clay	3*s.* ,, 4*s.*	2*s.* ,, 3*s.*
	Gravel or stiff clay	4*s.* ,, 6*s.*	3*s.* ,, 4*s.*
Pump dredging, 'suction or hydraulic dredging'	Mud, sand, or peat	6*d.* ,, 1*s.*	4*d.* ,, 9*d.*

Grab dredging can be done for as little as 2*d.* to 6*d.* per yd. cube, and sand pump dredging for 1*d.* ditto, but a great deal depends on the kind of spoil and distance to be carried.

Alluvial Mining.—Dredging for alluvial mining costs anything from 1*d.* to 1*s.* per yd. cube (pre-war), depending upon character of ground, depth, quantity treated, climate, transport, power, labour, management, types of dredges capacity of buckets, operating time, etc.

Mersey.—At the bar and in the entrance channels of the Mersey nearly fifteen million cubic yards of sand were removed annually at a pre-war cost of 6*d.* per yd. cube, exclusive of interest and depreciation on the plant.

Mississippi River (1907).—Three pipe-line suction-cutter dredges got up 1,279,000 cu. yds. including cost of idle time, for 4½*d.* per yd. cube. In 1908 the same dredges lifted 989,000 cu. yds., ditto, for 5½*d.* per yd. cube.

Newry.—After blasting the bottom in a depth of 6 to 8 ft. at low water, the material was removed by dredging at 4*s.* to 5*s.* per yd. cube.

New York Harbour (1909).—Four sea-going suction dredgers removed 10,786,638 cu. yds., including dredging, surveys, contingencies, and supervision, for 2*d.* per yd. cube. Another dredger disposed of 1,675,562 cu. yds. at 2½*d.* per yd. cube.

Panama Canal (1910).—Costs of removal of various quantities by different kinds of dredgers worked out at:

Removal of	6,083,902	cu. yds. by	two sea-going suction dredgers	3¾*d.* per yd. cube
,,	1,372,665	,, ,,	one ladder dredger	8*d.* ,,
,,	560,622	,, ,,	,, ,,	1*s.* 1¾*d.* ,,
,,	1,013,485	,, ,,	two small dipper dredgers	2*s.* 4*d.* ,,
,,	2,472,822	,, ,,	two 20-in. pipe-line suction dredgers	1*s.* 2*d.* ,,
,,	5,800,000	,, ,,	two dipper dredgers	2½*d.* to 5½*d.* ,,

Port Natal, Durban (1903–14).—Cost of pump dredging in sea, including all charges for working, maintenance, insurance, depreciation, and interest on capital, with coal at only 8*s.* per ton, 2½*d.* to 7*d.* per ton of material handled.

Severn.—Blasting and dredging the marl beds, 4*s.* per yd. cube.

Siam (1920).—Dredging heavy clay from irrigation canal in Siam, 85 ft. wide by 8 ft. deep, by a Lobnitz dipper dredge with buckets of 2½ yds. cube capacity, working at a speed of 30 to 40 seconds per dip. Total quantity raised was 312,130 cu. yds. in one place, running 9 hours per day, or an average of 26,010 cu. yds. per month. Average cost for year was 3*d.* per yd. cube, including labour, oil fuel, supplies, repairs, and every expense except interest and depreciation. Repairs, 4·3 per cent. of total cost.

Tees.—Blasting and dredging. In 1875 holes were drilled in the bed of the river Tees 7 to 9 ft. deep and 10 ft. apart, from a dredger; then dynamite blasting in 2 lb. charges. Average depth of rock blasted 4 ft. 6 in., oolite shale. 110,000 yds. cube were blasted at 3*s.* 1*d.* per yd. cube, so as to be lifted by the dredger.

Ladder dredging in the Tees, mostly in hard boulder clay, and lifting stones and tree trunks, cost 1*s.* 6¾*d.* per yd. cube.

Thames.—Contract cost of lifting ballast and clay, by ladder or bucket dredgers from the Thames below London Bridge has varied from 1*s.* 4*d.* to 2*s.* 6*d.* per yd. cube, depending on distance material had to be conveyed.

Rock-cutting and Dredging.

Rock-cutting Plant.—The sub-aqueous Lobnitz Patent Rock-breaker or Rock-cutter, made by Lobnitz & Co., Ltd., shipbuilders and engineers, Renfrew, Scotland, is usually employed. The rock-breaker chisel, or rock-cutter itself, consists of a compressed steel cone-shaped point fitted to a solid steel shaft, 15 to 18 in. diameter and 40 to 50 ft. long, depending on depth of water, and weighing from 6 to 25 tons. This drops 6 to 10 ft. on the same place, from a frame mounted on a barge fitted with steam winch, hoisting gear, steel wire ropes, mooring chains, etc. No explosives are used. The striking points are marked on a chart, and the holes are spaced about a yard apart. After a penetration of 3 ft. on one spot the rock is cut, and then dredged away in horizontal layers. The general result is 2 cu. ft. of rock broken per blow, suitable for bucket dredging afterwards to remove the pulverised stone. And 2 cu. ft. × 150 blows per hour on an average = 300 cu. ft. ÷ 27 = 11 yds. cube per hour for a single-cutter machine, or say 100 yds. cube per working day of 9 hours. Output of a double-cutter machine is about one-half more. A crew of four men is required on a barge with one cutter, and six men with two cutters, the barge being shifted after a hole has been made to the next position. Skilled labour is unnecessary. 1 ton of coal is consumed per working day by a single 10-ton cutter, and 1½ tons by a double 10-ton cutter. The cost of oil, stores, and repairs does not exceed the outlay for coal and wages.

Blyth Harbour, Northumberland (1906).—Rock-cutting in friable to very hard sandstone by the Lobnitz Rock-breaker steel ram weighing 15 tons, falling 8 ft. to a depth of 3 ft. in the rock in 8 or 9 blows, to allow of being dredged to a depth of 2 ft. 6 ins. Average quantity was 908 cu. yds. of rock broken by one machine per week, night and day, allowing for all stoppages, and based on six months' working. Details per week for 908 cu. yds. was:

	£	*s.*	*d.*
Wages for men on dredger	16	1	6
Coal, stores, and water	6	10	7
Estimated cost of repairs, renewals of rams, ropes, etc.	22	0	0
Insurance at 30*s.* per cent. = 120*l.* per annum	1	19	2
908)	46	11	3
Rate per yard cube	0	1	0¼
Add 4 per cent. interest on cost of one Rock-breaker at 6,800*l.*, and 2½ per cent. for depreciation, making a further	0	0	2¼
Total cost per yard cube	0	1	2½

As compared with blasting, the rock-cutting disintegrates and breaks the rock into smaller pieces, so that the quantity lifted by the dredgers in a given time is about 15 per cent. more than that of blasted rock. The comparative cost of removal under the two systems of blasting and rock-cutting is as follows:—

	s.	*d.*
Blasting:—		
Drilling and blasting rock in position	3	0
Dredging and removing débris, 2*s.* 6*d.*+ 8*d.* for interest and depreciation on the dredger	3	2
Cost per yard cube	6	2
Rock-cutting:—		
Cutting rock by rock-breaker as already detailed	1	2½
Dredging and removing débris, 2*s.* 2*d.*+ 7*d.* for interest and depreciation on the dredger	2	9
Cost per yard cube	3	11½

Difference in cost per yard cube in favour of the removal of rock by means of the rock-breaker was thus 6*s.* 2*d.* *less* 3*s.* 11½*d.*= 2*s.* 2½*d.* Saving on 500,000 cu. yds. executed was accordingly 55,200*l.*, after the drilling and blasting had been tried and abandoned.

Manchester (1905–6).—Deepening a length of 14 miles of the Manchester Ship Canal from 26 ft. to 28 ft. by a Lobnitz Rock-cutter No. 1, designed to work in sandstone of varying degrees of hardness, the average quantity broken per month (over a period of 10 months) being 6,403 cu. yds. at an average cost of 9*d.* per yd. cube.

The cost of rock-breaking and dredging on the same canal in April 1907, by the Lobnitz dredger was as below. The price of one Rock-cutter was 6,800*l.*, and if 4 per cent. is allowed for interest, and 2½ per cent. for depreciation, the additional cost per yard cube is 2¼d., as added. 17,060 cu. yds. were broken, the rock being sandstone of different hardness, with some stiff marl.

	s.	*d.*
Breaking, excluding cost of plant, but including maintenance and repairs	0	5½
Dredging the broken rock	0	5½
Barging and tipping	0	6¾
Towage	0	2¼
	1	8¼
Add interest and depreciation on cost of plant	0	2¼
Total cost per yard cube	1	10½

Suez Canal.—Deepened in 1884 by a Lobnitz dredger, with cutting rams of 3½ tons, rock removed in layers 20 in. thick, and the breaking up cost was 7*s.* 3½*d.* per yd. cube.

The canal was further deepened to 31 ft. in 1897 by a Lobnitz Rock-cutter with cutting rams of 13 tons, the average thickness of rock shattered was 31 in., and the average cost, including repairs and depreciation, was 1*s.* 4*d.* per yd. cube. Cost of raising the rock by dredgers and carrying it away in addition.

ELECTRIC POWER STATIONS.

Town.	Total Cost.	Cost of Land and Buildings.	Kilowatts Housed.	Cost of Buildings per Kilowatt.	Total Cost per Kilowatt.
	£	£	No.	£	£
Brighton	717,738	149,431	5,400	28	133
Bristol	671,616	53,550	9,050	6	74
Chichester	20,000	600 not land	200	3	100
Glasgow	1,368,134	211,172	10,993	19	124
Hackney	280,982	42,394	3,372	13	83
Metropolitan	1,802,383	435,777	18,500	24	97
Newcastle	1,192,442	169,675	20,000	8	55
Tonbridge	22,420	7,228	120	6	187
Tunbridge Wells	81,134	13,207	1,170	11	69

	£	£	
Electric light installations for cities and towns, with buildings, plant, and mains complete		80 to 130	per kilowatt.
	3 ,,	5	per c.p. lamp.
Steam plant power stations		24	per kilowatt.
Oil plant power stations	23 ,,	25	,,
Capital cost of generating stations		20	,,
Power and electric stations generally, brick, exclusive of plant and machinery. 10*l.* to 25*l.* per kilowatt			6*d.* to 8*d.* per f.c.
Aldershot electric lighting station (1902). Brick walls and slated roof			6½*d.* ,,
Curragh electric light generating station (1913–14). For three 100-h.p. Diesel oil engines, 66 kilowatts per engine, with a 750 ampere hour battery and accessory machinery. Contains a basement for engine beds, silencer chamber, and oil-fuel store, and above, on ground level, an engine-room, battery-room, store-room, two offices, and lavatory. Brick walls in cement mortar, 1 to 3, concrete foundations, engine beds and floors, steel sashes, steel roof, boarded and covered with asbestos slates. 99,700 feet cube. Building only, exclusive of plant and machinery. Total, 2,080*l.*			5*d.* ,,
Building, including plant and machinery. Total, 7,456*l.*			1*s.* 6*d.* ,,
Cost of foregoing per kilowatt housed, 10*l.* to 38*l.*			
Extension to Curragh electric light station (1920–21). For one additional 100-h.p. Diesel oil engine and plant, 66 kw., the extension of engine-room and exhaust pit being 12 ft. long, and same width and height as existing building. Construction also similar. A corrugated iron workshop, 18 ft. × 12 ft., is attached to the end of the brick extension, and included in total cost and cubing. Lump sum tender for both together. 17,600 ft. cube total. 17*l.* external services in addition. Building only, exclusive of plant and machinery. Total, 2,188*l.*			2*s.* 5½*d.* ,,
Building, including plant and machinery. Total, 5,188*l.*			5*s.* 10¾*d.* ,,
Cost of foregoing per kilowatt housed, 33*l.* to 79*l.*			
East Ham electric power station, including tramway sheds and offices. Built by direct labour under the borough engineer. Total, 26,020*l.*			5½*d.* ,,
Salisbury engine-room and boiler-house (1904). Brick walls and slated roof, concrete floors			6½*d.* ,,
Usual charges for current for electric lighting			2*d.* to 3*d.* per unit.
Ditto for heating and power purposes			1*d.* to 2*d.* ,,

Engineering Factories.

Engineering factories, brick, exclusive of machinery (1930)			1*s.* to 1*s.* 6*d.* per f.c.
Material		50	per cent.
Labour		40	,,
Power		5	,,
Other costs		5	,,
		100	,,

Esplanades.

Marine Drive, Scarborough (1905).—Road 40 ft. wide, parade 20 ft. wide, and sea-wall 20 to 40 ft. high. Of concrete, on deep filling at base of cliff. Length, 1,400 yds. Total, 102,400*l.*, or 73*l.* per yd. run.

Excavating (Mechanical).

Methods.—Mechanical excavating greatly saves in labour, time, and cost over hand digging by pick and shovel, and may be classed under ten heads:

1. Steam shovels.
2. Trench-diggers.
3. Grab buckets.
4. Chain bucket excavators.
5. Drag-line excavators.
6. Scrapers.
7. Derricks and cranes.
8. Radial loaders.
9. Paving breakers.
10. Soil Transport.

1. *Steam Shovels.*—Steam shovel or steam navvy excavating for large masses of earth or rock in canals, docks, railway cuttings, etc., including filling wagons for removal. The buckets scoop

upwards with a general capacity of 1 to 3 cubic yards. Average output and pre-war costs of a 1½ cubic yard bucket steam excavator :—

	Yds. Cube
Ordinary soft soils	60–90 per hour, at 3*d.* to 4*d.* per yd. cube.
Stiff clay or gravel	40–50 „ „ 4*d.* „ 6*d.* „
Strong ground or soft rock	30–40 „ „ 7*d.* „ 9*d.* „

Manchester Ship Canal (1887-94).—Forty-two million cubic yards of excavating in sand, clay, gravel, etc., and 10 million cubic yards in sandstone rock. Average quantity for each steam navvy was 700 yds. per 10-hour day, or 70 yds. per hour, and cost 2d. per yd. cube, exclusive of cost of plant, depreciation, repairs, superintendence, etc.

Pre-war costs excavating and loading into wagons, 2*d.* to 6*d.* per yd. cube ; the harder the material the greater the saving. In easy soil as low as 1*d.* per yd. cube, including interest, depreciation, and repairs, but this is exceptional ; in stiff clay 2*d.* or 3*d.* ditto. Costs per yard cube are higher for small machines owing to the comparatively small output, and less for larger machines with big output.

Analysis of Costs.—Assuming a steam-driven crane navvy with 20 ft. jib centre and ¾ cu. yd. bucket capacity is getting out clay at the rate of 30 yds. cube per hour, the costs (August 1922) would work out to approximately as follows, including excavating and dumping into clients' wagons :

Output per day = 30 yds. cube per hour × 8 hrs. per day = 240 yds. cube.

	s.	*d.*
Working costs per day—		
One driver upon the machine	12	0
One boy as fireman	5	0
Two men assisting on ground at 8*s.* each.	16	0
8 cwt. coal at 35*s.* per ton	14	0
Oil waste and stores	3	0
	50	0
240 yds. cube.		
Rate per yd. cube	0	2½

	s.	*d.*
Add capitalised cost, including depreciation at 10 per cent. per annum, interest 5 per cent. per annum, and maintenance 2½ per cent. per annum, making a further	0	1½
Total capitalised cost per yd. cube	0	4

With an output of 80 yds. cube per hour for loose material the cost would be 2*d.* to 3*d.* per yd. cube.

On bigger productions the cost per yard cube comes down owing to the working rates being practically the same with the exception of a slight increase in coal consumption, so that on the larger machines the value would be well below the figure given. Fewer men are needed with electric power, or if caterpillar tracks are fitted.

Removing earth cover with a No. 6 ' Ruston ' crane navvy for the United Brick Co., Peterborough, came to 3¾*d.* per yd. cube. This previously cost 1s. 6d. per yd. cube by hand labour.

A 45-ton Wilson's steam-crane ' Jubilee ' excavator will work a clear 22-ft. face, and drive a gullet 45 ft. wide, and can travel forward in 30 seconds as each advance is made.

Weight of Machine.	Capacity of Bucket.	Lift of Crane.	Depth of Cut.	Output per Hour based on 10-hour day.
tons.	cub. yd.	tons.	ft.	yds. cub.
17	¾	3	16	25–45
20	¾	4	16	30–50
25	1	5	18	40–70
30	1½	8	20	60–100
45	2½	15	22	100–140

Pre-war cost was 1*d.* to 6*d.* per yd. cube delivered into wagons.

The ' Erie ' revolving steam shovels are built in two sizes, and can be fitted with broad traction wheels, rail wheels, or caterpillar tread, and will climb 25 per cent. grades under their own power.

Type ' A ' is used for stiff clay, gravel, or lighter shales, and Type ' B ' for stiffer clay, gravel, heavy rock shales, or blasted rock. The following rates of output are conservative and have often been exceeded. The yield depends upon plant management, and to keep the shovel busy ample hauling equipment is essential. To dig and load by pick and shovel costs seven times as much as the same amount with an ' Erie.'

Standard Type.	Capacity of Bucket.	Working Weight.	Approx. Price, August, 1922.	Output per hour, ordinary conditions.
	cu. yd.	tons.		cu. yds.
A	½	13	£1,800	15–25 shallow cuts (10 in. to 24 in.) 30–40 deep cuts (3 ft. or over)
B	¾	19	£2,070	25–35 shallow cuts (10 in. to 24 in.) 50–60 deep cuts (3 ft. or over)

Two hundred to 300 yds. cube per day is good going in a trench cut, which may be 19 ft. deep. One ' Erie ' ditcher does as much as a 50-man gang in clearing sides and keeping cuts ditched.

Analysis of Costs.—The following approximate working costs per day (August, 1922) are based on 200 working days per annum of digging and filling wagons or trucks with a Type 'A' ' Erie ' revolving shovel. The figures do not include any team or truck time.

	£	s.	d.
One driver or operator	0	15	0
One fireman	0	10	0
Two groundmen at 12*s.* 6*d.* each	1	5	0
Oil and fuel	1	0	0
Daily interest on capital invested, cost of machine £1,800 at 6 per cent.	0	10	10
Daily depreciation, say 10 per cent.	0	18	0
Total capitalised cost per working day	4	18	10

And £4 18*s.* 10*d.* per day ÷ 200 yds. cube output per day . . 6*d.* per yd. cube.

(Output variable according to soil, shallow or deep cuts, wagon service, etc.) 300 " " " . . 4*d.* "
400 " " " . . 3*d.* "

For a Type ' B ' shovel the approximate working costs per day would similarly be:

	£	s.	d.
One driver or operator	0	15	0
One fireman	0	10	0
Two groundmen at 12*s.* 6*d.* each	1	5	0
Oil and fuel	1	1	0
Daily interest on capital invested, cost of machine £2,070, at 6 per cent.	0	12	6
Daily depreciation, say 10 per cent.	1	2	0
Total capitalised cost per working day	5	5	6

And £5 5*s.* 6*d.* per day ÷ 300 yds. cube output per day . . . 4½*d.* per yd. cube.

(Output variable according to soil, shallow or deep cuts, wagon service, etc.) 400 " " " . 3½*d.* "
500 " " " . . 2½*d.* "
600 " " " . . . 2*d.* "

The ' B ' ' Erie ' with 4-men crew (driver, fireman and two groundmen) is equal on deep cuts to 50 or more labourers, and on rock loading to 20 or 30 men. Assuming a labourer gets 1*s.* 3*d.* per hour, and can dig and throw into wagons ½-yd. cube per hour, this is equivalent to 2*s.* 6*d.* per yd. cube for hand labour. That is, 50 labourers × 4 or 5 yds. cube each per day = 200 to 250 yds. cube per day for the gang, whereas the Shovel will excavate 500 to 600 yds. cube per day and lift into wagons at a cost of about 2½*d.* per yd. cube.

Two loads of a ' B ' ' Erie ' dipper or bucket will completely fill an ordinary 1¼ cub. yd. wagon and three ditto a 2 cub. yd. wagon.

Wagon Service.—There is no waiting with steam shovels, as the buckets or dippers lift ¾ to 2, or even up to 6 cu. yds. in one bite, and the spoil is at once tipped into trucks or wagons and removed on rails with a minimum loss of time. One steam excavator requires at least one locomotive to look after it, with a speed not exceeding 8 to 10 miles per hour for sets of 30 to 40 wagons, and stops of 15 to 20 minutes not more than 4 miles apart for 'fatting.' Allow ½ hr. at one end and ½ hr. at the other for running round the wagons, getting coal and water, and shunting out crippled ones. Steam-driven end-tipping wagons hold 2½ cu. yds. and side-tipping wagons 3 cu. yds., measured in the solid. Maintaining them in good running order costs, pre-war, ¼*d.* to 1*d.* per yd. cube per mile.

Generally, it takes 45 side wagons to keep a steam excavator going, if the haul is not more than 1¼ miles, and gradients are such that a 12- or 13-in. locomotive can handle 20 wagons in a set.

Expenses of operating a locomotive depend on water (mostly), gradients, and coal. The pre-war labour cost of driving and repairing a locomotive is £1 per shift, including fitters and smiths. To this add foreman, boy turning points, general expenses 30 per cent., interest and depreciation, 1 ton of coal, repair accounts, water, oil, and stores, or £2 10*s.*, making a total cost per shift of £3 10*s.* Washing and repairing engine while in shed adds 20 per cent. to wages paid for actual driving.

2. *Trench-diggers.*—The advantages compared with hand labour are economy, speed, ease of control, and ready supervision. The cost is less than half that of hand labour, and the deeper the trench the greater the saving. If a labourer can excavate and throw out by hand ½ yd. cube per hour in trench work it will take 120 men to yield a total output equal to one No. 36 machine, which can dig 1 yd. cube per minute or 60 yds. cube per hour.

A 'Parsons' trench excavator, operated by petrol or steam, can dig a trench as detailed below. The prefix numbers indicate the maximum width in inches to which that particular machine will cut, and it will excavate any depth up to that mentioned.

Machine.	Working Weight.	Width of Trench.	Depth.	Rate per Minute.
	tons.	ins.	ft.	
No. 24	8	15, 18, 24	10	8 ins. to 10 ft.
No. 36	14	20 to 36	15	6 ins. to 10 ft.
No. 78	24	28 to 78	20	1½ ins. to 4 ft.

3. *Grab Buckets.*—Grab buckets are employed when it is impracticable to use a steam shovel or trench-digger, and they can be operated by a special crane, or by attaching to an ordinary steam crane or derrick. The common type is known as the clam-shell grab, which has jaws opening in and out. Capacity varies from ¾ to 3 cu. yds., and 20 to 30 cycles per hour of grabbing, raising and swinging round, and discharging can be performed. The cost is about half that of hand labour, and the saving is greater when under water. When dredging Limerick harbour with a Priestman's grab the average pre-war cost of dredging and discharging was 1⅞*d.* per ton.

4. *Chain Bucket Excavators.*—This excavator consists of a bucket ladder which scoops up material and discharges it into wagons either above or below the engine. The machine is best for a definite profile, such as a railway cutting or reservoir, but the ground must be free of rock or boulders. A good size weighs 40 tons, and digs to 20 ft. deep or more at theoretically 190 cu. yds. per hour, but the actual output is about 66 per cent. of this rate.

5. *Drag-line Excavators.*—Generally speaking, drag-line and cable-line excavators are used for cuttings of large size for drainage purposes, for canals, handling sand and gravel, and in some instances for preliminary road work over rough ground. These excavators can proceed in any direction from railway station to works or from one job to another.

Bucket capacity 1 cu. yd., weight of machine 26 tons.
" 2 " " " 62 "
" 3 " " " 75 "

The cableway drag-line excavator has two masts, known as the head tower and tail tower, with track cable between, along the line of which the bucket runs. Buckets, 1½ cu. yds. With an average haul of 225 ft. cost was 5*d.* per yd. cube, including maintenance, repairs, fuel, supervision, and depreciation.

6. *Scrapers.*—Two types, 'drag scrapers' and 'wheel scrapers.' These are widely used in America for shallow excavation and grading work, and are drawn by one horse and one man, or by two horses and two men. The surface is first cut up with an ordinary plough, and the scraper then dragged over the ground to remove the loosened soil. Capacity of a drag scraper is about $\frac{1}{3}$ cu. yd.—wheel scraper much greater, about $\frac{1}{2}$ cu. yd.

7. *Derricks and Cranes.*—Contractors' locomotive or 'loco' steam cranes may weigh from 10 to 60 tons, and lift loads up to 25 tons, with a jib radius of 16 to 25 ft. The cranes have 'all motions,' *i.e.* lifting, lowering, revolving, jib adjusting, and travelling on rails with a gauge of 4 ft. 8½ ins. to 12 ft.

Stationary or travelling steam cranes may be employed to work iron or flat-bottom wooden skips, which are like large buckets or open boxes respectively, and hold $\frac{1}{4}$, $\frac{1}{2}$, $\frac{3}{4}$, 1 cu. yd., or more, made to discharge from the crane over a wagon. Woodford's patent steel bucket skips are excellent, self-righting, and with safety catch.

8. *Radial Loaders.*—These are used for raising loose soil from the bottom of an excavation to the surface level, and have a chain of buckets attached to an inclined arm or boom, and a shoot, and are driven by an electric motor or a petrol engine. The usual capacity is 1 cu. yd. per minute, and the machine can load a 7-ton wagon in 7 mins. Cost of handling soil by this method is 1*d.* to 6*d.* per yd. cube.

The Haiss wagon-loader is a labour-saving device for dealing with loose materials, supplied by Millar's Machinery Co., Ltd., Pinners' Hall, London. It is a self-propelled petrol-driven machine fitted with a bucket elevator, operated by one man, and can load 5 cu. yds. into a truck in five minutes.

The saving in loading cost will enable a contractor to quickly regain the expenditure on these appliances.

9. *Paving Breakers.*—Breaking up old macadam, concrete, or paving was formerly done by steel wedges, sledge hammers, and crowbars by hand labour, but these are tedious and expensive compared with mechanical means.

Pneumatic paving-breakers show a cost saving of as much as 65 per cent., and a time saving of 75 per cent., over hand methods. These percussion tools are operated by compressed air from a portable compressor, driven either by petrol or electricity, which will work two to five large picks.

Weight.	Length overall.	Size of Steel.	Hose.	Air pressure.
lbs.	ins.	in. hex.	in.	
60	24½	1½	¾	Over 80 lbs.
69	26¾	1⅝	¾	Under 80 lbs.

The steel cutters are of two kinds—one with a chisel edge, and the other with a moil point. An ordinary gang consists of three men—one compressor man and two breaker operatives. Each operative can break up an average of 245 yds. sup. of 9-in. concrete per day. Cost of running plant in oil and fuel is about 2*s.* per hour. Breaking up concrete surfaces, 8 to 20 ft. wide and 9 ins. to 12 ins. thick, in London streets, was 2*s.* 6*d.* per yd. cube, based on one portable compressor working four or five large picks.

One man with one paving breaker can smash up as much old tarmac on concrete roads for cutting trenches, etc., as nine men handling wedges and sledge hammers. Removing the top layer of a road came to 6*d.* per yd. sup. against 2*s.* per yd. sup. for burning and picking off.

10. *Soil Transport.*—It is impossible to maintain a big output with excavating equipment unless the surplus soil can be removed as quickly as the digging is done. Speed obviates the large number of trucks or wagons caused by slowness. Careful planning is required for rails, wagons, dumps, etc., and good organisation.

The types of transport may be classified as—

(*a*) Hand labour small receptacles, such as wheelbarrows or buggies.
(*b*) Horse-drawn vehicles, wagons or carts.
(*c*) Steam-driven wagons.
(*d*) Petrol-driven wagons.
(*e*) Narrow-gauge track with wagons.
(*f*) Standard-gauge track with trucks.
(*g*) Electrically driven trucks and vehicles.

(*a*) *Hand Labour.*—Net unit cost per ton-mile about 7*s.* 6*d.* (February 1922), for removal including loading up, wheeling, levelling, and return of empty, reckoning interest on initial outlay, depreciation, and maintenance.

Assume one man will do the equivalent to 10 ton-miles per week of 44 hrs. Output low and unit cost high, as many journeys are needed to transport a ton.

(*b*) *Horse-drawn Vehicles.*—1*s.* 6*d.* per ton-mile under ordinary average conditions, so this is very economical compared with hand labour, or only one-fifth cost. It is also a flexible system, as horses and vehicles can be hired, but is not sufficient for extensive excavation schemes. Tip-carts or tip-wagons are necessary, and the tipping can be done by a lever at the seat of the driver.

(*c*) *Steam-driven Wagons.*—The haulage costs of these may be divided into standing charges, which are constant, and running costs, which are governed by mileage. The costs per mile (July 1921) work out as below, including standing charges of insurance, rent, interest on first cost, licence, registration, and wages; and running costs of fuel (coal, coke, or petrol), lubricants, tyres, repairs, maintenance, depreciation on 'life,' etc., based on 200 miles per week, although 400 miles or more may be accomplished. The fuel-consumption basis, plus 20 per cent. for wastage, is 55*s.* per ton for coal or coke, and 2*s.* 8*d.* per gallon for petrol.

Unit.	4-ton Steam.	4-ton Petrol.	6-ton Steam.	6-ton Petrol.	10-ton Steam and Trailer Combination.
Cost per ton-mile	4¾*d.*	6*d.*	3½*d.*	4¾*d.*	2½*d.*
Fuel mileage .	20 miles per cwt.	7 miles per gal.	15 miles per cwt.	5 miles per gal.	13 miles per cwt.

Life of a steam wagon without trailer is taken as 12,000 miles.
" petrol lorries is 25 per cent. less, say . 9,000 "
" wagon tyres with trailer . . . . 7,000 "
" trailer tyres . . . . . 12,000 "

In calculating the depreciation allowance—

Life for a steam wagon without trailer is taken as 150,000 miles.
" steam wagon with trailer . . . . 120,000 "
" petrol lorries . . . . . 100,000 "

The cost shows considerable saving over horse-drawn vehicles at 1*s.* 6*d.* per ton-mile. In fact, the expense of transporting 4 tons 1 mile by steam wagon, 4¾*d.* × 4 = 1*s.* 7*d.*, is practically equal to that of 1 ton only the same distance by horse wagons.

The output of a 4-ton steam wagon with a speed of 8 to 10 miles per hour will run to 800 ton-miles per week, equivalent to 80 men by hand labour. To justify the initial outlay of an end- or side-tipping wagon at £1,200 over a horse and cart, it should go 24,000 ton-miles to make the necessary saving, which is easy considering that its estimated life is 300,000 ton-miles of work to be performed. With several steam wagons the gain is enormous, and the economy of employing them on large works amply justified.

(*d*) *Petrol-driven Wagons.*—A steam wagon is best for transporting a maximum tonnage over a maximum distance in a minimum time at a minimum cost; but for maximum mobility with lighter loads and for special suitability for intermittent work, the petrol vehicle has undisputed claims, though its working costs are necessarily higher.

An investigation into the actual working records of 100 steam wagons by five different makers, and of 100 petrol vehicles by nine leading makers, showed that the average cost (1920) for hauling 5 tons per mile worked out at:—

For steam wagons, 1*s.* 5½*d.*, or 3½*d.* per ton-mile.
For petrol vehicles, 2*s.* 10½*d.*, or 7*d.* per ton-mile.

These rates included all overhead charges, 12½ per cent. depreciation, interest, fuel, wages, repairs, etc., over 15 million ton-miles. The annual saving of each steam wagon compared with petrol was over £1,000. Steam wagons gave steadily consistent figures, whereas petrol lorries varied widely.

Usual capacities of tipping petrol wagons are 1½, 2, 3, 4, and 5 tons, but 2 tons is the most useful. Cost per ton-mile, 6½*d.*, based on 300 miles per week, including depreciation, interest on capital, wages, tyres, renewals, fuel, etc.

(*e*) *Narrow-gauge Track with Wagons.*—Transporting surplus soil in side-tipping wagons on a narrow-gauge track is, under general conditions, more economical and far superior to ordinary manual labour with small receptacles or to horse-drawn carts. The track is commonly known as 'Jubilee,' industrial, or contractors' track, and the wagons as 'Jubilee' wagons. It is a portable railway with steel rails (10 to 20 lbs. per yd.) and corrugated steel sleepers, and the lay-out should be well planned. The tip wagons (½ to 7 cu. yds.) can be managed by hand labour, horses, or light locomotives.

Cost for hand-operated wagons on a level railway is 1*s.* 3*d.* per ton-mile.
„ horse-drawn „ „ „ 0*s.* 3*d.* „

Comparative costs for the different methods of transport:—

	s.	*d.*	
(*a*) Hand labour with wheelbarrow	7	6	per ton-mile
(*b*) Horse and cart	1	6	„
(*c*) Steam wagons, average	0	3½	„
(*d*) Petrol vehicles, average	0	7	„
(*e*) Hand-operated wagons on industrial track	1	3	„
(*f*) Horse-drawn „ „ „	0	3	„

Steam wagons are, therefore, cheapest for large quantities over considerable distances, and the industrial track (with horse labour) for small or large quantities over short distances.

(*f*) *Standard-gauge Track with Trucks.*—This system is restricted, as the first outlay is expensive, and the method is not so flexible as light railways. Large tipping wagons will be necessary, and no cost figures can be given, as the mode of transportation is for special circumstances.

(*g*) *Electrically-driven Trucks and Vehicles.*—These have not been extensively developed up to the present, but they are useful for short haulages and when continual stopping and restarting are unavoidable.

Taking a standard make of 5-ton electric vehicle, the 'fuel' cost at 2*d.* per unit works out at 2⅝*d.* per mile, with loads from 4 to 5 tons. Running costs, including interest at 6 per cent., depreciation at 10 per cent., tax, insurance, wages, charging current, rubber tyres, oil, grease, batteries at 5 years' life, etc.:—

Weekly mileage.	Load carried.	Annual Cost.	Running Cost.
	tons.	£	per ton-mile.
100	2	482	11½*d.*
100	3	482	7½*d.*
140	4	515	4½*d.*
140	6	515	3*d.*

Electrical traction on industrial or light railways, running costs 1½*d.* per mile.

Electric locomotives (costing about £450 per vehicle, complete with battery) will haul a load of 10 tons on a level track, and one charge of battery will give a normal run of 20 miles. One charge takes 14 units of electricity at 2*d.* per unit, which works out to 1½*d.* per mile run.

An electric locomotive can economically pull wagons on a narrow-gauge track in lieu of manual or animal labour, with great saving of time. The comparatively small capital will soon be recovered. The advantages are—reduced first cost, easy control, efficiency, flexibility, and low expenses of operation, upkeep, maintenance, and depreciation.

GAS WORKS.

Gasworks.—Gasworks complete, including buildings, plant, and distributing mains:

Estimated costs, 1930.	Small Towns.	Large Towns. or Cities.
Per head of population	40*s.* to 60*s.*	50*s.* to 80*s.*
Per ton of coal carbonised per annum	7*l.* to 10*l.*	9*l.* to 12*l.*
Per million cubic feet of gas produced per annum	700*l.* to 900*l.*	900*l.* to 1,200*l.*

Land.—Area of land for the erection of works at least 2 acres per million cubic feet of gas per maximum diem.

Proportionate Capital Expenditure.—Including works, distribution plant, etc.:—

On buildings and works	38	per cent.
On district mains	55	„
On Parliamentary, promotion and engineer's expenses	7	„
	100	

Proportionate Cost of Parts.—Including buildings and plant :—

Retort house, including benches, fireclay retorts, etc.	25 per cent.
Purification buildings, with purifiers, gas scrubbers, etc.	30 ,,
Gasholders, iron or steel, with guide framing, brick-lined tanks, concrete bottoms, and excavation	40 ,,
Station meter-house, meter, and miscellaneous	5 ,,
	100

Retort-houses.—Including foundations and brickwork to benches, etc. :—

Exclusive of plant or machinery (1930)	9*d.* to 1*s.* per f.c.
Including retorts, ironwork, and fittings (1930).	2*s.* to 4*s.* ,,

Glover-West's system for vertical gas retorts, Belfast (1913) :—
70,950*l.* for a plant to carbonise 480 tons of coal per 24 hours,
=148*l.* ,, ,, ,, 1 ton ,, ,,

Gasholders.—Steel or iron telescopic holders, including guide framing, etc. :—

Estimated costs, 1930.	Per 1,000 cub. ft. of capacity. £ £
Single-lift holders, up to about 50,000 cub. ft.	50 to 70
If of the spiral type, 25 per cent. less	38 ,, 53
Two-lift holders, about 250,000 cub. ft. capacity	30 ,, 40
If spirally guided, about 50 per cent. less.	15 ,, 20
Large holders, from 2 to 3 million cub. ft. capacity	18 ,, 30
The 10½-million holder at Manchester cost (pre-war)	5
The 12-million holder at East Greenwich Works cost (pre-war)	5 guineas.

GROYNES.

There are four types of groyne construction in this country—timber, concrete block, reinforced concrete, and chain cable. Cost of timber groynes for the seashore, comprising creosoted fir piles with horizontal sheeting bolted thereto, and stiffened with ties and struts. Piles well driven into sand or shingle, and executed as tide-work. Pre-war costs :

Light timber groynes, 1*l.* to 3*l.* per yard run.
Heavy ,, ,, 3*l.* ,, 6*l.* ,, ,,

The following particulars have been condensed from *Marine Works*, by E. Latham, M.Inst.C.E. (1922. 16*s.* Crosby Lockwood & Son, London), and the other books mentioned in each case. The first five examples are pre-war, and the last post-war.

	Cost per yd. run. £ *s.* *d.*
Bridlington.—Low timber case groynes, constructed of pitch pine piles, shod with 18-lb. pile-shoes, driven 8 ft. into the boulder clay in pairs, at 7 ft. 6 in. centres. 11 in. × 4 in. pitch pine planking inserted between the piles, so that a depth not exceeding two planks projects beyond the sand level. Timber all creosoted. Groynes spaced at a distance apart not exceeding their length, which was about 200 ft. First four groynes cost	2 14 0
Most of the others cost	1 16 9
For details see *Coast Erosion and Protection*, pp. 114–115, by E. R. Matthews, A.M.I.C.E. (1918. Charles Griffin & Co., London.)	
Eastbourne.—Timber groynes erected about 1886–96, some 70 yards apart, at right angles to the foreshore. Constructed of 9 in. × 9 in. pitch pine and oak piles, 6 to 16 ft. long, driven to an average depth of 9 ft., set 4 ft. 6 ins. apart, centre to centre, with walings 8 ins. × 6 ins. and 6 ins. × 6 ins. Planks, 9 ins. × 2 ins., are spiked to the piles in lengths of 20 ft. and over, to span four bays and break joint. Large oak land-ties fixed at right angles to the piles, and fastened to a smaller pile at back. The majority are high groynes, and were originally 8 to 10 ft. in height at shore end, and about 4 ft. at low end, and did not extend to low-water mark. The more recently constructed groynes vary from 4 ft. high at the land end to 2 ft. at low-water mark. Average length 100 yards, and cost 375*l.* each, or	3 15 0
Seven case groynes erected to the east of the pier in 1897. Pairs of 9 in. × 3 in. pitch pine piles, spaced 7 ft. 6 in. centres, with 9 in. × 3 in. planks inserted between. Average length 100 yards, and cost 68*l.* 15*s.* each, or	0 13 9
For details see *Coast Erosion and Protection*, pp. 104–106, by E. R. Matthews, A.M.I.C.E. (1918. Charles Griffin & Co., London.)	
Lowestoft (1901–1903).—North Denes. Four timber main groynes, three having a length of 290 ft., and the fourth 340 ft., spaced 1,500 ft. apart. Shoreward ends 4 ft. above high-water level, and seaward ends 3 ft. below the same level. For details see *Proceedings of the Institution of Civil Engineers*, vol. clxxxv. p. 106	22 1 3

	£	s.	d.
Southwold.—Timber groynes. For details see *The Protection of Sea Shores from Erosion*, p. 23, by A. E. Carey, M.I.C.E.	4	4	0
Withernsea (1873).—Six timber groynes consisting of 13 in. piles, 22 ft. long, 5 ft. apart, driven down 12 ft., with 11 in. × 4 in. planks, 13 in. × 6½ in. struts, and foot piles. Four are 300 ft. in length, and two 350 ft. Only three planks project above sand-level. For details see *Proceedings of the Institution of Civil Engineers*, vol. clix. p. 73	3	7	6
Blue Anchor (1921).—On the Bristol Channel, and in front of the Minehead to Watchet Road, north Somerset. Groyne 600 ft. long and two timber stockades, each 1,500 ft., parallel to existing sea-wall. Stockades consisted of 10 in. × 10 in. main piles of pitch pine, with 9 in. × 4 in. intermediate piles bolted to continuous walings. Half of total length of stockades had to be constructed by bedding the piles in mass concrete, necessitating trenching in rock. Engineer, E. Latham, M.I.C.E. Contractors, Messrs. John Cochrane & Sons, Westminster. Total about 30,000*l.* Cost per yd. run includes stockades	25	0	0

Harbours.

Artificial harbours complete, including reclamation of land, dredging, enclosure walls, breakwaters, landing stages, lighting, buoys, cranes, etc. Estimated costs, 1930 :—

10,000*l.* to 20,000*l.* per water acre enclosed.

Colombo Harbour (1874–1914).—On the west coast of Ceylon and entirely artificial. It ranks twelfth among the greatest seaports, and there is ample space for 50 steamers besides smaller craft. The enclosed area is 650 acres (over a square mile), and depth 30 to 36 ft. at low water. Rise of tide at Ceylon is only 2 ft.

South-west breakwater, 1874–85, is 4,212 ft. long, built of concrete blocks weighing 18 to 30 tons on the sloping bond system, and cost 705,200*l.*, or 167*l.* per ft. run. Extension to ditto as a further protective arm, 1907–12, is 1,800 ft. long, also of concrete blocks, and cost 380,000*l.*, or 211*l.* per ft. run. North-east breakwater, 1894–7, is 1,100 ft. long, and constructed as a rubble embankment; the north-west or island breakwater, 1900–6, is 2,657 ft. long, and made of concrete; the two together cost 500,000*l.*, or 133*l.* per ft. run. Total length of the four breakwaters about 9,770 ft., with 5 lighthouses on them, and total cost 1,585,200*l.* Graving dock, 1900–4, is 700 ft. long, 85 ft. wide, and cost 400,000*l.*; patent slip, 1900–3, is 800 ft. long, and cost 43,000*l.*; coaling depot, on 24 acres of reclaimed land, has a frontage of 2,200 ft. with 18 jetties, and cost 163,000*l.* Other services, 821,900*l.* Total expenditure from 1874 to 1914 (forty years) was 3,013,100*l.* The small Fishery Harbour immediately to the northward has a rubble arm 800 ft. long, and cost 12,000*l.* The engineers were Sir John Coode, Son, and Mathews; and the resident engineers, Sir John Kyle, M.I.C.E., Mr. J. H. Bostock, M.I.C.E., Mr. John Kyle. M.I.C.E. (son of Sir John), and Mr. A. D. Prouse, M.I.C.E.

4,636*l.* per water acre enclosed; 162*l.* per ft. run, average of breakwaters.

Dover Harbour (1898–1909).—Original Admiralty Pier, 1,000 ft. long, was built in 1847–71. Dover Harbour Board then constructed a small commercial harbour of their own in 1890, between the existing Admiralty Pier and a new east pier, of 56 acres net area, afterwards increased to 75 acres by reclaiming the foreshore. The National Harbour was proceeded with in 1898–1909, and comprised the extension of the western breakwater, or old Admiralty Pier, from 1,000 to 3,000 ft., an eastern arm, 3,320 ft., and an isolated breakwater in the middle, 4,200 ft. Total sheltering works, 9,520 ft., or nearly 2 miles long, with two entrances 740 ft. and 650 ft. wide and 42 ft. deep at low water, enclosing 610 acres, exclusive of the 75 acres in the Commercial Harbour. Height of breakwaters from bottom of foundation to parapet top 60 to 120 ft. or an average of 90 ft., with 50 ft. average depth below low-water level, and 50 ft. average thickness, built of concrete blocks, weighing 10 to 40 tons each and faced with granite above low-water level. Works include a sea-wall, 3,850 ft. long, to protect base of cliffs, a lesser sea-wall, 700 ft., and reclaiming about 20 acres of land. Engineers, Sir John Coode, Son & Mathews. Contractors, Messrs. S. Pearson & Son, Ltd. National Harbour, 610 acres, cost 4,000,000*l.*, and inside Commercial Harbour, 75 acres, 1,250,000*l.*, or a total of 685 acres and 5,250,000*l.*

National Harbour :	6,557*l.* per water acre enclosed ;	420*l.* per ft. run of breakwater.
Commercial Harbour :	16,666*l.* " " " ;	131*l.* " " "
The two combined :	7,664*l.* " " " ;	551*l.* " " "

Heating.

The fewer the radiators, and the longer the run, the higher the cost per radiator. 15 ft. sup. of radiator should be supplied for every 1,000 cubic feet of space to be heated. Cost includes fittings complete, but is exclusive of boiler-house and buildings (1930).

Heating buildings with boilers, pipes, and radiators	2*l.* to 8*l.* per 1,000 f.c.
Low-pressure hot-water heating and connections	15*l.* ,, 20*l.* per radiator.
Hot-water work inside a building, 3-in. and 4-in. pipes . . .	2*s.* ,, 3*s.* per ft. run.

Landing Slips.

Landing Slips, Milford Haven (1894).—Formed of Portland cement concrete, with shingle supplied by contractor, and 8 ft. wide × 3 ft. high, on 18-in. foundations. In tidework, and not exceeding 40 yards from shore. Excavation in boulders and shingle on beach. 30*s.* per foot run.

Lighthouses.

Lighthouses. (All of Granite.)	Date.	Height of Focal Plane above High Water.	Engineer.	Total Cost.	Cubical Contents.	Cost per Ft. Cub. of Masonry.
		Ft.		£	f.c.	£ *s.* *d.*
Ar'men, Finistère, France	1867–81	94	L. Reynand	37,692	32 400	1 3 3
Bell or Inchcape Rock, Forfarshire	1808–11	117	R. Stevenson	55,820	28,530	1 19 0
Bishop Rock, Scilly Isles	1851–58	110	J. Walker	34,560	35,209	0 19 8
Bishop Rock, strengthening and part reconstruction	1883–87	146	Sir J. N. Douglass	64,889	45,080	1 8 9
Dhu Heartach, West of Scotland	1866–72	145	D. & T. Stevenson	72,584	42,050	1 14 6
Eddystone (third lighthouse), off Plymouth	1756–59	72	J. Smeaton	40,000	13,343	3 0 0
Eddystone (fourth lighthouse)	1878–82	133	Sir J. N. Douglass	59,255	65,198	0 18 2
Fastnet, S.W. Ireland	1899–1904	158	W. Douglass	79,000	62 600	1 5 6
Great Basses, Ceylon	1870–73	108	Sir J. N. Douglass	63,560	47,808	1 6 7
Hanois, Alderney	1862	100	J. Walker, Burges, and Cooper	25,296	24,542	1 0 7
Longships, Land's End	1872	110	Sir J. N. Douglass	43,870	47,610	0 18 5
Minot's Ledge, Boston, Mass.	1855–60	89	General Barnard	62,500	36.322	1 17 2
Skerryvore, Argyllshire	1838–44	158	A. Stevenson	72,200	58,580	1 4 8
Smalls, Bristol Channel	1856–61	114	J. Walker	50.125	46,386	1 1 7
Spectacle Reef, Lake Huron	1870–74	93	General Poe	78,125	42,742	1 16 2
Wolf Rock, Land's End	1862–69	110	J. Walker and Sir J. N. Douglass	62,726	59,070	1 1 3

Lighthouse, Northern Point, Northern Province, Ceylon (1914–16).—Tower is supported on reinforced concrete platform over hardwood piles, and built of dressed coral stone in lime mortar, cement rendered externally. Walls 2 ft. 7½ ins. thick above plinth, carried up with a uniform thickness at a batter of 1 in 48. Internal diameter of tower 8 ft. 3 ins., and height from top of foundation to reinforced concrete cap 91 ft. Fourth order light visible for 16 miles, and focal plane 97½ ft. above mean sea level. Cost, exclusive of lantern and ironwork above the cap, 1,730*l.*, or 1*s.* 6*d.* per foot cube.

Lighting.—A complete gasworks plant for a lighthouse costs about 800*l.*

Annual expenditure for gas lighting, where works are installed . . . 72*l.* per light.
" " " " where gas is supplied from elsewhere . 32*l.* "

A first-class electric lighthouse installation, including optical apparatus, lantern, dynamos, engines, air compressor, siren, etc., but not buildings, costs about 6,000*l.*

Maintenance.—Average annual cost of maintenance of an English shore lighthouse, with two keepers, is 280*l.*

Average annual cost of maintenance of an English shore lighthouse, with three keepers and a siren fog signal, is 450*l.*

Annual maintenance of a rock lighthouse, with four keepers and an explosive fog signal, is about 800*l.*; and of an electric light station, 1,100*l.*

Lightships.—Steel lightships, of average size, with power-driven compressed air siren, cost about 16,000*l.*

Annual maintenance of ditto, including periodical overhaul, 1,320*l.*

Usual limits.—Length . . . . . 80 to 114 ft.
Beam . . . . . 20 " 24 "
Depth moulded . . . . . 13 " 16 "
Tonnage . . . . . 155 " 280 "

LIGHTING.

Buildings.—Cost of gas and electric lighting inside buildings, including service pipes, stopcocks, ordinary brackets, pendants, switches, wiring, and other articles, fixed complete, but excluding meters, external gas mains, generators, etc.

	s.	s.	
Acetylene	20	to 40	per light.
Coal gas	30	„ 50	„
Electric	25	„ 50	„

Wiring, including all necessary wire, switches, fuses, lamps, holders, casings, etc., but not electroliers or shades, 15*s.* to 40*s.* per lamp.

Installing artificial lighting per 100 points, including oil-engine, dynamo, wires, lamps, etc., but excluding buildings:

Acetylene.	Petrol Gas.	Electric Light.
100*l.* to 200*l.*	150*l.* to 200*l.*	200*l.* to 300*l.*

Acetylene.—Cost of upkeep consists mainly in the purchase of carbide with occasional renewal of burners. A 25 c.p. burner of ordinary type consumes about 1 f.c. of gas per hour, which demands ½ lb. carbide. With calcium carbide at 15*s.* per cwt. the cost of such a burner works out at ¾*d.* per hour. Total upkeep of a 100-light installation is between 20*l.* and 30*l.*, according to number of hours lights are used.

Gas Lighting of the Castle, Carlisle (1896).—Supply from City of Carlisle Gas Company's mains through a 600-light dry meter supplied by Company. 4 in., 3 in., and 2 in. cast-iron mains and 1½ in. to ½ in. galvanised wrought-iron service pipes, underground pipes 3 to 3½ ft. deep. Fittings include 19 governors, 10 valves, 20 stopcocks, 10 dry meters, 572 burners, 10 Wenham lights, 12 Schulke lights, 20 Deimel lights, 18 Welsbach incandescent lights, 12 Sugg's lights, etc. Occupation of barracks, 731 adults and 72 married quarters.

Total, 1,144*l.* 36*s.* per light or point. 28*s.* per head.

Gas Lighting of Strensall Camp, near York.—Completed 1915, in extension of the York Gas Co.'s mains. Cost includes 3-in. external steel mains, 2 in. to ½ in. wrought-iron branches, brick meter-house and meters, siphons, stopcocks, burners, excavation, fixing, etc., complete, in huts and 40 married quarters for about 1,260, total of officers, N.C.O.'s, and men. 376 incandescent burners and 584 flat-flame burners, or 960 lights altogether.

Total, 1,780*l.* 37*s.* per light or point. 28*s.* per head.

Electric Lighting, Army Pay Offices, York.—Feb. 1915. 84 electric lights, mostly of 100 c.p. each, including opal glass shades, wiring, cords, switches, fuses, etc., but not cable into building or any plant, as taken from the Corporation electric main in road. 18 officers and 300 clerks.

Total, 91*l.* 22*s.* per light or point. 6*s.* per head.

Electric Lighting, Army Pay Offices Extension, York.—Aug. 1916. One-storey corrugated iron building, lined with plaster slabs. 136 electric lights, mostly of 100 c.p. and 50 c.p. metal filament lamps, including pendants, opal and holophane glass shades, wiring, cords, 109 switches, fuses, distribution boards, etc. Direct-current supply on the three-wire system taken from the Corporation electric main in adjacent road. 24 officers and 400 clerks.

Total, 137*l.* 20*s.* per light or point. 6*s.* 6*d.* per head.

Railway Coaches.—	£
Cost of fitting a railway coach with electric light	130
„ „ „ „ incandescent gas	34
Annual maintenance of an electric lit railway coach	13
„ „ a gas „ „	7

STREET LIGHTING AT CROYDON.

(1910.)

Type of Lamp.	Number of Lamps.	Distance Apart.	Total Cost.	Average c.p. per Mile.	Cost per c.p. per Annum.
	No.	Yds.	£	c.p.	*d.*
Incandescent gas	2,137	80	7,062	839	15·86
Incandescent electric	90	66	288	1,373	13·71
Electric arcs	428	65	7,212	10,537	11·32

Locks (1930).

Including foundations, invert, walls, gates, sills, machinery, lock-keeper's cottage, etc.

Brick and concrete	Per yd. cu. of internal capacity 50*s.* to 80*s.*
Stone and concrete	,, ,, ,, 80*s.* ,, 120*s.*
Canal locks	each 10,000*l.* to 20,000*l.*
River locks	,, 30,000*l.* ,, 50,000*l.*

Marine Piers.

Timber and iron piers for seaside places, with creosoted fir or iron screw piles, averaging 40 ft. long and driven up to 15 ft. in sand or shingle, including braces, ties, timber decking, railings, tarring, gates, ticket office, seats, etc., but exclusive of pavilion or band-stand.

Gangway or promenade portion	8*s.* to 12*s.* per ft. sup.
Landing stage or head to pier	10*s.* ,, 16*s.* ,,

Military Pier, Tilbury (1896).—450 ft. long by 16 ft. wide, or 7,200 ft. sup. Height above h.w.o.s.t. level, 4 ft., and above l.w.o.s.t. level, 20 ft. Foundation, 3 ft. of mud over 20 ft. of chalk, the distance bored. Substructure of wrought-iron piles, 10 ft. apart, with rod iron braces, on which are fixed wrought-iron girders with intermediate joists supporting timber decking. 18-in. gauge tramway rails, 24 lbs. per yard, and steam crane to lift 10 tons. Total cost, including 500*l.* for hoisting machinery, 3,200*l.*, or 8*s.* 11*d.* per ft. sup.

Batticaloa Jetty, Eastern Province, Ceylon (1900).—Timber pier, 50 ft. long, with return portion 22 ft. long, and 10 ft. wide. Cost, 5*s.* 6*d.* per ft. sup.

Galle Jetty, Southern Province, Ceylon (1916).—Jetty is 162 ft. long × 20 ft. wide, carried on concrete cylinders 6 ft. diameter, spaced 20 ft. centre to centre longitudinally and 14 ft. transversely, end pair being taken down to a depth of 20 ft. Special cylinders, 8 ft. diameter, form seats for two 3-ton cranes. Jetty is roofed for a length of 137 ft. Cost, 4,923*l.*, or 1*l.* 10*s.* per ft. sup. Cost of cylinders sunk, 6 ft. diameter, 3*l.* 19*s.*, and 8 ft. diameter, 5*l.* 3*s.*, per ft. run.

Kalkudah Jetty, Eastern Province, Ceylon (1901).—Jetty of iron, with wooden joists and planking, 165 ft. long × 28 ft. wide, and landings. Piles of H-iron section, 9 ins. × 7 ins., carried through a stratum of sand about 8 ft. into the coral bed of 1 to 2 ft. thick. Cross bracing, walings, fenders, etc. Corrugated iron roof over 122 ft. of length, supported on iron principals carried by H-iron standards, 8 ins. × 6 ins. Water 8 ft. deep at end of pier. Cost, including crane seat, 2,516*l.*, or 11*s.* per ft. sup. of platform area.

Paddiruppu Jetty, Eastern Province, Ceylon (1900).—70 ft. long with a tee head 30 ft. × 6 ft. wide, constructed of Palu timber with brick abutment. Cost, 8*s.* per ft. sup.

Mining.

(See Section XXXIX, p. 2555.)

Pile Driving.

(See p. 507.)

Railways.

Average cost of railways, main lines, 4 ft. 8½ in. gauge, double track, total capital expenditure including land and all charges :—

	£	£	
In England	35,000 to	45,000	per mile.
,, Scotland	25,000 ,,	35,000	,,
,, Ireland (5 ft. 3 in. gauge)	20,000 ,,	25,000	,,
,, Europe	15,000 ,,	25,000	,,
,, India (5 ft. 6 in. and metre gauge)	12,000 ,,	25,000	,,
,, Norway, light railways, 3 ft. 6 in. gauge	5,000 ,,	10,000	,,
,, Queensland	7,000 ,,	15,000	,,
,, United States	8,000 ,,	12,000	,,
Single line railways in poor districts of Scotland and Ireland	5,000 ,,	7,000	,,

(See also Section XIII. p. 529.)

PROPORTIONATE COST OF PARTS.

Earthworks	24 per cent.
Tunnels	10 ,,
Permanent way, including ballast	12 ,,
Sidings	3 ,,
Junctions and signals	1 ,,
Bridges and viaducts	9 ,,
Culverts and drainage	5 ,,
Stations and buildings	6 ,,
Construction only	70 ,,
Land and compensation	15 ,,
Rolling stock	10 ,,
Legal expenses	5 ,,
	100

	£	£	
Main lines, construction work and stations only	10,000 to	20,000	per mile.
Light railways ,, ,, ,,	1,000 ,,	3,000	,,
Permanent way, single line, light construction, 4 ft. 8½ in. gauge, in flat country, cuttings and banks averaging 3 ft., rails 55 lbs. per yard, ballast 15 ins. deep, including stations and permanent works, but excluding land and rolling stock	5,000 ,,	6,000	,,
Ditto, heavy construction, 75-lb. rails, and 24 ins. ballast	6,000 ,,	8,000	,,
Yearly maintenance of permanent way	200 ,,	250	,,
Rolling stock, single line, depends on kind of traffic	700 ,,	1,000	,,
Permanent way, double line, in excess of above figures	50 ,,	75	per cent.
Rolling stock ,, ,, ,, ,, ,,	10 ,,	30	,,

Working expenses in England average 50 per cent. of the gross receipts.
,, ,, Scotland ,, 45 ,, ,, ,,
,, ,, Ireland ,, 40 ,, ,, ,,
Average gross receipts, 4*s.* to 6*s.* per train mile.

Railway Footbridges.—Of iron or steel, including foundations, supports, girders, floor, staircases, lattice sides, etc., erected complete, 5 ft. wide.

250*l.* to 350*l.* each, 30 to 40 ft. span, or 15*s.* to 20*s.* per ft. sup.

Railway Viaducts.—On the Great Western Railway there are a great many viaducts, especially in Cornwall, commencing at 50 to 60 ft. deep, and being as much as 90 or 100 ft., and even up to 150 ft., in the centre of the gorge. They are, as a rule, about 60 ft. spans between the piers, which is found to be the economical width. Segmental arches nearly, but not quite, semi-circular, 3 ft. thick at centre, and 3 ft. 9 ins. at the springing. Piers, 7 ft. 6 ins. wide at the top, tapering on every side at 1 in 32. This gives a pressure of about 8 tons per square foot on the masonry. The cost comes out for such a structure, carrying two lines of way, at 1*l.* per yard forward per foot high, measured from ground to rail. That is, for example, a viaduct 200 yards long, and averaging 70 ft. in depth, will cost 14,000*l.*

Avon Viaduct.—On the Midland Railway, and of brick. 720 ft. long, 51 ft. high, and 11 arches of 50 ft. span. Cost, 14,000*l.*, or 19*l.* per ft. run.

Congleton Viaduct.—On the Manchester and Birmingham Railway, and of stone. 3,080 ft. long, or over ½ mile, and 106 ft. high. Cost, 113,000*l.*, or 37*l.* per ft. run.

Dove Viaduct.—On the same line, and of brick. 1,716 ft. long, 88 ft. high, and 23 arches of 63 ft. span. Cost, 54,000*l.*, or 32*l.* per ft. run.

Viaduct over the River Eye, Berwickshire (1889).—Six openings of 50 ft. clear span, 55 ft. 3 ins. between centres of piers, and extreme height above bed of stream 60 ft. Total length of bridge 331 ft. and 15 ft. wide; single track. Abutments of 1 to 7 concrete. Five piers, 45 ft. high and 12 ft. wide at top, with 9-in. and 18-in. brick face and concrete hearting, and 9-in. brick wall up the centre. Superstructure of lattice main girders 8 ft. apart, with cross girders of 10-in. × 5-in. rolled iron joists, 40 lbs. per ft. run. Flooring of 4-in. planking, and the rails carried by longitudinal beams, 14 ins. × 12 ins. Cost, 3,700*l.*, or 11*l.* per ft. run, or 15*s.* per ft. sup.

Viaduct on the Gifford Light Railway, Haddingtonshire (1900).—Four spans of 42 ft. 8 ins., centre to centre of three piers, with 1 to 6 concrete foundations. Height to rail level above the bed of stream is 50 ft., and length over all 194 ft. by 18 ft. wide; single track. Masonry piers, haunches, and abutments, and brick arches over openings. Cost, 2,043*l.*, or 11*l.* per ft. run, or 12*s.* per ft. sup.

New York Elevated Railroad (1872–80).—Steam traction. Two lines of way supported on square wrought-iron lattice-work columns, let into cast-iron base blocks founded on brickwork and concrete, at distances of 37 to 44 ft. apart. Rails of Vignoles or flange pattern, of Bessemer steel, weighing 50 lbs. per yd. run, spiked to cross timber sleepers and guarded by longitudinal timbers. Sharpest curve is 90-ft. radius. Steepest gradient, 1 in 50 for a length of 800 yds.

Stations about ⅛ mile apart, with platforms 200 ft. long and 13 ft. wide. No payment was made for way-leave along the streets, nor for compensation to frontages. Length, 35 miles. Total cost, 2,870,000*l.* Cost per mile of double track :—

	£
Foundations, columns, girders, superstructure, and permanent way	57,696
Railway stations and platforms	12,000
Rolling stock, 5 locomotives and 12 cars	11,680
Per mile of double track	81,376

Tube or Underground Railways.—Electric traction, as constructed in London. Two tracks 4 ft. 8½ in. gauge, each laid in a separate tunnel, 11 to 12 ft. diameter. 300,000*l.* to 600,000*l.* per mile of double track.

	£
London (Central), 1891–1900, 6 miles, 13 stations	580,000 per mile of double track.
London (Waterloo and City), opened 1898, 1½ miles	340,000 „ „ „

Steam Traction.—Two tracks, 4 ft. 8½ in. gauge, laid in elliptical tunnel, 29 ft. wide and 17 ft. high, of 27 in. brickwork on concrete foundations.

	£
London (Metropolitan), 1863–84	600,000 per mile of double track.
London (Metropolitan District), 1864-84	250,000 „ „ „

Both the foregoing have since been converted to electric traction, completed in 1905. The Metropolitan has 32 electric miles, and the " District " 28 electric miles.

Cost of electrification only, exclusive of power stations, sub-stations, rolling stock, etc., 5,000*l.* to 10,000*l.* per mile of double track.

Cost of Indian Railways.—For the year 1895 the capital cost (rupee at 1*s.* 3¼*d.*) of Indian railways was :—

	£	
Standard gauge, 5 ft. 6 ins.	10,095	average cost per mile.
Metre gauge, 3 ft. 3⅜ ins.	4,519	„ „
Special gauges, 2 ft. 6 ins. and 2 ft.	2,093	„ „

London Railway Stations.

Particulars.	Cannon Street, London, South-Eastern Railway.		Charing Cross, London, South-Eastern Railway.	
	Total.	Per F.S.	Total.	Per F.S.
		£ *s.* *d.*		£ *s.* *d.*
Area, exclusive of forecourt	F.S. 152,632	—	F.S. 103,672	—
Cost of substructure	£ 74,962	0 9 10	£ 41,422	0 8 0
Cost of superstructure	£157,262	1 0 7	£111,604	1 1 6
Total costs	£232,224	1 10 5	£153,026	1 9 6
Cost of land	£572,370	3 15 7	£302,808	2 18 5

Charing Cross Station roof has since been rebuilt with a different type (1906).

LIGHT RAILWAYS.

Narrow-gauge Railways.—Approximate cost, including works and rolling stock, cuttings and embankments 2 ft. deep, and ballast 6 ins. to 14 ins. deep, but excluding cost of land.

Gauge.	Weight of Engines exclusive of Tender.	Max. Load per Wheel.	Weight per Yard of each Rail.	Cost per Mile, Single Track.
Ft. Ins.	Tons.	Tons.	Lbs.	£ £
2 0	6	1	20	1,500 to 2,000
2 6	9	1½	25	2,500 „ 3,000
3 0	12	2	30	3,000 „ 3,500
3 3½	17	3	40	3,500 „ 4,500
3 6	20	3½	50	4,000 „ 5,000

BELGIUM.—Over 1,000 miles of minor lines have been constructed at a cost, including rolling stock and all outlay, of rather less than 3,000*l.* a mile. These are generally on the metre gauge (3 ft. 3⅜ ins.), and as many of them appear to be laid on the roads the cost of land is very small. They earn about 6*l.* per mile per week, and working expenses are 66 per cent., or 4*l.* per mile per week. Net earnings of 2*l.* per mile per week give a return for over 3 per cent. per annum on the capital.

ENGLAND.—In England a line on the standard gauge, with no heavy work, would cost twice as much as the Belgian metre gauge railways, or say 6,000*l.* per mile. As regards land, a light railway can seldom do with less than an average width of 1 chain (66 ft.) or 8 acres per mile, and this at 100*l.* per acre comes to 800*l.* per mile. Hence to run the line on a strip of green alongside a road would save a good deal of expense, but this might involve bad gradients and sharp curves and the maximum speed would be reduced. The following are some actual costs :—

Clogher Valley Railway, Ireland (1885–87).—Length 37 miles, 3 ft. gauge, 45-lb. rail, and runs for the most part on the side of the road. Cost, 3,300*l.* per mile. It earns about 4*l.* per mile per week, but this is mostly absorbed in working expenses.

Listowel and Ballybunion Railway, Ireland (1888).—Length 9 miles, mono-rail supported on steel trestles, fastened to steel sleepers resting on wooden sleepers, with gradients up to 1 in 45, and worked by locomotives. Cost, including rolling stock, 3,666*l.* per mile.

Wisbech and Upwell Light Railway, Cambridgeshire.—Length 5¾ miles, with 2 miles of sidings, standard gauge, 50-lb. rail, and runs mostly on the edge of the road. Cost, excluding sidings, 7,000*l.* per mile of line. The earnings are about 9*l.*, and the expenses 7*l.* per mile per week.

Agrails.—The Decauville tracks of the new light railways, or 'agrails' as they are called, did admirable service at the front, both in France and Belgium. They were laid at a cost of 2,500*l.* per mile, and are now being put forward as a rival or as an adjunct to motor lorry traffic, to save the costly wear and tear of roads and to relieve congestion. They can be run on the edge of existing roads, with sidings on the various farms, but are only suitable for comparatively flat districts. The old narrow-gauge railways were too solidly and expensively built, but the agrails are of very light construction and should prove most useful.

RIVER CANALISATION, see Section XXIX.

RIVER REGULATION, see Section XXIX.

ROADS.*

(Prices, August, 1928.)

FORMATION SURFACE AND FOUNDATIONS.

(For general application: Prices include about 17 per cent. establishment charges and profit. Wages, labourer, 1*s.* 4*d.* per hour.)

		s.	*d.*
Excavating over areas in ordinary loose ground not exceeding 5 ft. deep, throw up, and form bottom to required levels or slopes	per y.c.	2	0
Spreading and levelling in layers not exceeding 12 ins. thick	,,	0	4
Add for well ramming, including watering as required	,,	0	5
Filling barrows and removing spoil not exceeding 50 yds., and depositing not exceeding 5 ft. high above starting-point	,,	1	3
Add for each 25 yds. beyond 50 yds., and up to 100 yds. total	,,	0	5
Removing over 100 yds., but not exceeding ¼ mile, and depositing	,,	2	8
Filling into carts or lorries, removing not exc. 1 mile, and depositing	,,	5	0
Surface digging in ordinary loose soil not exceeding 6 ins. deep, throwing into heaps, and forming bottom to camber or slopes	per y.s.	0	7
Ditto, ditto, 6–12 ins. deep, and ditto	,,	0	10
Levelling and ramming common earth for pavings, etc.	,,	0	2
Ditto, ditto, to camber or falls	,,	0	3
4 in. Hard core, or dry filling of clean broken brick, mason's spalls, etc., spread, levelled, and rammed to a true surface	,,	1	2
6 in. ditto, ditto	,,	1	9
9 in. ditto, ditto	,,	2	8
12 in. ditto, ditto	,,	3	6
4 in. Portland cement concrete, 1 to 6, of clean pit gravel or fresh-water ballast, spread and levelled, with spade-finished surface	,,	4	0
6 in. ditto, ditto	,,	6	0
9 in. ditto, ditto	,,	9	0
12 in. ditto, ditto	,,	12	0
Face of concrete worked up to an even surface to receive granite setts, wood blocks, asphalt, etc., including any additional cement	,,	0	10
Floating with wooden hand-float, or steel trowelling, to a fair and even surface, including additional cement	,,	1	3
¾ in. floated or trowelled surface, 1 to 2, as a screed or bed	,,	2	0

* For Cost of Road Maintenance, see 'HIGHWAY ENGINEERING,' Section XV.

Water-bound Macadam Roads.

The operations include materials, labour, steam rolling to finished thickness as below, and forming to slopes or camber, complete. Digging taken under 'Formation Surface and Foundations' in last section. 'Sub-base,' or 'sub-grade,' means the lower portion; 'bottoming,' or 'pitching,' the heavy stone foundation; 'crust,' the upper section; and 'carpeting,' the surface or wearing layer.

Cost of Measured Work—

		s.	d.
3 in. Furnace ashes, or ¾ in. gauge hard clinker bed, well rolled with a 10-ton roller	per y.s.	0	9
4 in. ditto, ditto	„	1	0
6 in. ditto, ditto	„	1	6
6 in. Bottoming or foundation of hard rubble stone, hand-packed, upper interstices filled with spalls, formed to camber, and rolled with a 2-ton roller	per y.s.	2	6
9 in. ditto, ditto, and rolled with an 8-ton roller	„	3	9
12 in. ditto, ditto, and rolled with a 10-ton roller	„	5	0
3 in. Gravel, good binding, in two thicknesses, 2 in. lower layer coarse screened to 1½ in. gauge, and 1 in. upper layer fine screened to ¾ in. gauge, well rolled with an 8-ton roller	„	2	6
4 in. ditto, but 3 in. lower layer, and 1 in. upper layer, ditto	„	3	0
3 in. Macadam surface of 2 in. broken granite, rolled once dry with a 10-ton power roller, then well rolled whilst being watered by a water-cart having a rose spray pipe, and a surface binding of fine chippings, ⅛–¾ in. gauge, of the same material, to fill the interstices during subsequent rolling, until the whole is thoroughly consolidated (sand and road sweepings prohibited)	„	5	6
4 in. ditto, ditto	„	7	0
6 in. ditto, ditto	„	10	0

3 in. whinstone,	4*s.* 0*d.* per y.s.		3 in. hard limestone,	3*s.* 9*d.* per y.s.	
4 in. „	5*s.* 0*d.*	„	4 in. „ „	4*s.* 9*d.*	„
6 in. „	7*s.* 3*d.*	„	6 in. „ „	7*s.* 0*d.*	„
			3 in. blast-furnace slag,	3*s.* 0*d.* per y.s.	
			4 in. „	3*s.* 10*d.*	„
			6 in. „	5*s.* 9*d.*	„

		s.	d.
Spreading and levelling gravel or macadam before rolling, including wheeling up to 50 yds. from stacked heaps, not exceeding 3 in. thick.	„	0	3
Ditto, ditto, 4 in. thick	„	0	4
Ditto. ditto, 6 in. thick	„	0	5
Steam scarifying old gravelled or macadamized surfaces, not exc. 2 in. deep	„	0	3
Ditto, ditto, 2 in. to 4 in. deep	„	0	4
Hand-picking old gravelled or macadamized surfaces, not exc. 2 in. deep	„	0	5
Ditto, ditto, 2 in. to 4 in. deep	„	0	8
Add for removing up to 50 yds. and stacking for re-use, not exc. 2 in. deep	„	0	1
Ditto, ditto, 2 in. to 4 in. deep	„	0	2
Steam or power rolling with a 6- to 8-ton roller (including driver, attendance, fuel, water, oil, depreciation of machinery, etc.), gravel 3–4 in. thick, and 500–1,000 y.s. area, within a radius of one furlong	„	0	3
Ditto, ditto, but exceeding 1,000 y.s.	„	0	2
Ditto, with a 10-ton roller for macadam 3–4 in. thick, and 500–1,000 y.s. area, within a radius of one furlong	„	0	4
Ditto, ditto, but exceeding 1,000 y.s.	„	0	3
Hire of steam or power roller, including driver, fuel, water, oil, depreciation of machinery, etc., per working day	50*s.* to 60*s.*		
Ditto, ditto, per day standing by	20*s.* „ 25*s.*		
Watering roads, including horse, driver, van, and water, per 1,000 y.s.	5*d.* „ 9*d.*		
Ditto, ditto, but exclusive of water	2*d.* „ 4*d.*		

Complete Estimated Costs of Water-bound Macadam Roads, based on the foregoing, for three ordinary types, carriage-way only, and exclusive of drains, channels, kerbs, and footpaths. For cost of these to add see later section on 'Footpaths,' which vary in width and materials.

	Approximate. Per y.s. *s.*	Per Mile. £
By-Roads for light traffic, 20 ft. wide, including digging in ordinary ground, forming bottom to camber, removing spoil ¼ mile and depositing, 3-in. bed of furnace ashes, 6 in. bottoming of hard rubble stone, 3 in. whinstone metalling, 2 in. gauge, and a surface binding of fine whinstone chippings, ⅛–¾ in. gauge, watered and rolled, 12 in. total thickness	9	5,300
Ordinary Roads, or highways for moderate traffic, 30 ft. wide, digging, etc., as before, but 4 in. bed of ashes, 9 in. stone bottoming, 4 in. whinstone metalling, and surface chippings, 17 in. total thickness	12	10,600
Main Roads for heavy traffic, 40 ft. wide, digging, etc., as before, but 6 in. bed of ashes, 12 in. stone bottoming, 6 in. whinstone metalling, and surface chippings, 24 in. total thickness	17	20,000

Tar Macadam Roads.

Tar is used on roads for—Tar macadam (surface layer); grouted macadam (surfacing by penetration method); tar spraying (hot surface dressing); and tar emulsions (cold surface dressing and patching). *Rocmac*, *Tarmac*, *Tarvia*, etc., are proprietary articles. Digging, bed, bottoming, etc., will be found under previous sections.

Cost of Measured Work—

		s.	d.
3 in. Tar macadam of hard limestone, basalt, etc., in two layers: the lower of two parts of 2 in. gauge tarred stone as a sub-crust, and one part of 1½ in. gauge as a wearing surface, well rolled with a 10-ton power roller. During the subsequent rolling apply an upper or 1 in. binding layer of tarred stone, ¼–¾ in. gauge (equal in bulk to 10–15 per cent. of main layer), well rolled in to properly fill the interstices. Heat the coal-tar to about 270° F. before mixing with the perfectly dried and heated stone, allowing 9–12 gals. per cu. yd. of stone, and to comply with the Ministry of Transport Specification No. 3 (*b*) for Standard Tar No. 2. (After one to two months' traffic the surface to be tar sprayed and blinded as a sealing coat, separate payment for this being made)	per y.s.	6	0
4 in. ditto, ditto	,,	7	6
5 in. ditto, ditto	,,	9	0

If blast-furnace slag is used instead of limestone deduct—

3*d.* for 3 in. thick; 4*d.* for 4 in.; 5*d.* for 5 in., per y.s.

Tar spraying first time (1 gal. to 5–7 y.s.) with No. 1 standard dehydrated coal-tar under pressure, heated on the spot to about 270° F., including preliminary sweeping and afterwards blinding with ¼ in. fine pea gravel, or limestone or granite chippings, free from dust, and not mixed with tar, and light rolling .	,,	0	6
Ditto, if sprayed previously (1 gal. to 8–10 y.s.), ditto, ditto	,,	0	5
Resurfacing macadam in repairs by roughing and sweeping surfaces of tar or ordinary macadam, applying one coat of No. 2 Standard Tar, heated to boiling point on the spot, and whilst surface is hot rolling in tar macadam, ¼–¾ in. gauge, averaging 1 in. thick. Finished surface to be then tar sprayed (1 gal. to 5–7 y.s.), and blinded with fine limestone siftings during rolling	,,	3	6
itto in 'pot-holes' and small patches, 5–25 y.s. in the aggregate at one time .	,,	4	0

Complete Estimated Costs of Tar Macadam Roads, based on the foregoing, for three ordinary types, carriage-way only, and exclusive of drains, channels, kerbs, and footpaths. For cost of these to add see later section on 'Footpaths,' which vary in width and materials.

	Approximate. Per y.s.	Per Mile.
	s.	£
By-Roads for light traffic, 20 ft. wide, including digging in ordinary ground, forming bottom to camber, removing spoil ½ mile and depositing, 3 in. bed of furnace ashes, 6 in. bottoming of hard rubble stone, 3 in. tar macadam of hard limestone, and an upper or binding layer of tarred stone, ¼–¾ in. gauge, to fill the interstices, all well rolled, 12 in. total thickness (including tar spraying and blinding after one to two months' traffic) . . .	11	6,400
Ordinary Roads, or highways for moderate traffic, 30 ft. wide, digging, etc., as before, but 4 in. bed of ashes, 9 in. stone bottoming, 4 in. tar macadam, and binding layer of small tarred stone, 17 in. total thickness (including tar spraying and blinding after one to two months' traffic) . . .	14	12,300
Main Roads for heavy traffic, 40 ft. wide, digging, etc., as before, but 6 in. bed of ashes, 12 in. stone bottoming, 5 in. tar macadam, and binding layer of small tarred stone, 23 in. total thickness (including tar spraying and blinding after one to two months' traffic)	19	22,300

Reinforced Concrete Roads.

Reinforced concrete should be—6 in. thick for residential areas and light traffic.
7 ,, ,, ordinary roads and streets.
9 ,, ,, main roads heavily trafficked.
12 ,, ,, very heavy city traffic.

Digging, bed, bottoming, etc., will be found under previous sections.

Two-course concrete, 6 in. total combined thickness, including preparing, wetting, and rolling formation surface to proper gradient and cross-section with a light 2- to 5-ton roller, placing a 4½ in. under layer of British Portland cement concrete, 1 : 2 : 4, of 1½ in. gauge limestone, reinforced 1½ in. from bottom surface with No. 9 B.R.C. steel fabric (4·71 lbs. per y.s., and having 2 in. side lap and 12 in. end lap) and a 1½ in. upper layer of hard-wearing fine granitic concrete, 1 : 1½ : 3, of ½ in. gauge granite chippings, screened free from dust. Each concrete layer to be well tamped, and the upper to follow the lower as rapidly as possible, and in *no* case more than 20 minutes later, before the latter has had time to set, so that the two may firmly unite as one mass. The final top surface to be screeded off and finished with a canvas band to remove excess moisture, then covered with 2 in. of sand or loam kept well wetted for 21 days till the concrete

	s.	d.
is cured and properly set. (Ordinary Portland cement hardens in 7 days, Ferrocrete in 3 days, and Ciment Fondu in 1 day.) Camber, 1 in 40 to 1 in 50 . per y.s.	12	6
Add for each additional inch of concrete in 1 : 2 : 4 under layer, with 1½ in. gauge limestone (up to 9 in. combined thickness) . . . . . ,,	1	3
Solution of Brunner Mond's P. 84 silicate of soda, sprayed and brushed in 3 separate coats, for hardening and dust-proofing the surface of concrete roads (1 gal. tin, 6s.), in the proportion of 1 gal. silicate to 4 gals. water . . . ,,	0	1½

Complete Estimated Costs of Reinforced Concrete Roads, based on the foregoing, carriage-way only, and exclusive of drains, kerbs, and footpaths. For cost of these to add see later section on 'Footpaths,' which vary in width and materials. The carriage-way should be extended 12–18 in. under the kerbs.

	Approximate. Per y.s. s.	Per Mile. £
By-Roads for light traffic, 20 ft. wide, including digging in ordinary ground, forming bottom to camber, removing spoil ¼ mile and depositing, 3 in. bed of furnace ashes or clinker and rolled (if the formation surface is clay), and 6 in. total thickness of reinforced concrete (4½ in. lower layer, 1 : 2 : 4, of 1½ in. gauge limestone, and 1½ in. upper layer, 1 : 1½ : 3, of ½ in. gauge granite chippings—sometimes 4 in. lower, and 2 in. upper) . . .	15	8,800
Ordinary Roads, or highways for moderate traffic, 30 ft. wide, digging, etc., as before, but 4 in. bed, and 7 in. total thickness of reinforced concrete (5 in. lower, and 2 in. upper)	17	15,000
Main Roads for heavy traffic, 40 ft. wide, digging, etc., as before, but 6 in. bed, and 8 in. total thickness of reinforced concrete (6 in. lower, and 2 in. upper)	19	22,300
Arterial Roads have 8–9 in. reinforced concrete carriage-ways 30–70 ft. wide, on 4–12 in. bed of clinker, hard core, or furnace ashes, with grass margins, and footways 10–15 ft. wide, or 100–120 ft. total width between fences (for very heavy traffic double reinforcement is sometimes used, weighty steel at bottom and light at top, 5–7 lbs. per y.s., and 2 lbs. per y.s.), per mile	£ 50,000	£ to 100,000

Actual Costs of Reinforced Concrete Roads.—The Greater London Arterial Road Scheme (1920–27) comprises new wide reinforced concrete by-pass roads surrounding London, to enable traffic to go round the city and avoid congesting busy areas. These include the North Circular Road, the South Circular Road, the Eastern Avenue, the Western Avenue, the Great West Road, the Great Cambridge Road, and others. Some stretches may be seen at Edmonton, Enfield, Southwark, and Willesden. They are 100–120 ft. wide between fences, but a carriage-way about 24 ft. wide is only done first, the remainder as funds permit. Other towns are constructing similar great highways.

North Circular Road (1922–23).—Planned to form a complete circuit from Kew Bridge over the Thames on the south, passing round London at a radius of 6½ miles from Charing Cross into Essex at Walthamstow, and from thence to the Thames Ferry at Woolwich, and also connecting with the roads to the Eastern Counties. Length about 17 miles, of which 6 miles are existing roads widened and reconstructed, and 11 miles of new road, including bridges over 12 railways and many streams. New road built in three lengths, 100 ft. wide, with two 24 ft. reinforced concrete carriage-ways, 20 ft. in the middle for tramways, and two 16 ft. footpaths. Construction varied, but at Willesden it was 4–6 in. clinker bed, 5 in. lower course of concrete, 1 : 2 : 4 and 1 : 2½ : 5, of 2 in. gauge ballast graded down to ⅜ in., reinforced with B.R.C. or Walker-Weston reinforcement, 12 lbs. per y.s., placed 2 in. from surface, 3 in. top course of concrete, 1 : 1 : 3, of crushed river ballast, ⅜–$\frac{3}{4}$ in. gauge, deposited immediately to form one homogeneous mass of 8 in. total thickness. Concrete was machine-mixed, and no expansion joints, but only 40 ft. whole width lengths were laid at one time to avoid cracks. Camber 1 in 48, surface treatment tar and grit. 12 × 6 in. reinforced concrete kerbs, with carriage-way extended beneath. Direct labour. Cost, excluding excavation, 11*s.* 4*d.* to 12*s.* 10*d.* per y.s. Whole work, about 1,200,000*l.*, or 71,000*l.* per mile.

Western Avenue (1922–23).—Leaves London at Shepherd's Bush, passes through Acton and Greenford, and rejoins the present Oxford Road in Bucks at Denham. It has 5 bridges over railways, one over the river Brent, and one over an industrial siding. Length 12 miles, with two 24 ft. carriage-ways and an intervening 20 ft. space for future developments, but only one carriage-way was made first. From Acton to Greenford 30 ft. central carriage-way, tar-painted gravel footways, and grass margins with trees. Subsoil very wet clay throughout the whole length; little cutting or filling, except filling at approaches to bridges. Carriage-way of 8 in. reinforced concrete, granitic faced, laid in alternate bays. On deep fills hard core, and tar macadam surfacing. Footways generally of gravel tar-painted, with 12 × 6 in. granite kerbs. Surface-water drains in one margin only, with cross drains connecting to gullies on both sides of carriage-way. Estimate for Middlesex section of 12 miles was 666,000*l.*, or 55,500*l.* per mile. 4½ miles cost 285,000*l.*, or 60,000*l.* per mile.

Great West Road (1919–24).—In Middlesex, from Chiswick High Road to the main Bath and West of England Road, near Hounslow Heath. Length 8½ miles, but only 5½ miles done up to 1924. Total width between fences 120 ft., the carriage-way at first only 50 ft. (but capable of being widened to 80 ft.), with two grass margins 25 ft. wide, trees 60 ft. apart, and two 10-ft. footways. Parts of grass strip available for future widening of footways. Subsoil generally of brick earth overlying sandy gravel. Foundation of 12 in. hard core and ashes, overlaid with 9 in. reinforced concrete, 6 in. lower course, 1 : 2½ : 5 of washed gravel, 1½ in. gauge graded down to ⅝ in., with B.R.C. fabric 2 in. from bottom, and 3 in. upper course, covered with 2 in. rock asphalt treated with bituminous dressing and granite chips. Footpaths of gravel, tar-painted, but slab paving in built-up areas, with 12 × 8 in. granite kerbing laid on the extended sides of the reinforced concrete road-bed, and concrete channels. The work included 4 new bridges, 80 ft. wide, one bridge widening, sewers and surface water drains on both sides, subways for cross mains, one subway being 33 ft. wide to take one 42 in. and two 66 in. mains, costing 11,730*l.*, the deepest cut being 18 ft. and the heaviest fills 17 ft., demolition of 60 houses and building new ones instead, etc. Concrete laid in alternate bays with butt joints, part having expansion joints. Gross expenditure 834,482*l.*, of which construction costs, including duplicate sewers and surface water drains, represented 670,152*l.*, or 130,760*l.* per mile.

Great West Road Extension (1922–23).—Links up the Brentford By-pass with the Basingstoke Road. Length, 3 miles. Highway 100 ft. wide, and 30 ft. central carriage-way with grass margins, trees 60 ft. apart, and two gravel footways tar-painted, with 12 × 6 in. granite kerb. Subsoil mainly gravel and old pits filled up. 6 in. clinker bed well rolled, 8 in. concrete carriage-way, 1 : 2 : 4, aggregate of washed pit ballast, 2 in. gauge down to ½ in., and some of ¾ in. shingle, machine-mixed, and reinforced with steel fabric. Surface partly asphalt carpet, and partly tarred. Half a mile laid without expansion joints, the remainder in alternate bays. Surface-water drains on both sides. Work included a ferro-concrete bridge, 60 ft. wide and 30 ft. span, and two smaller ones. Total, 124,000*l.*, or 41,330*l.* per mile.

Great Cambridge Road (1921–23).—An entirely new road from near Finsbury Park running through Tottenham, Edmonton, and Enfield to join the Hertford Road at Wormley. Length, 7 miles. Total width between fences 100 ft., with one prepared carriage-way 24 ft., but actually planned for two with an intervening space of 20 ft. for future tramways, etc., and trees planted 60 ft. apart. Two 6-ft. footpaths on either side adjoining the boundaries, gravelled and tar-painted, with 12 × 8 in. and 12 × 6 in. concrete and granite kerbs. Subsoil mostly brick earth overlying gravel, and southern mile of very bad clay. An old gravel pit, 30 ft. deep, on line of road, was filled up. 3–6 in. clinker bed, 8 in. reinforced concrete carriage-way, 1 : 2 : 4, of ballast and shingle, 2 in. gauge graded down to ¼ in., and cambered 1 in 48. Surface-water drains on both sides. Scheme comprised three ferro-concrete bridges over rivers, and over a railway. Total, 330,000*l.*, or 47,140*l.* per mile. First sections cost 13*s.* per y.s., but later sections only 10*s.* 6*d.* per y.s. (June 1922). A continuation of 3¼ miles by contract came to 12*s.* 4*d.* per y.s., excluding excavation (August 1923).

Central Drive, Westhoughton, Lancashire (1922).—120 yds. long and 25 ft. wide. Foundation of 4 in. furnace ashes rolled. Concrete in two courses—4 in. lower, 1 : 2 : 4 local stone, with fabric reinforcement 2 in. from bottom, and 2 in. upper, 1 : 1½ : 3 granite chippings, with tamped surface finish. Camber, 1 in 40. No expansion joints, concrete cured with damp sand, allowing three weeks to set. Cost, 12*s.* 9*d.* per y.s.

Ipswich Housing Estate (1922).—Total length of roads, 1,890 yds.; average width between kerbs, 21 ft. Concrete in two courses, totalling 5–6½ in. Lower course, 1 : 2 : 4 sea shingle, reinforced 1 in. from bottom, and upper course, 1 : 2 : 2 granite chips, rammed to form a granolithic surface finish. Camber, ¼–1 in. No expansion joints, concrete cured with wet sand. Direct labour. Cost, 13*s.* 4*d.* per y.s.

South Cliff, Hartlepool (1922).—New concrete road 200 yds. long and 24 ft. wide between kerbs, in one 6 in. course only. Excavation steam rolled and properly shaped, with bed of fine destructor ashes to receive the concrete, 1 : 1⅓ : 3½ slag, reinforced 2 in. from bottom, and machine-mixed. Work done in full widths, surface finished with screed or straight edge. Camber, 1 in 40. No cross expansion joints, but a longitudinal expansion joint of 1½ in. fine tarred slag alongside each kerb. Direct labour. Cost, 10*s.* per y.s.

Streets in Sheffield (1922).—31 streets reconstructed in 6-in. concrete from existing macadam. Combined length, 3,411 yds.; average width between kerbs, 19 ft. Two courses: 5 in. bottom, 1 : 5 broken gritstone, and 1 in. top, 1 : 2½, surface tamped. No reinforcement except over trenches. Concrete machine-mixed, and work in whole widths. Crossfall, 3 in. No expansion joints, and 7 days before traffic allowed. Direct labour. Cost, 16*s.* 6*d.* per y.s.

Streets in Southwark (1922–23).—Original granite macadam roadway excavated 9 in. deep, and rolled to receive 6 in. two-course concrete—4 in. lower, 1 : 6, with ⅜ in. interlaced rod reinforcement 2 in. from bottom, and 2 in. upper, 1 : 3, laid quickly before the lower course had time to set. Aggregate was the material from the old macadam road. Concrete cured with 2 in. damp sand kept wet at least 10 days, and no traffic allowed for three weeks. Surface lightly treated with tar or silicate of soda. Cost, including excavating, watching, materials, and labour, 10*s.* to 14*s.* 6*d.* per y.s.

Sett Paving.

Sometimes called pitcher paving. Digging, and concrete or rubble stone foundation, will be found under previous sections.

		s.	d.
4 × 4 × 5 in. deep, squared and hammer-dressed granite paving cube setts, joints squared 3 in. back from faces, bedded in screened gravel or hard screened clinker, and grouted flush with cement mortar, 1 to 3 (exclusive of digging and foundation)	per y.s.	24	0
8–9 in. long × 3 in. wide × 5 in. deep, ditto, ditto	„	26	0
Add or deduct for each inch of greater or less depth	„	4	0
Deduct if grouted in pitch mixture instead of cement mortar	„	0	3
Add if sett paving is laid in straight gutters or channels separate or detached from other paving	„		
Add if sett paving is laid over concrete foundation (excluding same), including bedding in cement mortar, 1 to 3, and grouting flush	„		
Taking up setts 5 in. deep, cleaning, and relaying, including gravel bed and grouting flush with cement mortar, 1 to 3	„	5	0
Complete Estimated Cost, based on foregoing, for 4 × 4 × 5 in. granite setts, grouted and bedded in cement mortar, 1 to 3, on and including 9 in. concrete foundation, floating to an even surface, and digging in ordinary soil	„	36	0
8–9 in. × 3 in. × 5 in. granite setts, ditto, ditto	„	38	0

Wood Block Paving.

Digging and concrete foundation will be found under previous sections.

		s.	d.
Wood block paving of selected Baltic fir or yellow deal, 9 × 3 × 4 in. deep, creosoted in vacuum with 8–10 lbs. creosote oil per cu. ft. of timber, the lower ends of blocks dipped in pitch mixture whilst being laid close together end grain up, then grouted by squeegee with the same mixture, a $\frac{1}{16}$ in. layer being left on the top surface and blinded with fine pea gravel. Mixture is 1 cwt. coal-tar pitch, as described in British Standard Specification No. 76, slightly thinned with 1 gal. creosote oil and heated to 300° F., but cooled to 270° F. at time of dipping and grouting. Cross fall, 1 in 40. (Price is exclusive of digging and concrete foundation)	per y.s.	14	0
Ditto, 9 × 3 × 5 in. deep, ditto, ditto	„	16	0
Taking up old wood blocks, 3–5 in. deep, removing 20 yds. and stacking	„	0	9
Ditto, including cleaning and trimming for relaying	„	1	6
Puddled clay expansion joints, 1 in. wide, at edge of road	per f.r.	0	1½
Pitch and tar ditto, ¾ in. wide	„	0	2
Cutting and waste to splays and rounded angles	„	0	6
Cutting only to wood blocks	„	0	2
Complete Estimated Cost, based on foregoing, for 9 × 3 × 4 in. wood blocks, dipped and grouted with pitch mixture, and surface blinded with fine gravel, including 9 in. concrete foundation, floating to a fair surface, and digging in ordinary soil	per y.s.	25	0
9 × 3 × 5 in. deep, ditto ditto	„	27	0

Asphalt.

1–1½ in. thick for pavements and footpaths; 1½–2 in. for roads and carriage-ways. Prices laid in London, by the Asphalt Company's own workmen. Digging and concrete foundation will be found under previous sections.

		s.	d.
1 in. Asphalt paving in two ½-in. layers, quantities not less than 100 y.s.	per y.s.	7	6
1½ in. ditto, in 1 in. and ½ in. layers, ditto	„	10	3
2 in. ditto, in 1 in. and 1 in. layers, ditto	„	12	9
Taking up old asphalt, any thickness, and removing	„	0	8
Labour only in forming channels and gutters, 6–9 in. girth	per f.r.	0	7
Complete Estimated Cost, based on foregoing, for 1 in. asphalt on footpaths, including 3 in. concrete foundation, floating to a fair surface, and digging in ordinary soil	per y.s.	12	0
2 in. Asphalt on roads, including 9 in. concrete foundation, floating, and digging	„	24	0

Footpaths.

The following prices are exclusive of digging or foundations, which latter may be of 3–4 in. furnace ashes, clinker, rubble stone bottoming, hard core, concrete, etc., according to the support required by the paving material; see previous sections.

		s.	d.
2 in. Gravel, good binding, coarse screened, with ¾ in. fine screened gravel on top, rolled down to a finished thickness with a 30-cwt. roller	per y.s.	2	0
3 in. ditto, in two layers: 2 in. coarse and 1 in. fine, rolled with 2-ton roller	„	2	4
2 in. Tar paving in one main layer of 1½ in. gauge tarred limestone, rolled with a 2-ton roller, then a binding coat of ¼–¾ in. tarred stone to fill the interstices, after which final rolling with a 30-cwt. roller	„	4	6
3 in. ditto in two layers: the lower 2 in. of 1½ in. gauge tarred stone well rolled before the upper 1 in. of ¼–3 in. tarred stone is applied and also rolled. Finished surface is then tar sprayed under pressure (1 gal. to 5–6 y.s.), and blinded with fine limestone siftings during final rolling	„	6	0

If blast-furnace slag is used instead of limestone deduct—

2*d.* for 2 in. thick; 3*d.* for 3 in. thick, per y.s.

		s.	d.
Vitrified blue Staffordshire pressed paving bricks, scored diagonally on face, 10 × 5 × 1½ in. thick, laid flat and straight on 1 in. layer of sand, or furnace clinker screened to ½-in. mesh, and grouted with cement	„	9	0
Ditto, ditto, laid and jointed in cement mortar	„	10	6
1 in. Stuart's granolithic paving laid in London area.	„	4	0
2 in. ditto, ditto	„	6	6
2 in. Yorkshire or Caithness smooth stone flagging, not less than 4 f.s. area each, edges squared and jointed, laid in lime mortar and jointed with cement mortar, 1 to 3, including forming the ground to required levels and removing surplus earth	„	19	6
2½ in. ditto, ditto	„	23	0
3 in. ditto, ditto	„	26	6
2 in. Pre-cast granitic concrete flagging, 1 cement to 3 granite chippings and fine stuff, laid as last	„	13	0
2½ in. ditto, ditto	„	15	6
Add to last five items if laid in cement mortar instead of lime mortar	„	0	6
1 in. Granitic surface, 1 cement to 2 granite chippings, incorporated whilst concrete is unset, and trowelled to a smooth face	„	4	0
Spike rolling, or flute rolling, surfaces of new concrete	„	0	4

Channel stones, not less than 3 ft. long, finely quarry scabbled upon upper surface, and on side and end joints 3 in. from face, including setting complete in lime mortar:

Size of Channel Stones.	Granite.	Whinstone.	Limestone.	Pre-cast Granitic Concrete, 1 : 2 : 4.	
	s. d.	s. d.	s. d.	s. d.	
10 × 4 in. section, per ft. run	4 0	3 4	3 2	2 2	
12 × 5 in. „ „	5 6	4 8	4 2	2 9	s. d.
Add to any of foregoing if set and jointed in cement mortar instead of lime mortar					per f.r. 0 1

Kerb stones for footpaths, not less than 3 ft. long, finely quarry scabbled on top, 6 in. one side, and on end joints and remaining side joint for 3 in. from face, including setting complete in lime mortar:

Size of Kerb Stones.	Granite.	Whinstone.	Limestone.	Pre-cast Granitic Concrete, 1 : 2 : 4.	
	s. d.	s. d.	s. d.	s. d.	
5 × 10 in. edge, per ft. run	4 6	3 8	3 5	2 4	
6 × 12 in. „ „	6 0	4 10	4 4	3 2	
10 × 6 in. flat „	5 6	4 6	4 3	2 10	
12 × 6 in. „ „	6 6	5 4	4 9	3 3	s. d.
Add to any of foregoing if set and jointed in cement mortar instead of lime mortar					per f.r. 0 1

(See also 'Highway Engineering,' Section XV.)

Roofs.

	£	£	
Roofs, light, corrugated iron, 18 to 22 gauge, exclusive of iron or steel trusses		2 to 3	per square.
Roofs for light cheap sheds and buildings, including trusses, corrugated sheeting, gutters, etc., fixed complete		5 ,, 8	,,
Roofs, ordinary steel and iron, including purlins, boarding, gutters, etc.		5 ,, 10	,,
Roofs, railway station platform, with glass		10 ,, 20	,,
,, ,, termini, over 150 ft. span, fixed complete		30 ,, 45	,,
Roof trusses, delivered only, for galv. iron		4*s.*	per foot of span.
,, ,, ,, for slates		8*s.*	,, ,,

Railway Station Roofs.

Railway and Station. (Old titles of Railways.)	Span of Roof.	Covered Area.	Cost per Square of 100 F. S.	Additional for one Gable.
	Feet.	Sq. Yds.	£ *s.* *d.*	£ *s.* *d.*
London, Brighton and South Coast Railway, Victoria Station (1860)	120	—	27 13 4	with gable
London & North-Western Railway, New Lime Street Station	212	—	30 0 0	—
Midland Railway, St. Pancras Station (1868)	240	18,822	31 11 0	4 7 0
South-Eastern Railway, Cannon Street Station (1867)	190	13,875	43 10 0	6 0 0
South-Eastern Raily., Charing Cross Station (1864)	166	8,888	34 0 0	6 0 0

Victoria Station and Charing Cross Station have since been entirely rebuilt.

Sea Cables.

The cost of a light submarine cable laid on the ocean bed is 200*l.* to 400*l.* per nautical mile laid (1928). Heavy trunk lines, 400*l.* to 800*l.* ditto. One nautical mile = 2,027 yds.

Seaside Resorts.

Based on actual costs, 1927.

Bathing Pools, open, including accommodation for spectators, require a capital outlay of about 1*l.* per head of population, or 20*l.* to 30*l.* per square foot of water area.

Bathing châlets, with gas and water laid on, frequently fetch a rental of 35*l.* to 50*l.* a year.

Foreshore Lay-out is commonly 2*l.* per head of population, and in one case was 17*l.* per head.

Groyning, to form and maintain beaches, may cost 15,000*l.* per mile of beach.

Hard Tennis Courts, red or green, 6*s.* to 7*s.* per square yard of surface.

Sea-Walls.

Sea-walls to protect coast line, formed of 1 to 6 Portland cement concrete, 10 ft. high from base to top, 4 ft. average thickness, including foundations and excavating in clay, in tide-work, pre-war average cost—

6*l.* per yd. run, or 2*l.* per ft. run.

Pre-war actual costs from *Marine Works*, by E. Latham, M.I.C.E. (1922. 16*s.* Crosby Lockwood & Son, London.)

	Cost per yd. run. £ *s.* *d.*
Bridlington.—Alexandra sea-walls (1879). 340 yards long, of mass concrete 5 ft. thick at base and 3 ft. at top, curved on face. Concrete apron, 9 ft. wide, and 2 to 2½ ft. thick, the toe-piece being 6 ft. deep and 18 ins. thick. Back of wall is battered 1½ in. per foot. Counterforts at 20 ft. centres. Wall is 20 ft. high, with projecting cornice, coping, and handrail. Chains embedded to act as bonds for the concrete. For details see *Coast Erosion and Protection*, pp. 49–50, by E. R. Matthews, A.M.I.C.E. (1918. Charles Griffin & Co., London); and *Proceedings of the Institution of Civil Engineers*, vol. clix. p. 65. Total, £5,657	16 12 9
Bridlington.—Beconsfield sea-wall (1888). 340 yards long, with Bramley Fall rock-faced stone and concrete backing. 1¾ in. wrought-iron tie-rods at intervals. 12 ft. 3 ins. thick at base, and 4 ft. at top, stepped and curved on face. Total height, 25 ft. 7 ins., with projectng cornice, coping, and handrail. Counterforts, 3 ft. 6 ins. by 3 ft. 6 ins, and 20 ft. apart, centre to centre. For details see *Coast Erosion and Protection*, pp. 47-48, by E. R. Matthews, AM.I.C.E. (1918. Charles Griffin & Co., London.) Total, £10,000	29 8 3

Cost per yd. run.

	£ s. d.
Bridlington.—Sewerby sea-defence (1896). 305 yards long. Combination of rubble and timber, with two rows of 12 in. × 12 in. pitch-pine piles, 10 ft. apart. Front piles are 15 ft. long driven 9 ft. into the clay, and back piles 23 ft., driven 10 ft., with tie piece, 9 ins. × 4 ins. Face of 11 in. × 3 in. planking, and back planking 11 ins. × 4 ins., with 9 in. × 6 in. bearers and 11 in. × 4 in. ribs. Puddle clay backing, 2 ft. 6 ins. thick. Block chalk filling between the timber work. Foot-path on top. For details see *Coast Erosion and Protection*, pp. 86–88, by E. R. Matthews, A.M.I.C.E. (1918. Charles Griffin & Co., London); and *Proceedings of the Institution of Civil Engineers*, vol. clix. p. 65. Total, £3,000 . . .	9 16 8
Extension of ditto in 1899, by 82 yards, £788	9 12 2
Chatham (1913).—Reinforced concrete revetment on the De Muralt system on the banks of the Medway. Protects a length of 90 yds. of bank between two wharves	7 15 5
Frinton (1912–13).—Reinforced concrete apron 200 yds. long, to protect a stone pitched sea-wall at Frinton in Essex, which was being undermined by the waves. Apron 12 ft. wide, sloping 1 in 12, and 4 ins. thick, reinforced with Clinton fabric and ½ in. steel bars. Front toe wall, 9 ins. thick and 2 ft. high, with ½ in. steel bar reinforcement. 6 in. × 3 in. reinforced concrete ribs, with ½ in. steel bars .	2 5 0
Hornsea.—Reinforced concrete sea-wall. For details see *Proceedings of the Institution of Civil Engineers*, vol. cxxxv. p. 151	39 0 0
Lowestoft (1901–1903).—North Denes. Mass concrete sea-wall on steel piles, including steps, filling, and excavation. Total length with returns, 5,744 ft. run. Cost per yd. cube, 21*s.* 2*d.* For details see *Proceedings of the Institution of Civil Engineers*, vol. clxxxv. p. 106	5 11 6
Maintenance.—Maintenance of earthen embankments along river fronts or sea shores, about 18 ft. high to top of wall and protected by stone pitching or light reinforced concrete revetment; average cost of maintenance per annum (1910–13) when in normal repair	4*d.* to 4½*d.*

SEWAGE WORKS.

Sewage disposal works for cities and towns . . .	3*l.* to 6*l.* per inhabitant.
Bacteria filter-beds with earth embankments and floors, completed with clay puddle, and filled in with clinker or coke breeze 3 ft. deep	5*s.* to 10*s.* per yd. sup.
Ditto with concrete walls, and clinker or coke breeze 4 ft. deep	20*s.* to 30*s.* ,,

Precipitation and sludge tanks in the north-west of England, to treat a million gallons of sewage daily, built in 1910. From actual tenders received.

	£
Scheme with cast-iron tank plates	7,000
,, ,, brickwork in cement	6,000
,, ,, reinforced concrete	4,500

Bio-aëration System.—Complete works on the bio-aëration system, including aëration or circulating tanks, settling tanks of the Dortmund type, gauging chambers, sludge wells, motor houses, pump houses, etc., cost from 25,000*l.* to 30,000*l.* per million gallons, d.w.f., according to size and local circumstances (1924).

Activated Sludge Process.—The estimated cost for a complete activated sludge scheme, dealing with all storm water and treating all the sludge, is 1*l.* per head (1924).

Reading.—Activated sludge plant on the diffused air method, completed in 1924, designed to treat the whole flow of the city, amounting to 3½ million gallons d.w.f. per 24 hours, from an estimated future population of 120,000, cost 72,813*l.*, or 12*s.* 1½*d.* per head.

Strensall Camp Sewage Works, near York (1914–16).—For 1,000 nominal number of troops at 20 gals. per head per day, or 20,000 gals. sewage per day. Lift system, ground being low and sandy. Drainage connections comprised replacing 86 e.c.'s by w.c.'s, 18 additional w.c.'s, or 104 total w.c.'s, 234 gullies, and 80 manholes, with 4-in. and 6-in. stoneware drains gravitating to the ejectors. Brick engine shed, 15 ft. by 14 ft., for air compressor plant with 2 gas engines, and 4 ejector pits, with 12-in. concrete walls lined with ⅜-in. cast-iron plates, and concrete floors with expanded metal; pits octagonal in plan, 12 ft. internal diameter and 12 ft. average depth, containing 5 pneumatic ejectors, 25 gallons each, 1 pit holding 2 ejectors. 2½ in. steel compressed air mains from engine-house to ejectors, which raise sewage to the 5-in. and 6-in. cast-iron rising main drains and thence 1,700 yards (about a mile) to the outfall works. Total lift for sewage, 19 feet. Outfall works include two reinforced 6-in. concrete tanks, 20 ft. × 10 ft. × 6 ft. each, containing 7,500 gallons, or 15,000 gallons altogether; two brick sludge beds, 20 ft. × 10 ft. × 3 ft. each, with 10-in. screened ashes at bottom and 4-in. agricultural drains on 6-in. concrete floor; two circular filter beds, 46 ft. diameter, with rubble sides and reinforced concrete floors

filled with 5-ft. furnace clinker, ½-in. to 4-in. gauge, the liquid sewage being distributed by a revolving sprinkler delivering 84,000 gallons per day. Special filter tiles on floor for aeration, concrete surface channel all round, and effluent then taken by waterways for land irrigation prior to discharging into existing ditch. Cost under five separate contracts :—

	£
Drainage connections, w.c.'s, stoneware drains, and manholes	2,710
Brick engine shed, and four concrete and iron ejector pits	775
Five ejectors, compressors, two gas engines, 2½-in. steel air mains, and 5-in. and 6-in. cast-iron main drains	1,837
Outfall works, including tool house, wire fence, and entrance road	2,035
Preparation of 13 acres of land, waterways and ditches for effluent	493
Total cost	7,850

Cost per head, 7*l.* 17*s.*, or 392*l.* 10*s.* per 1,000 gals. per day treated.

Stands.

Permanent stands for athletic or race meetings, displays, processions, reviews, etc., constructed of brick, concrete, or reinforced concrete, with refreshment rooms, kitchens, offices, weighing and saddling rooms, lavatories, drainage, water supply, electric light, seats, etc., roofed or partly roofed—

2*l.* to 3*l.* per spectator, or head of accommodation.

Imperial Stadium, Wembley, near London (1920–1923).—Major axis, overall length, 900 ft. and maximum overall width 650 ft. Oblong of grass for football and sports in centre 492 ft. × 260 ft., with cinder running tracks, encircled by a wide stepped slope for spectators beyond which begins the structural work with terraces of concrete steps, roofed in for 650 ft. on each side of the stadium and provided with seats. Total accommodation, including seated grand stands, ring seats, standing enclosures and terraces, 125,200 spectators. A space of 16 ins. is allotted per ring seat, and 17½ ins. for a grand stand seat. Royal box, banqueting hall, 215 ft. × 70 ft., for 1,000 people, dressing-rooms, shower baths, lavatories, board-room, box office, gymnasium with seats for 250 spectators, recreation room with billiard tables and reading tables, buffets, cloak-rooms, tea-rooms for 600 people at a time, observation gallery, offices, press gallery, special heating, 96 turnstiles, telephones, electric lighting and cooking, wireless, good water supply and drainage, etc. The whole structure is of reinforced concrete on a steel skeleton, and there are 25,000 tons of concrete, 1,400 tons of structural steelwork, 31 terraces of concrete steps on steel joists, girders, and stanchions, and steel-framed roofs covered with corrugated asbestos sheets. Height from ground to roof eaves of stand is 50 ft. Two towers 160 ft. high, flank the royal entrance, and high walls on main frontage, with arched openings with an average span of 50 ft. The whole structure is larger than the Coliseum, Rome, which held 80,000 people. Architects, Sir John W. Simpson, F.R.I.B.A., and Mr. Maxwell Ayrton, F.R.I.B.A. Engineer, Sir Evan O. Williams, B.Sc., A.M.I.C.E. General contractors, Sir Robert McAlpine & Sons, and numerous sub-contractors. Total cost, 300,000*l.*, or 2*l.* 8*s.* per spectator.

Submarine Cables, see ' Sea Cables,' p. 2882.

Tanks.

Estimated Costs, 1930.

	Per Gal. of Capacity.	Per Cu. Ft. of Capacity.
	d. *d.*	*s.* *s.*
Underground concrete tanks, 5,000 to 10,000 gals, capacity, for storage of rainwater to barracks, laundries, mansions, public institutions, etc., complete with concrete bottom, walls, and roof, rendered in cement with smooth trowelled face, including digging in ordinary ground, clay, or gravel, and removing surplus soil	4 to 6	2 to 3
Cast-iron tanks above ground, rectangular or octagon, 5,000 to 10,000 gals. capacity, with cast-iron flanged plates bolted together to form bottom and sides, and painted, supported on brick or concrete base	5 to 7	2½ to 3½
Steel tanks, rectangular, 5,000 to 10,000 gals. capacity, with steel plates riveted together to form bottom and sides, painted, and fixed in position	6 to 8	3 to 4
Steel cylindrical tanks, circular on plan, 40,000 to 80,000 gals. capacity, comprising steel plates riveted together to form bottom, sides, and top, painted, and fixed in position	4 to 6	2 to 3

TELEPHONES.

Estimated Costs, 1930.

		£	£
Private telephones for indoor use in houses, offices, and public buildings, complete, including batteries, wire, staples, terminal instruments, etc.	each	20 to 25	
Add for every extra telephone connection, including indicator, switches, etc.	,,	8	,, 10
Overhead land wires, comprising one copper line weighing 150 lbs. per mile, with poles, stays, brackets, and insulators, but excluding terminal batteries and instruments, fixed complete	per mile	60	,, 70
Add for every additional wire and fixing	,,	25	,, 30
Extra for terminal batteries and instruments complete	per telephone line	20	,, 25
Construction of municipal telephone systems in large towns and cities, including all wires, poles, batteries, and instruments, complete,	per line or subscriber	40	,, 60

TRAMWAYS.

Approximate Cost per Mile (1930).	Single Track. £ £	Double Track. £ £
Electric tramways, overhead trolley system, with span wires, including buildings and equipment, but exclusive of generating station for electricity	15,000 to 25,000	30,000 to 40,000
Ditto, ditto, with side brackets or centre poles	15,000 ,, 20,000	28,000 ,, 36,000
Overhead equipment, including supports :—		
Span wire construction	2,500 ,, 3,000	3,000 ,, 4,000
Bracket arm construction	1,800 ,, 2,400	2,500 ,, 3,500
Permanent way, granite setts :—		
3 ft. 6 in. gauge, 100 lb. rails	8,000 ,, 10,000	15,000 ,, 18,000
4 ft. 8½ in. gauge, 100 lb. rails	10,000 ,, 12,000	17,000 ,, 20,000
Conduit system, centre slot, with underground wires, including special work	25,000 ,, 30,000	45,000 ,, 55,000
Cable tramways, mechanical, with underground cables, etc.	30,000 ,, 40,000	50,000 ,, 60,000
Horse tramways, with stables, cars, etc.	12,000 ,, 18,000	25,000 ,, 30,000
Steam tramways, with locomotives and cars	15,000 ,, 20,000	26,000 ,, 32,000

The cost of line per mile varies according to power station, plant, number of cars, cables, class of paving for permanent way, gauge and weight of tram-rails, car-sheds, etc.

PROPORTIONATE COST OF PARTS.

	Per cent.
Permanent-way construction	50
Overhead construction	10
Street alterations, etc.	5
Cables, feeder pillars, etc.	10
Power stations, buildings, plant, etc.	8
Tramcars	7
Supervision and incidental expenses	5
Compensation and legal charges	5
	100

Cars per mile of single line	3 to 6
Motor terminal voltage	500 to 550

Trolley vehicle system of electric railless traction (1930) :—	£ £
Capital cost per mile of route, double wire	4,000 to 6,000
Good overhead work, per mile	2,400 ,, 3,000
Vehicles, cost of each	1,500 ,, 2,000
Obtaining Parliamentary powers	2,000 ,, 5,000
Cost of working per car mile	1*s.* ,, 1*s.* 3*d.*
Traffic receipts per car mile	1*s.* ,, 1*s.* 8*d.*

ACTUAL COSTS OF TRAMWAYS PER MILE (PRE-WAR).

Parts of Construction.	Edinburgh Cable Conduit.	South London (1903). Electric Conduit.	Electrical Overhead Trolley System. Glasgow.	Huddersfield.	Leeds.	Sunderland
	£	£	£	£	£	£
Permanent-way construction	14,431	13,460	5,849	5,589	5,600	6,810
Overhead construction	—	—	776	928	1,173	720
Street alterations, etc.	890	134	—	245	—	522
Cables, feeder pillars, etc.	221	2,301	3,068	904	1,000	312
Power stations, buildings, plant, etc.	7,994	3,933	7,188	1,440	3,477	1,543
Tramcars	1,780	4,363	2,573	1,125	1,875	2,240
Supervision and incidental expenses	1,000	915	348	50	—	380
Total cost per mile of single track	26,316	25,106	19,802	10,281	13,125	12,527
Cars per mile of single track	4·1	6·1	5·25	2	2·91	3·4
Length of single track in miles	35	16	130	35	81	21
Motor terminal voltage	—	550	500	500	500	500

Aberdeen, overhead trolley system, 4 ft. 8½ in. gauge, exclusive of generating station, cost per mile of double track . . . . . £ 13,400

Darwen, overhead trolley system, 4 ft. gauge, exclusive of power station, total 35,000*l.*, per mile of double track 13,000

Dover, overhead trolley system, 3 ft. 6 in. gauge, exclusive of power station, total 28,000*l.*, per mile of single track 7,000

Manchester (1905), overhead trolley system, current provided by the lighting station, cost per mile of track 12,500
Permanent way, cables, and electrical equipment only, per mile of track . . . 6,575

Portsmouth, overhead trolley system, 4 ft. 7¾ in. 'Scotch' gauge, exclusive of power station, but including taking up old tram lines and concrete foundations and new ditto, cost per mile of double track 21,000
Ditto, including power station and machinery, ditto 23,900

Staffordshire, Bristol, and Coventry, capital expenditure on electric tramways per mile of single track, average 12,000

Wolverhampton, permanent way, electrical equipment, and cables, on the Lorain surface contact system, per mile 8,600

WORKING EXPENSES.	Per car mile. d.
Aberdeen, electric overhead trolley system, including interest on sinking fund	9·00
Birmingham, 1890–1, electric accumulator	9·90
" " cable traction	6·33
" " horse traction	9·79
" " steam traction	10·99
Dover, overhead trolley system, including interest on sinking fund	8·75
London County Council conduit tramways worked on purchased current, in 1905–6	8·02
Staffordshire, Bristol, and Coventry electric trams	6·30
Modern electric trolley lines generating their own current work	5*d.* to 6*d.*
" " " where current is purchased	6*d.* to 7½*d.*

The Peak Cable Tramway, Hong-Kong (1888).—Rises 1,207 ft. in a total length of 4,690 ft. Maximum gradient 1 in 2, and minimum 1 in 25. Three intermediate stations, and top and bottom termini. Single track 5 ft. gauge of two 30-lb. rails, and steel sleepers 24 lbs. per yard. Midway crossing place 130 ft. in the clear, above which the track is formed of three rails. Power-house at summit, with two engines, boilers, etc. Steel haulage cable, 3½ in. circumference, weighing 2 lbs. per ft. run. Rolling stock consists of two cars, each holding 45 passengers, and one ascends while the other descends, 9 minutes per trip. Total cost, 20,000*l.*, or at the rate of 22,516*l.* per mile.

(See p. 689.)

Tube Wells.

Driven tube wells sunk in ordinary ground, including use of tackle, well driver's time, and wrought-iron lap-welded tubes, approximate costs as detailed below; add pumps. The quantity of water obtainable depends on the water-bearing strata, and can be increased by connecting two or more wells together. See also 'Artesian Wells,' p. 2840.

Depth of Tubes, etc.

Estimated Costs, 1930.

		Internal Diameter. 1½ in.	2 in.	3 in.
Not exceeding 15 ft. deep	each	8*l.*	15*l.*	25*l.*
15 ft. to 20 ft. deep	„	10*l.*	17*l.*	28*l.*
20 ft. „ 25 ft. „	„	12*l.*	20*l.*	35*l*
25 ft. „ 30 ft. „	„	15*l.*	24*l.*	43*l.*
30 ft. „ 35 ft. „	„	18*l.*	28*l.*	48*l.*
35 ft. „ 40 ft. „	„	21*l.*	32*l.*	52*l.*
If beyond this depth, per foot deep		8*s.*	15*s.*	25*s.*
Sand strainers	each	40*s.*	60*s.*	80*s.*
Galvanised perforated driving points in 3 ft. lengths	„	2*l.*	3*l.*	4*l.*
Special point of tough yellow metal for waters impregnated with iron	„	3*l.*	4*l.*	6*l.*
Apparatus for driving tube wells, including sheer legs, ram, pully block, rope, and windlass	per set	20*l.*	25*l.*	30*l.*
Average quantity of water obtained, gallons per hour:		150 to 900;	300 to 1,500;	450 to 3,000.

Description of Pump.

	Size of barrel. 3 in.	3½ in.	4 in.
Standard pump, with valve door, supplied only, each	6*l.*	8*l.*	10*l.*

Tunnels.

Estimated Costs, 1930.

Cost of all tunnelling varies considerably according to locality, nature of excavations, pumping, size and depth of shaft, lining, distance to remove spoil, etc.

Adits.—Headings or tunnels at bottom of shafts, including excavating, removing, and raising material from a depth not exceeding 100 ft., and length of tunnel not exceeding 50 yds. from mouth of shaft, timbering, etc., complete, but exclusive of pumping:—

4 ft. × 6 ft. in clay, gravel, or soft chalk	per ft. run	2*l.* to 3*l.*
do. in hard rock requiring blasting	do.	4*l.* „ 5*l.*

Aqueduct Tunnels.—As constructed in connection with large waterworks:—

Tunnels lined with concrete, 10 ft. diameter	per yard run	70*l.* to 100*l.*

Canal Tunnels.—As for inland navigation by ordinary canal boats, 72 ft. long × 7 ft. wide, average dimensions of tunnels being 16 ft. × 18 ft.:—

Excavation in clay, and lining with brickwork and concrete	per yd. run	50*l.* to 80*l.*
Excavation in rock, and lining with masonry and concrete	do.	80*l.* „ 120*l.*

Irrigation Canal Tunnels.—As constructed in India to convey water from the mountainous districts to the lowlands, the average dimensions being 18 ft. wide by 12 ft. high. The longest is the Swat Canal Tunnel, 11,235 ft., and the next, the Pariyar Tunnel, 5,704 ft.

Irrigation canal tunnels in hard rock	per yd. cube	2*l.* to 3*l.*
„ „ „	per yd. run	40*l.* „ 60*l.*

Railway Tunnels.—Cost depends upon nature of excavations, lining required, etc.:—

For narrow-gauge or metre railways (3 ft. 3⅜ ins.), single track	per yd. run	50*l.* to 100*l.*
For standard-gauge railways (4 ft. 8½ ins.), single track	,,	80*l.* ,, 150*l.*
,, ,, ,, ,, double track	,,	150*l.* ,, 300*l.*

Ordinary Tunnels.—Ordinary tunnelling, cubic rate. It is generally cheaper to tunnel than to make a cutting over 50 ft. deep, but a good deal depends upon whether earth is required for embankments, and upon the kind of ground.

Tunnelling or driving headings, 20–50 ft. sup. sectional area, not exceeding 50 ft. below surface of ground or 50 yds. from shaft or adit, including use and waste of timbering, and drawing spoil to foot of shaft ready for removal :	Per yd. cube. *s.*	*s.*
in ordinary ground, sand, or gravel	9 to	11
in stiff clay or shingle	12 ,,	14
in hard, solid chalk and flints	17 ,,	20
in soft rock not requiring blasting	45 ,,	50
in hard rock requiring blasting	55 ,,	60
Hoisting spoil from foot of shaft to ground surface, for any depth not exceeding 25 ft.	2 ,,	3
Extra only for every additional 25 ft. in height	1 ,,	2
Removing not exceeding ¼ mile, and depositing	4 ,,	5
Filling in, packing, and ramming, including transporting not exceeding 50 yds.	3 ,,	4
Excavating in large tunnels not less than 18 ft. wide × 20 ft. high, including fixing and removing temporary timbering and shoring, and removal of spoil not exceeding 1 mile :		
in ordinary ground, clay, or gravel	20 ,,	30
in soft rock not requiring blasting	30 ,,	40
in hard rock requiring blasting	40 ,,	70

Open Shield Tunnelling.—For subways or tunnels under rivers, with open headings and ordinary atmospheric conditions. Cost includes providing, fixing, and removal of timbering, shield, hydraulic jacks, pumping machinery, electric light, lifting excavated material to surface of ground, etc., but *exclusive* of concrete, masonry, or iron lining to tunnel ; for quantities not less than 3,000 yds. cube.

Open shield tunnelling, 100–200 ft. sup. sectional area, not exceeding 50 ft. below surface of ground nor 300 yds. from mouth of tunnel :	Per yd. cube. *s.*	*s.*
in ordinary ground or sand	50 to	70
in stiff clay or gravel	60 ,,	80
Open shield tunnelling, 200–400 ft. sup. sectional area :		
in ordinary ground or sand	70 ,,	90
in stiff clay or gravel	80 ,,	100
Add for every additional 25 yds. in length	1 ,,	2

Pneumatic (or Closed Shield) Tunnelling.—For excavations in water-logged ground under air pressure, the closed shield having bulk-heads and air-locks, to admit and remove materials, workmen, etc. Cost includes providing, fixing, and removal of timbering, shield, hydraulic jacks, pumping machinery, air-compressing plant, electric light, raising excavated material to surface of ground, etc., but *exclusive* of concrete, masonry, or iron lining to tunnel ; for quantities not less than 3,000 yds. cube.

Pneumatic shield tunnelling, 100–200 ft. sup. sectional area, not exceeding 50 ft. below surface of water or ground nor 300 yds. from mouth of tunnel :	Per yd. cube. *s.*	*s.*
in ordinary ground or sand	100 to	120
in stiff clay or gravel	110 ,,	130
Pneumatic shield tunnelling, 200–400 ft. sup. sectional area :		
in ordinary ground or sand	120 ,,	140
in stiff clay or gravel	130 ,,	150
Add for every additional 25 yds. in length	2 ,,	3

Tunnel Shields.—The cast-iron lining of a shield-driven tunnel is equivalent to about one-third total cost of tunnelling. The shield was first used by Brunel at the Rotherhithe tunnel in 1825, then by Mr. Greathead at the Tower subway in 1869, and the improved 'Greathead' shield in

1884 on the City and South London Railway. It consists of a steel ring with cutting edge, which is pushed forward by hydraulic jacks about 3 ft. apart. Tube railways and subways vary from 11 to 30ft. diameter, with 8–28 jacks per shield, and the tunnelling progresses at the rate of 30–60 ft. per week.

Costs complete with bolts, nuts, hydraulic jacks, etc., but *exclusive* of hydraulic machinery, piping, etc.:

Steel open tunnelling shields complete, 10–20 ft. external diameter, 40*l.*–50*l.* per ton each 1,000*l.* to 4,000*l.*

Steel closed tunnelling shields complete, 12–25 ft. external diameter, 50*l.*–70*l.* per ton each 4,000*l.* to 12,000*l.*

Tunnel Maintenance.—Cost of reinstating 3- to 6-ring double-track railway tunnels, in sand, clay, or chalk, including cutting out the inner ring and replacing by a brindle ring in cement, sometimes an additional ring, or part of the tunnel inverted, toe wall built to the footings, parts of the wall relined, or spaces behind walls and over arches filled with concrete, etc., the work being carried out day and night during traffic by contract or departmentally, a single line run at night, and taking various periods of from three to eight years to complete.

20*l.* to 40*l.* per lineal yard:

Other tunnels requiring more extensive repairs, such as shoring, bad portions cut out, and completely rebuilt, use of cement gun, etc.:

60*l.* to 500*l.* per lineal yard.

TUNNELS (RAILWAY).

TUNNELS. Actual Costs.	Date. Years.	Length. Yds. Run.	Ft. Per Day.	Total in thousands	Per Y.R.
				£	£
Arlberg (6¼ miles), Austria. 2 tracks in single tunnel, 25 ft. 3 ins. wide by 21 ft. high, partly lined with masonry	1880–84	11,210	28	1,204	107
Hoosac (4⅓ miles), Massachusetts, U.S.A. .	1855–76	8,390	7	2,014	240
Loetschberg (9 miles), Berenese Oberland .	1906–11	15,973	36	2,556	160
Mont Cenis (7⅞ miles), France to Italy. 2 tracks in single tunnel, 26 ft. 3 ins. wide by 24 ft. 7 ins. high. Through solid granite, brick and stone lined .	1857–71	13,712	9	2,880	210
Severn (4⅓ miles). 2 tracks in single tunnel, 26 ft. wide by 24½ ft. high, lined with 3 ft. brickwork. Water-logged ground and expensive pumping. Ventilated by fan 40 ft. diam. Hawkshaw, Hayter, and Richardson, engineers	1879–86	7,674	6	2,250	293
Simplon (12½ miles), Switzerland to Italy. 2 single-track tunnels, each 16 ft. 5 ins. wide by 19 ft. 6 ins. high, 56 ft. apart axis to axis. Enlarged in 1911, at 1,600,000*l.* A. Brandt, engineer .	1898–1906	21,578	28	3,120	145
St. Gothard (9⅛ miles), Switzerland to Italy. 2 tracks in single tunnel, 26 ft. 3 ins. wide by 24 ft. 7 ins. high, in granite and lined with masonry. Louis Favre, engineer	1872–82	16,426	18	2,300	140

WATERSHEDS.

Afforestation at Torquay (1923–28).—Watersheds at South Teign and Tottiford, near Torquay, for the intake of the waterworks conduit. 720 acres, 40 acres being planted per annum, and 278 acres previously since 1899. The cheapest and best way of obtaining trees was to buy two-year-old seedlings, ranging from 7*s.* to 15*s.* per 1,000, which were placed in the corporation nurseries for eighteen months at a net cost of 4*s.* per 1,000. The trees were then planted out into permanent positions in the watersheds at a cost (in 1928) of 10*l.* to 12*l.* per acre, towards

which the Forestry Commission made a grant of 2*l.* per acre. 120,000 seedlings were purchased each year, and the varieties were :—Japanese larch, Douglas fir, Sitka spruce, Norway spruce, Austrian pine, Scotch fir, silver fir, ash, beech, and sycamore. The soil of the watersheds was of a light loamy nature, and fairly sheltered by hills. The whole cost of afforestation was paid out of revenue, and the work done by the corporation's regular staff under a forester, the areas being protected by wire-netting because of rabbits, which added to cost. When the trees make progress a considerable revenue is made of the poles from the thinning out of the plantations each year, as there is a good demand for this class of timber from nurserymen for general rustic work.

Water Towers.

Water towers, brick, exclusive of tanks and pipes, 1*s.* to 1*s.* 4*d.* per ft. cub., 2*s.* to 5*s.* per gal.
Water supply inside a building, including service pipes, taps, etc. 2*l.* 10*s.* to 3*l.* per tap.

Beverley Water Tower, Yorkshire (1915).—Built of reinforced concrete on the Hennebique system. Two circular tanks, one within the other, the inner one holding 4,000 gals. and the other 7,200 gals., or a total of 11,200 gals. Extreme diameter 34 ft., and 10 ft. internal depth, with three 6-in. cast-iron outlets, 4-in. sides, 2-in. curved dome top, 4-in. bottom, with beams under, supported on three 12-in. × 12-in. columns or legs, 30 ft. high, with 12-in. × 8-in. braces to latter and half-way up. Total height from ground level to top of tank, 42 ft. 6 ins. Concrete foundations, 2 ft. deep. Cost, including materials and erection, 312*l.*, or 27*l.* 17*s.* per 1,000 gals.

Bramham Moor Water Tower, near Tadcaster, Yorkshire (1916).—Designed by the Trussed Concrete Steel Co. Two united rectangular tanks, 4,000 gals. and 6,000 gals., or a total of 10,000 gals.; two iron ladders inside the tanks for cleaning purposes, and 3-in. cast-iron outlets; 5-in. sides, 3½-in. flat top, 6-in. bottom, with 24-in. × 10-inch beams under, supported on four 10-in. × 10-in. columns or legs, 20 ft. high, with 10-in. × 6-in. braces to latter and half-way up. Foundations 2 ft. 6 ins. deep in addition. Cost, including materials and erection, 298*l.*, or 29*l.* 16*s.* per 1,000 gals.

Curragh Water Tower (1900).—Heavily built, 34 ft. square externally and 94 ft. high above ground. 6 stories. Concrete foundations 6 ft. thick, brick walls in cement 45 ins. thick below to 22 ins. at top. Flat concrete roof, asphalted. Contains two cast-iron tanks, 20,000 gals. each, or 40,000 gals. total. Lower floors used as a fire engine station. Building only, 9,300*l.*; cast-iron tanks and pipes ,1,630*l.*; special fittings, 70*l.* Total, 11,000*l.*
1*s.* 4*d.* per ft. cube. 275*l.* per 1,000 gals.

Waterworks.

Waterworks, comprising reservoirs, filter beds, mains, valves, hydrants, etc., complete:

	£	
Birmingham	3	per head of population.
Dublin	2	" "
Edinburgh	3½	" "
Glasgow	3	" "
Liverpool	4	" "
Manchester	3	" "
Generally speaking	2*l.* to 4	" "
Small waterworks in Ireland (20 to 25 gals. per head.)	2*l.* to 2½	" "

Batticaloa Water Supply, Eastern Province, Ceylon (1915–16).—Supplies a population of 6,000 with about 20 gals. per head per day. One central well 10 ft. diameter, and two auxiliary wells same diameter, connected by adits 500 ft. long, yielding 180,000 gals. per day. From the central well the water is pumped through a 7-in. steel delivery main to the reservoir, 4 miles distant, from which the town is supplied by gravitation by 2-in. and 1½-in. distributing mains to 75 standpipes. 11¼ miles of pumping and service mains altogether. Pumping machinery in duplicate, 3-in. centrifugal pumps, belt-driven by 18 B.H.P. oil engines, each pump capable of delivering 10,000 gals. per hour against a total head of 60 ft., or a total pumping capacity of 100,000 gals. per diem working 10 hours. Wells lined with 12-in. concrete, roofed with concrete slabs. Storage reservoir of 100,000 gals., with concrete walls faced in brick cement rendered and roofed with corrugated iron on steel trusses. Cost, 9,533*l.*, or 95*l.* per 1,000 gals., or 1*l.* 12*s.* per head of population supplied.

New Magdalen Barracks Water Supply, Oxford (1896).—Supply from City of Oxford Water Company's mains, constant service, Siemens' meter supplied by Company. Davey's lift and force pump worked by steam power, and 6 in. cast-iron mains, 3 in. cast-iron domestic supply mains, and 1½ in., 1 in., ¾ in., and ½ in. galvanised wrought-iron service pipes, 3 to 4½ ft. deep. Reserve tank of 13,000 gals. for fire purposes, 60 ft. high on Armoury roof, 18 fire hydrants, 20 sluice valves, 40 stop valves, 280 bib valves, 50 3-gal. flushing cisterns to w.c.'s, and 6 200-gal. cisterns to latrines. Occupation of barracks, 500 adults, 100 children, and 5 horses. Total cost, including 800*l.* for machinery and erection, 3,500*l.*, or 5*l.* 16*s.* 8*d.* per head of occupation, exclusive of horses. 6*s.* 6*d.* per yd. run of excavation, exclusive of machinery, but inclusive of all fitments.

(See also p. 1418.)

IMPOUNDING RESERVOIRS.

Impounding reservoirs for gravitation waterworks to large towns, including excavations, embankments, etc.

200*l.* to 700*l.* per million gallons storage.

	£	
Arnfield, for Manchester waterworks	432	per million gals. storage.
Crowden " "	561	" "
Hollingworth " "	458	" "
Tellowfold " "	722	" "
Monmouth	912	" "
Rooden, for Oldham	491	" "
Sheffield waterworks, structure only	210	" "
Ditto, including Parliamentary expenses, payments for compensation, etc.	485	" "
Swansea	550	" "
Swinshaw, for Ashton-under-Lyne and Stalybridge	405	" "
Walshaw Dean, for Halifax	469	" "

Curragh Old Waterworks Pumping Station.—Different dates, 1859, 1907, 1915, and 1921. Includes concrete reservoir, corrugated iron fence, brick engine house, boiler house, chimney shaft, workshop, coal yard, engineer's cottage, and new corrugated iron engine shed, engineer's hut, concrete tanks, old and new wells with bored Artesian tubes, 112 ft. and 276 ft. total depths respectively, barbed wire fencing, water-main connections, and 1,550*l.* worth of machinery, complete . . . . . 9,550*l.* total cost.

13½-in. Artesian bored tube well (1907), including boring 52 ft. deep sunk in sand and gravel from the bottom of existing dug well which is 8½ ft. diameter and 59 ft. deep . . . . . 3*l.* 5*s.* per ft. in depth.

Ditto including men's time, fixing plant, testing, and carriage, and hire of compressor . . . . . 7*l.* 10*s.* ditto.

(The cost of this Artesian well is included in the 9,550*l.* total.)

Curragh Old Waterworks Reservoir (1887–93). In connection with pumping station. Uncovered, with puddle clay embankments, stone-lined. Steel fence all round. Size, 320 ft. × 260 ft. × 6 ft. deep, and holds 3,100,000 gals. Total, 2,760*l.* 1½*d.* per foot cube.

Reservoir now disused . . . . . 18*s.* per 1,000 gals.

Selby Waterworks Reservoir (1908). 800,000 gals. 12-in. concrete floor, brick walls, 50½ ins. thick at base and 22½ ins. at top, 6-in. concrete roof in arches of 7 ft. span. Total, 4,800*l.* . . . . . 6*l.* per 1,000 gals.

In the London Metropolitan area—

Cost of pumping averages 43 per cent. of the entire maintenance expenditure.
Cost of pumping alone averages 50*s.* per million gallons.
Repairs and renewals of water-mains and pipes, 34*s.* per million gallons.
Maintenance of water-mains or pipe-way, 24*l.* per mile.
Cost of filtration, 7*s.* 5*d.* per million gallons.

Assuan Dam (1898–1902 and 1907–12).—The Assuan or Aswan Dam was built for irrigation purposes at the head of the first cataract of the Nile, about 600 miles from Cairo and 350 miles above the Assiout barrage. It was formed across 5 channels of the river, which flowed at the rate of 16 miles an hour, and constructed of local granite masonry in Portland cement, on a similar rock foundation which had dykes of softer igneous material. The original dam was 1½ miles or 6,600 ft. long, 100 ft. thick at bottom, tapering to 24 ft. at top, with a maximum height of 130 ft. from base. Total weight of stonework over a million tons; 180 sluice openings, closed by steel gates; 1,000 Europeans and 20,000 natives were employed. The barrier affected the Nile for 140 miles higher up, and turned it into a regular lake. The greatest difference of level of water above and below the wall was 67 ft., and navigation was provided by a ladder of five locks, each 260 ft. long and 32 ft. wide. This first dam was completed in 1902, and held about 250,000 million gallons, which brought over a million acres of desert under cultivation. Engineer, Sir Wm. Willcocks, and contractors, Sir John Aird & Co. Cost was 2,500,000*l.* constructional work and 2,000,000*l.* for land and compensation charges, or 4,500,000*l.* altogether. This latter is equivalent to 18*l.* per million gallons of storage capacity, or 4*l.* 10*s.* per acre of irrigation area.

Between 1907 and 1912 the foregoing original dam was raised an additional 23 ft., thickened by 16 ft. on the downstream side, lock walls widened, and new dock gates provided, so that the impounding reservoir or lake was now 240 miles long, and the storage capacity increased by an extra 310,000 million gallons. Another million acres were also brought under cultivation. The greater depth submerged the Philae temples well up the river. Sir Benjamin Baker was consulting engineer. Cost of extension, including land and compensation charges, was 1,500,000*l.*, equivalent to 4*l.* 17*s.* per additional million gallons, or 1*l.* 10*s.* per additional acre.

Total present capacity as enlarged, 560,000 million gallons, bringing two million acres under total irrigation, at a total cost of 6,000,000*l.* This is equivalent to an average of 10*l.* 14*s.* per million gallons, or 3*l.* per acre of irrigation, as finally completed.

Burrinjuck Dam (1909-20).—Or Barren Jack. The highest dam in the Empire and in the Blue Mountains of New South Wales, at the confluence of three rivers—the Murrumbidgee, the Goodradigbee, and the Yass, which the dam ponds up 40 miles, 13 miles, and 19 miles respectively. The artificial lake so created has a surface area of 20 square miles, or 12,800 acres, and is 220 ft. greatest depth, giving a capacity of over 33,000,000,000 cubic ft., or about 206,250,000,000 gallons. The dam is a massive wall in a deep gorge with cliffs 1,000 ft. high and only 300 yds. apart, the sides being so steep that it was impossible to establish workshops upon them, and it was also across the bed of a river subject to floods. The granite masonry is 236 ft. high from rock to crest, set in cement, and 186 ft. thick at base, tapering to 18 ft. at top. The dam is of the arch type with convex side turned up stream; radius of curve 940 ft. and length of arc 900 ft. Flood-water tunnel 14 ft. × 13 ft., and two culverts cut through the cliffs at the ends of the dam itself. The water had to be conveyed 260 miles, and then distributed by miles of canals over 1½ million acres of parched land, allowing the conservation of water for some 2,000 farms. Total cost about 3,000,000*l.*, or 14*l.* 11*s.* per million gallons of storage capacity, or 2*l.* per acre of irrigation area.

Croton Dam (1896–1911).—Or the Cornell Dam, which holds up the waters of the Croton River to form the great Croton lake reservoir which supplies New York with water, 33 miles away. A diversion canal, 1,000 ft. long and 2,000 ft. broad, was first excavated to divert the river before the dam itself could be commenced. The latter is about 1,200 ft. long, a maximum width of 216 ft., and greatest height 300 ft. from rock to crest, making it the highest in the world. It contains 833,000 cubic yards of masonry, and required 1,500,000 cubic yards of excavation. Owing to the rock being rotten, the foundations were carried in places 130 ft. below river-bed. Spillway built of granite blocks, some weighing 10 tons each. The lake behind the dam is 20 miles long, and contains 32,000,000,000 gallons of water. Three villages and numerous isolated buildings and farms were cleared away, and 75 miles of stone wall built round the reservoir. The outlay was excessive, as a large part was due to alterations and work which had to be afterwards undone, including the removal of the first core wall and taking up the earth on either side of it. Total cost about 1,600,000*l.*, or 50*l.* per million gallons of storage capacity.

Wentwood Reservoir, Newport (1894–1904).—Situated at Llanvaches, some ten miles from Newport, Monmouthshire. It has a water area of 41 acres, a depth of 86 ft., and a capacity of 410,000,000 gallons. The great dam, which is 456 ft. above sea level, is 97 ft. high, 800 ft. long, and 700 ft. wide at base, gradually tapering to 30 ft. wide at top. The inside slope is pitched from top to bottom. The valve tower and outlet are placed quite independently of the dam at the eastern end, and the culvert, which runs for a length of 900 ft. through the dam, is 6 ft. diameter and carries two water mains. Water is drawn into the reservoir from two streams in the Newchurch valley by means of a tunnel, which ranges from 3 ft. diameter at the Newchurch end to 6 ft. at the reservoir end; length of tunnel, 3,683 yds. Total cost, about 390,000*l.*, or 951*l.* 4*s.* per million gallons of storage capacity.

Scraping Water Mains.

Ayr.—Scraping 14-in. and 16-in. cast-iron water mains, 14½ miles long, at Ayr, between Loch Findas and the service reservoir at Dalrymple, in 1915. Pipes laid 28 years before, but incrustation reduced capacity by 30 per cent. Scraper weighed 3 cwts., and was driven by a pressure of 15 ft. head of water. When this pressure was unobtainable the instrument was pulled through by a chain. Pipes were divided into sections with inspection boxes, and each section was done twice, the scraper travelling at the rate of 2 miles an hour. Capacity of mains was at once increased by 37 per cent.

For 14-in. pipes the scraping alone cost 13*l.* 8*s.* per mile, or 2*d.* per yd. run.
„ 16-in. „ „ „ 12*l.* „ „ 1⅜*d.* „
„ 14-in. pipes the total cost was 34*l.* 5*s.* „ „ 4⅜*d.* „
„ 16-in. „ „ „ 46*l.* 10*s.* „ „ 6½*d.* „

Scraping is necessary every 3 years, owing to rapid reincrustation.

Prestwick.—Scraping 6 in., 8 in., and 10 in. water mains leading from the filters to the town, in 1923. In all 7,100 lin. yds. of pipes were scraped, and over 8 tons of corrosion removed. The pressure heads after scraping, as taken at air valves along the mains, showed an increase overall of 20 ft. The cost worked out at 1*s.* 5*d.* per yd. run.

Cleaning.—Cleaning 4 in. water main by the patent Powrcleer apparatus, supplied by the British Pipe Cleaning Co., W. & J. Glossop, Hipperholme, Halifax, for a minimum quantity of 1 000 yds., 1*s.* 4*d.* per yd. run (August 1922).

Torquay (1927).—Old water mains, cast-iron spigot and socket pipes, varying from 3 in. to 12 in. diameter, and total length 200 miles. Scraping cost 15*s.* per mile, at a usual rate of 3 miles per hour.

Various Towns.—Actual cost by pipe scraper driven by water pressure and supplied by Glenfield & Kennedy, Ltd., Kilmarnock:—

Year.	Place.	Diam. of Main.	Length of Main.	Total Cost.	Cost per Mile.	Cost per Yd. Run.	Obstructions removed, other than corrosion.	Gain in Delivery after Scraping.
		Ins.	Miles. yds.	£	£ s. d.	s. d.		Per cent.
1880	Bradford	18	4 1,100	634	137 1 7	1 6¾	Stones, lead, crowbar, etc.	56
1883	Whitehaven	11	3 1,223	516	81 19 5	11¼	Mussels, stones, and lead	28
1885		13	2 1,056					
1887	Omagh	6	2 792	53	21 12 8	3	Lead and defective castings	300
1887	Roubaix	24	4 1,162	425	91 3 11	1 0½	Calcareous matter	—
1890	Dumfries	8	4 0	113	22 12 0	3	Stones	—
		9	1 0					
1891	Lanark	7	6 880	75	11 10 9	1½	Lead, wood, and stones	34
1891	Newport	..	0 75	8	187 14 8	2 1½	High-pressure hydraulic main	—
1892	Burntisland	8	3 1,320	252	67 4 0	9¼	Lead, wood, and stones	43
1894	Waterford	13	8 0	212	26 10 0	3½	Broken pipe, rope	40
1895	Cupar-Fife	7	3 880	67	19 2 10	2½	Broken pipe, rabbit	52
1895	Merthyr-Tydvil	12	6 770	178	27 13 0	3¾	Broken pipe, lead, and stones	82
1896		14	5 617	318	59 8 8	8	About 400 stones	30
1897	Alfreton	7	4 0	57	14 5 0	2	Pieces of pipe and stones	30
1897	Cumnock	5	1 880	52	25 15 4	3½	Bad joints, stones, etc.	50
		6	0 912					
1898	Kirriemuir	6	6 0	78	13 0 0	1¾	Bad joints	75
1899	Dunfermline	12	14 0	270	19 5 8	2¼	Lead and stones	30
1899	Plymouth	24	1 880	300	200 0 0	2 3½	—	—
1899	Tarapaca, South America	5		714	12 15 0	1¾	Chisel and piece of metal	27
		7	56 0				Stones and metal	27
		9					30 tons of incrustation	22
1900	Innerleithen	6	0 1,300	60	81 4 7	11	Stones and piece of wood	60
1900	Pontypridd Water Works	9	4 880	277	23 1 8	3¼	Stones, 36 lbs. lead, cast iron. Piece of wood, gate hinge. Steel bar and a bolt	Gain of 100 ft. head.
		10	4 880					
		12	3 0					
1900	Cockermouth	15	7 880	470	62 13 4	8½	Stones and 3 tons corrosion	99
1900	Mossel Bay, South Africa	4	8 1,170	785	36 4 8	5	Lead, yarn, and metal	88
		4½	1 0					
		5	12 0					
1902	Pelotas, Brazil	12	9 830	450	47 10 2	6½	Large stones, etc.	44

SECTION XLVI

DEPRECIATION OF PLANT AND MACHINERY—INCOME-TAX ALLOWANCES

(Contributed by Alfred B. Searle, Vice-Pres. Valuers' Inst., Licensed Appraiser and Valuer.)

SECTION XLVI

DEPRECIATION OF PLANT AND MACHINERY—INCOME-TAX ALLOWANCES.

(Contributed by Alfred B. Searle, Vice-Pres. Valuers' Inst., Licensed Appraiser and Valuer.)

Depreciation is a term applied to the reduction in value which occurs in plant or machinery, no matter whether they are in use or stored. The amount of depreciation is affected by the condition in which they are kept and by the availability or otherwise of more suitable appliances.

The loss in value is most properly met by an annual charge (to which the term 'depreciation' is commonly applied); the total of such charges should, when the appliance is worn out, discarded, or replaced by another, be equal to the difference between the original cost and the sum obtained (if any) by the sale of the worn-out or discarded appliance. Thus if a machine costing 1,000*l.* is discarded as useless after ten years and then is sold for only 10*l.* the total depreciation is 990*l.*, the annual depreciation is 99*l.*, and the residual value is 10*l.* The depreciation may be affected by any sums spent on renewing parts of a machine or plant, though these are usually charged separately in the accounts, as *renewals*.

Deterioration is the loss in value due to the lapse of time and to the use of the appliance. It occurs more rapidly when a machine is used than when it is stored under good conditions, but may be just as serious if a machine is stored under unsuitable conditions. In deterioration is also included the loss in value which occurs directly the ownership of anything has been transferred from the seller to the buyer, the 'second-hand value' being always less than the market price when new; the difference forms part of the loss by deterioration.

Obsolescence is the loss in value which occurs when a more suitable appliance can be used for the same purpose. It occurs when a machine is discarded, long before it is worn out, because another machine is more profitable to use.

Renewals are the cost of replacing worn-out or damaged portions of machinery or plant. To some extent they reduce the depreciation, but for convenience are usually treated quite separately. The Inland Revenue officials, for some purposes, take renewals instead of depreciation when calculating allowances for income-tax assessments. Thus, instead of allowing 3*l.* per annum as the depreciation on a typewriter, they allow nothing until the typewriter is replaced by another one; then they allow the cost of the new machine less the sum obtained by the sale of the old one, or a sum estimated to be obtainable for it.

Residual Value is the value of a machine or plant at the end of a given period—usually, but not necessarily, when it is worn out or discarded. In dealing with the estimated life of plant or machinery, the residual value of such plant or machinery when its period of usefulness has expired must be taken into account, and with plant of special nature it may generally be assumed that in proportion to its special character its residual value will be low. Plant of general character, engines, boilers, shaftings, pulleys, etc., carries a high residual value in the event of displacement.

Diminishing Value is the reduction in the value of plant or machinery as the result of depreciation or obsolescence, or both. The term is also used to indicate the value obtained by deducting depreciation for one year at a given rate, and then applying the same rate to the reduced value so obtained. Thus, if a machine costing 1,000*l.* is depreciated at 10 per cent. per annum on the diminishing value it will be worth 900*l.* at the end of the first year, 900*l.* — 90*l.* or 810*l.* at the end of the second year, 810*l.* — 81*l.* or 729*l.* at the end of the third year, and so on. This method is largely used because of its convenience, but it requires a larger rate of depreciation than when the rate is calculated throughout on the original value. In recent years there has been a tendency to revert to the basis of original cost. For instance, a rate of 10 per cent. on the original cost will

exhaust the value in ten annual instalments; but a rate of about 16⅔ per cent. would be required to exhaust the value in the ten years if calculated on the diminishing value remaining after the deduction of depreciation in each year.

The argument in favour of basing the calculation on the diminishing value is that the sums written off in the earlier years were greater when the repairs were small, and this tended to equalise matters. Too often, however, the two factors of rate and basis have not been settled together, and consequently the basis of diminishing value has had the effect of leaving a considerable value in the books when the plant has disappeared. For instance, a 5 per cent. rate generally carries the impression of a twenty years' life, yet taking 100*l.* as the cost, and writing off on the diminishing value, the balance left at the end of the twentieth year is 37*l.* 14*s.* After twenty-one years' service the residual value of the plant would be, roughly, one-third of the first cost, a figure far too high in most cases. Besides, by adopting the basis of original cost, the position is much clearer for any revision of the rate at any subsequent period.

Rate of Depreciation.—This is usually expressed as a percentage of the original value of the plant or machinery, or on what is known as the diminishing value. The rate of depreciation is based on the assumed durability of the appliance, on its residual value, and on whatever allowance should be made for its becoming obsolescent before it is worn out. The determination of the rate of depreciation of various kinds of machines, etc., requires special skill and experience, and has given rise to specially trained appraisers or valuers, each of whom has expert knowledge of certain classes of plant or machinery; a valuer who is an expert on brewing appliances would probably know little of cotton-spinning, bricks, or presses, and *vice versâ*. It is, therefore, important that valuations should be made by men of *special* knowledge and not (as is usually the case in valuations for rating assessments and some other purposes) by men with only a general knowledge.

As different parts of a plant and different machines wear out or become obsolescent at different speeds, it is desirable to decide on a rate for each separate machine or appliance, though some grouping is desirable.

There is much difference of opinion as to what are suitable rates of depreciation, especially when accurate data on which to base them are not available. The following figures are only approximate, but they give a useful *general* idea, which may be greatly modified in particular cases.

Motive Power, Boilers, Engines, etc.—Some authorities place the depreciation on engines as high as 12½ per cent., and boilers at 10 to 15 per cent. Much depends on stress of working and the water. Mathieson places the life of a boiler at fifteen years, with renewal of the furnace at the end of ten years, and recommends a rate of 7½ per cent. per annum on the diminishing value on this class, no provision being made for obsolescence. This is, however, an extreme view.

Power Transmitters.—Shafting, pulleys, and plant under this head have, as a rule, a high residual value, even if displaced, and a rate of 5 per cent. will usually be found sufficient. Belting should be classified as loose tools. In this class a rate of 10 per cent. would be desirable. No provision is made for obsolescence.

Process Plant, Fixed.—Under this head would be placed all heavy process plant, the renewal portions of which are classified under their separate heading. The deterioration would then be met by a rate of 2½ per cent., the renewal rate at 2½ per cent. An obsolescence rate depending on the nature of the processes will also be needed; it may be 6 per cent.

Process Plant (Wearable).—For this it is necessary to provide a heavy renewal rate. By the dissection of this plant the separation of the different details can be secured, and it is easy to apply to each a distinctive rate for renewals applicable to its character and requirements, based on experience. It will usually be at least 10 per cent.

Process Plant (Loose).—In the case of loose tools, the yearly valuation gives the most reliable results. In some cases there is an annual increase in this class, but this should verify itself. It is to be remembered that at a going-concern valuation this cannot, in the nature of the plant, work out at more than half the original cost value.

Accessory Plant.—Usually has a high residual value due to general utility, and consequently a rate for deterioration of 2½ per cent. and a renewal rate of 5 per cent. should suffice.

Horses, Carts, Harness, etc.—The risks and depreciation on this class are too great to include in any average rate. A revaluation annually gives probably the best results.

The following rates of depreciation have been authorised by the Government of Australia in connection with the Federal income-tax on industrial establishments:—Boilers, 5 per cent.; boring machinery, 10 per cent.; cables, aerial, 7·5 per cent.; cables, underground, 2·5 per cent.; condensers, 5 per cent.; cranes, 5 per cent.; elevators, electric, 7·5 per cent.; elevators, hydraulic, 10 per cent.; engines, steam, 5 per cent.; foundry cupolas, 15 per cent.; furniture, 5 per cent.;

generators, 5 per cent.; machine tools, 5 per cent.; machinery used in ordinary manufacturing, 5 per cent.; meters, switchboard, 7·5 per cent.; motor trucks, 10 per cent.; patterns, 15 per cent.; pumps, 5 per cent.; galvanised iron tanks for rain water, 5 per cent.; galvanised iron tanks for other water, 10 per cent. The figures are based on the *diminishing value* of the assets and not on their original cost.

The following table, by Preece, gives (A) the approximate life in years of the various classes of machinery, etc., in an electrical undertaking if properly maintained, and (B) the probable residual value at the end of that term in per cent. of original cost. It affords a useful means of checking any suggested rate of depreciation on the appliances, etc., mentioned, but is very incomplete and tends to include figures in column A which are too large for present-day conditions.

	A.	B.		A.	B.
Foundations	100	nil	Cables (solid in wood trough)	40	12
Buildings	80	nil	Buildings (sub-station)	50	nil
Boilers (water tube)	25	5	Sub-station equipment	25	12
Boilers (Lancashire)	22	3	Accumulators	15	10
Dynamos and alternators	30	8	Arc lamp-posts	40	5
Engines and other machinery	25	6	Arc lamps	12	5
Cables (armoured)	35	15	Electric motors	25	9

Methods of Estimating Depreciation.

Appraisers and valuers have various methods of estimating the depreciation of plant and machinery, and two men of equal skill may not agree in their estimate because they use a different method, or because they have different ideas as to the durability of a particular appliance; the latter affects the rate of depreciation. The two methods generally adopted are: (1) To assume a definite rate of depreciation on either the plant and machinery as a whole or definite rates for each part thereof, and to deduct this from either the original cost or the book value. (2) To value each individual part of the plant and machinery each year and write off the difference in value from that of the previous valuation. The Inland Revenue officers usually employ a third method, based on renewals.

All these methods are faulty. In the first the result is hazardous, especially if based on a single rate of depreciation on the whole plant and machinery, because different parts depreciate at different rates, and no single figure or 'average' can be accurate. Even if the plant and machinery are divided into groups or each individual item is taken separately errors may still occur, as it is by no means easy to find an accurate rate of depreciation for some appliances. In the second method an independent valuer is essential, and the method is therefore costly. A common device is to use the first method, applied to the plant and machinery after dividing it into five or six groups, and to adjust this by a revaluation by an expert every five years. This is probably the best compromise between accuracy and cheapness.

When the first method alone is used, a distinction must be made in the rate of depreciation, according as this is deducted from the original cost of the appliance or from its diminished value (see *ante*).

Fire Insurance and Depreciation.—Whilst it is a sound, general policy to write as much off for depreciation as possible, such a course has an adverse effect on insurance against fire, as the value in the case of a fire would be affected in the claim for replacement. It is usual in some offices to append a declaration on the face of the policy that the depreciation written off in the books shall not be taken for the purpose of assessing a fire loss. Usually the assured is covered for the value of the plant at the time of the fire, and that the original cost of the plant, less depreciation, is not necessarily the basis of the claim. There appears to be no scale of depreciation in force for different kinds of plant recognised by the insurance companies, but these losses are generally settled as between the assessors and surveyors acting for both parties, or, failing their agreement, by arbitration: no special rates are applicable, but each case is settled on its merits. The principal factor appears to be the possession of definite and specific information on the value and cost of the plant, and the rate of depreciation should not in any way govern the claim in case of fire.

It is a general provision in fire insurances and policies that the company may elect to replace the plant; instances of this have rarely occurred, if ever, and probably the replacement clause was never intended as other than a protection to the insurance company.

Depreciation and Income-Tax.—The rate of depreciation allowed by the inspectors appointed under the Income-tax Acts differs considerably in different localities. For instance, the rate of depreciation allowed to engineers in Leicester is 7½ per cent. on the full value, while in Cardiff the rate is only 5 per cent. on the diminished value. Hosiery machinery in Leicester obtains 7½ per cent. as against 5 per cent. on lace machinery in Nottingham. Dyers' machinery in Leicester can get 10½ per cent., if justified on inquiry. Generally speaking, a distinction is made between motive

power and process plant, and the general practice assumes 5 per cent. on motive power and 7½ per cent. to 10 per cent. on process plant; these allowances are in respect of both deterioration and renewals.

The Inland Revenue authorities have now reverted, in a number of cases, to the basis of original cost, and where, say, 5 per cent. is allowed on original cost, there is an alternative rate of 7½ per cent. allowed on diminishing value.

On *Fixtures* no allowance for depreciation is made, unless they are included under the head of plant. Any replacements are allowed in full against the year's working.

Renewals.—The practice adopted by the income-tax authorities provides for the charging up of renewals to capital account where depreciation is allowed, and they usually decline to allow both renewals and depreciation out of the year's profits. They will, however, permit, as an alternative to the allowance for wear and tear and obsolescence of plant and machinery, the cost of renewing plant and machinery, under Schedule D. When this course is adopted, the amount to be allowed is the actual cost of the new plant and machinery (excluding any part of such cost which is attributable to additions or improvements, *i.e.* to an increase in capital) after deducting the scrap value or realised price of the plant and machinery replaced.

Example (*a*).—A machine which originally cost 1,000*l.* is worn out and replaced by a machine of similar power or size or capacity which now costs 1,500*l.* The whole of this expense of 1,500*l.* less, say, 100*l.*, the scrap value of the old machine (making a net amount of 1,400*l.*), is allowable from the profits of the year in which it is incurred.

Example (*b*).—A machine which originally cost 1,000*l.* is worn out and replaced by one of greater power or size or capacity costing 2,500*l.* The amount to be allowed as an expense is in this case not the full 2,500*l.*, but only the cost of replacing the old machine by one of similar power of capacity, say, 1,500*l.*, less the scrap value of the old machine.

Although this method of allowance is alternative to the wear and tear allowance for the same class of plant, the two principles may run concurrently for different classes of assets in the same business. For example, the wear and tear allowance may be applied to fixed machinery, and the renewal method used for loose plant.

Example (*c*).—Assuming the total written down value of the trader's plant and machinery before the sale of the obsolete machine to be 10,000*l.*, the new machine to cost 1,400*l.* and the old one to be sold or 400*l.*, future wear and tear allowances will be made upon a value of 11,000*l.*, thus :—

	£
Written down value of total plant, etc.	10,000
Deduct written down value of the obsolete machine . .	400
	£9,600
Add cost of new machine	1,400
	£11,000

The *Wear and Tear Allowance* was formerly based on the average of the preceding three years' trading, but in 1927 this was stopped and since then only the preceding year is taken into account.

If the allowance for wear and tear exceeds the assessable profits, the excess may be carried to the following year's assessment. Thus :—

	£
Gross assessment, 1927–8	6,000
Wear and tear allowance, 6,500*l.*, of which 6,000*l.* is allowed for 1927–8 and 500*l.* is carried forward . . .	6,000
Net amount on which tax is payable for 1927–8 . .	nil
Gross assessment, 1928–9	7,600
Wear and tear allowance, 6,800*l.* for 1928–9 plus the unexhausted balance carried forward for 1927–8, 500*l.* .	7,300
Net amount on which tax is payable for 1928–9 . .	£300

The 'renewal allowance' has been made statutory by each Finance Act since 1918.

ADDENDUM.

DEPRECIATION TABLE FOR PLANT AND MACHINERY, SHOWING THE TOTAL AMOUNT WRITTEN OFF IN A GIVEN NUMBER OF YEARS IF DEPRECIATED AT THE RATE OF 5 TO 10 PER CENT. PER ANNUM ON DIMINISHING VALUES.

Years	5 Per Cent.	6 Per Cent.	7 Per Cent.	8 Per Cent.	9 Per Cent.	10 Per Cent.
1	5·00	6·00	7·00	8·00	9·00	10·00
2	9·75	11·64	13·51	15·36	17·19	19·00
3	14·26	16·94	19·56	22·13	24·64	27·10
4	18·55	21·92	25·19	28·36	31·42	34·39
5	22·62	26·61	30·43	34·09	37·59	40·95
6	26·49	31·01	35·31	39·36	43·21	46·86
7	30·16	35·15	39·84	44·21	48·32	52·17
8	33·66	39·04	44·05	48·67	52·97	56·95
9	36·97	42·70	47·97	52·78	57·20	61·26
10	40·12	46·14	51·61	56·56	61·06	65·13
11	43·12	49·37	55·00	60·03	64·56	68·62
12	45·96	52·41	58·15	63·23	67·75	71·76
13	48·67	55·26	61·08	66·17	70·65	74·58
14	51·23	57·95	63·80	68·87	73·29	77·12
15	53·67	60·47	66·33	71·37	75·70	79·41
16	55·99	62·84	88·69	73·66	77·88	81·47
17	58·18	65·07	70·88	75·77	79·87	83·32
18	60·28	67·16	72·92	77·70	81·68	84·99
19	62·26	69·14	74·81	79·48	83·33	86·49
20	64·15	70·99	76·78	81·12	84·83	87·84
21	65·94	72·73	78·29	82·63	86·20	89·06
22	67·65	74·36	79·81	84·02	87·44	90·15
23	69·26	75·90	81·22	85·30	88·57	91·14
24	70·80	77·35	82·53	86·42	89·60	92·02
25	72·26	78·71	83·76	87·51	90·53	92·82
30	78·54	84·37	88·70	91·77	94·10	95·76
40	87·15	91·58	94·46	96·42	97·70	98·52

(*Supplied by Fuller, Horsey, Sons & Cassell, Mechanical Auctioneers and Valuers, 11, Billiter Square, London, E.C. 3.*)

SECTION XLVII

LEGAL NOTES FOR ENGINEERS

Contributed by W. Valentine Ball, O.B.E., M.A., Barrister-at-Law, A Master of the Supreme Court.

Author of 'The Law Affecting Engineers' (Constable & Co.); 'The National Form of Building Contract' (published by The Local Government Journal), and Joint Editor of 'Emden's Building Contracts,' 4th Edition (Butterworth & Co.).

SECTION XLVII

LEGAL NOTES FOR ENGINEERS.*

(Contributed by W. Valentine Ball, O.B.E., M.A., Barrister-at-Law, A Master of the Supreme Court.)

(1) Legal Status of an Engineer. (2) Fees. (3) Employment of an Engineer. (i.) *Generally*; (ii.) *Employment Abroad*; (iii.) *Wrongful Dismissal*; (iv.) *The Restrictive Covenant*, (v.) *Engineers and Compensation for Accidents*; (vi.) *Engineers and the National Insurance Act*; (4) Secret Commissions. (5) Disclosure of Secrets. (6) Rights of an Engineer to the Inventions of a Servant. (7) The Engineer as Witness. (i.) *Generally*; (ii.) *Arrangements as to Fees*; (iii.) *Evidence before Parliamentary Committees.* (8) Liability for Negligence. (9) Rights, Powers, and Duties of an Engineer under a Contract for Works. (i.) *Generally*; (ii.) *Specifications*; (iii.) *Plans and Drawings*; (iv.) *Extras*; (v.) *Certificates*; (vi.) *Penalties and Bonuses*; (vii.) *Defects, Repairs, and Maintenance*; (viii.) *Subcontractors and Specialists*; (ix.) *Whether the Employer's Engineer can act as Arbitrator in Relation to Disputes Arising under a Contract.* (10) The Engineer as Arbitrator. (i.) *Generally*; (ii). *Who may be Arbitrator*; (iii.) *Proceedings before the Arbitrator*; (iv.) *The Award*; (v.) *Time for Making the Award*; (vi.) *The Fees of an Arbitrator*; (vii.) *Stating a Case for the Opinion of the Court*: (viii.) *Costs of a Reference.* (11) Notes on Sale and Purchase of Machinery.

(1) LEGAL STATUS OF AN ENGINEER.

Any person may describe himself or practise as an engineer, as the word engineer is not a term of art in the legal sense. Although the Institution of Civil Engineers may make rules of professional etiquette to be observed by its members, compliance with such rules will not be enforced in a court of law, save in so far as they involve a restatement of the ordinary law. For instance a rule which states that no consulting engineer shall advertise could not be enforced, whereas infringement of a rule that 'he shall not pay by commission or otherwise anyone who introduces clients' might offend against the common law (*Panama Telegraph Co.* v. *India Rubber Co.* (1875) L. R. 10 Ch. 515) or against the Secret Commissions Act.

(2) FEES.

If an engineer is consulted, there is an implied contract to pay him reasonable remuneration (*Manson* v. *Baillie* (1855) 2 Mac. H. L. Cas. 80). It is more satisfactory, however, to provide for payment of fees beforehand. In the case of a contract to design and supervise erection of works it is well to provide for payment of ⅓ on signing the contract, ⅓ when half the contract price has been paid to the contractor, and ⅓ when the contractor has been fully paid. Provision should also be made for (1) payment for plans and drawings in case the works are abandoned; (2) travelling and hotel expenses. There is, of course, no fixed scale of remuneration for the engineer, but in deciding what is a reasonable charge, the Court would have regard to (i.) the standing of the engineer; (ii.) the nature and extent of the work undertaken and carried out. It is no answer to a claim for fees (*a*) that a particular undertaking has proved to be a failure if the failure was not due to any fault of the engineer—as where, for instance, an engineer is employed merely to *supervise* the erection of a dam which bursts on completion; or where an engineer is employed to draw plans for a newly invented machine which fails to work (*aliter* if the employer was relying on the engineer's skill and judgment: *Duncan* v. *Blundell* (1820) 3 Stark. N.P. 6); (*b*) that the work in question is not completed, if the failure to complete is not due to the engineer. Where an engineer is to be paid on completion and the work is abandoned before the time, the engineer is entitled to what is due for services rendered, and to what he has lost by not being allowed to complete the contract (*Horton* v. *Hemsley* (1908) *Times*, February 19).

* See also 'Legal Points for Road Engineers,' p. 827.

(3) EMPLOYMENT OF AN ENGINEER.

(i.) **Generally.**—Where an engineer is to be employed for more than a year certain, the agreement must be in writing; and this is so, even if the employment is terminable at any time by six months' notice on either side (*Hanau* v. *Ehrlich* (1912), A.C. 39). The document must set forth all the terms. It is safer, however, to have the terms of an agreement always put into writing. Special care should be taken to (*a*) specify the times when the salary is to be paid; (*b*) provide for the contingency of illness, as illness does not necessarily determine the contract, nor will it justify dismissal without regular notice, unless the employee is so smitten with disease that he can never be expected to work again (*Cuckson* v. *Stones* (1858) 1 E. & E. 248); (*c*) provide for the notice which may be given on either side to determine the contract. If nothing is specified in the agreement, an engineer employed at a yearly salary is entitled to a year's notice (*Buckingham* v. *Surrey and Hants Canal Co.* (1882) 46 L. T. 885). An engineer wrongfully quitting his employment without notice forfeits the salary due for the part of the current year which he has served. The fact that a contract of employment for an indefinite period contains no express term as to notice to be given for its determination does not exclude the implication that such a notice is required (*Payzu* v. *Hannaford* 34 T. L. R. 442). The suspension of servant from his duties which prevents his earning his commission may amount to a wrongful dismissal (*Rubel* v. *Vos*, 34 T. L. R. 171). Dismissal without notice is only justifiable in a case of gross misconduct. If the 'employer' is a company or a municipal corporation or other local authority it is safer to insist that the contract be under seal. The Public Health Act, 1875, s. 174 (1) provides that every contract made by an urban authority whereof the value or amount exceeds 50*l.* shall be in writing and sealed with the common seal of such authority. A surveyor who prepared plans for a local board under a verbal contract was held entitled to recover nothing (*Hunt* v. *Wimbledon Local Board* (1878) L. R. 4 C. D. 48). Close examination of a contract between an engineer and a local authority may, however, show that it is not within this section (see *Douglass* v. *Rhyl U.D.C.* (1913) 2 Ch. 407). Where the engineer is in the employment of a local authority he must have no interest in any contract into which that body enters (see *Melliss* v. *Shirley Local Board* (1885) L. R. 14 Q.B.D. 911). An attempt was recently made to raise the question whether a consulting engineer is liable to pay excess profits duty under s. 39 of the Finance Act, 1915, but the Court of Chancery refused to decide the point, leaving it to the King's Bench Division (*Smeeton* v. *Attorney-General* 35 T. L. R. 706).

(ii.) **Employment Abroad.**—Where an engineer is about to accept employment abroad, he should be careful to see that provision is made for the payment of travelling and other expenses. No agreement to pay travelling expenses would be implied from an undertaking, say, to employ an engineer at a salary of 1,000*l.* a year at Malacca as from January 1, 1913. It is also well to provide in the agreement that any disputes which may arise shall be determined either by arbitration or according to English law.

(iii.) **Wrongful Dismissal.**—If an engineer is wrongfully dismissed, it is his duty to reduce the damages as far as possible by taking other employment if he can procure it (see *Storey* v. *Fulham Steel Works* (1908) 23 T. L. R. 306). The damages are then ascertained by subtracting what he has earned from what he would have earned.

(iv.) **The Restrictive Covenant.**—When employing an assistant, an engineer may find it necessary to bind him not to set up close by in competition. A restrictive covenant is legal, if it is not too wide; it must be only what is reasonably necessary for the protection of the parties. A covenant to prevent a dentist practising within 200 miles of York was held to be too wide (*Horner* v. *Graves*, 7 Bing. 735). On the other hand, a covenant which bound a canvasser not to enter an employment 'within twenty-five miles of London where the plaintiffs carry on business' was held by the House of Lords to be unreasonable (*Provident Clothing and Supply Co.* v. *Mason* (1913) 29 T. L. R. 727). Care should be taken not to make the restriction perpetual unless that is absolutely necessary, for the Court criticises a perpetual covenant more closely (*Eastes* v. *Russ* (1915) 30 T. L. R. 237). In a recent case the validity of a covenant affecting a man who was employed as works engineer at a bottleworks was considered (*Forsters* v. *Suggett* 35 T. L. R. 87).

(v.) **Engineers and Compensation for Accidents.**—An engineer whose remuneration (*e.g.* money, board and lodging, etc.) amounts to less than 250*l.* in the service of an employer is probably entitled to the benefits of the Workmen's Compensation Act, 1906.

(vi.) **Engineers and the National Insurance Act.**—A nice question may arise as to whether, say, a station engineer must be treated as an employed contributor within the meaning of the National Insurance Act, 1911. That act covers employment in any contract of service subject to the proviso that persons earning more than 160*l.** otherwise than in manual labour need not be insured. It is submitted that the station engineer earning less than 160*l.* must be insured, but that the consulting engineer who receives an annual fee of 100*l.* for advising a firm or company need not be insured. His is a contract not 'of service' but 'for services'—a distinction which has been drawn in the Court of Appeal.

* By the National Health Insurance Act, 1919, now raised to 250*l.*

(4) SECRET COMMISSIONS.

The receipt of a secret commission or a bribe by an engineer would clearly justify his dismissal (*Boston Deep Sea Fishing Co.* v. *Ansell* (1888) 39 Ch. D. 339). In such a case there is a presumption that he was influenced, against his employers' interest, by the bribe (see *Hovenden* v. *Millhoff* (1900) 83 L. T. 11).

(5) DISCLOSURE OF SECRETS.

An engineer may be employed to do work of a confidential nature for others; or he may have to employ workmen to do confidential work for him (*e.g.* by employing a workman to make a model of a new and patentable machine). The law implies an agreement by the servant to keep his master's secrets, and he may be restrained by injunction from disclosing them (*Alperton Rubber Co.* v. *Manning and Others* (1917) 116 L. T. 499; 33 T. L. R. 205).

(6) RIGHTS OF AN ENGINEER TO THE INVENTIONS OF A SERVANT.

If a servant invents a machine in his employer's time and even when using his employer's materials, the invention does not become the property of the employer so as to prevent the servant taking out a patent for it (*Heald's Case*, 8 R. P. C. 430). But if the servant were to steal an idea and endeavour to patent it either before or after the termination of the service, the employer could oppose the grant of a patent in respect of it (*Thwaites' Application*, 9 R. P. C. 515). And if the servant obtains a grant of a patent in respect of the invention, he can only hold it as trustee for his employers (*British Reinforced Concrete Engineering Co.* v. *Lind* (1917) 33 T. L. R. 170). The engineer when employing a servant should take care to provide, by the agreement, that the inventions of the servant shall belong to him; but when himself entering into a contract of service he should be careful to sign nothing which will abrogate his rights.

(7) THE ENGINEER AS WITNESS.

(i.) **Generally.**—As the engineer frequently comes into contact with the law and the lawyers in the capacity of a *witness*, a few notes of a practical character on the duties of a witness may be found useful.

When invited to give evidence as an expert the engineer should: (*a*) arrange for a qualifying fee to be paid in any event; (*b*) examine the plans and documents, and, if necessary, obtain a view of the *locus in quo* before consenting to give evidence; (*c*) warn the parties consulting him of all the difficulties which are manifest to him; (*d*) ascertain, if possible, the contentions which are going to be put forward by the other side, and make himself familiar with any views which they may have published upon the matters in dispute. Having consented to give evidence, the engineer should (*a*) be ready to satisfy the lawyers with regard to any technical point which may be raised in consultation; (*b*) listen to the arguments and evidence in Court, and be prepared to deal with any points made by witnesses on the other side; (*c*) bring with him any entries in his diary or any notes which he may have made *at the time* with reference to the matters in dispute, for he may use such notes to refresh his memory. Thus, if an engineer has to report as to the condition of a structure, he may use in the witness box the notes which he made on the spot. But he could not use a copy of such notes unless he was prepared to produce the original; (*d*) make and bring to court a plan or model of the structure or machine under discussion, as this will save much wordy explanation; (*e*) search in libraries and in the published proceedings of the various institutions for any books and papers dealing with the matters in hand—especially any papers written by himself, in order that he may not be confronted with an adverse opinion given on a former occasion. A witness should bear in mind that it is his duty to state facts and express opinions, not to advocate the cause. As soon as he appears to be taking a biased view or attempting to fight the case it is more than likely that the tribunal will pay little heed to his evidence. A candid answer to an awkward question is often less dangerous than a long and evasive statement. In *Crosfield and Sons, Ltd.* v. *Techno-Chemical Laboratories Ltd.* (1913, 29 T. L. R. 378) Neville, J., expressed some views as to the function of an expert witness which may be thus summarised:—

(*a*) It is not competent in any action for witnesses to express their opinion upon any of the issues, whether of law or fact, which the Court or a jury has to determine.

(*b*) In patent cases, expert evidence is often necessary (i) to explain words or terms of science or art, or to inform the Court in case the import of a word in a document to be construed differs from its popular meaning; (ii) to instruct the Court in the laws of science; (iii) to describe the state of public knowledge at the date of the patent. 'It is rare,' added his lordship, 'to find any substantial difference of opinion between eminent experts upon matters of science whenever it is possible to dissociate the question from immediate connection with the issues in the action.'

(ii.) **Arrangements as to Fees.**—On being *subpœnaed* one is bound to attend as a witness provided sufficient conduct money is paid with the subpœna. Failure to attend might lead to an action for damages, while in the County Court the judge has power to fine a witness for non-attendance. A witness may refuse to give evidence until his expenses have been

paid (*Newton* v. *Harland* (1840) 1 M. & G. 956). It should be noticed that the person liable to pay the fees is the party to the litigation, not the solicitor, unless he has expressly made himself liable (*Lee* v. *Everest* (1857) 26 L. J. Ex. 334). In order to save trouble it is best to arrange for and if possible secure payment of fees in advance. The fees 'allowed on taxation,' that is to say, the fees which an unsuccessful litigant will be ordered to pay in respect of witnesses called by the other side, are very small, but the party calling the witness will have to pay his agreed fee whatever it is. Where a case is tried at assizes in the town where the engineer lives or in London (when the engineer lives in London), the fee allowed is 1*l.* 1*s.* *per diem*, or where the engineer lives away from the town 1*l.* 1*s.* to 3*l.* 3*s.* But the taxing masters may also allow a qualifying fee of so much a day varying with the magnitude of the case. In 1875 a fee of 7*l.* 7*s.* was not considered excessive payment to a scientific witness. It is probable that a larger sum would be allowed at the present day. In addition he is allowed the travelling expenses reasonably and actually paid. In County Courts the fee allowed for qualifying to give evidence on Scale B is 1*l.* 1*s.* to 3*l.* 3*s.*; on Scale C, 1*l.* 1*s.* to 5*l.* 5*s.*; for attending Court on trial (*per diem*), Scale B, 1*l.* 1*s.* to 2*l.* 2*s.*; Scale C, 1*l.* 1*s.* to 3*l.* 3*s.*

The above scales were fixed before the war, and have not been altered. So far as they can the taxing masters will probably grant the maximum owing to high prices.

(iii.) **Evidence before Parliamentary Committees.**—When an engineer is summoned to appear as a witness before a committee of either House of Parliament he looks for his fees to the party who summons him—whether he is the promoter or an opponent. According to May's 'Parliamentary Practice' (1906) p. 434, an engineer summoned as a witness before a committee should report himself to the committee clerk on his arrival in London, or he will not be allowed his expenses for residence prior to the day of making his report; 3*l.* 3*s.* a day is the parliamentary allowance, but of course as between the engineer and the party summoning him the *quantum* of the fee is a matter of agreement.

(8) LIABILITY FOR NEGLIGENCE.

The liability of an engineer for negligence is similar to that of any other professional man. To render him liable it is not enough that there has been a less degree of skill than some other professional man might have shown. A fair average degree of skill is all that can be insisted upon (*Lamphier* v. *Phipos* (1838) 8 Car. & P. 475). If an engineer were so careless in the preparation of a tender that the cost of the works was grossly underestimated, he would probably be liable to pay damages, and he certainly could not recover his fees (*Moneypenny* v. *Hartland* (1826) 2 C. & P. 379). An engineer should be acquainted with any local bye-laws appertaining to the work in hand (*Monks* v. *Dillon* (1884) 12 L. R. Ir. 321). On the principle of *respondeat superior*, an engineer is responsible for the acts or defaults of assistants employed by him (*Mackersy* v. *Ramsays* (1842) 9 C. & F. 818). The question 'To whom is the engineer liable for negligence?' depends upon the conditions of employment. If he is engineer in relation to a large contract, he is not liable for anything done by him in his capacity as arbitrator between the employer and the contractor; but he is liable for damage resulting from negligence in supervision, *e.g.*, failure to examine and test work before it is hidden from view (*Leicester Guardians* v. *Trollope* (1911) 75 J. P. 197). The contractor, however, could not sue the engineer for an error in the specifications on the basis of which he made his tender. Where, however, the 'quantities' are prepared by the engineer, and his fees for their preparation are paid by the contractor, it is conceived that he could be sued for damages resulting from an error in those quantities (*Bolt* v. *Thomas*, 'The Law Affecting Engineers,' p. 58). If an engineer order extras without authority, the contractor may possibly sue him for damages for breach of warranty of authority (*Randall* v. *Trimen* (1856) 18 C.B. 786). The amount of damages recoverable for negligence is to be measured by the loss which the plaintiff has suffered. In *Saunders* v. *Broadstairs Local Board* (1890) 2 H. B. C. 160, two engineers claimed fees for preparing surveys, obtaining tenders, and superintending the execution of a contract for certain drainage works. The board counterclaimed for negligence and were awarded 2,046*l.*, the amount overpaid to the contractor by reason of the negligence of the engineers in over-certifying the quantities; 2,400*l.* estimated cost of doing bad work over again; 240*l.* actual cost of repairs already done to defective work. Damages recovered by a workman against his employer for an accident might also be recovered from the engineer if it were shown that the accident was caused by the engineer's negligence (*Mosdell* v. *Mitchell*, 'Times,' January 20, 1891).

(9) RIGHTS, POWERS, AND DUTIES OF AN ENGINEER UNDER A CONTRACT FOR WORKS.

(i.) **Generally.**—Common forms of contract for large works usually set forth the powers, etc., of the engineer very clearly. The preparation of the contract is generally left to a lawyer, but the engineer should be on his guard against allowing a common printed form to be applied indiscriminately to the construction of works of a special nature. When difficulties arise it may be found that the common form does not provide for their elucidation.

(ii.) **Specifications.**—The contract should contain a clause providing that the employer will not be responsible for any mistakes or inaccuracies in the specifications which form part of the contract. Such a clause will, in the absence of fraud, afford him protection (*Pearson* v. *Dublin Corporation* (1907) A. C. 357). There should also be a clause calling the would-be contractor's attention to the fact that he must make surveys and borings for himself before making his tender. In that case, if unforeseen difficulties are met with the employer will not be responsible (*Thorn* v. *Mayor of London* (1876) L. R. 1 A.C. 120). Again, where a man undertakes to make *e.g.* sound hydraulic cylinders according to a specification, capable of withstanding a certain pressure, he takes upon himself the risk of the cylinders being able to withstand the named pressure when made according to the specification (*Hydraulic Co.* v. *Spencer* (1886) 2 T. L. R. 554). Carelessness in the preparation of a specification may involve the employer in added expense, for which the engineer may be liable (*Moneypenny* v. *Hartland* (1828) 2 C. & P. 378).

(iii.) **Plans and Drawings.**—These form part of the contract, and may, in the absence of a disclaiming clause, operate as a warranty that dimensions, measurements, etc., are correct. A clause disclaiming responsibility for accuracy should therefore be inserted in the contract. The engineer has no implied authority to alter plans and drawings so as to increase or lessen the burden imposed upon the contractor (*Cooper* v. *Langdon* (1841) 9 M. & W. 60). Delay in supplying the necessary plans and drawings may excuse the contractor from carrying out the work within the contract time (*Roberts* v. *Bury Commissioners* (1869) L. R. 5 C. P. 310). It is for the engineer who prepares them to see that they conform to local bye-laws (*Cubitt* v. *Smith* (1864) 11 L. T. 298). Plans and drawings made for the purpose of carrying out works are the property of the employer—not of the engineer (*Gibbon* v. *Pease* (1905) 1 K.B. 810). As to copyright in plans, see the Copyright Act, 1911, s. 5.

(iv.) **Extras.**—The duty of deciding what are 'extras' within the meaning of the contract is usually entrusted to the engineer. He has no implied authority to order extras (*Cooper* v. *Langdon* (1841) 9 M. & W. 60), although an agreement to work *to the satisfaction* of an engineer has been held to give him implied authority to order extras (*Robertson* v. *Jarvie* (1910) 45 S. L. R. 260). It has been held, however, that where the plaintiffs were employed by an architect (whose functions, for this purpose, correspond to those of an engineer) to supply door handles and door furniture for a building, and shortly before the delivery of the goods he informed the defendants, who were the builders, that the goods were being supplied by the plaintiffs, the defendants who used the goods in the building were liable to pay for them (*Ramsden* v. *Chessum* (1914) 78 J. P. 49). Extras should be authorised in the manner provided by the contract, which usually provides that all extras shall be ordered in writing (*Russell* v. *Saida Baudeira* (1862) 13 C. B. N. S. 149); but, in the case of a contract with a local authority, which must be under seal, the order for extras need not be under seal (*Williams* v. *Barmouth Urban Council* (1897) 77 L. T. 383). If a written certificate is rendered necessary by the contract, nothing else will suffice; but in a recent case a contractor proceeded with extra work as required by an engineer without any written order. On his claiming payment, an arbitrator found that the engineer was wrong and unreasonable, and that the work and materials required by him were not included in the contract. After much litigation the House of Lords eventually held that this decision was correct (*Brodie* v. *Cardiff Corporation* (1919) 120 L. T. 417). Although the whole scheme of works cannot be altered by virtue of the extra clause (*R.* v. *Peto*, 1 Y. & J. 37) power to *vary* the work will give the engineer authority to substitute one piece of machinery for another of which it is well known to be the mechanical equivalent (*Stevens* v. *Mewes & Davies*, Court of Appeal, June 7, 1910; Emden's 'Building Contracts,' Supplement 1911). The engineer should therefore see that he has power to *vary* the work. The engineer's decision as to what are extras may or may not be subject to revision in accordance with the arbitration clause. So far as the contractor is concerned, the fact that the engineer has given his final certificate (see *sub tit. Certificates, post*) may involve a final decision as to what is and what is not an extra (*Connor* v. *Belfast Water Commissioners* (1871) 5 L. R. Ir. C. L. 55). Power to extend the time for completion is generally given to the engineer. It may be that owing to the action of the Government acting under the Defence of the Realm Acts, the performance of a contract for works becomes illegal during the continuance of the War. In such a case it is conceived that no extension of time by the engineer will avail to keep the contract alive (see *Metropolitan Water Board* v. *Dick, Kerr & Co.*, 34 T. L. R. 111).

(v.) **Certificates.**—It is the duty of an engineer in relation to a contract for large works to grant or withhold the certificates upon which the contractor is paid. 'Progress certificates' are given from time to time as the work proceeds. They generally authorise payment to the extent of 80 per cent. upon the contract value of the work done from time to time. The remaining 20 per cent. is kept back usually, as to 15 per cent. until the final certificate is granted and as to 5 per cent. until the end of some stated period after the final certificate. This 20 per cent. is called the retention money (see *West Yorkshire Bank Ltd.* v. *Isherwood Bros.* (1912) 28 T. L. R. 593). The 'final certificate' is granted when the work is finished to the satisfaction of the engineer, and the contractor can sue for nothing until it has been granted (*Wallace* v. *Brandon and Byshottles U.D.C.* (1903) 2 H. B. C. 392). In deciding as to certificates the engineer acts as an arbitrator, and his decisions cannot be called in question (*Wadsworth* v. *Smith* (1871) L. R. 6 Q.B. 332); nor can he be compelled, and is better advised not to give reasons for his decision (*Stevenson* v. *Watson*

(1879) L. R. 4 C. P. D. 148). No action lies against him for negligence in relation to certificates (*ib.*) (but see the case of *Goddard* v. *Ferguson*, *infra*). In exercising his discretion he must act as an arbitrator between employer and contractor (*Chambers* v. *Goldthorpe* (1901) 1 K.B. 624). It may, however, be that the decision of the engineer as to a certificate is subject to the arbitration clause, which has been held to be the case in relation to the form of contract approved by the Royal Institute of British Architects (*Robins* v. *Goddard* (1905) 1 K.B. 294). A fraudulent agreement between the employer and the engineer would however affect the validity of a certificate (*Kellett* v. *New Mills U.D.C.* (1900) 2 H. B. C. 330). The certificate of the engineer may be made a condition precedent to the contractor's right of action against the employer. If so an action cannot be maintained unless there has been improper dealing between the engineer and the employer (*Eaglesham* v. *McMaster* (1920) 2 K. B. 169). The statement that an engineer is not liable for negligence in relation to a certificate does not appear to be supported by the case of *Goddard* v. *Ferguson* (referred to by Mr. Saxon Snell in a paper read by him at the Royal Institute of British Architects, April 15, 1912). It was there held that an architect who grants a certificate which is successfully challenged by an employer in an action brought against him by a builder to recover the amount so certified to be due to him, is liable to repay to the employer the costs incurred by him in resisting the builder's claim. The Official Referee held the architect liable on the ground that when acting as agent for the building owner in granting certificates, he must be held responsible for the reasonable and probable consequences of giving a certificate subsequently held to be inaccurate or excessive. Certificates should always be given in writing in a formal manner. No engineer should pass any contract which does not contain a provision to this effect. As to when an engineer is disqualified from granting certificates, see paragraph ix., *infra*. An engineer has no right to delegate the power to grant a certificate to a subordinate, *e.g.* to the resident engineer (*Freeman & Sons* v. *Middleton Electric Traction Co.*, per Hamilton, J., July 15, 1910; Court of Appeal, January, 1911; Emden's 'Building Contracts,' Supplement, 1911).

(vi.) **Penalties and Bonuses.**—The question whether the contractor must pay penalties or not depends almost entirely upon the engineer. Penalties are imposed if the work is not carried out in accordance with the provisions of the contract as to time. The fact that extras are ordered will excuse the contractor from compliance with the conditions as to time (*Dodd* v. *Churton* (1897) 1 Q.B. 562), but the jurisdiction of the engineer does not extend to deciding this point (*Gallivan* v. *Killarney Urban Council* (1912) 2 Ir. R. 356). To avoid complication, however, it is best to insert a clause in the contract giving the engineer power to extend the time. With regard to the penalty clause generally nice questions sometimes arise as to whether it is capable of enforcement. If it is unreasonable on the ground that the punishment is too severe, the Court may refuse to enforce it; and it is for this reason that the words 'the sums to be regarded as liquidated damages and not as a penalty' are usually inserted after the sum named. The breach of an engineering contract involves damage of so uncertain a nature that the courts will generally treat the sum which the parties themselves have named as liquidated damages. Penalties will not be imposed if (*a*) the delay was caused by the act or default of the employer (*Thornhill* v. *Neats* (1860) 8 C. B. N. S. 831); (*b*) if there is delay in setting out the site or in delivering plans (*Roberts* v. *Bury Improvement Commissioners* (1870) L. R. 5 C. P. 310). The fact that the contractor has been delayed by a strike of his workmen, however, affords no excuse for delay (*Budgett* v. *Rimington & Co.* (1890) 25 Q.B. D. 320), nor, apparently, is the employer to be held liable for the acts of third parties causing delay (*Porter* v. *Tottenham U.C.* (1915) 1 K. B. 776). While a penalty clause is usually optional, it is obligatory in contracts made with certain local authorities (see Public Health Act (1875) s. 174 (2)). A bonus clause is sometimes inserted by virtue of which a contractor is enabled so much extra for expedition. If a contract contains such a clause, and the engineer has power to extend the time, it seems that his power to extend must be exercised not to enable the contractor to earn the bonus but only to avoid penalties (*Ware* v. *Lyttelton Harbour Board* (1882) 1 N. Z. R. 191).

(vii.) **Defects, Repairs, and Maintenance.**—In contracts relating to the erection of machinery clauses should be inserted providing for any defects which appear either before or after the issue of the final certificate. A useful form of this clause will be found in the printed conditions of contract issued by the Institution of Electrical Engineers. Its importance will be recognised when it is pointed out that when it is absent the final certificate of the engineer may possibly bar any further claim by the employer (*Lord Bateman* v. *Thompson* (1875) 2 H. B. C. 23). The time for the remedy of defects should be prolonged as much as possible, inasmuch as the very fact of there being a defects clause impliedly releases the contractor from liability for defects appearing after the stated time (*Sharp* v. *Great Western Railway Co.* (1841) 11 L. J. Ex. 17). The contractor can only be called upon to repair and maintain when compelled to do so by a repairing clause and a maintenance clause. The employer must give notice of want of repair. Maintenance should be provided for in cases where the machinery or work is of a special character which is best known to the contractor. In the absence of any clause relating to defects and maintenance, it does not follow that on payment of the full price the employer thereby waives all claim for damages for work improperly done. It is a question of fact having regard to all the circumstances of the case whether the settlement was final and complete.

(viii.) **Subcontractors and Specialists.**—Most modern contracts involve the employment of subcontractors or specialists. A clause should be inserted in the head contract prohibiting the employment of any subcontractor or specialist unless approved by the engineer. In the absence

of such a clause, the contractor might, possibly, farm out portions of the work to all and sundry (*British Waggon Co.* v. *Lea* (1880) 5 Q. B. D. 149). The engineer, apparently, has no implied authority to employ subcontractors (*Cowan* v. *Goderich* (1859) 10 U. C. C. P. 87). It is often well to specify the names of manufacturers of the raw materials which are to be used in the body of the contract. In order to preserve his right to supervise the work of any subcontractor or specialist, the engineer should secure the insertion in the head contract of a clause providing that payments to subcontractors shall be made by the contractor only upon the certificates of the engineer. It is also well to secure the insertion of a clause to the effect that, in the event of the contractor becoming insolvent, the subcontractor or specialist may be paid directly by the employer. All responsibility for delay or negligence on the part of a subcontractor is generally, by a special clause, placed upon the contractor (see, generally, *Crittall* v. *L. C. C.* (1910) 75 J. P. 203; *Hampton* v. *Glamorgan County Council* (1916) 115 L. T. R. 426, and the case of *Ramsden* v. *Chessum*, cited, *supra*, s. (9) (v.) (i.)).

(ix.) **Whether the Employer's Engineer can act as Arbitrator in Relation to Disputes Arising under a Contract.**—The position of the engineer in relation to a large contract has been rendered very difficult by certain recent decisions of the courts. In theory he is agent for the employer in the preparation of plans and specifications and in the supervision of the work, and he is arbitrator or *quasi* arbitrator between the employer and contractor in granting certificates, approving work done, granting extensions of time, etc. Sometimes he is made sole arbitrator to decide all disputes between the employer and the contractor, and a serious question has arisen in several recent cases as to how far a contractor can be forced to accept his decision whether in the matter of granting certificates, etc., or as arbitrator. In *Jackson* v. *Barry Railway Co.* (1893) 1 Ch. 238, it was held that the fact of the employer's engineer having expressed a view of the matter in dispute hostile to the contractor was not a sufficient justification for refusing to allow him to act as arbitrator in relation to that dispute. The Court of Appeal pointed out that the contractors had agreed to this form of reference, and what they relied on was the professional honour, practice, and intelligence of the engineer. There was nothing in the circumstances of the case to render it improbable that he would not arrive at an honest decision.

In *Ives* v. *Willans* (1894, 2 Ch. 478) the same result was arrived at, although in giving judgment Lord Justice Davey said: 'No doubt in a sense the engineer will be the judge of his own conduct, and no doubt that is a position which *prima facie* raises some surprise in a judicial mind; but that is the contract of the parties.' As already indicated later decisions appear to have practically altered the law, although the judges always took care to say that *Ives* v. *Willans* is still good law. The result of the recent cases may be thus summarised. A contractor will not be bound to submit to the arbitrament of the employer's engineer (*a*) if the matter in issue is an unseemly personal dispute raising a vindictive feeling between the engineer and the contractor, and the engineer has so strongly expressed his view that it amounts to a prejudgment (*Nuttall* v. *Manchester Corporation* (1893) 8 T. L. R. 513); (*b*) if the nature of the dispute is such that the cross-examination of the engineer is essential (*Freeman* v. *Chester R. D. C.* (1911) 1 K.B. 783); (*c*) if the matter in issue is something outside the original agreement, e.g., a dispute as to whether the engineer and contractor had agreed to vary the original agreement (*Aird* v. *Bristol Corporation* (1913) A. C. 241); (*d*) if the conduct of the engineer himself is practically the only point in dispute (*Blackwell* v. *Derby Corporation* (1909) 75 J. P. 129). Finally, it was held in *Roberts* v. *Hickman* (1913) A. C. 229 that the grant of a certificate by an architect will not be a condition precedent to the builder's right to sue if, by writing letters to the employer, and allowing himself to be influenced by their views, the architect has disqualified himself from acting judicially although guilty of no fraud or improper conduct.

If the method of arbitration provided for in many existing forms of contract is declared to be impossible for any of the above reasons the only alternative is litigation, a method of settling disputes which employers, contractors, and engineers always seem to desire to avoid. What can be done to avoid such a calamity in relation to future contracts? The following courses are open. Insert a clause (*a*) referring all disputes to an arbitrator to be mutually agreed or to be nominated by the president of one of the Institutes (see, *e.g.*, the form approved by the Institution of Electrical Engineers, clause 47); or (*b*) referring disputes to a *named* arbitrator; or (*c*) refer disputes to the president of one of the Institutes whoever he may be. The first of these suggestions appears to offer the best practical solution of the difficulty. The writer, however, ventures to make the following suggestion which might possibly work in practice. Frame the arbitration clause so as to leave the decision of every question arising under the contract to the engineer but subject to a proviso that should it appear to the Court on an application to stay any action or proceeding brought in relation to the contract that the engineer is disqualified from acting as arbitrator the matters in dispute will be referred to an engineer to be agreed between the parties, or failing agreement to be nominated by the president of the Institution of Civil Engineers. Another alternative is suggested by Mr. E. J. Rimmer in his Arbitration Clause in 'Engineering Contracts' (Constable & Co.). It is that a proviso should be added to the arbitration clause stating that if any action is brought the defendant may either (*a*) have the action stayed or (*b*) have the matter referred to an independent arbitrator appointed in one or other of the manners above suggested.

(10) THE ENGINEER AS ARBITRATOR.*

(i.) **Generally.**—Engineers are frequently called upon to act as arbitrators to decide disputes of a technical nature wholly unconnected with any contract in which the engineer has been employed. Arbitrations are conducted in accordance with the provisions of the Arbitration Act, 1883. An arbitrator may be appointed (*a*) by the Court; (*b*) by the terms of the contract itself; or (*c*) by a person nominated for that purpose in the contract, e.g., the president of the Institution of Civil Engineers. As to appointment by the Court, this occasionally takes place when a prolonged scientific enquiry is necessary. A 'referee' of this kind may have power either to wholly dispose of the dispute or make an enquiry and report to the Court, in which latter case the Court itself gives the judgment after considering the report. As references of this kind are rare in practice they will not be further dealt with.

As a rule the engineer finds himself made arbitrator pursuant to the arbitration clause in an agreement. Any dispute arising under any contract may thus be referred, and the parties will as a general rule be bound to proceed to arbitration for the purpose of settling a dispute instead of bringing an action at law. Where a contract contains an arbitration clause, if one party commences an action, the other may apply to the Court to stay the action pending a reference, and the action will generally be stayed, unless there is some good reason for not allowing the arbitration to proceed. For instance, a contractor alleged that he had been induced by fraudulent representations as to the nature of the soil to enter into a particular sewerage contract, and brought an action for fraudulent misrepresentation. It was held that such an action ought not to be sent to arbitration (*Monro* v. *Bognor Urban District Council* (1915) 84 L. J. K. B. 1091). But the Court will stay an action, and allow an arbitration to proceed, although the question at issue is one which might be more suitably tried in a court of law—*e.g.* the construction of a contract (*Metropolitan Tunnel, etc.*, v. *London Electric Ry.* (1926) 1 Ch. 371). (See also par. 9 (ix.), *supra*.)

(ii.) **Who may be Arbitrator.**—Any person upon whom the parties agree may act as arbitrator. Thus if in a contract with a local authority it is specified that the borough engineer shall be arbitrator there is no objection to his so acting, and the contractor can be compelled to accept his award (*Ives* v. *Willans*, *supra*, 2 Ch. 478). That is the broad rule, but the Court, as has been shown, always has power to refuse to enforce the arbitration clause in a case where it is obvious that the arbitrator ought not to act owing to bias, etc. Frequently the clause provides for the appointment of an arbitrator by each party with power to the arbitrators in case of differences to appoint an umpire.† When this is done the umpire and arbitrators generally find it convenient to sit together to decide the case.

(iii.) **Proceedings before the Arbitrator.**—Witnesses must be sworn in the usual way. Due notice must be given to either party of the time when it is proposed to hold the sitting. In a case of great complexity it is well to have a preliminary sitting in order to require the parties to formulate the points in difference between them. By this means the issues may be considerably narrowed. The arbitrator should hear both sides, who may be represented by solicitor or counsel. He must never hold private communication with one party on the subject matter of the reference. He has no power to examine witnesses in the absence of the parties. An arbitrator who takes evidence in the absence of, and without notice to, a tenant, is guilty of legal misconduct, for which his award may be set aside (*In re O'Connor*, 88 L. J. K. B. 1242; *Ramsden* v. *Jacobs* (1922) 1 K.B. 640). It is for the arbitrator to decide every question whether of law or fact. He may, however, consult a solicitor or, for that matter, any other expert on legal points. Indeed the writer was recently concerned in a case in which, with the consent of the parties, the arbitrator had his solicitor sitting by his side. He should take full notes of the evidence, unless, as is frequently the case, the parties agree to have a shorthand note.

(iv.) **The Award.**—If the case is complex, the engineer is well advised to consult his solicitor as to the form of the award. The following are the features of a valid award, which should be in writing. (*a*) *It must not exceed the submission.* Thus if the question submitted was whether the stone used for building a dam was according to specification, the arbitrator would have no power to award that the stone used was better than as specified and that the employer must pay for it. (*b*) *It must extend to all matters in dispute.* If several distinct questions are referred and the arbitrator omits to decide upon any one of them, the whole award is bad. If in the case mentioned

---

* The Institute of Arbitrators, 28 Bedford Square, London, W.C. 1, and the Court of Arbitration of the London Iron and Steel Exchange, 28 Essex Street, Strand, London, W.C. 2, are always willing to suggest the names of qualified arbitrators with special technical knowledge. Reference to either of these bodies, therefore, tends to ensure that the arbitrator has the requisite technical knowledge as well as the legal knowledge and experience, which are so essential to the satisfactory conduct of an arbitration.

† An engineer who is appointed by one party to a dispute, the other party appointing another arbitrator, will usually find it advisable to nominate a Fellow of the Institute of Arbitrators with special knowledge of the subject in dispute as the umpire.

he were asked to decide (i.) was the stone as specified ; (ii.) was it suitable for the purpose, he would have to decide both questions. (*c*) *It must be certain.* The award must be so framed that its meaning is clear to the parties ; but according to a legal maxim, that is certain which is capable of being rendered certain. Thus to award that A. and B. shall pay a debt in the proportion in which A. and B. hold shares in a ship is a sufficiently definite award (*Watson* v. *Watson* (1671) Sty. 28). (*d*) *It must be final.* The award must peremptorily state the arbitrator's finding. A mere expression of opinion that A. ought to pay B. such sum as C. thinks proper would not be a final award. Finally (*e*) *it must not be impossible, unreasonable, inconsistent,* or *illegal.* An award which directed that one party should do something illegal, *e.g.* trespass upon the land of another, would be invalid.

(v.) **Time for Making an Award.**—In the absence of agreement, an award must be made (*a*) within three months of the date of entering on the reference, or (*b*) within three months after having been called on to act by notice in writing from one of the parties, or (*c*) on or before any later day to which the arbitrator may by writing under his hand extend the time for making the award (Arbitration Act, 1889, Schedule 1 (c)). The power to extend the time is important in the case of a long and complicated reference, but it must be exercised within the period of three months above referred to. Unless the arbitrator extends the time, or the parties agree to allow him to do so, an application must be made to the Court to extend the time ; and such an application may be made whether the time for making the award has expired or not (*ib.*, s. 9).

(vi.) **The Fees of an Arbitrator.**—The arbitrator should name his fee beforehand. It is best fixed at so much *per diem* for the hearing. If no preliminary arrangement is made, he is entitled to charge what is fair and reasonable, and he can keep back his award until the fees are paid (*Ponsford* v. *Swain*, J. & H. 433). When deciding in the award as to costs he should state who is to pay his fees ; but he may insist upon their being paid by the party who takes up the award. He should not reveal his decision until after these formalities have been observed. Sometimes the *quantum* of an arbitrator's fees is left to the decision of a taxing master. On such a taxation, if the evidence goes to show that, in the opinion of persons in the same profession as the arbitrator, that his charges are fair and reasonable, the taxing master must allow them. In *Mason* v. *Lovatt* (1907, 23 T. L. R. 486) a quantity surveyor was allowed to charge 527*l.* 10*s.* for a reference which lasted 22 days, the arbitrator subsequently taking 13 days to consider his opinion. In *Llandrindod Wells Water Co.* v. *Hawksley* (1904, 20 T. L. R. 241) the fees of an umpire and two arbitrators all being engineers were allowed at 476*l.* 12*s.*, in respect of a reference which lasted 2 days. The arbitrator should state how his fees are made up.

(vii.) **Stating a Case for the Opinion of the Court.**—Where a specific question is referred to an arbitrator, and he decides it, the fact that his decision is erroneous in point of law does not make the award bad on its face so as to admit of its being set aside (*In re King and Duveen* (1913) 2 K. B. 32) ; nor will the mere fact that the dispute involves questions of law induce the Court to stay a reference to arbitration (*Rowe* v. *Crossley* (1913) 108 L. T. 11). But where a point of law arises during a reference, the arbitrator, if he does not care to decide it, may state a case for the opinion of the Court (Arbitration Act, 1889, s. 19), or the Court may compel him to state a case (*ib.*). He can also make his award in the form of a special case, leaving it to the Court to decide how the judgment shall be entered. Refusal to state a case may constitute misconduct on the part of an arbitrator (*Czarnikow* v. *Roth Schmidt & Co.* (1922) 38 T. L. R. 595).

(viii.) **Costs of a Reference.***—On the question of costs, and as to the form of that part of his award which relates to costs, an arbitrator should consult his solicitor, because (*a*) highly technical questions arise with regard to costs and their taxation which no layman can be expected to understand, and (*b*) it is desirable that in this matter an arbitrator should observe as nearly as possible the principles which are recognised in courts of justice. Again, the defendant in the proceedings may have preferred a counter-claim in which he may have been successful wholly or in part. This further complicates the question of costs. It may be observed that there is no objection to an arbitrator employing a lawyer to advise him in the matter of costs or with reference to any other question which arises in a reference. The importance of taking this course is that a layman has not, and in the nature of things cannot have, the same mental outlook as a court of justice in dealing with the question of costs. Let a recent example suffice. An arbitrator made an award of £200, but as the plaintiff had claimed £1,200 and only succeeded as to £200, he made each side pay its own costs. In the result the plaintiff got less than nothing at all, as his costs far exceeded £200. Such a decision as to costs was, from a lawyer's view-point, entirely wrong. The arbitrator wholly overlooked the fact that the plaintiff was compelled to sue to recover

* The Court of Arbitration of the London Iron and Steel Exchange, 28 Essex Street, Strand, London, W.C. 2, has a simplified Scheme of Arbitration whereby if the necessary documents are placed before the arbitrator and no hearing is required, a fee of only two to five guineas is charged. This scheme is only applicable to simple cases ; further particulars can be obtained from the Secretary of the Exchange.

anything at all. In any event, and whether a solicitor is employed or not, the question of costs should be carefully considered, as the expenses of a reference are often wholly out of proportion to the sum in dispute. In litigation the general rule is 'costs follow the event'—an application of the maxim *vœ victis*. Unless the submission contains some special term as to costs, the arbitrator has full power to decide who is to pay them. He may award costs to be taxed 'as between solicitor and client'—which means that the successful party is nearly indemnified—or he may award costs to be taxed as between litigants in the ordinary way. He may also, in his discretion, award a fixed sum for costs (Arbitration Act, 1889, Schedule I. (i)), or he may declare that each party shall pay his own costs. In deciding whether to make a defendant pay costs, the arbitrator should (it is submitted) consider the nature of any offer made before the proceedings were commenced. Thus supposing a contractor claimed 5,000*l.* and the building owner made a firm offer of 4,000*l.* which was refused. After hearing the evidence, the arbitrator awards 4,000*l.* or less. In such a case he would be justified in disallowing the contractor any costs. An arbitrator can only make a successful plaintiff pay costs in exceptional circumstances as when (*e.g.*) it appears that although successful he should not have prosecuted his claim. It was held, however, in a case arising under the Small Holdings Act, 1908, the arbitrator in the exercise of his powers under that Act had wider powers in the matter of costs than a High Court judge (*Gray* v. *Baron Ashburton*, 115 L. T. R. 729).

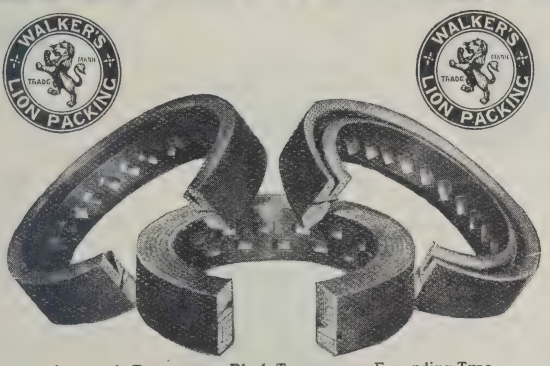
(11) NOTES ON SALE AND PURCHASE OF MACHINERY.

The question which most frequently arises on the sale of machinery, motor cars, etc., is what is the warranty of fitness? Sometimes, of course, there is an express written warranty, but in the absence of that the matter is regulated by section 14 of the Sale of Goods Act, 1893, which provides (in effect) that there is no implied warranty or condition as to the quality or fitness for any particular purpose of goods supplied under a contract of sale, except (1) where the buyer expressly makes known the purpose for which the goods are required so as to show that the buyer relies on the seller's skill or judgment and the goods are of a description which it is in the course of the seller's business to supply (whether he be the manufacturer or not) there is an implied condition that the goods shall be reasonably fit for such purpose; provided that in the case of the sale of a specified article under its patent or other trade name, there is no implied condition as to its fitness for any particular purpose; (2) where the goods are bought by description from a seller who deals in goods of that description (whether he be the manufacturer or not) there is an implied condition that the goods shall be of merchantable quality; provided that if the buyer has examined the goods there shall be no implied condition as regards defects which such examination ought to have revealed; (3) an implied warranty or condition as to quality or fitness for a particular purpose may be annexed by the usage of trade; (4) an express warranty or condition does not negative a warranty or condition implied by the Sale of Goods Act unless inconsistent therewith. The result of these provisions is that in certain cases the vendor is made liable for latent defects, that is to say, for defects which would not be discoverable on an ordinary examination of the article. Sometimes a contract for the sale of machinery contains provisions for payment of the price by instalments after production of the certificate of the engineer that such instalments are due and payable. It was held in a Scotch case (*Howden* v. *Powell Duffryn Steam Coal Co.* (1912), S. C. 920, Ct. of Sess.) that where, in such circumstances, part of the machinery was rejected, but the manufacturers refused to accept the rejection, the production of the engineer's certificate was not a condition precedent to the right to recover payment.

In the case of an engine which is attached to the freehold, a question may arise, on a sale, as to when the property passes. In *Underwood Ltd.* v. *Burgh Castle Brick and Cement Syndicate* (1922, 1 K. B. 343), a horizontal condensing engine was sold at a price free on rail, London. It was bolted to and embedded in a flooring of concrete. Before it could be delivered on rail it had to be detached and dismantled. The sellers detached it, but in loading it on a truck they damaged it by accident so that the buyers refused to accept it. In an action by the sellers for goods bargained and sold it was held that the property in the engine had not passed to the buyers, inasmuch as the sellers had failed to do something necessary to put it in a deliverable state. Consequently the loss fell on the sellers and not on the buyers.

PATENT

"LION" PACKING



Automatic Type Block Type Expanding Type

Reliable Packing for all Pressures

"LION" AUTOMATIC Packing for all Hydraulic work. It functions like a ram leather, and will stand the highest pressures.

"LION" BLOCK Packing for general purposes and for use under solid metal rings.

"LION" EXPANDING Packing for High Pressure and Superheat Steam work. Automatic in action, and a very popular type of Packing.

The KING of PACKINGS

JOINTING A SPECIALITY

Write for Catalogue No. 161.

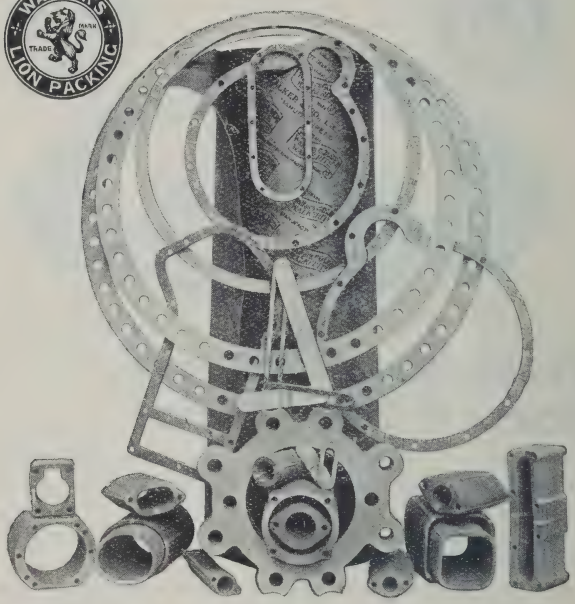
JAMES WALKER & CO. LIMITED

"LION" WORKS, WOKING, Surrey.

'Phone : Woking 1040. 'Grams : "Lioncello."

LONDON OFFICE : 96 Leadenhall Street, E.C.3.

WALKER'S
"GOLDEN WALKERITE"
JOINTING



The best Compressed Asbestos Sheet Jointing obtainable for Superheat Steam, Hydraulic, Gas, Air, etc. Millions of "Golden Walkerite" Joints are in use throughout the world. Supplied in Sheets of any size and thickness.

Joints cut to any shape.

JAMES WALKER & CO. LTD.

"LION" WORKS, WOKING, Surrey.

'Phone: Woking 1040. 'Grams: "Lioncello."

LONDON OFFICE: 96 Leadenhall Street, E.C.3.

INDEX

ABD

A.B.D.C.M., chain gearing, 899
A. Abney reflecting level, 68
Abrasive materials, 1103
——— wheels and blocks, 435, 1105
———, ———, dressers, 1113
———, ——— dust collection, 1113
———, ——— for wear, device of adjusting,
——— ——— guards for, 1112 [926
——— ——— lubrication of, 1107
——— ——— mounting, 1108
——— ——— rests for, 1113
——— ——— selection of, 1107
——— ——— speed of, 1108
——— ——— storage of, 1109
Absolute strength of materials, 182
——— zero of temperature, 107
Absorption dynamometer, 37
———, losses by, in reservoirs, 1387
———, ——— rainfall, 1402
Absorptive powers of coloured glass, 1500
A.C. electric welding system, 2786
Acceleration, 28
——— ——— train, 587
Accumulated work, 29
Accumulators, electric, 2256
——— ——— acid for, 2257
——— ——— battery rooms, 2259
——— ——— charging, 2257
——— ——— discharging, 2257
——— ——— Edison nickel-iron, 2260
——— ——— Hart, 2256
——— ——— hydraulic leathers for, 1265
——— ——— steam, 1853
Accuracy, cost of unnecessary, 779
A.C.E.C. ia Cour motor converters, 2275
Acetylene, 699, 1491. 1493, 2171, 2173, 2483, 2485, 2759, 2763
——— blow-pipe welding, 2767
———, dissolved, 699, 1491, 2485, 2763
——— for buoys, 1491
——— internal combustion engines, 2171
——— lighthouses, 1493
——— lighting, 2759
——— purification of, 2764
——— rail welding, 699
——— supply, 2763
——— welding, 2759
Acid, carbonic, refrigerating machine, 1213
——— for electric accumulators, 2257
Acid-resisting alloys, 2728
——— ——— iron castings, 2690
Activated sludge process, 1376
Acts, London Building, 370
Adams, Prof. H., on tall chimney design, 522

AER

A.D.C. aero engines, 2052
Adhesion of locomotives, 588
Adhesive strength of mortars, 175
Adiabatic expansion, gas, 1863
——— ——— heat-drop, 1777-1780
Adie's cement-testing machines, 402
Administration of roads and streets, 711
Admiralty coefficients, 2365
——— ——— gun-metal, 2731
——— ——— oil fuel for Diesel engines, 2166
——— ——— pier, Dover, 1456
——— ——— specification for oil fuel, 2166
——— ——— tests for iron, 188
——— ——— steel, 186
——— ——— rivets, 186
——— ——— white metal, 2737 [gravity, 126
Adulteration of metals, to discover, by specific
A.E.G. pulverised fuel locomotives, 622
Aerial ropeways, 834
———, ———, cost of, 2839
———, ———, limitations of types of, 843
Aerodynamics, 1988
——— ——— wind channels, 1988
Aero engines, 2045
——— ——— A.D.C., 2052
——— ——— Armstrong-Siddeley, 2052, 2057
——— ——— Bristol Jupiter, 2063
——— ——— carburation, 2068
——— ——— Cirrus, 2062
——— ——— Condor, 2048
——— ——— Eagle, 2047
——— ——— Genet, 2058
——— ——— Isotta Fraschini, Italian, 206
——— ——— Jaguar, 2052
——— ——— Lion, 2048
——— ——— Napier, 2048
——— ——— Nimbus, 2062
——— ——— Packard, American, 2066
——— ——— radiators for, 2070
——— ——— recent progress, 2073A
——— ——— Rolls-Royce, 2045
——— ——— tables of details, 2071
——— ——— water cooling, 2069
——— ——— weight of, 2070
——— ——— Wright Cyclone, 2060
——— ——— Whirlwind, 2058
Aerofoils, 1990
——— airscrews for, 2090
———, ———, characteristics of, 2002
——— ——— curves, 2017
——— ——— diameter required, 2006
——— ——— dynamic similarity and scale effect, 2011
——— ——— form of blades, 2006

AER

Aerofoils, airscrews, horse-power, 2006, 2018,
——— manufacture, 2007 [2020
——— materials for, 2007
——— number of blades, 2007
——— speed, 2007
——— theories, 2003
——— biplane, 1995
——— factors affecting, 1995
——— pressure distribution over, 1997
——— slotted, 1993
——— triplane, 1995
Aeroplane components, 1984
——— performance, 2015, 2040, 2041 [2016
——— at different altitudes,
———, curves, 2017
——— standard atmosphere,
Aeroplanes and seaplanes, 1983 [2015
———, geometrical particulars of, 2037–
Afforestation, 2889 [2039
Air and oxygen required for combustion of gas,
—— blast, cupola, 2662 [2170, 2475
—— blowers and exhausters, 2520
——, compressed, horse-power in, 1129 [1129
——— driving power-hammers by,
——— mining drilling by, 2568
——— water raising by, 1276
—— compression, 1187
—— compressors, 1190
——— Belliss high-speed forced lubrication for, 1193
——— Brotherhood's, 1190, 1193
——— cooling water for, 1200
——— design of, 1190
——— Hardy, 2568
——— Holland, 1198
——— Holman, 1196, 1197
——— lubrication of, 1199
——— packing for, 1199
——— Weir, 2439
——— Westinghouse, 1193
——— constant volume efficiency
—— coolers, Heenan, 2267 [of, 1867
—— cooling, 2514
——— machines, 2267
—— density, 1181
—— discharge nozzles, 2665
——, distribution of, for ventilation, 2515
—— ducts, 2519
—— efficiency, 1867
—— ejector, Mirrlees, 1846
—— engines, hot-, 1931
——, expansion of, by heat, 108, 1190
—— filtering, 2514
—— filters, Heenan, 2268
——— 'Ventex' 2269
——— 'Visco,' 2268
——, flow of, through ducts, 2515
——— nozzles, 1184
——— pipes, 1082
—— friction, 1184
——, gas, petrol, 2487
—— heaters for boilers, 1648, 2393
—— lift pumps, 1276
——, liquid, 1182
—— mains, 1081
—— metering, 1435
——, moisture in, 1865
——, movement of, by fans, 2518
——— caused by difference of temperature, 2519

ALL

Air passing through nozzle, weight of, 1184
—— petrol gas, 2487
—— pipes, 2517
—— preheating, 1648, 2393
—— pressure, forge, 2665
—— propellers, 2520
—— pump ventilator, Boyle's, 2523
——— water, measurement of, 1849
—— pumps, 1840
———, Bailey, 1200
——— circulating, 1842
——— Dexine valves for, 1264
——— Edwards', 1842
——— ejector or jet, 1845
——— Hick-Breguet, 1845
——— Leblanc, 1844
——— Mirrlees ejector, 1846
——— Paragon, 2408
——— Radojet, 1848
——— Weir ejector, 1849
——— paragon, 2408
—— receivers, mine drill, 2571
—— regulators, mining, 2616
—— required for combustion, 2078
——— of coal, 1181
——— gas, 2169, 2475
——— oil fuel, 2166
——— mine drilling, 2564
——— ventilation, 2513
—— resistance of, to plane surface, 1183
——— train, 582
——, saturated, vapour in, 1866
—— ships, 1985
——, velocity and pressure of, 1183
—— velocity and volume of, 2664
—— velocity of sound through, 117
—— washing, 2514
—— weight of, 1181
——, work stored up in, 1085
Aircraft, 1983, 2043A
——— aeroplanes and seaplanes, 1983
——— airship drag, 2043
——— airships, non-rigid, 1985
——— rigid, 1986
——— semi-rigid, 1986
——— and balloons, 1985
——— balloons, 1985
——— kite, 1985
——— design, 2023
——— Armstrong, Whitworth, 2023
——— Blackburn Aeroplane, 2026
——— Boulton & Paul, 2026
——— Bristol Aeroplane Co., 2029
——— Calcutta flying boat, 2035
——— Gloster Aircraft Co., 2031
——— Short Bros., 2034
——— Vickers, 2036
——— flying boats, 2012
——— water performance, 2023
Alcohol, 2155
——— reduction of volume under pressure, 117
——— water mixtures, calorific value of, 2155
Alden power brake, 38
Alderney breakwater, 1456
Alidade, Zeiss sectioning, 72
Alignment of crankshafts and engine parts,
——— pins, 1741 [1905
Allen oil engines, 1899, 1911
Allen's link motion valve gear, 1764
Allen West lifting magnets, 824

ALL

Alloy Welding processes, A.W.P. electrodes, 2784
——————— Daysohm's Arc Welding machines, 2784
Alloys, 2678
Alloys, acid-resisting, 2729
——- ferro, 2693
———, general, 2744
——- heat-resisting, 2729
——- high temperature, effect of, on, 2747
——- mechanical treatment of, 2757
——- melting-points of, 104
——- non-ferrous, 2721
——- non-magnetic, 2743
——- resistivities of, 2204
——- tensile strength of, 170
Alluvial mining, 2564, 2642, 2646
Alternating-current commutator motors, 2294
——————— generators, 2263
——————— motors, 2286
——————— traction motors, control of, 683
——————— turbo-generators, 2263
Alumina, melting-point of, 430
Aluminium, 2719
——————— alloys, 2730, 2732, 2734, 2735
——————, annealing sheet, 2720
——————— bar and strip, weight and electrical
——————— brass, 2730 [value of, 139
——————— bronze, 2732
——————— solder for, 2751
——————— cables, steel-cored, 2219
——————— conductors, 2213
———————, corrosion of, 2720
——————— flat bars, 139
———————, impurities in, 2719
——————— machining, 1094
——————— overhead lines, 2220
——————— paint, 418
———————, sheet, annealing, 2720
———————, weight of, 140
——————— soldering, 2751
——————— strip, 139
——————— tubes, weight of, 159 [2218
——————— v. copper for electrical conductors,
——————— welding, 2774, 2793B
——————— working, 2815
Amatols, explosive, 2798
American canal and river locks, 1523
——————— locomotives, 576
——————— iron castings, 2689
——————— Portland cement, 405
——————— railway electrification, 675
——————— rails, 533
——————— screw threads, 1139, 1145, 1146
——————— sewers, 1360
——————— specification for waterproofing material for Portland cement concretes, 405
——————— standard gas pipe threads, 1152
——————— nuts and bolts, 1153
——————— pipe flanges, 1168
——————— screw threads, 1145
——————— standards for cast-iron, 2689
——————— steel sheet piling, 506
——————— turbines (water), 1341
——————— wire gauge, 15 & 16
Ammeters, Weston standard, 2312
Ammonia absorption machines, 1217
———————, compression, 1211
———————, properties of anhydrous, 1211

ARE

Ammonia, recovery, 2099
———————, sulphate of, 2473
Ammoniacal liquor, gas, 2463, 2465, 2474
Amsterdam ship canal, 1528
Anchorage of overhead structures, 2349
Anderson, Thomas, Ltd., cork insulation, 1222
Anderson's Rok roofing, 316, 326, 327
Anemometer, hot wire, 1445
——————— Robinson, 1184
Aneroid, Steward Hypsometric, 69
Angle and tee iron, Henderson & Glass', 141
——————— steel ——————— 142
——— iron, weight of, 141
——— measurement by foot rule, 8
——— steel, weight of, 142
———, twisting, of shafts, 1005
Angles and leads of railway crossings, 557
——————— lengths of slopes, 492
——— British standard, 238
——— bulb, British standard, 239
——— equal, British standard, 238
——— of repose of soil, 494
——————— slopes, 493
———, roof, 316, 365
——— tangential, for chords, 563
——— trisection of, 92
———, unequal, British standard, 238
Angold magazine flame arc lamp, 2326
Angular contact slabs for boilers, 1607
Angus Smith's composition, 2808
Anhydrous ammonia, properties of, 1211 [1211
——————— refrigerating machine,
Animal power, 44
Animals, work done by, 45
Annealed and bright iron wire, sizes, weights, and breaking stresses of, 176
Annealing connecting rod bolts, 1747
——————— wrought-iron chains, 199
Annison's side-groove steel piling, 505
Annual compass variation, 74
Anthracite, 2103
——————— wastes, 2095
Anti-corrosion paint, Carson's, 2802
Anti-friction bearings, 923
——————— Delta metal, 2739
——————— Fry's metal, 2739
——————— metallic packing, 1750
Anti-rust paint, 2802
Apothecaries' measures, 7
Appleby-Bentley gas producer, 2179
Appliances, lifting, arrangements in, 825
——————— timber, for quays, 1473
Arbitrator, engineer as, 2910
Arbitrator's fees, 2911
Arc lamps, 2326
——— welding, 2780
'Arca' compact valve, 1678
Arch bridges, 301
Arched bridge, reinforced concrete, 471
Arches, 470, 523
——— centering of, 525
——— keystones of, 524
——— reinforced concrete, 470
——— setting out, 523
——— thickness of, 525
——— to find depth of keystone of, 524
Archstones, pressure on, 525
Area computation, Simson's rule for, 53
——— of boiler chimneys, 1610
———, grate, of locomotive boilers, 599

ARE

Area, of safety valves, Board of Trade rules for, 1661
——— Bureau Veritas rules for, 1660
——— Lloyds rules for, 1660
——— steam cylinders, 2372
——— steam passages, 1748
——— valves, 1748
Areas, calculation of, from co-ordinates, 52
———, safety valve, 1661
Armature shafts, gearing on, 918
Armco' ingot iron, 2679
Arms, pulley, 1174
Armstrong-Siddeley aero engines, 2052
Arrangement of refrigerating chambers in ships, 1224
——— steam turbines in ships, 2376
Arrol, Sir W., & Co.'s giant cranes, 813
Artesian well pumps, 1251
——— wells, 1409
Artificial bacterial methods, sewage, 1372
——— daylight, 2546
——— seasoning of timber, 359
——— stone, 407
Asbestos Cement Building Products Ld., 323, 324, 412, 414, 420, 1827, 2479
——— building sheets, 395
——— cement sheets, 'Everite,' 323
——— 'Fibrent,' 324
——— 'Italit,' 327
——— 'Poilite,' 323
——— 'Turnall,' 324
——— cushioned flue cover, 1606
——— seam inspection blocks, ——— seating blocks, 1606 [1607
——— jointing, 1690
——— packing for tubes, 1827
Ascension pipes of gasworks, 2463
Ash and moisture content in fuel, 2082
Ashdown industrial stroboscopes, 956
——— rotoscope, 957
Ashlar masonry, 380
Asphalt flooring, 407
——— paving, 743
Asserson's formula for circular sewers, 1360
Association electricity supply cables, 2211
———, British, screw-thread, 1139, 1143
——— of British Driving Chain Manufacturers, 899
Assuan dam, 1415 [
Astronomy, 76
——— in field work, 77
Atlantic cargo steamers, 2368
——— liners, 2371
Atmosphere, standard, 1181
——— pressure of, 1182
——— temperature of, 106
Atomic weights, 123
Atomisers, liquid fuel, 2162
Attemperator, brine, Webb's, 1229 [2336
Attenuation and formulæ for telephones,
Audible sea signals, 1500
Audibility of signals at sea, 1500, 1501
Auld's reducing valve, 1671
Australian mines, 2594
——— timber, strength of, 356
Austrian electric railways, 671
Auto Recorder Co.'s CO_2 recorder, 2083
Autocopyist, 94
Automatic brakes, 638
———, Westinghouse, 638

AUT

Automatic control gear, electrical, 2299
——— forced lubrication, 989 [2508
——— regulation of heating apparatus,
——— reversing drive for planing machines, 1021
——— temperature regulator, 1642
——— tools, lubrication of, 985
——— trip clutch, 893
——— vacuum brake, 642
——— water recorder for steam engines, Lea's, 1849
Automobile engines, 1935
——— air filters for, 1948
——— anti-knock substances,
——— bearings, 1965 [1949
——— brake M.E.P., 1956
——— Bianchi, 1942
——— breakdowns, 1979
——— carburation, 1951
——— carburettors, 1948
——— intake, 1948
——— Citroën, 1942
——— compression, 1961
——— crankcase corrosion, 1954
——— Daimler, 1938
——— detonation and anti-knock substances, 1949
——— efficiency of, 1954
——— firing order, 1954
——— Franklin, 1939
——— fuel distribution to cylinders, 1948
——— gear-box gears, 1966
——— heat losses, 1952
——— horse power, 1960
——— ignition, 1958
——— inlet pipes, 1961
——— Lanchester, 1937
——— lubrication, 1969
——— mechanical losses, 1951
——— Morris, 1936
——— Packard, 1940
——— piston clearance, 1955
——— rings, 1957
——— pistons, 1956
——— practice, 1940
——— radiator tablets (Boiler-
——— shafts, 1966 [ine), 1953
——— starting, 1948
——— supercharging, 1944
——— valve timing, 1964
——— valves, 1962
——— material for, 1964
——— water pumps, 1953
Automobiles, 1935
——— brakes for, 1971
——— stopping distance of,
——— breakdowns of, 1979 [1971
——— British and Foreign, 1941
——— design, progress of, 1935, 1979
——— friction between tyres and road surface, 1971
——— lubrication of, 1969
———, power required to negotiate gradients, 1975
——— springs for, 1974
——— steering wheels, 1968
——— supercharging, 1944
——— transmission data, 1967
——— tyres, 1976

AUT

Automobiles, tyres, N.A.P. unpuncturable, —— —— weight of, 1970 [1977
'Avery' coal weigher, 1448
—— —— truck weigher, 1440
—— —— water weigher, 1439
—— —— weighbridge, 1440
Avonmouth docks, 1462
A.W.P. electrodes, 2784
Axle boxes, railway, 632
Axles, crank, of locomotives, 596, 611
—— diameter of, 1003
—— fatigue cracks in, 536
—— railway carriage and wagon, 612
—— straight, of locomotives, 611
—— tramway, 683
Azimuth, determination of, 79, 80, 84

B. Th.U. (British Thermal Unit), 107, 1531, 2449, 2458, 2474, 2491
Babbitt metal, 2737
Babcock & Wilcox boilers, 1596, 2397, 2398, 2403
—— —— —— (Clayton) crude oil engine, 1917
—— —— —— electric crab crane, 805, 806
—— —— —— mechanical stoker, 2112
—— —— —— —— superheater, 1617
Back pressure and extraction engines, 1731,
—— —— turbines (steam), 1810 [1734
Bacterial treatment of sewage, 1372
Bailey, W. H., & Co.'s vacuum pump, 1200
Baker oil separator, 1681
Baking ovens, electric, 2333
Balancers, static, 2304
Balata belt ropes, 881, 884
—— belting, 856, 863
Ball and socket joint, 607, 1084
—— bearings, 963
—— —— lubrication of, 975
—— —— permissible loads on, 968
—— —— Skefko, 964, 970, 972
—— cast iron, weight of, 133
—— mills, Krupp, gold mining, 2641
Ballast, 542
Ballistic pendulum, 34
Balls, cast-iron, weight of, 133
—— metal, weight of, 135
Baltic yellow fir girders, 330
—— —— joists, 331
Bandsaws, 1059
Bar aluminium, weight of, 139
—— cast iron, weight of, 133
—— copper, weight of, 137
—— brass, weight of, 137
—— iron, tests for, 188
—— lead, weight of, 137
—— metals, weight of, 132, 137, 139
—— steel, tests for, 188
Barff process for protecting iron, 2714
Barimar, welding repairs, 2776
Barnes' formula for water velocity, 1507
Barometer, height of, in millimetres, 111
Barometric height, 1182
Barrow and cart runs, 494
Bars, fire, 2109, 2395
—— for reinforced concrete, 451, 452
——, steel, Admiralty tests, 186
—— T, 240
Bascule bridges, 293

BEL

Base line, to measure, across a river, 53
——, measured, to reduce to level of sea, 58
Basic bricks, 431
Baths, tempering, for cutting tools, 1070
Batteries, G.E.C., 2255
—— —— Ironclad-Exide, 707
—— —— primary, 2255
—— —— Siemens, 2255
Battery rooms for accumulators, 2259
—— vehicles, 707
Baumé gravity, 131
Bauxite bricks, 430
Bazin's formula for channels for irrigation, 1390
—— —— —— flow of water in pipes, 1298
Beam, to cut stoutest, from log, 357
—— —— stiffest, from log, 357
Beams, 218, 317, 446, 453, 458
—— British standard, 238
—— broad flange, 235
—— building up, 313
—— continuous, 219
—— deflection of, 219, 315, 462
—— fishing, 311
—— I, 235, 236
—— inverted, 458
—— jointing, 311
—— of various sections, values of co-efficient, 181
——, pine, breaking weight of, 185
—— rectangular, 455, 460
—— —— solid, 216
—— reinforced concrete, 442, 453, 455
—— safe loads on, 236
—— scarfing, 312
—— straining-joint of, 312
—— strength and stiffness of, 314
—— stresses on, 207–211, 458
—— T, 442
—— timber, weight of, 364
—— yellow pine, breaking weight of, 185
Beardmore Tosi Diesel engines, 1898
Bearers, rail and road, 281
Bearing metals, 2443, 2736, 2740, 2744
—— pressures, 961, 1738
—— springs, 613, 632
—— surface for girder ends, 496
—— surfaces, locomotive, 596
Bearings, ball, 963
—— —— lubrication of, 975
—— —— bridge, 270
—— —— crane, 815
—— —— engine, 1741
—— —— heating of, 962
—— —— locomotive, 596
—— —— lubrication of, 975
—— —— manufacture of, 975
—— —— Michell, 2409
—— —— motor-car engine, 1936
—— —— mounting, 974
—— —— permissible loads on, 968
—— —— roller, 970
—— —— rubber, 1741
—— —— Skefko, 964, 972
—— —— thrust, 967
—— —— turbine (steam), 1796
Beck water meter, 1438
Bedplates and bearings, bridge, 270
Beldam Asbestos Co.'s 'Lascar' V packing, 1755

BEL

Beldam Packing & Rubber Co.'s packings, 1753, 1754
——— 'Excelsior' ring packings, 1753
——— 'Pilot' jointings, 1690
——— 'Pilot' packing, 1753
——— 'Serpent' packing, 1754
Bell buoys, 1490
—— metal, 2731, 2744
Belliss high-speed forced lubrication air compressor, 1193
——— & Morcom Diesel engines, 1897
——— forced lubrication, 1731
——— Fullagar gas engine, 1902
——— high-speed engines, 1731
——— pneumatic hose couplings, 1084
Belt conveyors, 813
—— dressings, 862
—— driving, Tullis on, 863
—— driving with quarter twist, 865
—— joints, cement for, 862
—— pulleys, 864
—— tension, 858
Belting, 855
——— Balata, 856, 883
——— chain or link, 869
——— coefficients of friction of, 869
——— Dick's Balata, 856
——— German practice in regard to, 857
——— leather, 855
——— 'Lincona' Balata 'V,' 883
——— endless, 868
——— power transmitted by, 855
——— Reddaway or camel hair, 857
——— steel, 869
——— strength of, 864
Belts, compounding, 863
—— cross-sections of, 858
—— jointing, 861
—— leather, oiling, and greasing, 862
—— length of, 860
—— link or chain, 869
—— power transmitted by, 855, 857
—— putting on heavy, 867
—— rules for, 859
Bending moments, 210
Bends for steam pipes, 1685
———, corrugated steel, 1686
——, unbalanced pressures on water pipe, 1431
Benjamin Electric Ltd. reflectors, 2544
Benn friction clutch, 892
Bennis chain grate stoker, 2114
——— coking stoker, 2114
——— compressed air furnace, 2114
——— conveyors, 845
——— self-cleaning stoker, 2114
——— sprinkler stoker, 2113
Benzine, 2150
Benzol, 2154
——— recovery, 2099
Berk, F. W., & Co's accumulator acid, 2257
Beton, 388
Bevel wheel teeth, 924
Bianchi motor car engines, 1942
Bills, railway, standing orders for, 664
Bins, reinforced concrete, 468
Birmingham gauge for sheets and hoops, 13
——— wire gauge, 11
Bismuth solder, 2750

BOI

Bits, steel mining, sharpening, 2565
Bitumen sheeting, Callender's, 368, 1412
Bituminous carpeting, 737
——— macadam roads, 737
——— materials, tests of, 750
——— suction gas producers, 2186
Bizerta breakwater, 1459
Black copying process, 94
Blackening brass, 2808
Blackett conveyor, 845
Blackett's wire-rope socket, 889
Blackstone oil engines, 1920
Blades and caulking pieces for steam turbines 1790
———, impulse, for steam turbines, pitch of, 1791
Blading efficiency of steam turbines, 1791
Blakeley copying process, 93
Blast-furnace gas, 2189
——— Portland cement, 405
—— furnaces, 2660
Blasting, 2572
——— machines, 2798
———, rock, 1487
Block-in-course masonry, 380
Blocks, abrasive, 435, 1105
——— and terra-cotta, 428
———, boiler, 429, 1606
———, furnace lining, 433
———, hollow, 429
Blow, energy of a, 1128
Blowers, portable electric, 2269
———, power variation of, 2520
——— rotary, for ventilation, 2520
——— Sturtevant, for ventilation, 2520
Blowpipe cutting, 2777
——— welding, acetylene, 2767
Blue bricks, 426
Blue photographic copying process, 94
Blueing steel, 2812
Board of Trade regulations for electric tramways, 701
——— rules for safety valve areas, 1661
——— springs, 1661
Boards, fireproofing and waterproofing, 2813
——— flooring, 332
———, weight of, 364
Bodies, falling, 27
——— impact of, 27, 28
——— rotating, energy of, 34
Boiler air heaters, 1648
—— blocks, 429, 1606
—— butt straps, 1567
—— casings, 2403
—— chimney and flue dimensions, 1610
—— chimneys, sizes of, 1611
—— cleaners, mechanical, Hulburd's, 1682
—— combustion and draught rate, 1543
——— chambers, iron for screw stays of, 1565
—— construction, 1560
—— covers, Pearson's patent, 1606
—— design, 1541
—— distance thermometer, 1667
—— draught, 1542, 2392
—— draught gauges, 1545
—— efficiency, 1553, 1555
——— maintenance of, 1554
—— evaporation rate, 1553
—— expansion joints, 1609

Boiler explosions, 1654
—— feed and service pumps, 1254
—— heating, Weir, 605, 2406
—— lines, removal of air from, 1680
—— water evaporators, 2405
——, treatment of, 1624
—— fittings, 1655
—— flanging, pressures required for, 1562
—— flat plates, 1568
—— flues, dimensions of, 1610
—— flues, working strength of, 1562
—— foundations, 1605
—— front expansion joints, 1610
—— furnace arches, 'Liptak,' 1608
—— furnaces, 1568
—— arches for, 1608
—— corrugated, 1568
—— girders, 1569
—— plain, 1598
—— plates, 1568 [1704
—— radiant heat transmission in,
—— securers, 1608
—— stays, 1568, 1569
—— girders, 1569
—— graphite, Foliac, 1627
—— heat transmission, 1545
—— heaters, tubular, 1549
—— improvements, modern, 1605
—— injectors, 609, 1655
—— inspection, 1654
—— lagging, 1695
—— legislation, 1655
—— output, 1542
—— plant efficiency, 1555
—— maintenance of, 1554
—— plants, typical, working figures of, 1556
—— plates, multitubular, thickness of, 1576
—— plugs, fusible, 1609
—— pressure required for pumping, 1262
—— rate of combustion and draught, 1543
—— rivet bars, 1563
—— riveting, pressures required for, 1562
—— rules and regulations, 1563
—— safety valves, 1659
—— scale, 1622, 1704A
—— seams, flanged, 1562
—— welded, 1563
—— setting, 1604
—— settings, leaky, 1605
—— shells, 1560, 1566
—— strength of, 1561
—— thickness of plates of, 2397
—— soot removers, 1683
—— stays, 1563
—— steel, 1561
—— surfaces, 2391
—— tests, table of, 1558 [distance, 1667
—— thermometers, Negretti & Zambra's
—— tube brushes and scrapers, 1684
—— cleaners, 1684
—— tubes, 600, 616, 1571, 2395
—— charcoal iron, lapwelded, 1571
—— heat transmission through, 1550
—— locomotive, life of, 600
—— soot removers, 1683
—— steel, 1572
—— surface of, 1571
—— weight of, 1571
—— water gauges, 1662
Boilerine, 1626

Boilerine radiator tablets, 1953
—— 'Rosseline' compound, 1887
Boilers, 599, 787, 1542, 2391
—— air heaters for, 1648
—— Babcock & Wilcox, 1596, 2397
—— Clarke-Chapman, 1588, 2400
—— Clarkson, 2196
—— coal combustion in, 2391
—— Cochran, 1582, 2502
—— construction of, 1560, 1563
—— Cornish, 1573
—— corrosion in, preventives, 1624
—— cotton mill, 787
—— cross-type, 1598
—— cylindrical, 1565
——, length of, 2395
—— marine, 2492 [of, 2395
——, diameter of tubes
—— Danks-Netherton vertical, 1595
—— effect of altitude on draught of, 1544
—— embrittlement of, 1627
—— evaporation of, 1553, 1582
—— 'Express,' 2397
—— Galloway, 1574
—— gas-fired, Kirke, 2191
—— grate area of, 599
—— heat transmission in, 1545
—— heating surface of, 600, 1552, 2395
—— Howden H.P., 1599
—— John Thompson, 1576, 1593, 1594
—— Kirke, 2191, 2194
—— Lancashire, 1573
—— life of, 1653, 2400
—— locomotive, 599, 1576
—— fire boxes, 600
—— grate areas and heating surfaces of, 600
—— iron castings for, 617
—— plates for, 617, 619
—— rivets for, 617 [617
—— specification of plates for,
—— steel castings for, 615
—— tubes for, 600, 615, 616
—— main, cylindrical type, 1565
—— marine, 1563, 2391
—— draught of, 2392
—— furnaces and firebars of, 2395
—— heating surface of, 2395
—— length of, 2395
—— life of, 2400
—— mill, 787
—— multitubular, 1576
—— Niclausse, 1584
—— Paxman 'Economic,' 1581
—— portable, 1598
—— preventives of corrosion in, 1623
—— scale in, 1622
—— riveted joints of, 1567
—— Robey, 1576, 1579
—— scale in, 1622
—— setting, 1604
—— settings, leaky, 1605
—— Simon Carves, 1592
—— Spencer-Hopwood, 1600
—— steamship, 2391
—— coal combustion, 2391
—— draught, 2392
—— furnaces and firebars, 2395
—— heating surface, 2395

BOI

Boilers, steamship, length of cylindrical, 2395
——— tube diameter, 2395
——— steel for, 1561
——— Stirling, 1590
——— Sulzer water-tube, 1601
——— Thompson, 1573, 1577, 1578, 1593
——— Thornycroft, 2399
——— transmission of heat in, 1545
——— types of, 1573
——— steamship, 2391
——— vertical, Cochran patent, 1582 [1595
——— Danks, H.&T.(Netherton) Ld.
——— Vickers water tube, 1584, 1586
——— waste heat, 2194
——— water purifying for, 1632
——— water softening for, 1628
——— water-tube for land service, 1584
——— marine service, 2397
——— tubes for, 1704
——— welded seams in, 1563
——— White-Foster, 2397
——— Woodeson, 1588, 2400
——— Wollaston, 1584, 1585
——— working pressure to be allowed in, 1566,
——— Yarrow, 1602, 2399, 2401 [2397
Boiling-points of saturated solutions of salts,
——— water, 111 [111
Bolection mouldings, 410
Bolinder oil engines, 1913
Bolt and nut machines, horse-power for, 1054
—— fixing in walls, 421
—— forcers, hydraulic, 1354
—— heads, British standard hexagonal, 1154
——— round, 1159
——— weight of, 161
——pins, French, 1158
—— securers, 1174
—— spacing, 1175
Bolted sectional dock, 1477
Bolts and nuts, 1152
——— American, 1153 [tion for, 538
——— for railway plant, specifica-
——— French standard, 1158
——— Whitworth's, 1152, 1153
—— fixing, in walls, 421
—— foundation, 1740
—— French, 1157
Bombay harbour and docks, dredging, 1485
Bonds for brickwork, 367
——— electric tramway rails, 698
——— masonry, 381
Bone, to make permanently white, 2814
Books and paper, sizes of, 19
Booster transformers, 2305
Booth, J., & Co.'s 'Duralumin,' 2735
Booth's formula for strength of boiler flues, 1562
——— report on blast-furnace gas, 2189
——— table of properties of gases, 112
Bore-holes, 1564
Boring (mining), 2563
——— cost of, 2564
——— diamond rock-, 2563
—— machines, 1054
Bosh tuyeres, 2662
Bosses and pulley arms, 1174
Bostock enveloping worm gear, 815
Bottles, glass, to divide, 2813
Boulton water elevator, 1253
Bower-Barrf process, 2712
Box keys, 1164

BRE

Box sextant, 67
—— spanners, 1164
Boxes, axle, 632
Boyer long-stroke pneumatic hammer, 1079
Boyle's air pump ventilator, 2523
Brackets, 341
Bradford steam trap, 1699
Brake, driver's equalising valve for, 639
——— horse-power, 1707
——— linings, 819, 850, 897
——— rims, cooling, 37
——— thermal efficiencies of gas engines, 1884
Brakes, 626, 638, 691
——— automatic, vacuum, 642
——— automatic, Westinghouse, 638
——— Ohekko surfaces for, 819, 850, 897
——— crane, 818
——— electric locomotive, 678, 691
——— electro-mechanical, 2298
——— Ferodo surfaces for, 819, 850, 897
——— locomotive, 626, 638
——— power measurement, 34
——— Alden, 38
——— Brotherhood fluid friction, 38
——— fan, 41
——— Froude's, 38
——— Hirn's rope, 34
——— Prony, 35
——— Ringelmann's, 36
——— water-pipe ab- [sorption, 57
——— railway, 626, 638
——— railway, power required by, 636
——— tramway, 691
——— Westinghouse automatic, 638
Brass, aluminium, 2730
—— balls, weight of, 135
—— bars, weight of, 137
——, blackening, 2808
—— cast, 2723
—— ductile, 2723
——, effect of certain metals on, 2724
—— naval, 2724
—— hot-forgeable, 2723
—— melting point of, 2732
—— pipes, weight of, 146
——, rolled, weight of, 137
—— screw thread, British Standard, 1144
—— sheets and bars, weight of, 137
——, to clean, 2810
—— tubes for locomotive boilers, 616
——, seamless, weight of, 158
——, weight of castings in, 158
—— welding, 2775
—— wire, weight of, 161
Braziers' copper rivets, weight of, 163
Breakages in gas and oil engines, 1889
Breakdown cranes, 809
Breakers, rock, 1488
Breaking ground (mining), cost of, 2592
——— load on steel rails, 536 [wire, 176
——— stresses of annealed and bright iron
——— galvanised high purity iron telegraph line wire, 177
——— hard drawn telegraph copper wire, 177
——— steel wire ropes, 170
——— wire and wire ropes, 176
——— weights of pine beams, 185

BRE

Breakwaters, 1455
——— construction of, 1457, 1459
——— ——— cost of, 1456, 2841
Breeze concrete, 388
Breguet ejectair, 1845
Bremen river weir, 1511
Brett's Patent Lifter Co., Ltd., drop forging [equipment, 1135
Brick floor paint, 2804
——— noggings, 369
Brick retaining-walls, 517
——— roads, 739
Bricks, 366
——— basic, 426
——— bauxite, 430
——— blue, 426
——— building, specification for, 427
——— cement-concrete, 426, 428
——— chromite, 430
——— chuff, 426
——— colliery, 426
——— common, 426
——— concrete, 428
——— crozzle, 426
——— crushing strength of, 175, 427, 441
——— cutter, 426
——— engineering, 426
——— facing, 426
——— fire, 430
——— fireclay, specification for, 432
——— furnace, 433
——— glazed, 426
——— grizzle, 426
——— kinds of, 426
——— lime-sand, 426
——— magnesia, 430
——— malm, 426
——— melting point of, 430
——— neutral, 431
——— paviour, 426
——— place, 426
——— porosity of, 428
——— pressed, 426
——— rattler, test for, 428
——— rubbers, 426
——— sand-lime, 426
——— semi-bauxite, 430
——— semi-dry, 426
——— semi-plastic, 426
——— silica, 431
——— ——— specification for, 432
——— sizes of, 366, 427, 432
——— specification for, 427
——— Staffordshire, 426
——— stiff plastic, 426
——— stock, 426
——— strength of, 175, 427, 432
——— tests of, 428
——— wire-cut, 426
Brickwork, 366
——— bonds, 367
——— of wells, 1407
———, reinforcements for, 336
——— strength of, 175
Bridge, arched, reinforced concrete, 471
——— Clyde, 258
——— expansion bearings, 270
——— foundations, 258
———, girder, specification for, 265
——— girders, camber of, 275
——— ——— economical proportions of, 248

BRI

Bridge materials, unit loads on, 252
——— piers, cylinder, 258
——— riveting, 275
Bridges, arch, 301
——— bascule, 293
——— bedplates and bearings of, 270
——— British standard steel for, 185
——— centrifugal force on, 267
——— cost of, 2842
——— cross girders of, 247
——— deformation and deflection of, 282
——— economical span of, 247
——— flooring of, 273
——— foundations of, safe loads on, 258
——— ——— skin friction on, 258
——— impact in, 268
——— lattice girder, 235
——— lattice girders of, 249
——— piers of, 526
——— plate-girder, 277
——— portal bracing of, 289
——— reinforced concrete, arches of, 470
——— rail and road bearers of, 281
——— rolling load on, 249
——— spans and standard clearances of, 249
——— steel, 247
——— stress on, 262, 264
——— suspension, 305
——— swing and bascule, 293
——— temperature effect on, 270, 307
——— train momentum on, 268
——— types of, 247
——— waterway of, 258
——— wind pressure on, 265
——— working stresses on, 295
Bridgework, British practice, 265
———, steel, 247
——— weights of, 261
Bridging joists, 331 [stresses of, 176
Bright iron wire, sizes, weights, and breaking
Brightness (light), 2527
Brine attemperator, Webb's, 1229
———, calcium, 1220
———, corrosion by, 1220
Brinell hardness numbers, 1100
Briquette fuel, 2106
Bristol Jupiter aero engines, 2063 [1888, 1903
'Britimp' metallic packings, 1199, 1751, 1793,
British Admiralty specification for oil for Diesel engines, 2166
——— specification for oil fuel for the Navy, 2166
——— Aluminium Co.'s aluminium bronze,
——— ——— conductors, 2213 [2732
——— weight of aluminium, 139, 140, 159
——— and Allied Manufacturers' Association, specification for turbine forgings, 1804
——— Arca Regulators, Ltd. 'Arca' Automatic Regulator, 1677
——— Architects, Royal Institute of, experiments on strength of brickwork, 175
——— ——— ——— Report on Reinforced Concrete, 444
——— Association screw threads, 1139, 1143,
——— canals, 1514 [1145
——— cylinder (steam) ratios, 1737
——— driving chain manufacturers, 899
——— electric railways, 672
——— Fottinger hydraulic transformer, 2388

British Fottinger marine transformers hydraulic drive, 2389
——— Hele-Shaw clutches, 890, 895
——— Isles, rainfall in, 1399
——— locomotives, recent additions, 571
——— types of, 578
——— Metallic Packings Co.'s packings, 1199. 1751, 1793, 1888, 1903
——— Oxygen Co.'s blowpipes, 2778 [2778
——— cast iron cutting results,
——— steel cutting results, 2778
——— Power Railway Signal Co.'s electric signalling (railway), 658
——— railway electrification, standards for, 673
——— rails, dimensions of, 529
——— Reinforced Concrete Engineering Co.'s system, 452
——— Standard aluminum conductors, 2213
——— bare copper wire, 2261
——— benzol, 2158
——— boiler tubes, 1572
——— bolts and nuts, and bolt heads, 1154
——— brass thread, 1144
——— bridge steel, 185 [1099
——— Brinell hardness numbers,
——— building construction steel,
——— bus lamps, 2323 [185
——— cable soldering sockets, 2243
——— carbon brushes, 2295
——— cast-iron, 2689
——— water, gas, and sewage pipes, 165
——— castle nuts, 1160
——— channels, 241
——— charcoal-iron lapwelded boiler tubes, 1571
——— conduits, electrical wiring, 2246
——— copper conductors, 2205, 2206
——— creosote, 363
——— derrick cranes, 808
——— drawing office practice, 92
——— ebonite, 2346
——— electric lamps, 2532
——— electrical machinery, 2260
——— electricity supply cables, 2205
——— meters, 2315
——— enamelled wire, 2309
——— fine screw threads, 1141, 1143, 1156
——— fire extinguishers, 2549
——— fuel oils, 2167
——— gas pipes, 165
——— heads for small screws, 1150
——— heating pipes, 2497
——— hexagonal nuts and bolt heads, 161
——— insulating oils, 2347
——— insulators (porcelain), 2225
——— keys, keybars and keyways,
——— limit gauges, 774 [1162
——— locomotive materials, 611
——— steel tubes, 1572
——— tyre contours, 597
——— low-pressure heating pipes, 2497
——— measuring instruments, 2310
——— metallic resistance materials,
——— motor fuel, 2145 [2205

British Standard overhead lines electrical transmission, 2221
——— pillars, 242
——— pipe flanges, 1166
——— threads, 1150, 1151
——— Portland cement, 403
——— blast furnace, 405
——— pressboard, 2346
——— rails, 529
——— railway electrification, 672
——— rolling stock material, 611–620
——— rivets, 227
——— rolled steel sections, 238
——— safe loads on bridge struts,
——— screw heads, 1150 [243
——— screw taps, 1170
——— screw-threads, 1140, 1145, 1156
——— sections, 238
——— sewer pipes, 1361
——— signalling steel wire strand, 178
——— socket joints for cables, 2241
——— solder, 2747
——— soldering sockets, electrical, 2243
——— spanners, 1164
——— specification of road tars,
——— steam turbines, 1816 [762A
——— steel for bridges, 185
——— steel for building construction, 185
——— for shipbuilding, 189
——— steel electrical conduits, 2246
——— for bridges and buildings
——— tubes, 1571 [ings, 185
——— street lighting, 2546
——— switch gear, 2297
——— telegraph and telephone poles, 2339
——— wire, 177
——— tensile test pieces, 1564
——— traction lamps, 2323
——— motors, 2282
——— train lamps, 2323
——— tramway poles, 695
——— rails, 684
——— tyres and axles, 683
——— trolley wheels, 693
——— work specifications, 703
——— tungsten gas-filled lamps, 2322
——— vacuum lamps, 2322
——— twist drills, 1039
——— vulcanized fibre, 2234
——— water pipes, 165
——— Whitworth screw threads, 1139, 1140, 1146, 1150, 1155
——— wire gauge, 12
——— Steel Piling Co.'s hammers, 502
——— steel piling system, 504
——— thermal unit (B.Th.U.), 109, 1542, 2449, 2476, 2494
——— Thomson-Houston electric brake controller, 691
——— Fabroil gears, 919
——— magnets, 1958

BRI

British Thomson-Houston Mazda lamps, 2323
——— motor coach equipment, 682
——— Pitot tube meters, 1435
——— self-ventilated, interpole traction motor, 2284
——— Thermostatic metal, 2816
——— traction motors, 2284
——— variable speed polyphase commutator motors, 2294
——— voltage regulator (Tirrill), 2266
——— water turbines, 1320
Broaching, 1031
Broad flange beams, Skelton's, 237
Bronze, aluminium, 2732
——— effect of certain metals on, 2731
——— manganese, 2723
——— melting-point of, 2733
——— phosphor, 2731
——— welding, 2775
Bronzes and lacquers, 2807
Bronzing liquids, 2808 [1047
Brooke's quick-change universal drill jig,
Brotherhood air compressors, 1190, 1193
——— fluid-friction dynamometer, 38
——— gas engines, 1877
——— oil engines, 1912
——— pressure gauge tester, 1666
Browett-Lindley gas engines, 1880
Brown Curtis steam turbines for ships, 2381
Brownlie, D., economiser (steam) data, 1646
——— working figures of boiler plants,
Bruell burner, 2131 [1556
Bruges and Crimp's formula for discharging capacity of sewers, 1359
Bruges ship canal, 1528
Brushes and scrapers, boiler tube, 1684
——— carbon, 2295
Büchi turbo-charging system, 2435
Buchli (link connection) drives for electric locomotives, 676
Buckets, Pelton wheel, 1310
Buffer springs, 633
Building Acts, London, 370
——— blocks, hollow, 429
——— bricks, specification of, 427
——— materials and construction, 347
——— effect of moisture changes
——— sheets, asbestos, 395 [on, 408
——— 'Poilite,' 323
——— walls, 368
——— walls, rules for, 368, 373
Buildings, British standard steel for, 185
——— effect of weight of, on retaining walls, 514
——— electrical equipment of, Institution of Electrical Engineers regulations, 2231
——— heating, 2491
——— shoring, 513
——— structural steelwork for, London County Council Act, 373
Bulb angles, British standard, 241
——— furnaces, 1568
——— plates, British standard, 241

CAM

Bull-head rails, 529
Bull's metal, tensile strength of, 2725
——— metals, 2441, 2725, 2744
——— Metal Co. metal for propellers, 2441
——— non-magnetic alloy, 2744
Buoy moorings, 1490
Buoyage, 1489
Buoyancy, centre of, in ships, 2355
Buoys, 1489
——— bell, 1490
——— channel demarcation by, 1489
——— gas, 1489
——— luminous, 1490
——— whistling, 1490
Burdon oil-gas heater, 1068
Bureau Veritas rules for safety valves, 1660
Burkli-Zeigler's formula for rate at which rainfall reaches a sewer, 1359
Burmeister & Wain Diesel engine, 2413
Burners, 2167
———, incandescent, 2476
Bursting strength of tubes, 190
Burtoning (cargo), 1471
Bus electric lamps, 2323
Bushes, steel, to reduce diameter of, 1091
By-product recovery gas-producer plant, 2185

CABLE, cab-tyre sheathed, 2244
——— length, 1
——— 'Maconite,' 2254
Cables, aluminium, 2213
——— electric, for collieries, 2612
——— soldering sockets, 2243
——— tramway, 700
——— electricity supply, 2207
——— flexible, 2207
——— for high voltage, 2350
——— Henley's, 2247
——— lead covered, 2210
——— sags on aluminium, 2220
——— Silvertown, 2211
——— three-core, 2247
——— two-core, 2247
Cab-tyre sheathed cable, 2244
Caissons, 508, 1468
Calcium brine, 1220 [1388
Californian experiments on evaporation, 1385,
——— type dredger, mining, 2648
Callendar's tables of properties of saturated steam, 1531
Callender, G. M., & Co.'s bitumen sheeting, 368, 1412
Callender's 'Kalibond' wiring system, 2252
Callendrite for reservoirs, 1412
Calorie, 109, 2474
Calorific value of coal, 2077, 2091
——— properties of fuels, 2077
——— value of alcohol-water mixtures, 2156
——— coal gas, 2172
——— gaseous fuels, 2168
——— liquid fuels, 2169
——— refuse, 2148
——— solid, liquid, and gaseous fuels, table of, 2169
Calorising iron, 2713
Camber of joists, 329
——— roads, 728
Cambridge Instrument Co., Ltd., 1435, 1445, 2087

CAM

Cambridge Instrument electrical CO_2 recorder, 2087
—— —— & Paul, Anemometer, 1445
—— —— —— meters, 1435
Camel hair belting, Reddaway's, 857
Camshaft control for motor-coach trains, 682
Canal, Amsterdam, 1528
—— bank protection, 1524
—— Bruges, 1528
—— capacity, development of, 1514
—— construction, cost of, 1516
—— curves, 1524
—— development, 1514
—— inclined planes, 1521
—— lifts, 1519
—— locks, 1519
—— New Welland, 1528
—— Panama, 1527
—— Suez, 1527
—— traction and haulage, 1525
Canalisation of rivers, 1511
—— —— cost of, 1513
Canals, American, 1516
—— British, 1514, 1516
—— capacities of, 1390
—— Continental, 1515
—— cost of, 1513, 2847
—— French, 1515
—— German, 1515, 1516
—— haulage on, 1525
—— lateral, 1513
—— loss of water on, 1389, 1524
—— Rhone-Marseilles, 1516
—— ship, 1526
—— Swedish, 1516
—— traction on, 1525
—— United Kingdom, 1505, 1515, 1526
—— —— —— —

CAS

Castings, to repair, 2754 [wood patterns, 161
——— weight of metal, compared with
Castle nuts, British standard, 1160
Caulking hammers, 1081 [turbines, 1790
——— pieces and blades for Parsons' steam
Ceiling joists, 331
Ceilings, expanded metal suspended, 337
Cellar drainer, Willcox-Penberthy, 1263
Cells, battery, dry, 2285
Celluloid cements, 397
———, flexure of, 185
———, rendering incombustible, 2816
Cellulose lacquers, 2807
Cement, belt joint, 862
———, concrete, 388
——— 'Fondu,' or 'Electric,' 397, 405
——— gun, 391, 394
——— injection (François system), 1412, 2581, 2582
——— joints for water mains, 1247
———, Keenes', 397 ['ferrocrete,' 405
——— Marketing Co.'s rapid hardening
——— mortar, 383
——— Portland, tinting, 387
——— Portland, 401
——— blast furnace, 405
——— testing, 401
——— rapid hardening, 405
——— refractory, 398, 432, 435
——— specification for, 403, 432
——— Roman, 397
——— testing machines, 402
Cementation process, François, 1412, 2581, 2582
Cements, 395, 435
——— bituminous, 396
——— calcareous, 397
——— casein, 396
——— celluloid, 397
——— ferro concrete, 405
——— fishglue, 396
——— flour and starch, 397
——— gypsum, 397
——— mastic, 395
——— metallic, 400
——— miscellaneous, 400
——— oxychloride, 397
——— Portland, 401
——— rapid hardening, 405
——— refractory and fire-resisting, 398, 435
——— water glass or silicate, 397
Centering of arches, 525
Centigrade-Fahrenheit conversion table, 99
Centre of buoyancy of ships, 2355
——— flotation of ships, 2356
——— gravity, 23
——— oscillation, 31
——— percussion, 31
Centreless grinding, 1117
Centrifugal clutch, 893
——— fans, 2520, 2663
——— for cupolas, 2663
——— smiths' fires, 2663
——— force, 31, 267
——— on bridges, 267
——— pumps, 1266, 2613
———, classification of, 1267
——— efficiency of, 1268
———, hot water, 1270
——— packing for, 1268
——— Pulsometer, 1269

CHI

Centrifugal pumps, solid matter, 1269
——— stresses in drums and discs, 1797
Chain belts, 869
——— gearing, 899
——— centre distance between shafts,
——— clearance, 901 [900
——— desirable conditions for, 900
——— disposition of drives, 901
——— erection, 901
——— lubrication, 901
——— maintenance, 901
——— roller type, Association standards, 902
——— cutter forms for,
——— wheels for, 904, 905 [902
——— grate stoker, Babcock & Wilcox, 2112
——— Bennis, 2114
——— Illinois, 2117
———, Gunter's, 1, 51
——— lubrication, 901
———, Morse, 905
——— or link belts, 869
——— rings, 194
——— speed (driving), 900
———, surveying, Gunter's, 1, 51
——— testing, 198
Chaining, 51
———, over sloping ground, 55
Chains, annealing, 199
——— crane, 817
——— driving, 899
——— iron, strength of, 193
——— strength of, 193
——— rocker joint, 905
Chambers, flushing, capacity of, 1366
Chamier's formula for flood discharge, 1404
Channel curves, 1509
———, demarcation of, by buoys, 1489
——— training works, 1508
Channels, British standard, 239
——— flow of water in, 1298
——— velocity of water in, 1298
Charcoal iron lapwelded boiler tubes, 1571
———, wood, 2106
Charging accumulators, 2257
Check rails, railway, 538
'Chekko' brake linings, 819, 850, 897
——— clutch linings, 850
——— friction linings, 897
Chemical elements, melting points of, 101
——— purification of water, 1419
——— treatment of foundations, 496
——— sewage, 1367
'Chezy's' formula for flow of water in pipes and channels, 1298
Chicago electric tramway rail bond, 698
Chilian mills (mining), 2640
Chimney area, boiler, 1610
——— gasworks, 2460
——— foundations, safe load on, 496
——— shafts, stability of, 521
Chimneys, 521
——— dimensions of, 521, 1610
——— draught required for, intensity of,
——— house heating, 2502 [522
——— table of sizes of, 1611
——— tall, 522
——— temperature of flues of, 1610
China clay, 425
Chisels, cold, 1090

CHL

Chloride Electrical Storage Co.'s Exide batteries, 707
——— ——— ———

COL

Cold sawing (steel), 1057
—— solder, 2751
—— starting oil engines, 1916
—— storage, 1219
—— —— insulation, 1219
Collar-beam roofs, 317
Colliery bricks, 426
——, electric cables for, 2612
—— haulage gear with Fabroil gear drive, [920
— —— tubs, 2598
Colour of smoke, 2112
Coloured glass, absorptive powers of, 1500
—— —— mortars, 387
Colouring Portland cement mortar, 387
Colours, light, 119
— — of different temperatures, 105
—— tempered steel, 1070
Columbia rail bond, 698
Columns, 222
—— cast iron, 226
—— reinforced concrete, 463
—— —— —— —— safe loads on, 463
—— safe load on, 497
—— strength of, 223
Combination of colours (light), 119
Combustion, 2077
—— —— and heat, 107 [fuels, 2168
—— —— value of usual gaseous
—— — computation, 2082
—— for different draught pressures, 1544
—— — heat generated by one pound of carbon, 107
—— —— indicators, 2083
——, internal, turbines, 1926
—— of carbon, 110
—— —— —— —— heat generated by, 107
—— —— coal, and smoke, 2110
—— —— —— in marine boilers, 2391
—— —— gas, weight of air or oxygen necessary for, 2170, 2475
——, spontaneous, in mines, 2618
— — —— steam generator, 2126–2133
——, table of heat of, 108
Commercial efficiency of direct-current motors, 2281
Commissions, secret, 2905
Common bricks, 426
—— rafters, 317
Communication, radio, 2821
Commutator alternating current motors, 2294
Compass, prismatic, 68
—— variation, annual, 74
—— —— —— —— —— to find, 73
Composition, Angus Smith's, 2808
—— —— nails, weight of, 163
—— —— of gases, 2458
—— —— of grinding wheels, 1105
Compound and simple locomotives, cylinders of, 592
—— dredgers, 1483
—— engines, steam pressure in, 2372
—— —— locomotives, 703
—— pendulum, 30
—— steel girders, 238
Compressed-air drilling plants, 2568
—— —— furnace, Bennis, 2114
—— —— hammers, 1129
——, horse-power of, 1188
—— —— pumping by, 1276

CON

Compressed-air, water-raising by, 1276
—— —— gas, heating by, 1068
Compression, air, 1187
—— —— —, cost of, 1086
—— —— motor-car, 1961
—— —, of materials, resistance to, 169
Compressors, air, 1190
—— —— —, Belliss, 1193
—— —— Brotherhood, 1190, 1193
—— —— Holland, 1198
—— —— Holman, 1196
—— —— lubrication of, 1199
—— packing for, 1199
—— —— Westinghouse, 1193
—— —, indicator diagrams of, 1216
Compressors, refrigeration, 1215
Concave lens, 119
Concrete, 388
—— beton, 388
—— blocks, 394
—— breeze, 388
—— bricks, 428
—— bridges, reinforced, 470, 2842, 2846
—— cement, 388
—— coal residues in, effect of, 393
—— colouring, 387, 393
—— condensation on, 392
—— cracking, prevention of, 393
—— dams, 1292
—— —, effect of pigments on, 392
—— —, effect of sea water on, 392
—— expanded steel, reinforcement for, 334
—— floors, 773
—— —— deadening noise through, 773
—— — dusty, treatment of, 393
—— — — oil stains on, 773
—— lime, 388
—— lined metal pipes, 1247
—— materials for, 389
—— mixers, capacities and leading dimensions of, 390
—— mixing, 390
—— mixtures for expanded metal, 340
pavements, filler for, 745
— —— permeability of, 393
—— piles, moulded-in-place, 499
— — —— reinforced, 463, 465, 501
— — —— —— —— Coignet, 498
—— pipes, reinforced, 340, 1303, 1363, 2855
——, placing, 391
—— prevention of cracking and scaling of, 393
—— proportions for, 389, 408
—— — rapid hardening cement for, 405
— —— raw materials for, 388
—— reinforced, 388, 452
—— —— —— floors, 341
— —— —— —— roads, 735
—— roads, 735, 737
—— —— —— cost of, 738
—— —, rust prevention in, 393
—— — scaling, prevention of, 393
—— — sea water, effect on, 392
—— ships, 2444
— —, slag, 388
—— strength of, 391
—— tiles, 394
—— —, volume of, 391
—— waterproofing of, 394

CON

Concrete, weight of, 391
Condensation on concrete, 392
Condensed steam, measurement of, 1849
Condenser air pumps, 1840
——— circulating pumps, 1842
——— injection cocks, 1850
——— tube packing, 1827
——— tube plates, 1826
——— tubes, 1826
Condensers, 797, 1825
——— air in, 1834
——— cooling water for, 1832
———, ejector, 1834
——— gas, 2453
——— jet, 1825
——— Mirrlees-Delas-Ginabat surface, 1835
——— refrigerating, 1215
——— steam, 1825
——— surface, 1826
——— heat transmission in, 1828
——— Weir regenerative surface, 1837
——— ——— uniflux, 2406
——— ——— water extraction, 1838
Condor aero engines, 2048
Conductivity of aluminium, 2213
——— ——— wires, electric tramway, 692
Conductors, electrical, aluminium, 2213
——— copper, 2205
——— British Standard, 2205
——— currents in, 2204
——— Henley's, 2247 [2237
——— jointing electric light,
——— testing, 2238
——— thickness of dielectric of, 2205
——— lightning, rules for, 2328
Conduits, electrical, 2245
——— dimensions of steel, 2246
——— steel, British standard specification for, 2245
——— expanded steel concrete, 340
Cones, Seger, 2674
Conic frustum, 24
Conical coverings, reinforced concrete, 469
——— pendulum, 31
——— round springs, 1014
Connecting pipes of steam indicator, 1717
——— rod bolts, annealing, 1747
Connections of iron pipes, 152
Considère reinforced concrete system, 452
Consolidated Brake & Engineering Co.'s electric brakes, 677, 678
Consolidated Pneumatic Tool Co.'s Boyer long stroke hammer, 1079
——— electric drill, 1043
——— giant drill, 1080
——— little giant air drill, 1044
——— portable electric blower, 2269
——— precision grinder, 1115
Constantan alloy, 2733
Construction cost of railways, 543
——— of breakwaters, 1455
——— cost of, 1455, 2841
——— lighthouses, cost of, 1495, 2870

COP

Construction of overhead electric tramways, 694
Consumption of coal in locomotives, 624
——— marine boilers, 2392
——— gas per head of population, 2451
——— lubricating oil in locomotives,
——— steam, 1708, 1809 [624
——— in turbines, 1779
——— winding engines, 1714
——— water for domestic purposes, 1421
——— in locomotives, 624
Content sizes of timber, 356
Continental canals, 1515
Continuous beams or girders, 219
Contract for works, engineer's rights and powers, 2906
Control gear, automatic, electrical, 2299
——— levers (railway signals), 646
———, tramway motor, 690
Convergence of meridians, 85
Conversion of degrees Centigrade into Fahrenheit, 99
——— tables, 2–10
Converters, motor, A.C.E.C., 2275
——— rotary, 'G.E.C.,' 2270
Convex mirrors, 119
Conveyors, 843
——— and elevators, gas works, 2464
———, belt, 843
——— Bennis, 845
——— Blackett, 845
——— bucket, 847
——— coal, 847
——— D.B., 846
——— pneumatic, 848
——— push plate, 846
——— scraper, 845
——— steel belt, 845
——— water, 848
Cooking, electric, 2332
Cooling brake rims, 37
——— compounds for hot bearings, 2809
——— machine, air, 2267
——— ponds, 1851
——— towers, Premier, 1852
———, water, 1851
——— water for air compressors, 1209
——— condensers, 1832 [1895
——— internal combustion engines,
Copes pump governor, 1680
——— Regulators, Ltd., feed water regulator,
Copings, 369 [1678
Copper, 2717
——— balls, weight of, 135
——— bars, weight of, 137
——— castings, 2756
——— conductors, 2221
———, electrolytic, modulus of elasticity of,
——— impurities in, 2717 [184
——— jointings, seamless soft, Hulburd's, 608
——— melting furnaces, 2669
———, milling cutters for, 1031
——— nails, weight of, 162
——— ore, 2562
——— overhead lines, electrical, 2221
——— pipes, weight of, 146, 156
——— plates for locomotive fire-boxes, 615
——— rivets, weight of, 163

COP

Copper rods for locomotive stay-bolts, 615
———, rolled, weight of, 137
——— sheet and bar, weight of, 132, 137
——— steam pipes, 1687
———, tensile strength of, at different temperatures, 173
——— tubes for locomotive boilers, 616
——— ——— ——— feed pipes, 608
———, seamless, weight of, 156
——— *v.* aluminium for electrical conductors,
———, weight of castings in, 161 [2218
——— welding, 2775
——— wire, enamelled, 2309
———, hard-drawn (telegraphic), strength of, 177
——— ——— weight of, 161
Copying processes, 92
Corbelling, 369
Cordite explosive, 2797
Cordoz, 2656
Core boring, 2563
Cork insulation for cold storage, 1223
———, Anderson, Thomas, Ltd., 1223
Cornish boilers, 1573
——— pumps, 2612
Corrosion by brines, 1220
——— in boilers, preventives of, 1623
——— of aluminium, 2720
——— iron, 174, 2710
——— pipes, 1419
——— ropes, 200
——— lead pipes, 420, 1419
——— metals, 174, 2710
——— soft steel, wrought iron, and nickel steel, 174
——— steel, iron, and Delta metal,
——— steel pipes, 1419 [174
——— zinc and lead water pipes, 420
Corrugated furnaces, 1568
——— -iron galvanised sheets, 234, 365
——— roofing, 231, 234, 365
———, strength of, 234
——— steel steam tube bends, 1686
Coslettising iron, 2712
Cost of aerial ropeways, 2839
——— accuracy (unnecessary), 779
——— artesian wells, 2840
——— boring (mining), 2841
——— breaking ground, mining, 2592
——— breakwaters, 1456, 2841
——— bridges, 2842
——— canals, 1513, 1516, 2847
——— in Great Britain, 1516
——— lateral, 1513
——— coal cutting, 2583, 2589
——— chlorinating water, 1419
——— compressing air, 1086
——— concrete, 2852
——— bridges, 2845
——— cotton mill electrical driving, 790
——— docks, 1459, 1474, 2856
——— drainage, 2857
——— dredging, 1484, 2857
——— plant, 1484, 2857
——— drilling rock, 1487
——— electric power stations, 2860
——— tramways, 689
——— engineering factories, 2861
——— esplanades, 2861
——— excavating, mechanical, 2861

COT

Cost of factories, 2861
——— floating docks, 1474
——— gas purifiers, 2468
——— gasworks, 2451, 2867
——— goldmill equipment, 2639
——— graving docks, 1474
——— groynes, 2868
——— harbours, 2869
——— haulage, 756
——— heating, 2869
——— impounding reservoirs, 2891
——— irrigation, 1394
——— landing slips, 2870
——— lateral canals, 1513
——— lifting by hydraulic power, 1242
——— light railways, 2875
——— lighthouses, 1495, 2870
——— lighting, gas and electric, 2871
——— locks, 2872
——— marine piers, 2872
——— mechanical excavating, 2861
——— milling, gold, 2639
——— mining operations, 2575, 2577, 2589,
——— oil engines, 1904 [2593, 2594, 2602
——— passenger vehicle, working, 759
——— paving, street, 751, 2880
——— quay walls, 1465
——— quays, 2855
——— railway track renewal, 664
——— railways, 543, 553, 2872
——— reinforced concrete bridges, 2846
——— reservoirs, 2891
——— river canalisation, 1513
——— regulation, 1510
——— road maintenance, 720
——— roads, 717, 2875
——— rock cutting, 2859
——— removal, subaqueous, 1487
——— roofs, 2882
——— scraping water mains, 2892
——— sea cables, 2882
——— seaside resorts, 2882
——— sea-walls, 2882
——— sewage works, 2883
——— sinking mining shafts, 2575
——— stands, 2884
——— steel bars, estimating, 780
——— street lighting, 2871
——— paving, 751
——— surface tarring roads, 733, 755
——— tanks, 2884
——— telephones, 2885
——— tramway working, 2886
——— tramways, 689, 2885
——— transport, 47
——— tube wells, 2887
——— tunnels, 2887
——— underground haulage, 2602
——— vehicle working, 756, 759
——— water lifting, 1242
——— towers, 2890
——— waterworks, 2890
——— working electric tramways, 2886
——— ——— English railways, 661
Cotton machinery, power required for, 789
——— mill boilers, 787
——— mills, 784
——— costs of electrical driving, 790
——— horse-power required for, 789
——— Lancashire, machinery in, 788

COT

Cotton mills, machine sizes, 792
—— ropes, 870
—— —— splicing, 879
——, silicate, 1220
—— yarn measurement, 788
Counterbalancing locomotive driving-wheels, [599
Counters, Veeder magnetic 1452
Country roads, 726
Couple roof, 317
Couplings, 1007, 1176
—— flexible, 1007
—— muff, 1176
—— pipe, for automatic brake, 639
—— railway, 633
—— —— tube, 1081
Course, damp, in walls, 368
'Coverine,' paint, Carson's, 2802
Coverings, domes and conical, reinforced concrete, 469
—— — roof, 322
—— steam pipe, loss of heat from, 1694
—— relative value of, 1695
Cracking process, 2151
Cracks in metal, repair of, 2754, 2816
Crane bearings, 815
—— brakes, 818
—— chains, 817
—— electro-lifting magnets, 822
—— gearing, 815
—— hooks, 816
—— lifting tackle, 817
—— navvies, 483
—— ropes, 816
—— shafts, 815
Cranes, 801
—— breakdown steam, 809
—— crab (Babcock & Willcox), 805
—— derrick, 808
—— electric, 802, 820
—— —— runabout, 809
—— electrical equipment of, 820
—— floating, 811
—— general data of, 815
—— —— notes on, 808
—— giant, 813
—— Goliath, 807
—— hand, 801
—— hydraulic, 801
—— locomotive, 808
—— overhead runways, 808
—— travelling, 802
—— quay, 1470
—— runabout, 809
—— shipbuilding, 811
—— structures, 812
—— travelling, 802
—— wharf, 810
Crank axles of locomotives, 596, 611
—— chamber, lubrication of, 984
—— connecting rod mechanism, 1745
—— effort and turning moment, 1746
—— pin, diameter of, 1003
—— shaft alignment, 1739, 1905
—— shaft-bearing lubrication, 984
—— shafts, 1004
—— —— of Diesel engines, 1904
—— —— steamers, standard rules for, 1005
Cranked gimbal dial, Stanley's, 66
Creosote, British standard, 363

CUR

Creosote, oil, 362
Creosoting timber, 361, 539
—— —— plants, 361
Cresswell's Asbestos Co.'s 'Chekko' linings, 819, 850, 897
Crewe electric railway signals, 650
Cribs, 511
Crimp and Bruges' formula for discharging capacity of sewers, 1359
Critical speed in steam turbines, 1802
—— velocity, irrigation canals, 1390
—— —— in rivers, 1507
Crockatt, W., & Sons' Simplex boiler tube brushes and scrapers, 1684
—— — —— —— turbine tube cleaners, 1684
—— —— —— —— Salinometer, 1637
—— —— —— valve reseaters, 1670
Cropping machines, power for, 1056
Crosby de-superheater, 1622
—— feed-water regulator, 1672
—— indicator (steam), 1715
—— —— link gear, 953
Crossings, railway, leads and angles of, 557
—— —— rule for leads of, 557
—— tables and leads of angles of, 561
—— —— tramway, 689
Crossley's gas engines, 1869
—— oil engines, 1917, 1925
—— 'Premier' gas engines, 1874
—— —— oil engines, 1918
—— —— refuse fuel plants, 2181
—— suction gas plant, 2180
Cross-overs, 538
Cross sections of belts, 858
Croton dam, the, 1413
Crowds, weight of, 44
Crowds, pressure of, 44
'Crown' bond for electric tramway rails, 698
Croydon street lighting, 2871
Crucible melting furnaces, 2666
Crucibles, 434
Cruickshank Smith, value of paints, 417
Crushing mills, gold, 2640
—— oil, 793
—— strength of bricks, 175, 427, 441
—— —— timber, 182
C.T.S. cables, 2244
—— wiring system, 2254
Culverts, discharge from, 1359
—— —— —— submerged orifices of, 1359
—— —— through long, 260
Cumberland electrolytic process for preventing corrosion, 1624
Cup leathers, friction of, 1265
Cupolas, 2662
—— centrifugal fans for, 2663
Cupro-nickel alloys, 2733
Current, alternating, motors, 2286
—— meters, 1292, 1506
Currents in conductors, laws of, 2204
Curvature, radii of, railway rails, 556
Curves, canal, 1524
—— of displacement of ships, 2354
—— polar (light), 2528
—— , radii of, table of, 564
—— railway, 555, 556
—— —— crossing, 557
—— —— ranging, 562

CUR

Curves, railway, resistance of, 585
——— setting out by offsets, 563
——— superelevation of outer rail for, 555, 556
——— tables for calculating, 564
——— transition, 567
——— vertical, 569
Cutter dredgers, suction, 1483
Cutters, coal, 2583
——, the 'Hardiax,' 2584
——— milling, 1024
——— speed of, 1026
——— pitch of teeth of, 1030
——— tube, 1089
——— wood, sharpening angles of, 1072
Cutting edges of drills, 1033
———, electric, 2792
——— lubricants, 1078
——— machines, metal, 1019
———, oxy-acetylene, 2777
——— plates, power required for, 1061
——— speeds of lathe tools, 1072
——— tools, angles of, 1072
——— tempering, 1062, 1066, 1700
Cuttings, railway, walls for, 514
——— to determine volume of earth in, 481
Cyanogen, 2469
Cycle engineers' screw threads, 1149
Cyclic variation recorder, Mirrlees, 1908
Cycloidal teeth, 910
Cyclostyle, 94
Cylinder area ratios (marine engines), 2372
——— bridge piers, 258
——— diameter (marine engines), calculation of, 2374
——— ratios, 1737, 2372
——— of marine engines, 2372
——— sinking by pneumatic process, 503
Cylinders, allowable pressure in, 192
——— engine, 592, 1740, 2372
———, for foundations, 503
——— hollow, allowable pressure in, 192
——— hydraulic, thickness of, 1244
——— strength of, 1244
——— of compound and simple locomotives of equivalent power, 592
——— locomotives, 592
——— gas and oil engines, lubrication of, 988
——— hydrostatic presses, thickness of metal of, 1349
——— locomotives, 592
——— marine engines, 2372, 2374
——— steam engines, 1740
———, sinking, by pneumatic process, 503
——— locomotive, 2 and 4, relative areas of, 592
——— ram, 1349
——— thick, allowable pressure in, 192
Cylindrical iron boilers, marine, length of, 2395
———, diameter of tubes of, 2397
——— sewers, brickwork for, 1408
——— valves for locomotives, 593

D.A. (acetylene), 699, 1491, 1493, 2171, 2483, 2485, 2763
——— rail welding, 699
Daimler motor-car engines, 1938
Dam, Assuan, 1415

DES

Dam, Croton, 1413
——— Vyrnwy, 1413
Damp course of walls, 368
———-proofing, 'Callendrite,' 368, 1412
Dampness in walls, to remedy, 407, 2804
Dams, 508, 1292, 1411
——— balk, 511
——— basic economy of, 1396
——— coffer, or puddle, 509
——— concrete, 1292
——— design of, 1294
——— François cementation process for, 1412
——— earth, 1292, 1411
——— masonry, 1292, 1411
——— Molesworth's rules for, 1411
——— puddle, 509
——— Rankine's rule for, 1411
——— recently built, 1398A
——— reinforced concrete, 1292
——— rock-fill, 1292
——— shrinkage of, 1398
——— single balk, 511
——— spillway, 1295
——— stability of, 1294
——— timber, 1292
——— water losses through, 1390
Danish regulations for working lifts, 1354
Danks-Netherton vertical boiler, 1595
Danube river training works, 1509
D.A.V. gas engine, Crossley's, 1875
Davey-Paxman (*see* Paxman)
Dawson, J., & Son's 'Lincona' Balata V Belt Drive, 881
——— endless driving band, 868
——— Philip, tensile strength of copper, 173
Day, C., on tar fuel for Diesel engines, 2160
Daylight, artificial, 2546
Daysohm's electric welding system, 2784
D.B. patent conveyors, 846
Deacon water meter, 1438
Dead-weight of ships, 2363
Deals, standards of, 356
De-aerator, Allen, 1639
——— Kestner, 1639
——— Neckar, 1640
——— Weir, 1640
Decomposition of light in the spectrum, 119
Dee Superheaters Ltd., 'Dee' Superheater, 1620
Deep mine shafts, 2575
——- wells, water-raising from, by compressed air, 1276
Deflection of beams, 219, 315, 462
Degassing plant, 1638
Degree, length of, in latitude and longitude, 75
Degrees of temperature, conversion of, 99
——— value of, 102
Delta metal, 170, 171, 174, 962, 2441, 2443, 2724, 2725, 2732, 2737, 2739, 2744
——— corrosion of, 174
Demagnetisers, 1052
Density, change of, by heat, 116
Depositing floating dock, 1480
Depreciation of plant and machinery, 2896
Depth of keystones for arches, 524
Derbyshire iron, 2682
Derrick cranes, 808
Designs, registration of, 2835
Despatch tubes, pneumatic, 1202
Destructors, refuse, 2148
——— Heenan, 2148

DES

Destructors, refuse Pontypridd, 2148
De-Superheaters, 1620
——————— Crosby, 1622
——————— Drayton, 1621
Detonators, 2573, 2798
Development of water-power, 1283
Dewar, greatest artificial cold produced by, 101
Dexine sheets, steam joints, 1690
——— pump valves, 1264
Dexonite, 2346
Diagram factor (steam indicator), 2373
——— of steam indicator, to interpret, 1717
Dial, cranked gimbal, Stanley's, 66
Dials, clock, size of, 2815
Diameter of axles, 1003
——————— crank pins, 1003
——————— marine engine cylinders, 2374
——————— rivet holes, 227
——————— shafting, 1004, 1005
——————— steam main for heating, 2493
——————— tubes of cylindrical marine boilers,
Diamond mesh expanded metal, 337 [2395
——— rock-boring, 2563
——— turning tools, 1087
Diaphragm gland loss in steam turbines, 1792
——————— of theodolites, 62
Dick, R. & J., balata belting, 856, 857
Dickbalata element driving ropes, 884 [1452
Dickinson F. E., Veeder magnetic counter,
——————— John, Jaquet indicator, 1450
——————————————— hand tachograph, 1451
Dick's balata belting, 856
Die-casting, 2756
Diesel engines (*see* Oil engines, Diesel)
Dilatation by heat, 116
'Dionic' method of water testing, 1637
Dip, magnetic, 74
—— of horizon, 57
Dipper dredgers, 1481
Direct-acting feed pumps, 1254
——— current generators, 2261
——————————————, Mather & Platt, 2261
——————— motors, 2278
——————————— efficiencies of, 2280
——————————————, speed regulation of,
——————————traction motors, 2282 [2281
Disc drum type turbines, 1792
——— friction clutch, 894
——————— in turbines, 1794
Discs, centrifugal stresses in, 1797
Discharge of rivers, 1286 [1252
——————— water from reciprocating pumps,
——————————————— submerged orifice of a culvert, 1359
——————————— in sewers, 1359
——————————— through long culverts, 260
——————————————— orifices, 1306
——————————————— pipes, 1298, 1364, 1419, 1423, 1426, 1429
Discharging accumulators, 2257
Disclosure of secrets, 2905
Displacement (ship), coefficients of, 2354
——————— curves of ships, 2355 [2485, 2763
Dissolved acetylene, 699, 1491, 1493, 2171, 2483,
Distance at which sounds can be heard, 118
——————— measurements, 53
——————— of objects at sea, 76
——————— sun from earth, 76
——————— visible from lighthouses, 1497
Distemper, 418

DRA

Distillation of coal, products from, 2459
Distillers, water, 797
Distribution of air, 2515
District Chemical Co.'s acetylene purifying material, 2763
——————————————— 'Suder' Bronze, 2771
——————————————— Welding wire, 2769
Diving, 1501
——— bells, 1502
Division plate of turbines (water), 1320
Dock, bolted sectional, 1477
——— depositing floating, 1478
——— gates, 1465
——————— stresses in, 1465
——— Havana floating, 1477
——— offshore, 1477
——— sectional box, 1478
——— sectional pontoon, 1477
——— walls, 1463
——————— stresses in, 1463
Docking condition of vessels, 2357
Docks, 1459
——, cost of, 1459, 1474, 2356
——— floating, 1473
——————— cost of, 1474
——— graving, 1473
——————— cost of, 1474
——— offshore, 1477
——— wet, 1459
Domes and conical coverings, reinforced concrete, 469
Domestic uses of electricity, 2330
Dorman, Long & Co., tables of I beams, 235, 236
——————— W. H., & Co.'s 'Flexstel' system of ball and socket joints, 1084
——————————————— oil engines, 1924
Double-acting force pump, 1250
——————————— steam engines, horse-power of.
——— floor, 328 [1707
——— helical gear curves, 1819
——————————— gearing, 927, 1818
——————————— gears, turbine, 929, 1818
Dover breakwater, 1456
Dowelling, 381
Dowson gas, 2174
——————— producers, 2174
Doxford Diesel engines, 2422, 2423
Draft of ships, 2353
——— gauges of steam boilers, 1545
——— tube (water turbine), 1330
Drag bar conveyors, 845
Dragline excavators, 484, 2864
Drain pipes, 429, 1379
——————— comparative value of, 1431
Drainage, cost of, 2851
——————— house, 1379
——————— road, 728
Drainer, cellar, Willcox-Penberthy, 1263
Drains, flushing, 1366
——————— protective distance from wells, 1409
Draught, mechanical, 2122, 2392
——————— boiler, 1542, 2392
——————— of ships' boilers, 2392
——————— pressures, rate of combustion for, 1543
——————— required for efficient combustion of fuel, 1544
Draughtsmen, kink for, 91
Drawbars, railway, 633
Drawhooks, railway, 633

Drawing, 87
——— appliances, 90
——— board edge, 91
——— illustrations of materials, 89
——— office practice, 92
——— paper, sizes of, 92
——— pen, Stanley's patent, 91
———, wire, 2816
Drawings for patent specifications, 90
Drayton, A. M., regulator, 2509
——— automatic temperature regulator,
——— boiler damper, 2508 [1642, 2509
——— de-superheater, 1621
——— feed-water regulator, 1673
——— gas regulator, 2475
——— oil-burner regulators, 2168
——— separator, 1704
——— pressure regulator, 1676
——— steam traps, 1700, 1702
Dredgers, compound, 1483
——— dipper, 1481
——— grab or grapple, 1482
——— hopper service, 1484
——— hydraulic, 1482
——— ladder, 1480
——— sand pump, 1483
——— suction, 1482
——— cutter, 1483
Dredging, 1480
——— Bombay Harbour, 1481
——— cost of, 1484, 2857
——— cost of plant, 1484
——— for alluvial minerals, 2646
——— Port Natal, 1486
——— pump, 1483
——— river, 1510
Dressing ores, 2619
Dressings, belt, 862
Drifts, mining, 2573
——— timbering, 2574
Drill, giant pneumatic, 1080, 1081
——— Hardy-Simplex rock, 2568
——— Little Giant air, 1044
——— Little Giant electric, 1043
——— hammer, 1043
——— protector, Brown's, 1041
——— sizes for tapping, 1171 [2566
——— steel sharpening machine, Holman,
———, Van Dorn, 1042
Drill-jig, quick-change universal, 1047
Drilling, 1033
——— and blasting rock, 1487
——— appliances, 1046
———, diamond, mining, 2563
——— glass, 2813
——— hard steel, 2815
——— lubricant for, 1041
——— machine, vice for, gravity, 1046
——— machines, power for, 1054
——— thrust bearings for, 1040
———, mining, 2564, 2593
——— notes on, 1039
——— plant (mining), compressed air,
———, power for, 1054 [2568
Drills, 1033
——— cutting edges of, 1033
——— Firth's speedicut, 1035
——— grinding, 1036
——— pneumatic, 1081
——— portable electric, 1042

Drills, rock, 2565
——— Hardy-Simplex, 2568
——— speed of, 1036
——— twist, 1034
———, British Standard specification for, 1039
———, Morse taper measures, 1038
Drive, automatic reversing, for planing machines, 1021
Driver's equalising brake valve, 639, 641
Driving belts, 855
——— Tullis on, 863
——— chains, standard, 899
———, Morse rocker-joint, 905
——— electric power, in workshops, 773
———, ropes, 788, 808, 870
——— sizes of, 870
——— wheels of locomotives, counterbalancing, 599
Drop-stamps, Brett's, 1135
——— Massey's, 1132
Drop-valve engine, Galloway, 1730
——— reversing and overwinding gear, Robey, 1726
Drought, effect of, on roads, 729
Drum disc type steam turbines, 1792
Drums, centrifugal stresses in, 1797
——— and pulleys, rope driving, sizes of, 870
Dry battery cells, 2255
——— grinding, 1107
——— measure, 7
——— purification plant, gas, 2455
Dryers, steam, 1699
Drying and heating, 2511
——— apparatus, timber, 359
——— of woods, 130
——— room temperatures, 2512
Dubbing for fire-hose, 2551
Ductile brass, 2723
Ducts, flow of air through, 2519
Dummy loss in steam turbines, 1793
Dumpy level, 66
Dunbar Scott's auxiliary top and side telescope, 63
Duplex packing, Beldam, 1755
Dura battery cells, 2255
Durability of timber, 355
——— wire ropes, 826
Duralumin alloy (Vickers), 2735
Dust, coal, 2617
Duty of engines, 1713
——— mining stamps, 2639
Dwellings, London Building Act, 370
Dykes, longitudinal, 1509
Dynamics, 27
Dynamite, 2797
Dynamo lubrication, 985
Dynamometer, Brotherhood fluid friction, 38
——— fan, 41
——— Froude's water, 38
——— waterpipe absorption, 37

EAGLE aero engines, 2047
Earth, angle of repose of, 494
——— dams, 1292, 1411
———, excavation of, 483
——— getting, 491
——— magnetic force of, 74
——— method of getting, 491

EAR

Earth, polar radius of, 76
—— pressure of, on retaining walls, 514
—— radius of, and distance from sun, 76
——, temperature of, 106
—— the, 76
—— transporting, 491
—— volume of, in cuttings, 481
Earths and clays, 425
—— specific gravity of, 127
Earthwork, 481
—— tables, 482
East Coast iron, 2681
——, true, to determine, 74
Eaves gutters, 'Everite,' 420
Ebonite, British standard, 2346
—— machining, 1095
—— sheet and rod, weight of, 144
—— specific gravity of, 130
Eccentric loads, 223
'Economic' boiler, Paxman's, 1581, 1606
Economical speeds of ships, 2366
Economiser data, 1647
Economisers, 1644
——, Green's, 1644
Economy of fuel in cargo steamers, 2368
—— high-pressure steam working, 1710
'Eddystone' boiler appliance, 1609
Eddystone lighthouse, 1493
Edison nickel-iron accumulator, 2260
Edwards air-pump, 1842
Efficiency of air, 1867
—— boilers, 1553
—— electric lamps, 2325
—— engines, 1709, 1710
—— hot-air engines, 1931
—— open fire grates, 2491
—— power plant, 1539
—— rope driving, 872
—— steam boiler plants, 1555
—— steam turbines, 1791
Effluents, sewage, standard of purity of, 1378
Egg-shaped sewers, 1363
Egyptian yarn, 786
Ejectair, Hick-Breguet, 1845
Ejector air pumps, 1845
—— Mirrlees, 1846
—— Weir, 1849
—— condenser, 1834
—— steam-jet pump, Willcox's, 1657
Elastic fluids, specific gravities of, 132
Elasticity, modulus of, Young's, 184
Electric accumulators, 2256
—— acid for, 2257
—— air coolers, 2267
—— filters, 2268
—— arc cutting, 2792
—— lamps, 2326
—— bare wire, 2261
—— batteries, primary, 2255
—— vehicles, 704, 760
—— battery locomotives, 707
—— blower, portable, 2269
—— brakes, 678
—— building equipment, 2231
—— cables for collieries, 2612
—— flexible, 2207, 2211
—— paper, 2210
—— vulcanised rubber, 2209, 2211
—— for installation work, 2243
—— cement, 397, 405

ELE

Electric circuits and fire streams, 2549
—— clocks and time service, 2327, 2350A
——, Gent's, 2327 [2327
—— Pul-syn-etic System,
—— conductors, aluminium, 2213
—— copper, 2205, 2221
—— control gear, automatic, 2289
—— converters, 2270
—— cooking, 2332
—— cranes, 802, 820
—— currents in conductors, laws of, 2204
—— direct current generators, 2262
—— drills, portable, 1042
—— driving of cotton mills, 790
—— enamelled wire, 2309
—— engineering, 2203
—— furnaces, 2667
—— power required for, 2671
—— fuses, 2224
—— generators, 2261
—— heating, 2331
—— installation cables, 2243
—— industrial trucks, 704, 760
—— insulating materials, 2346
—— lamp flexible cords, 2208, 2325
—— lamps, 2321
—— efficiency of, 2325
—— lifting magnets, 822
—— lifts, 831
—— light cables, 2209
—— Silvertown, 2211
—— testing, 2319, 2320
—— carbons, 2326
—— mains, testing, 2319, 2320
—— lighting of factories, 2530, 2544
—— railway carriages, 633
—— locomotives, 676, 2602
——, brakes for, 678
—— lorries, 705
—— machinery, brakes for, 678
—— bare copper wire for, 2261
—— British standards for, 2260
—— measuring instruments, 2310, 2350A
——, Evershed, 2319
——, Weston, 2312
—— mercury vapour rectifiers, 2277
—— meters, 2315
—— specification for, 2315
—— motors, 2278
—— overhead lines, 2220
—— power driving in workshops, 773
—— factor improvement, 2307
—— stations, cost of, 2860 [1539
—— fuel consumption of,
—— precipitation of smoke and dust, 2334
—— railways, 669
—— railway signalling, 645
—— Recorder, the Shakespear, 2087
—— resistance pyrometers, 2678
——, temperature effect, 2205
—— resistivity of metals and alloys, 2204
—— ship equipment, 2240
—— signalling in coal mines, 2619
—— supply cables, 2210
—— standard sizes of, 2210
—— switch gear, 2297
—— testing of electricity supply mains, 2244
—— time service, 2327
—— traction, 669
—— traction on canals, 1525

ELE

Electric traction on long-distance railways, 669
——— ——— suburban railways, 678
——— ——— tramways, 683
——— ——— railless, 703
——— ——— switchboard, 702
——— tramways, 683
——— ——— axles for, 683 [dum, 700
——— ——— Board of Trade memorandum regulations for, 701
——— ——— ——— rules, 701
——— ——— braking, 691
——— ——— cables for, 700
——— ——— cars for, 701
——— ——— cost of, 689
——— ——— equipment of, 692
——— ——— fishplates for, 684
——— ——— foundations, 689
——— ——— guard wire for, 697
——— ——— leakage on, 700
——— ——— overhead construction, 694
——— ——— paving, 689
——— ——— permanent way, 701
——— ——— points and crossings, 689
——— ——— poles for, 695
——— ——— rail bounding, 697
——— ——— joints, 688
——— ——— welding, 699
——— ——— rails for, 684
——— ——— sectioning of, 692
——— ——— span wire for, 697
——— ——— static balancers, 2304
——— ——— trolley wire for, 694, 697
——— ——— trolleys for, 693
——— ——— tyres for, 683
——— transformers, 2301
——— transmission, turbines (steam), 2390
——— trolley buses, Ransomes', 705
——— trucks, industrial, 704, 760
——— units, 2203
——— vehicles, battery, 704
——— working costs, 760
——— wall plugs and sockets, 2242
——— ware, 430
——— welding, 699, 2779
——— winding, 832
——— plants, mining, 2604
——— wiring, British standard specification for, 2245
——— Callender's 'Kalibond,' 2252
——— C. T. S. system, 2254
——— Henley system, 2247
——— Institution of Electrical Engineers rules for, 2231
——— Johnson&Phillipssystem,2251
——— Macintosh system, 2254
Electrically operated points and signals:
——— British Power Railway Signal Co.'s, low voltage system, 658
——— Crewe system, 650
——— Westinghouse systems,645
Electricity Commissioners, loans by, 2226
——— ——— regulations, 2227,
——— domestic uses of, 2230 [2229
——— generation by waste heat, 2200
——— of, in Great Britain,
——— in agriculture, 2335 [2226
——— meters, 2315
——— stations, gas and oil engine, 1884

ENG

Electricity supply, 2226
——— ——— cables, 2209
——— ——— for collieries, 2612
——— *v.* gas, 2476
——— works, refuse destructors, 2148
Electro-galvanising, 2711
——— magnetic and electro-pneumatic control for motor-coach trains, 682
——— magnets, lifting, 822
——— mechanical brakes, 2298
——— pneumatic railway signalling system, Westinghouse, 645 [184
Electrolytic copper, modulus of elasticity of,
Elevators, Giant, mining, 2644
——— grain, 1472
——— hydraulic, mining, 2644
——— mechanical, mining, 2645
——— Ruble, mining, 2645 [brakes, 2298
Elliston, Evans & Jackson's electro-mechanical
Elongation produced by stress, 134
Embankments, labour upon, 45
——— slopes of, 492
Embrittlement of boilers, 1627
Emery wheel guards, 1112
——— speeds, 1108
Enamel, to remove, 2817
Enamelled copper wire, 2309
Encasing, steel work, 337
Energy, luminous, 120
——— of a blow, 1128
——— rotating body, 34
Engineering bricks, 426
——— Developments Ltd., 'Fluxode system, 2787
——— factories, cost of, 2861
——— sanitary, 1359 [1669
——— Specialists Ltd., 'Vee-Reg' valve,
——— Standards Committee's specifications (*see* British Standards)
——— works, cost of, 2861
——— workshops, 765
Engineers as arbitrators, 2910
——— witnesses, 2905
———, employment of, 2904
——— evidence before committees, 2906
——— fees, 2903
———, legal notes for, 2903
——— status of, 2903
——— liability for negligence, 2906
——— rights *re* inventions, 2905
———, powers, and duties, under contract, 2906
———, road, legal points for, 714
Engines, aero (*see* Aero engines), 2045
——— back pressure and extraction, 1731
——— bearing pressures, 1738
——— bearings, 1741
——— Belliss and Morcom high-speed, 1731
——— cylinders, 592, 1740, 2372
——— Davey-Paxman steam, 1724, 1733
——— ——— gas, 1874
——— design of, 1737
——— design of ship's, 2372
——— Diesel, 1893, 2411
———, crankshafts, 1904
——— drop valve, 1726
——— duty of, 1713
——— economy of high-pressure working,
——— efficiency of, 1709 [1710

ENG

Engines, expansions in various types of, 1707
——— failures of parts of gas and oil, 1889
——— flywheels, 1742, 1888
——— foundation bolts, 1740
——— frames, 1740
——— Galloway steam, 1730, 1735
——— gas (*see* Gas engines), 1869
——— governor of, 32, 1743
——— high-speed Belliss & Morcom, 1731
——— hot-air, 1931
——— kerosene, 1922
——— locomotive, 571
——— power of, 587
——— marine, 1926, 2372, 2375, 2411, 2443
——— metals for marine, 2443
——— motor-car, 1936 (*see* Automobile engines)
——— oil, 1893
——— parts, alignment of, 1739
——— Paxman steam, 1724, 1733
——— pistons, 1756
——— portable Davey-Paxman, 1733
——— Robey high-speed steam, 1728
——— speed of locomotive, 586
——— steam expansion table, 1707
——— horse-power of, 1707, 1719
——— *v.* oil, 2439
——— stresses, 1737
——— tests, typical, 1713
——— traction, 722
——— triple-expansion, 2469, 2373
——— types for various duties 1721
——— uniflow, 1722
——— water-pressure, 1309
——— wind, 1187
——— winding, steam consumption of, 1714
English Electric Co.'s Diesel engines, 1896
——— direct current traction motors, 2285
——— electric brakes, 691
——— Fullagar gas engines, 1883
——— oil engines, 1898, 1902
——— induction motors, 2291
——— regulators, 2305
——— motor coach trains equipment, 682
——— Pelton wheel buckets, 1310
——— regulator, 1313
——— safety device for electric winders, 832
——— synchronous motors, 2287
——— traction motors, 2285
——— transformers, 2301
——— Ward-Leonard electric lengths, 1 [winding system, 832
——— standard screw thread, 1139
Entrance locks, 1465
Epicyclic wheel teeth, 910
Equation of time, 74
Equatorial parallax of sun, 76
——— radius of earth, 76
Equilibrium of retaining-walls, 515
Equipment for motor-coach trains, 682
——— of gold mills, cost of, 2639
Erith Roe mechanical stoker, 2117
——— tunnel dryers for timber seasoning, 359
Erosion of river banks, 1508

EXP

Esplanades, cost of, 2861
Estimate, railway, House of Commons rules
——— sheet for ships, 2367 [for, 664
Estimating painters' work, 417
Eternit Pietra Artificiale (London), Ltd., 'Italit' roofing, 327
Ether, reduction of volume under pressure, 117
Evaporative power of coal, 2095
Evaporation and humidity, 1384
———, annual, 1385
——— boiler, 1553, 1582, 2095 [1388
——— Californian experiments on, 1385,
——— Egyptian experiments on, 1384
——— factors, 2095
——— from ice, 1389
——— irrigation canals, 1389
——— soil, 1388
——— water surface, 1383
———, fuel cost of, 2103
——— losses by, 1387
——— monthly, 1386
——— of Cochran boilers, 1582
——— steam, 1582
——— ratio of pan evaporation to open water, 1398
Evaporators, 794
——— Caird & Rayner, 2405
——— for trade liquids, 795
——— Mirrlees Watson Co., 795, 796, 797
——— multiple-effect, 795
——— refrigerating, 1215
———, Weir, 2406
Everest theodolite, 61
'Everite' eaves gutters, 420
——— gas pipes, 2479
——— rain water pipes, 420
——— sheeting, 323
Evershed Dionic method of water testing, 1637
——— electrical measuring instruments, [2319, 2320
Excavation, 483
———, cost of, 2861
Excavations, slopes of, 492
Excavator machines, 483, 2861
———, Ruston (Ruston-Hornsby), 483
Excess, spherical, 68
Exhaust gases, 2081
——— steam, heating by, 2499
——— oil separator, 1682
——— pipes, 1689
——— turbo-charging, 2435
Exhausters, gas, 2453
——— Sturtevant, for ventilation, 2520
Exide battery, 707
'Exmet' reinforcement for brickwork, 334
'Expamet' expanded metal, 334
Expanded Metal Co.'s expanded metal, 334, 452
——— metallic mesh work, 334
———, concrete mixtures for, 338, 340
——— flooring, 335
——— lathings, 334, 335
——— partitions, 335
——— suspended ceilings, 339
——— steel concrete pipes, conduits, sewers
——— reinforcement, 334 [340
——— work encasing, 337
Expansion, isothermal and adiabatic, 1863
——— of air by heat, 117
——— fire bricks, 430
——— steam, 1705
——— substances by heat, 116

EXP

Expansions in various types of engines, 1707
Expenditure, railway, 661
Explosions, boiler, 1654
Explosive signals, sea, 1500
Explosives, 2797
——— mining, 2572, 2797
——— permitted, 2798
Express type water-tube boiler, 2397
Extraction engines, 1731
Extruded metal, 171, 2725
Extinguishers, fire, 2549
Eytelwein's formula for discharge from submerged orifice of a culvert, 1359
——— water pipes, 1419

FABRIC linings, 850
Fabroil gears, 919
Facing bricks, 426
Factories, cost of, 2861
——— electric power driving in, 773
——— engineering, 765
——— lighting of, 2530, 2544, 2545
——— reinforced concrete, 478
Factory room temperatures, 2512
——— windows, care of, 773
——— work, reinforced concrete for, 478
Fahrenheit-Centigrade conversion table, 99
Failures of boilers, 1654
Failures of parts of gas and oil engines, 1889
Fairbairn's rules for diameters of rivets, 227
Falk cast weld for rails, 699
Falling bodies, 27
Fan dynamometer, 41
Fans, centrifugal, 2520, 2663
—— cupola, 2663
—— movement of air by, 2518
—— ventilating, 2663
—— wing, 2521
Fatigue of metals, 200
Feed pipes, locomotive, 607, 608
—— pumps, 1254, 1256
—— pumps, Clarke Chapman direct-acting, ——— Warner Anthony, 1254 [1256
——— Weir, 1256
—— water, 1624
——, gas removal from, 1638
——— heaters, Drayton, 1642
——— Hudson, 1641
——— Row (Royles), 1641
——— Weir, 605, 2406
—— heating, 1640
———, oil elimination from, 1681
——— regulators, 1672
——— Arca, 1677, 1678
——— Copes, 1678
——— Crosby, 1672
——— Drayton, 1673, 1676
——— Mumford, 1677
——— Weir's, 1676
——— Thermofeed, 1674
—— softening, 1628
——— temperature regulators, 1642
——— Drayton, ——— treatment of, 1625 [1642
Fees, arbitrator's, 2911
—— engineer's, 2903, 2905
Feet, decimals of, inches reduced to, 2924
Felt roofing paint, 2805
—— underlining for roofs, 413

FIS

Felt wipers for machine slides, 1053
Ferodo friction linings, 819, 850, 897
Ferranti-Hopkinson stop valve, 1668
'Ferri-silicon paint,' Carson's, 2802
Ferrite, 2678
Ferro alloys, 2693
Ferro-arc welding, 2785
Ferro-concrete, 388, 405
——— tubbing, 2579
——— vessels, Lloyd's rules for, 2445
'Ferrocrete,' 405
'Ferrox' welding flux, 2770
—— welding wire, 2771
Fiat oil engine Diesel, 2430
Fibre, 2814
——— machining, 1095
——— tool steel for cutting, 1095
'Fibrent' asbestos cement, 324
——— sheets, 324
——— slates, 413
Field work, astronomy in, 77
Fielding & Platt gas engine, 1874
——— starter, 1886
——— hydraulic riveter, 1355
——— oil engines, 1914
Files, 1088
—— plating, 1089
—— resharpening, 1088
—— to clean, 1089
Filling materials for acetylene welding, 2769
Filters, Heenan air, 2268
——— pulsometer 'Torrent,' 1416
Filtration, 1416
——— of oil, 982
'Filtrator' system of scale prevention, 1637
Fine screw threads, British standard, 1141,
Fir beams, breaking weight of, 185 [1156
—— girders, 330
—— joists, 331
Fire bars, 2109, 2395
—— boxes, locomotive, 600
——— life of, 601
—— bricks, 430
——, gasworks, 433
——— expansion of, 430
—— extinction, 2547
——— in mines, 2619
—— extinguishers, 2549
—— grates, open, efficiency of, 2491
—— hose, dubbing for, 2551
—— proofing boards, 2813
—— resistance, reinforced concrete, 439
——— standards of, 2547
—— resisting buildings, 2551
——— cements, 435
——— floors, 333
——— glue, 399
——— materials, 341, 2547
——— paint, 2805
—— streams, 2548
——— and electric circuits, 2549
—— tube superheaters, 600, 601
Fireclay bricks, specification for, 432
Fireclays, 431
——— used on gasworks, 431
Fires, smith's, centrifugal fans for, 2663
Firth, Thos., & Sons' high-speed drills, 1035
——— steel for turbine blading, Fish glues, 396 [1805, 1806
—— plates, 537

FIS

Fishing beams, 311
Fittings, boiler, 1655
——— cast-iron pipe, 154
——— Callender's 'Kalibond' electric light, ——— C.T.S. electric light, 2254 [2252
——— Henley's electric light, 2247
——— hot water, British standard, 2497
——— Johnson & Phillips electric light, 2251
——— Mackintosh electric light, 2254
——— steam and gas, dimensions of, 150
——— trolley wire, 694
——— water pipe, 154
Flange beams, broad, 237
Flanged rails, 533
Flanges of pipes, 1166, 1667
Flanging, pressures required for, 1168, 1245
Flash temperatures, 117
——— of oils, 117
Flat bars, rolled aluminium, weight of, 139
—— bottomed rails, 533
—— plates of boilers, 1568
—— rolled iron, weight of, 135
——— steel, weight of, 137
—— ropes, 2611
—— steel, weight of, 137
Flattened strand ropes, 2607
Flexible cables, 2207
——— cords, electric light, 2207, 2325
——— couplings, 1007
——— metallic tubing, 608, 1081, 2571, 2766
——— paint, 2803
——— pipe lines, Victaulic, 1246
——— steel-wire ropes, 828
'Flexstel' ball and socket joints, 1084
Float method of measuring water flowing in stream, 1291
Floating cranes, 811, 1471
——— docks, 1473
——— cost of, 1474
——— list of, 1475
——— notes on, 1477
——— landing stages, 1502 [1506
Floats for measuring water flowing in streams,
Flood discharges, rainfall, 1403
Floor joists, 328, 329
——— ventilation of, 333
——, single joisted, 328
——— slabs, reinforced concrete, 459
——— space for machine tools, 772
Flooring, asphalt, 407
——— boards, 332
——— secret joints in, 332
——, bridge, 273
——— expanded metal, 335
——— lock-woven mesh, 453
——— weight of, 364
Floors, 328
——— concrete, 773
——— oil stains on, 773
——— double, 328
——— fire resisting, 333, 452
——— framed, 328
——— of engineering workshops, 770
——, paint for, 2804
——— reinforced concrete, 452
——— safe load on, 497
——— single joisted, 328
——— wall plates for, 320
——— wooden girders for, 329
Flotation, centre of, of ships, 2356

FOU

Flotation machines, mining, 2627
Flour milling, 783
——— weighers, 1440
Flow of air through nozzle, 1184
——— gas in mains, 2478
——— rivers, 1288, 1506
——— sand, 2817
——— sewage, hourly variation of, 1368
——— steam through orifices, 1538
——— pipes, 1691
——— turbines, 1767
——— water through orifices, 1396
——— pipes, 1298, 1423
——— flumes, and channels.
—— sheets, mining, 2628 [1298
Flue cover, asbestos cushioned, 1606
—— gases, 1542
—— composition of, 2080
Flues, boiler, dimensions, 1610
——— working strength of, 1562
Fluid friction dynamometer, Brotherhood, 38
Fluids, elastic, specific gravities of, 126
Flumes of turbines, flow of water in, 1293
Flushing chambers, 1366
——— sewers, 1366
Fluxes for acetylene welding, 2770
Fluxode welding system, 2787
Flywheels, 1742
——— gas engine, 1888
Focal length of concave lens, 119
——— convex mirror, 119
'Foliac' boiler graphite, 1626
——— flake graphite, 986
——— graphite paint, 2806
——— lubricating graphite, 986
'Fondu' electric cement, 397, 405
Foot-rule, angle measurement by, 8
Force by screw, 27
——, centrifugal, 31, 267
—— pumps, 1250, 1850
——, tractive, of locomotives, 587
Forced draught, 2122, 2392
—— lubrication, 989
Forcers, bolt, hydraulic, 1354
Forces, parallelogram of, 26
Foreign horse-powers, 1708
——— measures, 11
——— money, 16
——— weights and measures, 16
Forests and rainfall, 1406
Forge air pressure, 2865
Forgeable brass, 2723
Forgings, steam turbine, 1804
——— steel, locomotive, 614
Fosalsil (Moler), 429, 435, 1222
Foster-White water tube boiler, 2397
Fottinger, hydraulic steam turbine trans-
Foundation bolts, 1740 [mission, 1821, 1822
——— sinking by freezing process, 508
Foundations, 376, 495
——— boiler, 1605
——— bridge, 258
——— chemical treatment of, 496
——— chimney, safe load on, 497
——— electric tramway, 689
——— freezing process of sinking, 508
——— on different soils, 496
——, piled, 497
——— pneumatic, frictional resistance
——— retaining wall, 516 [of, 508

FOU

Foundations, safe loads on, 496
——— bridge, 249
——— skin friction on bridge, 258
——— steam hammer, 1128
——— tramway, 689
——— wall, 376
Founding and casting, 2753
Foundry ladle crane hooks, 816
Fox's corrugated furnaces, 1568
Fram fire-resisting floor, 452
Frame saw roller bearings, 973
Framed floors, 328
Frames of engines, 1740
Francis' table of shafting supports, 1003
——— turbines (water), 1317
François cementation process, 1412, 2581, 2582
——— ferro-concrete tubbing, 2579
Franklin motor car engine, 2579
Fraser & Chalmers coal dust conveyor, 2129
——— pulverised coal system, 2137
Freezing mixtures, 105
———, prevention of, 105
——— process of sinking foundations, 508
——— temperatures, 105
French bolts, 1157
——— canals, 1515
——— electric railways, 672
——— nuts, washers, and bolt pins, 1158
——— Government regulations with reference to reinforced concrete, 467
——— screw threads, 1145
——— standard bolts and nuts, 1151
Fret saws, speed of, 1061
Friction clutches, 889 [ropes, 809
———, coefficient of, or leather belts and ropes, 809
——— coefficients, table of, 28
———, cup leather, 1265
——— in gas mains, 2479
——— lifters for hammers, 1132
——— linings, 819, 850, 897
——— lubrication, 297
——— lubricants, coefficients of, 962
——— of air, 1184
——— per sq. ft. of plane, 1183
——— of bridge foundations, 258
——— surfaces, 818, 850, 897
———, table of coefficients of, 28
Frictional properties of standard oils, 981
——— resistance of pneumatic foundations, 508
Froude water dynamometer, 38
Frustum, 24
——— pyramidal, 24
Fry's anti-friction alloys, 2739
——— die casting alloys, 2756
——— heat transmission formulæ, 1549
Fuel and costs, electricity works, 1539, 1884, 2148
———, anthracite, 2103 [1885,
———, ash and moisture content, 2082
———, briquette, 2106
———, calorific values of, 2077, 2091
———, charcoal, 2106
——— coal, 2090
——— coal tar, 2156
——— coke, 2096, 2098, 2459
——— coke, as a boiler, 2101
——— combustion of, 2078. 2110
——— air required for, 2078
——— consumption of, in locomotives, 624
——— in electric power stations, 1539, 1884, 1885, 2148

FUS

Fuel cost of evaporation, 2103
———, Diesel engine, 2160
——— economisers, Green's, 1644
———, evaporative power of, 2095
———, gaseous, 2168
——— , air required for complete combustion of, 2191
——— calorific value of, 2169, 2191
——— , for engines, 2167, 2170
——— heating buildings, 2503
——— lignite, 2105
——— liquid, 620, 2149
——— atomisers, 2162
——— burners, 2161
——— motor, 1951
——— oil, combustion of, air required for, 2166
——— for locomotives, 620 [
——— for marine engines, 2404
——— for the Navy, 2166
——— properties of, 2157
——— peat, 2105
——— petrol, 2150
——— for engines, 2151
——— petroleum, 2149, 2156
——— pulverised coal, 2125
——— pumps for tar-oil engines, 2158
——— required for heating buildings, 2503
———, solid, 2090
——— thermal efficiency of, 1539
——— Wallsend-Howden patent liquid fuel system, 2163, 2404
——— wood, 2106
Fullagar gas engine, 1881
——— oil engine, 1902
Fuller coal pulverising plants, 2129
Furnace arches, 'Liptak,' 1608
——— blast, table of air for, 2664
——— bricks, 433
——— construction, 2671
——— efficiency, 2672
——— firing with liquid fuel, 2161
——— linings, 433, 1608
——— roof, 1608
——— ventilation of mines, 2614
Furnaces, 2659
——— and fire-bars of steamships, 2395
——— arches for, 1608
——— blast, 2660
——— gas from, 2189
——— bulb, 1568
——— corrugated, 1568
——— crucible, melting, 2666
——— electric, 2667
——— insulation of, 433
——— lift-out crucible, or pit type, 2666
——— lining for, 433, 1608
——— metallurgical, 2659
——— muffle, 2665
——— oil-fired, 2666
——— reverberatory, 2665
——— shaft, 2660
——— temperature control of, 2673
——— tilting, 2667
Fuses, electrical, 2224
———, mining, 2573, 2798
Fusible boiler plugs, 1609
——— metals, 2741
Fusion of metals, latent heat of, 2673
——— pyrometers, 2674
——— temperatures, 103

GAL

GALLAGHER & CROMPTON'S baffle bridge, 2101
Galloway boilers, 1574
——— extraction engine, 1735
——— drop piston valve engine, 1730
——— gas engine, 1880
——— reversing rolling-mill engine, 1731
——— steam accumulators, 1854
——— steam pressure regulator, 1735
——— superheaters, 1615
——— supermiser, 1651
——— uniflow steam engine, 1723
Galvanised iron corrugated sheets, 234, 365
——— ——— roofing, paint for, 234
——— ——— telegraph line wire, 177
Galvanising, 2711
Ganister, 430
Gardner gas engine, 1877
Gas and gasworks, 2449
——, acetylene, 1491, 1493, 2171, 2173, 2483
—— adiabatic expansion of, 1863
—— and oil engines, failures of parts of, 1889
—— ascension pipes, 2463
——, blast furnace, 2189
—— bulk of, 1862
—— buoys, 1489
—— burners, 2476
——, calorific value of coal, 2172, 2458
—— caps of safety lamps, 2617
—— coal, 2172
——, coal, air necessary for combustion of, 2169, 2475
——, coal, calorific value of, 2172
——— and coke, composition of, 2458
——— storage, 2452
—— coke oven, 2100, 2173
——— storage, 2452
——— used as fuel for, 2459
—— composition of, 2091, 2458
—— compressed, 1068
—— compressors, 1193
—— condensers, 2453
——, consumption of, per head of population, [2451
—— distillation products of, 2459
—— Dowson, 2174
—— dry purification plant, 2455
—— engine electricity stations, 1885
—— engines, 1869
——— breakages of parts of, 1889
——— Brotherhood, 1877
——— Browett-Lindley, 1880
——— cooling water for, 1886
——— Crossley, 1869
——— Davey-Paxman, 1874
——— Fielding & Platt, 1874
——— fixing, 1889
——— flywheels, 1888
——— Fullagar, 1881
——— Galloway, 1880
——— Gardner, 1877
——— gaseous fuels for, 2170
——— generation of electricity by, in Great Britain, 1884
——— Hornsby-Ruston, 1871
——— Lilleshall, 1878
——— Lindley-Browett, 1880
——— Lloyd's rules for marine, 1926
——— lubrication of, 988
——— National, 1871
——— packings for, 1888

GAS

Gas engines, parts of, 1887
——— Premier, 1874
——— prices of, 1883
——— Rosseline for, 1887
——— Ruston-Hornsby, 1871
——— starters, 1886
——— Tangye, 1875
——— thermal efficiency of, 1884
——— types of, 1869
——— valves of, 1888
——— Vickers, 1872
——— weights of, 1883
——, benzol from, 2154
——, cyanogen, 2469
—— Dowson, 2174
—— exhausters, 2453
—— filled lamps, lumen values of, 2532
—— fired boilers, Kirke, 2191
—— fittings, dimensions of, 152
——, flow of, in mains, 2478
—— high pressure, 2475
——— compressor for, 1193
——— in works, 773
—— holder tanks, 2457
—— holders, 2456
——, hot water supply by, 2511
—— impurities in, 2466
——, incandescent burners, 2475
—— isothermal expansion of, 1863
—— lighting, high pressure, 2475
——— of factories, 2544, 2545
—— liquor, 2463, 2465, 2474
—— made in Great Britain, 2451, 2488
—— mains, 2458, 2463
——, foul, 2463
——— friction in, 2478, 2479
——— Pole's formula, 2478
——— Unwin's formula, 2478
——— hydraulic, 2463
——— mantles, 2476
—— memoranda, 2474
—— metering, 1435
—— meters, 1443, 2479
——, Mond, 2183
—— naphthalene, 2469
—— natural, 2171
——, parliamentary regulations in reference to, 2449
—— petrol air, 2487
—— pipes, 2476
——— British standard, 165
——— threads of, 1150
——— weight of, 145, 2477
—— plant, Appleby-Bentley, 2179
——— Mond, 2183
——— suction, Crossley's, 2180
——— Dowson, 2174
——— Ruston, 2187
—— power units, 1861
—— pressures, 2479
——, producer, 2173, 2174
——— composition of, 2175, 2460
——— heat value of, 2175
—— producers, 2174
——— Appleby-Bentley, 2179
——— Crossley, 2180
——— Hornsby-Stockport, 2187
——— Ruston, 2187
——— Kerpely, 2177
——— Mond, 2183
—— production, coke required for, 2459

GAS

Gas purification, 2454, 2465, 2467
—— purifiers, cost of, 2469
—— referee's tests, 2481
—— removal from feed water, 1637
—— retort house, 2453
—— —— governors, 2464
—— retorts, 434, 2453, 2462
—— —— —, bench fittings, 2463
—— —— —, benches, 2461
—— —— —, specification for, 434
—— scrubbers and washers, 2454
—— storage capacity, 2456
—— suction plant, Crossley, 2180
—— —— —— Dowson, 2174
—— —— —— Ruston, 2187
—— tar, 2463
—— tests, 2481
—— *v.* electricity, 2476
—, volume of oxygen and air to burn, 2169,
—— washers and scrubbers, 2454 [2475
—, water, 2189, 2471
—— weight of, 124, 1861
—— works, capital expenditure on, 2451
—— —— coal storage, 2452
—— —— condensers, 2453
—— —— conveyors and elevators, 2464
—— —— cost of, 2451, 2867
—— —— exhausters, 2453
—— —— fire bricks used on, 433
—— —— grate and chimney area, 2460
—— —— land required for, 2452
—— —— mains, 2458, 2463
—— —— parliamentary Acts, 2449
—— —— retort house of, 2452
—— —— temperatures, 2460
—— —— terms, 2474
Gaseous fuels, 2168
—— —— for gas engines, 2171
—— —— —, heat value of, 2168, 2191
Gases and vapours, mass and bulk of, 1862
—— at high temperatures, specific heat [of, 107
—, composition of, 2558
—— exhaust, 2081
—— flue, 2081
—— mine, 2614
—— producer, heat values of, 2175
—— properties of, 109, 112
—— specific gravities of, 123, 133
—— specific heat of, 107
Gaskets, metallic, for steam-pipe joints, 1689
Gates, dock, 1465
—, head, 1297
—— sluice, 1343
Gathering-grounds for water, 1416
Gauge glasses, 1664
—— —— Tomey's, 1665
Gauges, limit, 774
—— boiler draught, 1545
—— pressure and vacuum, 1666
—— —— —, Brotherhood's tester, 1666
—— railway, 552
—— rain, 1406
—— water (boiler), 1662
—— —— —— defects in, 1664
—— widening on railway curves, 552
—— wire, 11
—— Woodite packing for, 1663
Gear, automatic control, electrical, 2299
—— calculations, spur, 911

GEC

Gear, continuously-variable chain driven change-speed, 951
—— cutters, involute, 910
—— H.-R. balance drive, 947
—— —— reduction, 946
—— Keenok pinion and, 951
—— P.I.V., 951
—— —— chain, 951
—— Robey drop valve, 1726
—— single reduction turbine, 933
—— slide valve, 1758
—— teeth, forms of, 908
—— —— grinding of, 922
—— —— lapping, 922
—— to determine the pitch line velocity of a, [911
—— variable speed, 948
Geared turbines, 930, 1818, 2381
Gearing, 907
—— calculations for worms, 940
—— chain, 899, 1818
—— critical speeds of, 922
—— design of worm, 940
—— helical, 927, 1816
—— Maag, 925
—— mill, 910, 914
—— noises, 922
—— Parsons, 2381, 2387
—— rawhide, 919
—— Seymour's, resilient, 921
—, spur, 911, 952
—— tooth, 908
—— turbine (steam), Parsons, 2381, 2387
—— wheel, helical, 899, 1818
—— worm, 937
Gears, bevel, 924
—— cycloidal teeth for, 910
—— double helical, 929, 1818
—— 'Fabroil,' 919
—— for machine tools, heat treatment of,
—— hardening of, 923 [923
—— heat-treated, 923
—— herring bone, 929
—— high carbon, 923
—— non-resonant, 919
—— reduction, 930, 946, 1818, 2381
—— —— resilient steel, Seymour's, 921
—— rolling mill, 910
—— strength of bevel, 924
—— to complete the circular pitch of a wheel, 911
—— triple helical, 930
—— turbine, 930, 1818, 2381
G. E. C. Angold magazine lamp, 2326
—— dry battery cells, 2255
—— Fraser & Chalmers coal dust conveying system, 2129
—— —— —— —— —— pulverised coal system, 2137
—— —— —— —— —— turbo-alternator, set, 2265
—— intercommunication telephones, 2341
—— rotary converter starting and controlling, 2273
—— 'Osglim' discharge lamps, 2325
—— 'Osram' lamps, 2324
—— Rotary converters, 2270
—— 'Studio' arc lamps, 2326
—— synchronous converters, 2270
—— tramway controller, 691
—— —— —— motors, 2283

GEC

G. E. C. turbo-alternator, 2263
——— 'Witton' direct-current motors, 2283
Gelatine blasting powder, 2797
——— dynamite, 2797
Gelignite explosive, 2797
Generators, alternating current, 2263
——— Mather & Platt, 2263
——— turbo, 2265
——— direct current, 2261
——— ———, care of, 2262 [2261
——— ———, Mather and Platt,
Genet aero engines, 2058
Gent & Co.'s electric clocks, 2327
Geodetic formulæ, 84
Geographical measures, 1
Geophysics for the location of minerals, 2564
German belting practice, 857
——— canals, 1513, 1515, 1516, 1518, 1526
——— electric railways, 672
——— waterways, 1515
Giant cranes, 813
——— elevators, mining, 2645
——— turbine (water), little, 1320
Gilkes' Pelton wheel regulator, 1312
——— turbine (water) governor, 1335
——— turbines (water), 1323, 1324, 1337
Gimbal dial, cranked, Stanley's, 66
Girder bridge specification, 262
Girders, Baltic yellow fir, 330
——— bending moments of, 212
——— boiler, 1569
——— circular, 299
——— compound steel, 236
——— continuous, 219
——— economical proportions of, 247
——— reinforced concrete, 453
——— riveted, 226
——— rolled steel, 238
——— safe loads on bearing surface for ends
——— steel, 238 [of, 496
——— steel, deflection of, 212
——— stresses on, 207, 211
——— weights of, 261
——— wooden, for floors, 329
Girder-work, riveted joints in, 226
Gladstone docks, Liverpool, 1462
Glands, water, steam turbine, 1794
Glass and glazing, 414
——— bottles, to divide, 2813
———, coloured, absorptive powers of, 1500
——— cutting, 2813
——— drilling, 2813
———, steam gauge, 1664, 1665
——— tubing, to cut, 2813
Glazed bricks, 426
——— tiles, 413
Glazier, constants of labour for, 415
Glazier's putty, paint, and solder, 416
Glazing and glass, 414
——— of lighthouses, 1500
Gleniffer Engines Ltd., cold starting oil engines, 1931
——— oil engine, 1931
Glue for fastening cloth to wool, 400
——— leather, 399
——— parchment, 399
———, holding power of, 176
Glues, 398, 399
Glutinous solutions, evaporators for, 796

GRI

Glycerine jelly process of copying, 92
Gold and silver tables, 2642
——— dredging, 2647
——— milling, 2638
——— mills, cost of, 2639
———, wire gauge for, 11
Goliath cranes, 804, 807
Gooch's link motion valve gear, 1763
Gordon's Mono Recorder, 2084
Governor Copes pump, 1680
——— of steam engine, 32, 1748
Governors, Pelton wheel, 1312
——— relay, 1748
——— retort house, 2464
——— turbine (water), 1335
Grab or grapple dredger, 1482
Grade and duty of sluices (mining), 2643
Gradients, railway, resistance due to, 585
Gradiometer, 67
Grain elevators, 1472
——— weighers, 1448
Graphite, for boilers, 'Foliac,' 1627
——— lubrication, 986
——— paint, 2806
Grapple or grab dredger, 1482
Grate area, boiler, 599
——— gasworks, 2460
——— in locomotives, 599
Graving docks, 1473
——— cost of, 1474
Gravity bucket conveyors, 847
———, centre of, 23
——— coefficient of, 30
——— drilling machine vice, 1046
———, specific, 125
——— of elastic fluids, 133
——— woods, 128, 129, 130
Greases, 2809
Greasing leather belts, 862
Great Britain and long distance railway electrification, 672
——— gas made in, 2451
——— generation of electricity in, 2226
——— Ordnance maps, 56
——— Western Railway carriages, 627, 628
——— locomotives, 571, 574, 576
Green's air heater, 1650
——— economiser, 1644, 1645, 1646
Gresham's hot water combination injector, 1610
Grey process broad flange beams, 237
Grinder, portable precision, 1115
——— Van Dorn tool post, 1114
Grinding, 1103
——— allowances, 1120
———, centreless, 1117
——— cylindrical, 1116
——— dry, 1107
——— examples of, 1120
——— external, 1116
——— internal, 1118
——— lubrication, 1107
———, machines, 1113
——— power required for, 1058
——— procedure, 1116
———, surface, 1119
——— wet, 1107
——— wheels, 1105
———, classification of, 1106
———, composition of, 1105

Grinding wheels, lubrication of, 1107
Grindstone speed, 1104
Grindstones, 1103
———, to true, 1105
Grizzle bricks, 426
Gromets, asbestos tube, 1827
Grooves for rope-driving, 870, 871 [871, 879
——— ——— ——— ——— Hart's improved, 870,
Ground glass, to imitate, 2803
——— sills, 1510
———, sloping, chaining over, 55
Groynes, 1508
Guard wire for electric tramways, 697
——— for rope drives, 876
——— for abrasive wheels, 1112
Guest & Chrimes rotary water meters, 1436
Gun-cotton, 2797
'Gunite' cement, 394
Gun-metal, 2731
——— tensile strength of, 174
——— weight of castings in, 161
Gun tap, 1172
Gunpowder, 2797
Gunter's surveying chain, 1, 51
Gunther & Sons, on Pelton wheels, 1311
——— regulation of turbines, 1333
Guyot's steam atomiser, 2162
Gwilt's rule for building walls, 369
Gwynne's pumps, 1429
Gyration, radius of, 32

HACK saws, 1090
Haematites, 2561, 2681
Hair belting, 857
Hall, J. & E., refrigerating plants, 1224
'Hall-Mark' photo copyer, 96
Hammer, Boyer long-stroke, 1079
Hammers, caulking, 1081
——— driving, by compressed air, 1129
——— electro-mechanical ('Kango'), 1086
——— Massey's, steam, 1125
——— pneumatic power, 1129
———, piling, 502
——— pneumatic power, 1129
——— Rigby's patent, 1135
Hamworthy oil engines, 1923
Hand cranes, 801
——— wheels, machine, proportions of, 1163
Handles, cock, proportions of, 1162
——— machine, proportions of, 1163
Harbours, 1455
———, cost of, 2869
Hard-drawn copper wire, strength of, 177
——— plaster, 406
——— rubber, 2814
——— solder, 2749
——— steel, to drill, 2815
——— woods, 348
Hardening and tempering steel, 1062, 1064, 1067
——— high-speed tools, 1062, 1067
'Hardiax' coal cutter, 2584
Hardness numbers, Brinell, 1099
——— of materials, 1098
——— water, 1421
——— tester, Herbert, 1101
Hardy air compressor, 2569
——— Patent Pick Co.'s coal cutter, 2568
——— Simplex air-hammer rock drill, 2568
Harling pantagraph, 90
Harrison's formula for strength of boiler flues,
Hart electrical accumulators, 2256 [1562
Hart's improved rope driving groove, 870, 871, 879, 880
Haslam refrigerating plants, 1226, 1230
Haswell's table of transverse strength of mate-
Haulage, 2593 [rials, 183
——— canal, 1525
——— cost of, 756, 762c
———, horse, cost of, 756
———, horse-power required for, 2601
——— mining, 2593
——— pulleys, 2605
——— road materials, 757
——— tail, 2599
——— underground, cost of, 2602
——— wire ropes for, 178, 885, 2606
Hauling and winding, ropes for, 2606
Head gates, 1297
——— races of turbines (water), design of, 1298
——— required to deliver water through pipes, 1426
Headgears, mining, 2605
Heading joints, 332
Heads for small screws, 1150
Heat, 99
———absorbed in the metamorphic conversions of combustion of carbon, 107
———and combustion, 109
———, change of density by, 116 [1778
———drop, adiabatic, in steam turbines, 1777,
———, expansion of air by, 117
——— or dilatation by, 116
——— generated by combustion of carbon, 107
——— indicating paint, 962, 2806
——— insulating materials, 435
——— value of roofing material, 322
———, latent, 107
——— loss of, from steam pipes, 1693
——— losses in heating buildings, 2492
——— of combustion, table of, 110
———superheated steam, 1535
——— resisting materials, 341, 430
———, specific, 106
——— of steam, 1538
——— transference, 1216
——— transmission in boilers, 1545
——— boiler tubes, 1546
——— surface condensers, 1828
——— through roofing materials, 322
——— walls, concretes, and plasters, 344
——— treatment of metals, 2697, 2758
——— springs, 1015
———unit table, 108
———, units of, 108, 109, 1542, 2449, 2476, 2494
——— value of gaseous fuels, 2168, 2169, 2191
——— producer gas, 2175
——— waste, recovery of, 2198
Heater, Drayton feed-water, 1642
——— Green air, 1650
——— Ljungström air, 1648
——— Hudson feed-water, 1641
——— Row (Royles) feed-water, 1641
——— tubular, 1549
——— Weir feed-water, 605, 2406
Heating, 2491
——— and drying, 2511 [2508
——— apparatus, automatic regulation of,

HEA

Heating, bearings, 962
——— buildings, 2491
——— ——— boilers for, 2502
——— ——— cost of, 2492
——— ——— fuel for, 2503
——— ——— plenum system, 2499
——— ——— radiators for, 2503
——— by compressed gas, 1068
——— ——— exhaust steam, 2499
——— ——— hot water, 2492
——— by open fire grates, 2491
——— steam, 2494
———, cost of, 2492
——— electric, 2330, 2336
——— feed water, 1640
——— fuel required for, 2503
——— horticultural, 2336, 2505
——— mains, size of, 2493
——— pipes, 2493, 2497
——— British standard, low pressure, 2497
——— railway carriages by steam, 633
——— surface of boilers, 600, 1552, 2395
——— locomotives, 600
——— superheaters, 1613
——— systems, 2492
——— water, 2509
——— by electricity, 2511
Hedley's mining surveying instruments, 66
Heel-straps of wooden roofs, 320
Heenan air cooler, 2267
——— filter, 2268
——— oil coolers for heat treatment of steel, 2704
——— refuse destructors, 2148
——— water cooler, 1232, 1886
——— & Froude's dynamometer, 38
Height of an object, to measure, 55
Heights, table of dip and distance of horizon,
Hele-Shaw clutches, 890, 895 [57
Helical gearing, 927, 1818
——— or spiral springs, 614, 1009
——— springs for locomotives, 614
——— tooth-gearing, 927
Helium, 120
Hematites, 2561, 2681
Hemp ropes, strength of, 199
Henderson & Glass' angle and tee iron, 141
——— steel, 142, 143
Hendry's hydraulic mining table, 2644
Henley wiring system, 2247
Hennebique reinforced concrete system, 451
Herbert, Alfred, Ltd., milling machines, 1057
——, E. G., Ltd., hardness tester, 1101
'Hercules' gauge glass, 1665
Hexagon nuts and bolts, American, 1153
——— British standard, 1154
——— Whitworth, 1152, 1153
——— rod metal, weight of, 135
Hick-Breguet Ejectair, 1845
Hick, Hargreaves & Co.'s condensing plants,
High-pressure compressors, 1193 [1845
——— gas in works, 773
——— steam, 1535
——— working, steam, economy of,
——-speed drills, 1035 [1710
——— engines, Belliss & Morcom, 1731
——— Robey, 1728
——— steels, 1063
——— steel scrap, to identify, 1063

HOR

High-speed tool steel, 1062, 2665
——— furnaces, 2665
Highway authorities, 711
——— bridges, loads on, 255, 256, 257
——— engineering, 711
——— statistics, 716
Hind's Rowan's pistons, 1756
——— slow action packing, 1756
Hirn's rope brake, 34
Hodgson fan dynamometer, 41
——— spring-controlled flexible diaphragm instruments, 1448
——— steam flow meter, 1447
Hoffmann ball bearings, 965, 967, 970
——— roller bearings, 972
Hoists, regulations for Danish, 1354
Holden's steam atomiser, 2162
Holding power of glue, 176
Holland B. & Co.'s, air compressor, 1198
——— vacuum pump, 1201
———, railway electrification in, 673
Hollow building-blocks, 429
——— cylinders, allowable pressure in, 192
——— ram hydraulic bolt forcer, 1354
——— shafts, 1002
——— walls, 368
Holm, Dr., on flash temperatures, 117
Holman air compressors, 1196, 1197
——— drill steel sharpening machines, 2566
——— light drift mining driller, 2570
——— mining drill grinder, 2566
——— sharpener, 2566
——— rotary air compressor, 1197
Holyhead breakwater, 1455
Hooks, crane, 816
——— foundry ladle, 816
Hoop-iron, dimensions and weight of, 160
Hopkinson-Ferranti stop valves, 1668
——— parallel slide valves, 1668
Hopkinson's torsion meter, 42
Hopper service dredging, 1484
——— weighers, 1440
Hopwood-Spencer boiler, 1600
Horizon, dip of, 57
Horizontal force of earth's magnetism, 74
——— sun-dial, 85
Hornsby-Ruston gas engines, 1871
———Stockport suction gas plant, 2187
Horse haulage, cost of, 756
Horse-power, automobile, 1960
——— brake, 1707
——— cotton machinery, 789
——— foreign, 1708
——— for haulage, 2601
——— formula for worm gears, 945
——— in compressed air, 1188
———, indicated, 1707, 1719
——— link belt, 869
——— of aerofoils, 2006, 2018
——— locomotives, 587, 589
——— shafting, 1001
——— steam engines, 1707
——— water, 1284, 1285
——— windmills, 1185
——— required for cupola fans, 2767, 2768
——— wood-working machinery, 1058
——— to move plane surface against the air, 1183

HOR

Horse-power required to pump water, 1240, 1241, 1242
——— raise water, table of, 1240
——— work pneumatic tubes, 1203
———, skin effective, of ships, 2363, 2364
——— standard for different nations, 1708
———, transmitted by belts, 855, 856
———, wave, effective, of ships, 2364
—— traction on canals, 1525
Horses, power of, 44
Horticultural heating, 2336, 2505
Hose clips, 'Jubilee' worm drive, 1085
—— couplings, 1084
——, fire, to preserve, 2551
Hot-air engines, 1931
—— bearings, cooling compound for, 2809
—— forgeable brass, 2723
—— water fittings, British standard, 2497
———, heating by, 2492
——— supply, 2509, 2511
—— wire anemometer, 1445
Hour angle, to find, 84
House drainage, 1379
——— heating, 2491
——— tubes, pneumatic, 1206
Houses, struts between, 513
Howden air preheater, 1648, 2393
—— ——— boiler, 1599
—— ——— cellular air heater, 2395
——— Ljungström air preheater, 1648, 2393
—— ——— Wallsend oil fuel system, 2163
Howden's forced-draught system, 2124, 2392, 2393, 2394
Howes turbines (water), 1320, 1321, 1322
'H.—R.' balance drive gear, 947
——— reduction gear, 946
Hudson exhaust steam feed heater, 1641
Hulburd's cartridge sight-glasses, 1663
——— —— force feed lubricator, 990
———— locomotive steam regulator, 601
——— —— mechanical boiler cleaners, 1682
——— —— seamless soft copper jointings, 608
Hull dock wall, 1460
Humidity and evaporation, 1384
Humphrey magnetic chucks, 1049
Hungary, railway electrification in, 673
Hunt synchronous motor, 2293
Hydraulic bolt forcers, Young's, 1354
———— cement, 383
———— cement, paint for, 2803
———— cylinders, thickness of, 1244
———— dredgers, 1482
———— drive, Vulcan Diesel, 2389
———— elevators, mining, 2644
———— gas mains, 2458, 2463
———— leathers, 1265
———— lifting and slewing machinery for bridges, 298
———— lifts, 1353
———— limes, 384
———— mains, gas, 2458, 2463
———— ———, pressure in, 1242
———— ———, size of, 1244
———— ———, mean depth, 1392
——— —— mining, 2643
———— monitors (mining), 2643

IND

Hydraulic mortars, 383
———— packings, 1264
——— —— pipes, thickness of, 150, 1243
——— ——— and cylinders, thickness of, 1242
———— rams, 1350
———— riveters, Tweddell's, 1355
———— steam turbine transmitter, Fottinger's, 1821, 2389
———— variable-speed gear, 948
Hydraulics, 1239
Hydro-electric plants, 1339
Hydrogen, 2170
——— ——— welding, 2792
Hydrometer scales, 131
Hydrostatic press, 1349
———— ———, thickness of cylinders of, 1349
Hypsometric Aneroid, Steward's, 69

I BEAMS, dimensions and properties of, 235
——— safe loads on, 236
Ice evaporation, 1389
—— making, 1219
——, supporting power of, 2817
Ignition temperatures, 101
Illinois chain grate stokers, 2114, 2117
Illuminated buoys, Pintsch's, 1490
Illumination, 2525
——— ——— calculations, 2531
——— —— lighthouse, 1497
Illumination, reflectors for, 2535
——— ——— requirements for, 2529
——— ——— standards of, 2532
——— ——— terms of, 2525, 2546
Impact of bodies, 27
——— on railway bridges, 268
Impounding reservoirs, cost of, 2891
Impulse blades, turbine (steam), 1791
——— of wind, 1184
——— turbines, steam, 1783
Inaccessible objects, to ascertain distance of,
Incandescent burners, 2475 [53
——— ——— mantles, 2476
Inches reduced to decimals of a foot, 4
Inclined plane, 26, 28
——— ———, work on, 28
Inclines, mining, 2599
Inclining experiment on ships, 2357
Inclinometer, Lister's, 61
Income tax, mining, 2656
Incrustation of water pipes, 1419
Indented bar for reinforced concrete, 451
India, rainfall in, 1406
——— survey maps of, 57
Indiarubber & Gutta Percha Co.'s electricity supply cables, 2211, 2212
Indicated horse-power of steam engines, 1707,
Indicator (steam engine), 1715 [1716
———— card, 1716
———— CO_2, 2083, 2086, 2087
———— connecting pipes, 1717
————, Crosby, 1715
———— diagram, to interpret, 1717, 1720
———— diagrams of air compressors, 1216
———— ———, piston, 1717
———— revolution, Jacquet, 1450
———— stability and trim, of ships, Ralston's,
Induction motors, 2289, 2291 [2359

IND

Induction regulators, 2305
Industrial Stroboscope, 956
———— trucks, electric, 704, 760
Inertia, moments of, 33, 217, 315
Ingot iron, 'Armco,' 2769
Injection cocks, condenser, 1850
Injector, Penberthy, 1655
Injectors, 1655
———— locomotive, 609
———— notes on, 1658
Inspection, boiler, 1654
Institution of Electrical Engineers regulations for electrical equipment, 2231, 2240
———— —— Mining and Metallurgy, standards, 2641
Instrument screw threads, 1144
Instruments, electric measuring, 2087, 2310
—— —— mining surveying, 66
—— —— surveying, 59
Insulated conductors, 2205, 2210, 2243
—— —— —— —, specification for, 2243
Insulating materials, electrical, 2346
———————————— Dexonite, 2346
———————————— Ioco Rubber Co., 2347
———————————— Maconite, 2254, 2346
———————————— Willesden paper, 2346
—— —— oils, 2347
Insulation, cold storage, 1220
—— —— —, cork, for cold storage, 1222
—— —— measuring instruments, 2319
—— —— of electrical conductors, 2204, 2205, 2209, 2243
—— —— —, paper, electrical, Willesden, 2346
—— —— —— for cold storage, 1222
—— —— ship refrigeration, 1223
—— —— silicate cotton, for cold storage, 1220
Insulators, porcelain, 2225
Intercommunicating telephones, 2341
Interlocking steel piles, 504
Internal combustion engines, cooling water for, 1895,
—— —— —— —— Lloyd's rules for
—— —— —— turbines, 1925 [1926
—— —— stresses, girders, 211
International candle, 2525
—— —— Combustion Ltd., 'Lopulco' pulverised coal system, 2125, 2137
—— —— Gas Detectors Ltd., CO_2 Indicators, 2086
—— —— system metrical screw threads,
Invar steel, 2703 [1146
Inventions, and engineers' rights, 2905
—— —— —, patents for, 2833
Inverted beams, 458
Involute gear teeth for spur gearing, proportions of, 909
—— —— teeth for gearing, proportions of, 909
—— —— —— gears, 908
—— —— —— —— setting out, 909
—— wheel teeth, 908
Inward-flow turbines (water), 1325
Ioco Rubber Co., electrical insulation, 2347
Iron, 2678
——, Admiralty tests for, 188, 189

IRR

Iron angle and tee, weight of, 141
——, 'Armco' ingot, 2679
—— balls, weight of, 133
——, Barff process for protection of, 2712
—— boiler tubes, weight of, 1571
—— boilers, 1541
——, calorising, 2713
—— cast, 2683
—— —— bar, ball, and sheet, weight of, 133
—— —— non-magnetic, 2348
—— —— pipe flanges, 1116, 1245 [2477
—— —— pipes, tables of, 145, 147, 150, 1430,
—— —— —— —, weight of, 145, 147, 150, 1430,
—— —— soldering, 2752 [2477
—— castings, 2698
—— —— locomotives, 619
—— chains, strength of, 193
—— chains, testing, 198
—— columns, cast, 226
——, corrosion of, 174, 2710
—— crankshafts, Bureau Veritas rules for,
—— ferrite, 2678 [1005
—— flat rolled, weight of, 135, 136
—— galvanised, corrugated, 231
—— —— painting, 232
—— galvanising, 2711
—— hoop, weight and dimensions of, 160
—— nails, weight of, 162
——, pig, 2680
—— pillars, cast, 375
—— pipes, cast, weight of, 145, 1430, 2477
—— —— soil, 1381
—— —— water, incrustation of, 1419
——, protection of, from rust, 2712, 2803
—— rings, strength of, 194
—— rivets, weight of, 163
—— rolled, flat, weight of, 135
—— rolled joists, 329
—— roofing, corrugated, 231, 234
—— ropes, corrosion of, 200
—— —, mining, 178
——, rustless, 2716
—— rust-proof coating for, 2710
—— shafting, 1001
——, sheet, weight of, 133
—— sheets, galvanised corrugated, 231, 234
——, sherardising, 2711
—— soil-pipes, 1381
——, specific gravity of, 126
—— stainless, 2716
—— straps for wood roofs, 320
——, strength of corrugated, 234
—— tee, weight of, 141
——, tensile strength of, at high and low temperatures, 171
—— tubes, boiler, 1550, 1571, 2397
—— tyres on roads, 726
——, weight of, castings, 126, 161
—— wire, annealed and bright, sizes, weights and breaking stresses of, 176
—— —— telegraph, breaking stresses of, 177
—— —— weight of, 126, 161
—— wrought, 134, 2680
—— —— —, corrosion of, 174, 2710
—— —— —— tubes, weight of, 148, 149
Ironclad Exide lead battery, 707
Ironstone, 2581
Ironwork, painting, 2802
—— —— preservative, 2803
Irrigation, 1383

Irrigation canals, critical velocity, 1390
——————— water losses from, 1389
—— —— costs, 1394
—— —— losses by degradation, 1396
—— —— recent literature, 1398A
—— —— works : projects in hand, 1398A
—— —— water, duty of, 1393
Isothermal expansion of gas, 1863
Italian railway electrification, 673
'Italit' roofing, 327
Ivory, to make permanently white, 2814

JACKETS, steam, 1711
Jaguar aero engines, 2052
Japan, weights and measures, 17
'Jaquet' revolution indicator, 1450
——— tachograph, 1451
Jet condensers, 1825
Jetties, 1508
Jig-back aerial ropeways, 843
Johnson & Phillips wiring system, 2251
Johnson friction clutch, 893
——— patent marine water-tube boiler, 2400, 2402
Jointing beams, 311
——— belts, 861
——— materials for pipes, weight of, 164
——— , 'Pilot,' Beldam, 1690
Jointings, copper, Hulburd's, 608
Joints, ball and socket, 607, 1084
——— belt, cement for, 862
——— boiler, strength of riveted, 1567
——— cement, for water mains, 1247
——— Dexine sheets for, 1690
——— edge, 332
——— heading, 332
——— house drainage, 1379
——— in boilers, riveted, 1567
——— leather belting, 861
——— , pipe, 1169, 1244
——— riveted, 226, 1567
——— secret, in flooring boards, 332
——— sewer pipe, 1381
——— soldered, strength of, 176
——— steam pipe, 1689
——— ——— metallic gaskets for, 1689
——— straining beam, 314
——— T, 312
——— tramway rail, 688
——— water pipe, 1244, 1429
——— wooden roof, 320
——— ——— , secret, in flooring boards, 332
Joists, bridging, 331
——— camber of, 329
——— ceiling, 331
——— floor, 328
——— floor, ventilation of, 333
——— rolled iron, 329
——— , wood, 331
J. & P. wiring system, 2251
'Jubilee' worm-drive hose clips, 1085

KAHN bar for reinforced concrete, 452
'Kalibond' wiring system, 2252
'Kango' electro-mechanical hammers, 1086
'Kapok,' 2486
Keenes' cement, 397, 406
'Keenok' pinion and gear, 951

Kempe straight line link motion, 955
Kennicott system of water softening, 1629
Kent-Hodgson meters, 1443, 1445, 1446, 1448
Kent's meters, 1435, 1436, 1437, 1438, 1447, 1448
Kenyon on coverings for steam pipes, 1696
——— table of cotton ropes for power driving, [873, 874
Kenyon's 'Kisol,' 1695
Kerosene, 2150
Kerpely gas producer, 2177
Kestner de-gassing plant, 1639
Kestner Evaporator Co.'s hydrometer scales, [131
Keys, keybars, keyways, 1162
——— box spanner, 1164
——— railway, 542
Keystones, 524
Kiel ship canal, 1526
Kilns, 2665
King George V Dock, 1459
King-post roofs, 318
Kirkaldy, strength of lead pipes, 192
Kirke gas-fired boiler, 2191
——— waste heat boilers, 2194
'Kisol,' 1695
Kjellberg system of electric welding, 2781
Klinger valve, 1659
——— water gauge, 1662
Knots, scaffolding, 512
Krupp ball mills, mining, 2641 [2432
Krupp-Diesel engines for submarines, 2430,
Kutter's formula, velocity of water in pipes and channels, 1299, 1419

LABOUR for glaziers, constants of, 415
——— produced by animal power, 44
——— upon embankments, 45
Labyrinth packings, turbine (steam), 1792
Laced butt-joint for driving belts, 861
Laces for belts, 861
La Cour motor converters, 2275
Lacquers and bronzes, 2807
———, cellulose, 2807
Ladder dredgers, 1480
Lagging, steam pipe and boiler, 1695
La Guayra breakwater, 1458
Lakes, temperature of, 106
Laminated springs for locomotives, 613
——— ——— motor cars, 1974
Lamps, electric, 2321, 2532
——— ——— Angold arc, 2326
——— ——— arc, 2326
——— ——— carbons for, 2326
——— ——— flexible cords for, 2207, 2325
——— ——— 'Mazda,' 2323
——— ——— metal filament, 2321
——— ——— 'Osglim,' discharge, 2325
——— ——— Osram, 2324
——— ——— Siemens, 2324
——— ——— standard tungsten, 2322
——— ——— 'Studio,' 2326
——— ——— traction, 2323
——— miner's safety, 2616
Lancashire boilers, 1573
——— cotton mills, machinery in, 787
Lanchester motor-car engine, 1937
Land required for railways, 552
——— treatment of sewage, 1372
——— type boiler, Babcock & Wilcox, 1596
Landing slips, cost of, 2870
Lang Lay wire ropes, 885

LAP

Lap joints for belting, 862
Lapwelded charcoal iron boiler tubes, 1571
'Lascar' piston rod packing, 1755
Lassen-Hjort water softening and purifying plant, 1633
Latent heat, 107
—— of fusion of metals, 2673
Lateral canals, 1513
—— cost of, 1513
—— strength of materials, 179
Lathe centres, 1020
—— tool holders, 1021
—— tools, cutting speeds of, 1073
Lathes, 1019
—— power required to drive, 1053, 1054
—— speed of, 1020
Lathing and plastering, 409
Lathings, expanded metal, 335
Latitude and longitude, length of a degree in, 75
——, to find, 78
Lattice girder bridges, 285
—— girders, 249
Laws relating to engineers, 2903
Lea automatic water recorder for steam engines, 1849
—— coal meters, 1449
—— cubi-meter, 1450
—— weir recorders, 1441
Lead alloys, 2731
—— balls, weight of, 135
—— bars, weight of, 137
—— covered cables, 2210
—— ore, 2562
—— pipes, formulæ for weight of, 146
—— soil, 1381
—— strength of, 192
—— weight of, 146, 1431
—— plumbing, 419
——, rolled, weight of, 137, 138
—— sheet, weight of, 138
—— reinforced, 174
—— sheets and bars, weight of, 137, 138
—— tensile strength of, 170
—— water pipes, corrosion of, 420, 1419
——, welding, 2779
Leading material, 492
Leads and angles of railway crossings, 557
Leakage, electric rail conductor, 700
—— through concrete, 394
——, water, tracing, 1420
Leaks in steam pipes, 1690
Leaky boiler settings, 1695
Leather belting, 855
—— friction, coefficients of, 869
Leather belting, joints in, 861
—— Reddaway's, 857
—— belts, oiling and greasing, 862
—— length of, 860, 862
—— hose, preservation of, 2551
Leathers, hydraulic, 1265
—— pump, 1265
—— ram or U, 1264
Leblanc air-pumps, 1844
Legal notes for engineers, 714, 2903
—— points for road engineers, 714
—— status of engineers, 2903
Legislation, boiler, 1655
Length measures, 1
—— of a degree in latitude and longitude, 75
—— driving belts, 860, 862

LIG

Lengths and angles of slopes, 493
—— ropes and cables, 1
Lens, apertures of, photographic, 119
—— concave, 119
Lentz valve gear, 595
Levees, 1509
Level, Abney reflecting, 68
——, dumpy, 66
——, reversing, Zeiss, 70
Levelling, 68
Levers, 25
Lewis, 377
—— formula, 915
—— modifications of, 916
—— testing machine for spur gearing, 952
—— working load on machine-cut spur wheels, 915
Liability for engineer's negligence, 2906
Life of boilers, 1653, 2400
—— boiler tubes, locomotive, 600
—— fire boxes, locomotive, 601
—— marine boilers, 2400
Lift-out crucible, or pit type, furnaces, 2666
Lifting appliances, arrangements in, 825
—— electro-magnets, 822
—— pumps, 1249
—— stones, 376
—— tackle, design of, 817
——, vacuum cup, 824
Lifts, 829
—— canal, 1519
—— Danish regulations for working, 1354
—— electric, 831
—— specification for, 831
—— hydraulic, 1353
—— steel-wire ropes for, 828
Light, 119
—— curves (polar), 2528
——, decomposition of, in the spectrum, 119
—— electric, reflectors, 2535
—— from gas-burners, 2475
—— railways, 552
—— cost of, 2875
—— electric locomotives for, 553
——, standards of, 2525
—— velocity of, 119
Lighthouse, Eddystone, 1493
—— illumination, sources of, 1497
—— lights, 1495
—— optical apparatus, 1495
Lighthouses, 1493
—— cost of, 1495, 2870
—— distance visible from, 1495
—— glazing of, 1500
—— lightning conductors for, 1500
Lighting, 2525
—— acetylene, 2759
—— coefficients, 2541
—— cost of street, 2871
—— distribution curves, 2528
—— electric, railway carriages, 633
—— factories and workshops, 2530
——, gas, in factories, 2544, 2545
—— incandescent gas, 2476
—— petrol air gas, 2487
—— street, 2546
—— systems, design of, 2531
—— terms, 2525, 2546
—— train, 633
Lightning conductors for lighthouses, 1500

Lightning conductors, Phœnix rules, 2330
——————— rules for, 2328
——— Rod Conference, 2328
Lights for lighthouses, 1493, 1495
Lightships. 1493
Lignite, 2105
——— tars, 2158
Lilleshall gas engines, 1878
Lime, 384
Lime-burning, 384
—— concrete, 388
—— Lias, 384
—— mortar, 383
—— purification of gas, 2467
—— wash, 419
Limes, hydraulic, 384
—— tentative specification for, 385
Limit arms for micrometers, 778
—— gauges, 774
——— British standard, 774
—— of safe loading of beams, 236
Lincolnshire iron, 2682
'Lincona' balata belting, 881
——— endless driving bands, 868
Line, wire (iron telegraphic), strength of, 177
Liners, Atlantic, 2371
Lining, brake, 819, 850, 897
——, felt, for slates, 413
—— for furnaces, 433, 1608
—— reservoirs, 1412
——, Willesden paper, 413
Link belts, 869
—— motion, 594, 953, 1758
——————— Allen's valve gear, 1764
——————— Gooch's valve gear, 1763
——————— locomotive valve gear, 1762
——————— Stephenson valve gear, 1762
—— motion straight line, 953
——— ———, Walschaert valve gear, 594
—— or chain belts, 869
Lintels, 332
Lion aero engines, 2048
'Lion' metallic packing, J. Walker & Co.'s, 1752, 1753
'Liptak' flat furnace arches, 1608
Liquid air, 1182
—— fuel oil-gas burners, 1068
—— fuels, 620, 2149
——, ——, furnace firing with, 2161
—— glues, 399
—— measures, 7
—— weighers, 1428
Liquids, reduction of volume of, under pressure,
——— specific gravities of, 126, 132 [117
Liquor, gas, 2463, 2465, 2474
Lister's inclinometer, 61
Little Giant air drill, 1044
——— —— electric drill, 1043
—— ——— turbine (water), 1320, 1321, 1322
Liverpool docks, 1462
——— —— landing stage, 1502
—— ——— Refrigeration Co.'s plants, 1226
Ljungström air preheater, 1648, 2393
——— turbine, 2385
Lloyd's rules for boiler safety valves, 1660
—— ——— ferro-concrete vessels, 2445
——————— marine internal combustion engines, 1926
——————— refrigerating machinery, 1234
——————— steam pipes, 1687

Lloyd's rules for (tentative) application of electric arc welding to ship construction, 2445
Load of different materials, equivalents of, 494
—— on foundations, 496
——— rails, breaking, 536
——— roofs, 365
——, safe, on chimney foundations, 497
——————— columns, 497
——————— floors, 497
——————— girders, 496
——————— materials, 496
——————— ordinary foundations, 496
——————— piles, 497, 502
———————, formulæ for, 502
——— ——————— reinforced concrete, 501
——————— walls, 497
—— which foundations will bear, 496
Loading beams, safe limit of, 182
——— coils for telephone circuits, 2338
Loads carried by piles, 500
——, eccentric, 223
——— of carts and waggons, 494
——— on highway bridges, 255
——— railway bridges, 249
——— reinforced concrete piles, 501
———, road regulations for, 722
——— safe, on struts, 243, 244
Lock nuts, 1159
Locking nuts, 1159 [453
Lock-woven mesh for reinforced concrete work,
Locked-coil ropes, 836, 2608
——————— Wright's, 2608
Locks, canal, cost of, 2872
——— entrance, 1465
——— on canals and rivers, 1519, 1523
Locomotive axles, 596, 611
——————— bearing surfaces, 596
——————— boiler tubes, 600, 616
——————— boilers, 599
——————— brass tubes for, 616
——————— copper plates for, 615
——————— ——————— tubes for, 600
——————— feeding, 605
——————— fire boxes of, 600
——————— grate areas and heating surfaces of, 600
——————— ——— iron for, 619
——————— rivets for, 615
——————— rods for, 615
——————— specification of material for, 611
——————— steel castings for, 615
——————— steel tubes for, 616
——— ——— brakes, 626
——— ——— coal consumption, 624
——————— construction, contract specification for, 620
——————— contract specification, 620
——————— coupling and connecting rods, 593
——————— cranes, 808
——————— crank axles, 596, 611
——————— cylinders, 592
———————driving-wheels, counterbalancing, 599
——————— feed pipes, 607, 608
——————— feed-water heater, Weir, 605
——————— fireboxes, 600
——————— ——————— copper plates for, 615

LOC

Locomotive fireboxes, life of, 601
—— —— flexible wheel base, 599
—— —— frames, steel for, 618
—— —— fuel consumption, 624
—— —— injectors, 609
—— —— jointings, copper, Hulburd's, 608
—— —— link motion, 594, 1758
—— —— motion work, 593
—— —— oil, lubrication, consumption, 624
—— —— smoke boxes, 601
—— —— straight axles, 611
—— —— superheaters, 600, 601
—— —— tenders, 625
—— —— tubes, 600, 616
—— —— types, 572, 578
—— —— tyres, 597, 612
—— —— water consumption, 624
—— —— —— troughs, 625 [249, 252
—— —— weight distribution on bridges,
—— —— wheel base, flexible, 599
—— —— wheels, 597
—— —— —— counterbalancing, 599
Locomotives, 571
—— —— adhesion of, 587
—— —— alloys for parts of, 620
—— —— , American, 576
—— —— brass tubes for, 616
—— —— British, recent additions to, 571
—— —— general types, 572, 578
—— —— tank, 574, 578
—— —— tender, 572
—— —— bearing surfaces of, 596
—— —— castings for, 615
—— —— compound cylinders of, 592
—— —— , 592
—— —— cylinders of, 592
—— —— consumption of fuel in, 624
—— —— water in, 624
—— —— copper plates for, 615
—— —— rods for, 615
—— —— tubes for, 600, 616, 617
—— —— coupling rods of, 593
—— —— crank axles of, 596, 611
—— —— cylinders of, 592
—— —— simple and compound of equivalent power, 592
—— —— design of, 579
—— —— electric, 676, 2602
—— —— for light railways, 553
—— —— examination of, in running
—— —— forgings for, 615 [sheds, 625
—— —— Great Western Railway, 574, 576
—— —— horse-power of, 587, 589
—— —— iron castings for, 619
—— —— liquid fuel for, 620
—— —— London Midland and Scottish Railway, 572, 574
—— —— London & North Eastern Railway, 572, 574
—— —— lubricating oil consumption in,
—— —— 'Mallet,' 581 [624
—— —— mechanical stokers, 622
—— —— mine, 2602
—— —— motion work of, 593
—— —— parts of, 592
—— —— power of, 587
—— —— pulverised coal for, 622, 2141
—— —— —— peat for, 2142

LOW

Locomotives, regulators, Hulburd's, 601
—— —— relation of power to weight, 589
—— —— resistance of, 586
—— —— Southern Railway, 574, 576
—— —— speed of, 586
—— —— springs of, 613
—— —— steel castings for, 615
—— —— forgings for, 615
—— —— plates for, 617
—— —— tubes for, 616
—— —— storage battery, 707
—— —— straight, axles of, 611
—— —— superheaters, 600, 601
—— —— tank, 574
—— —— tender tools for, 625
—— —— tractive force of, 587, 592
—— —— tyres for wheels of, 597, 612
—— —— valve gear, 594
—— —— valves of, 593
—— —— weight of, 589
—— —— —— restriction in, 589
—— —— wheel base of, 599
Log frame-saw roller bearings, 973
Log-line, 1
Logs, storing, 360
—— to cut stoutest beam from, 357
—— —— stiffest beam from, 357
London Midland & Scottish Railway carriages, 628
—— —— —— locomotives, 572, 574
—— —— —— wagons, 630
London & North Eastern Railway carriages, 628
—— —— —— locomotives, 572, 574
—— —— —— wagons, [630
—— Building Acts, 370
—— County Council regulations with reference to reinforced concrete, 443
—— —— regulations with reference to structural steelwork, 373
—— —— regulations for water
—— pavings, 743 [closets, 1380
—— —— cost of, 745
—— refuse, calorific value of, 2148
—— street paving, 743, 762c
—— tube railways, 679
Longitude and latitude, length of a degree in,
—— —— , to find, 81 [75
Longitudinal metacentre of ships, 2358
Longridge's formula for strength of boiler-flues,
Long stroke hammer, Boyer new, 1079 [1562
Lopulco pulverised fuel system, 2125, 2137
Lorries, electric, 705
—— petrol, 760
—— petrol-electric, 707
—— steam, 759
Loss of heat from steam pipes, 1693
—— power in machine shops, 1061
—— strength in steel rails, 535
—— water on canals, 1389, 1524
Losses, turbine (steam), 1792
—— steam heating, 2492
Low pressure steam heating, 2494
—— temperature distillation of coal, 2100
—— —— —— by-product recovery gas producers, 2185

LOW

Low water, to reduce a sounding to, 59
Lubricants, 2809
——— coefficients of friction of, 962
——— cutting, 1078
——— filtration of, 982
——— graphite, 987
——— safe pressures for, 962
Lubricating oil, consumption in locomotives,
——— filtration of, 982 [624
——— recovery of, 982
——— tests for, 982
——— oils, 977
——— properties of, 980
——— viscosity of, 978
——— specification of, 981
Lubrication, 984
——— air compressor, 985, 1199
——— automatic tool, 985
——— ball bearing, 975
——— broaches, 1033
——— chain gearing, 901
——— crank chamber, 984
——— crankshaft bearing, 984
——— drills, 1039, 1041
——— dynamo, 985
——— forced, 989
———, gas engine, 988
——— graphite, 987
——— grinding, 1107
——— high pressure, 986
——— locomotives, 624
——— oils, standard, physical properties of, 2809
——— ring, 984
——— roller bearing, 975
——— shafting, 985
——— steam turbines, 985
——— 'Tecalemit' system, 991
——— V.S.M., system, 989
——— wire rope, 889
Lubricators, 989
Lumen value of gas-filled lamps, 2532
——— vacuum lamps, 2532
Luminous buoys, 1490
——— energy, 120
Lyddite, or picric acid explosive, 2798

MAAG tooth gearing, 925
Macadam, tar, 732 [499
Macarthur moulded-in-place concrete piles,
Machine-cutters, sharpening angles of, 1072
——— handles, proportions of, 1163
——— handwheels, proportions of, 1163
——— moulded teeth for gears, 910
——— shop power losses, 1061
——— tools, 1019
——— floor space for, 772
——— power required for, 1053
——— wood, speed of, 1060
——— work tables of, 1172
——— vice, gravity drilling, 1047
Machinery, cane sugar, 794
——— cotton, power for, 789
——— depreciation of, 2896
——— electrical, British standards, 2260
——— in Lancashire cotton mills, 788
——— purchase of, 2912
——— refrigerating, 1211, 1217, 1223
——— Lloyd's rules, 1234

MAR

Machinery, sale of, 2912
——— sugar cane, 794
——— weight of, in ships, 2361
——— wood-working, 1058
——— worn parts, saving of, 1095
Machines, bolt screwing and nut tapping,
——— boring, 1054 [1054
——— cement testing, 402
——— excavator, 483, 2861
——— for cotton mills, sizes of, 792
——— grinding, 1113
——— metal cutting, 1019
——— milling, 1024
——— mortising, 1060
——— planing, 1021
——— tests, 1023
——— welding, 2776
Machining aluminium, 1094
——— ebonite and fibre, 1095
Macintosh, C., & Co., wiring system, 2254
——— tyres, 1977
'Maconite' insulation, 2254
Magnalium alloy, 2735
Magnesia or magnesite bricks, 430
Magnesium, 2720
Magnetic chucks, 1048
——— clutches, 896
Magnetic counter, Veeder, 1452
——— dip, 74
——— force, horizontal, 74
——— separation, mining, 2626
——— separators, 850, 2626
Magnetos, B.T.H., 1958
Magnets, electro lifting, 822
Main driving ropes, 788, 808, 810
——— line electric traction, 669
——— roads, 716
——— steam pipes, copper, 1687
——— tail rope haulage, 2599
Mains, electric-supply, testing, 2244
——— gas, 2458, 2463
——— hydraulic, pressure in, 1242
——— size of, 1244
Maintenance and renewal of waterworks, 1418
——— cost of roads, 717, 2875
Mallet locomotives, 581
Malm bricks, 426
M.A.N. oil engine, Diesel, 2427, 2432
Man, power of, 44, 46
Manchester ship canal, 1526
Manganese bronze, 2733
——— Bronze and Brass Co.'s alloys, 2733
———, influence of, on cast iron, 2685
——— steel, 2696
Manganin, 2733
Manila ropes, 199
——— testing and preservation of, 200
Mantles, gas, 2476
Maps of India Survey Dept., 57
———, Ordnance, scales of, 56
Marble, to clean, 2811
Marine audible signals, 1500
——— boilers, Babcock & Wilcox, 1598, 2397
——— Clarke, Chapman, 2400
——— combustion of coal in, 2391
——— cylindrical, 2395
——— draught of, 2392
——— furnaces and firebars, 2395

MAR

Marine boilers, life of, 1653, 2400
——————— Niclausse shells, 2397
——————— standard conditions for design and construction of, 1563
——————— surfaces of, 2395
——————— Thornycroft, 2397
——————— tubes, 2395
——————— Woodeson, 2400
——————— Yarrow, 2399
——— Diesel engines, 2411
——— engineering, 2353
——— engines, 1926, 2372, 2375, 2411
——————— cylinder diameters, 2374
——————— diagram factor, 2373
——————— metals for, 2443
——————— oil fuel for, 2404, 2438
——————— piston speed of, 2373
——————— recipro-turbine machinery, 2374
——————— standard rules for shafts of, 1005
——— gas and oil engines, Lloyd's rules, 1926
——— oil engines, 2411
——— piers, cost of, 2872
——— propellers, metals for, 2440
——— propulsion, 2441
——— reduction gears, 2381
——— steam turbines, 2375
——————— Brown-Curtis, 2381
——————— Parsons, 2378
——— superheaters, 2403
——— turbine gears, reduction, 2381
——— water-tube boilers, 2399
Marks, scribe, 357
Marsh gas, 2170
Marshall oil engines, 1911
Martin's cement, 406
Masonry, 376
——— ashlar, 880
——— block-in-course, 880
——— bonds, 881
——— dams, 1292, 1411
———, descriptions of, 377
——— dowelling, 381
——— measurement and valuation of, 381
——— proportions of stones and bond, 381
——— rubble, 379
——— strength of, 175
——— valuation of, 381
Mason's stair-treads, 343
Massey, B. & S., drop stamps, 1132
——————— pneumatic hammers, 1129
——————— power hammers, 1129
——————— patent compressed-air hammers, 1128
Mastic cements, 395 [
Material, leading, 492
Materials, abrasive, 1103 [182
——— absolute and ultimate strength of,
——— buildings, 347
——— drawing, illustrations of, 89
——— electric, 2346
——— equivalents of a load of, 496
——— fire-resisting, 341, 2547
——— hardness of, 1098
——— heat insulating, 435
——— insulating, electrical, 2346
———, for cold storage, 1219
——— lateral strength of, 179
——— locomotive construction, 620
——— refractory, 430, 435

MER

Materials reinforced concrete, 439
——— resistance of, to compression, 169
——— roofing, 322
——— safe load on, 496
——— shrinkage of, 483
——— strength of, 169, 182
——— tensile strength of, 170
——— transverse strength of, 183
——— turbine construction, 1802
Mather & Platt direct current generators, 2261
——————— geared alternators, 2263
'Mazda' drawn-wire electric lamps, 2323
McLaren-Benz oil engines, 1922
McPhail & Simpson superheater, 1613
Mean pressure of steam, 1706, 1718, 2372
——— refractions, table of, 83
Measure height of an object, to, 55
Measured base, to reduce to level of sea, 58
Measurement and valuation of masonry, 381
———, cotton yarn, 788
———, distance, 53
——————— power by brakes, 34
——————— water flowing in a stream, 1286, 1290
——————— velocity in a stream,
———, sluice, 1290 [1606
——————— subtense, by tacheometer, 65
———, weir, 1286
Measures, geographical and nautical, 1
——— and weights, 1
——————— foreign, 16
———, foreign and English, 11
Measuring instruments, electric, 2087, 2310
——————— Evershed, 2319, 2320
——— open streams, 1290
Mechanical boiler-cleaners, Hulburd's, 1682
——— draught, 2122, 2392
——————— Howden system, 2124, 2393, 2394
——————— Musgrave system, 2122
——————— Sturtevant system, 2123
——————— drilling, mining, 2564, 2593
——————— efficiency of steam engines, 1709,
——————— elevator, mining, 2645 [1710
——————— excavation, 423, 2861
——————— stokers, 622, 2112
———————, Babcock & Wilcox, 2112
———————, Bennis, 2113
———————, Erith Roe, 2117
———————, Illinois, 2114
———————, Underfeed, 2119
Melbourne and Sydney electrification systems, 681
Melting furnaces, crucible, 2666
——— metals, power required for, 2666,
Melting-point of alloys, 104
——————— brass, 2732
——————— bricks, 430
——————— bronze, 2733
——————— chemical elements, 103
——————— refractory materials, 104, 430,
Melting-points, 103 [435
Men, power of, 44
——— work done by, 46
Mensuration, 51
Mercedes automobile supercharger, 1945, 1946
Mercury, reduction of volume under pressure,
——— vapour rectifiers, 2277 [117
Meridians, convergence of, 85
Mersey river dredging, 1487

MES

Meshwork, metallic, 452
Metacentre, longitudinal, of ships, 2358
——— transverse, of ships, 2356
Metal, Babbitt, 2737
——— balls, weight of, 133
——, bell, 2731, 2744
——— Bull's, 2440, 2441, 2725, 2744
——— for propellers, 2440
——— cracks in, to stop, 2754, 2816
——— cutting machines, 1019
——, Delta, 170, 171, 174, 962, 2441, 2443, 2724, 2725, 2732, 2737, 2739, 2744
——— expanded, 334, 452
——— extruded, 171, 2725
——— filament electric lamps, 2321
——— for marine engines, 2443
——, gun, 2731
——— tensile strength of, 174
——— Monel, 2729
——— phosphor bronze, 2731
——— road, stones used for, 749
——— sawing, 1057
——— sheet, specification of, 138
——— sheets and bars, weight of, 135
——, Spence's, 2743
——— spraying, 1096
——— thickness of, for hydrostatic press cylinders, 1349
——— type, 2743
——— white, Admiralty, 2737
——— work, to clean, 2810
Metallic flexible tubing, 608, 1081, 2571, 2766
——— gaskets for steam pipes, 1689
——— mesh, lock knot, 452
——— meshwork, reinforced concrete, 452
——— piston-rod packing, 'Beldam,' 1753
——— 'Britimp,' 1199, 1751, 1793, 1888
——— 'Lascar,' 1903
——— 'Lion,' 1752, 1753
——— 'Reefer,' 1754
——— United Kingdom, 1750
——— United States, 1749
——— Walker, 1752, 1753
——— resistance materials, 2205
——— spraying, 1096
Metalliferous minerals, 2555
Metallurgy, 2659
——— Admiralty white metal, 2737
——— air velocities and volumes for cupolas and metals, 2678 [las, 2664
——— alloys, general, 2744
——— aluminium, 2719
——— alloys, 2719, 2730, 2732, 2734, 2735
——— Babbitt metal, 2737
——— bearing metals, 2736
——— bell metal, 2731, 2744
——— blast furnaces, 2660
——— brass, 2723
——— bronze, 2724, 2730
——— Bull's metals, 2449, 2725, 2744
——— case-hardening, 2705
——— casting and founding, 2752
——— cast iron, 2683
——— chromium, 2719
——— cobaltron steel alloy, 2704
——— constantan, 2733
——— copper, 2717

MET

Metallurgy, corrosion of iron, 174, 2710
——— crucible furnaces, 2666
——— cupolas, 2632
——— cupro-nickel, 2733
——— Delta metals, 170, 171, 174, 962, 2441, 2724, 2725, 2732, 2737, 2739, 2744
——— die-casting, 2756
——— Duralumin (Vickers), 2735
——— electric welding, 699, 2779
——— furnaces, 2667
——— extrusion process, 2725
——— ferrite, 2678
——— foundry practice, 2683
——— furnace construction, 2671
——— efficiency, 2672
——— furnaces, 2659
——— fusible metal, 2741
——— gun metal, 2731 [metals, 2673
——— heat energy required to melt
——— heat treatment of metals, 2697, 2758
——— hematites, 2681
——— Invar, 2703
——— iron (ferrite), 2678
——— latent heat of fusion of metals,
——— lead, 2731 [2673
——— manganese bronze, 2733
——— Magnalium, 2733
——— magnesium, 2720
——— Magnolia, 2733
——, manganin, 2733 [and alloys, 2757
——— mechanical treatment of metals
——— 'Melloid,' 2726
——— metals and alloys, 2678
——— Monel metal, 2729
——— muffle furnaces, 2665
——— nickel, 2718
——— alloys, 2726
——— chrome steel, 2695
——— chromium, 2727
——— silver, 2727
——— nickel steel, 2694
——— non-ferrous alloys, 2721
——— non-magnetic alloys, 2743
——— pattern composition for castings,
——— Permalloy, 2726 [2754
——— phosphor bronze, 2731
——— platinite, 2729
——— platnam, 2727
——— pyrometry, 2673
——— reverberatory furnaces, 2665
——— Séger cones, 2674
——— solders and soldering, 2747
——— Spence's metal, 2743
——— standards, 2641
——— steel, 2691
——— steel specifications, 2693
——— type metal, 2743
——— Vickers' Duralumin, 2735
——— welding, 2759
——— zinc, 2718
Metals, 2678
——— adulteration of, to discover, 126
——— bearing, 2443, 2736, 2740, 2744
——— corrosion of, 174, 2710
——— expansion of, 116
——— fatigue of, 200
——— fusible, 2741
——— heat energy required to melt, 2673
——— treatment of, 2697, 2758
——— latent heat of fusion of, 2673

MET

Metals, marine engine, 2443
——— propeller, 2440
——— resistivity of, 2205
——— sheet, specification of, 138
——— specific gravity of, 132
——— strength of, 170 [171, 2769
——— ——— ——— at different temperatures,
——— weight of square foot of, 132
——— ——— ——— cubic inch of, 132
Metcalfe's exhaust steam injector, 609
Metering water, air, gas, steam, and coal, 1435
Meters, coal, 1448
———, current, 1292, 1506
——— electricity, 2315
——— ——— specification for, 2315
——— for measuring water flowing in stream,
——— gas, 1443, 2479 [1292, 1506
——— steam, 1446
——— torsion, Hopkinson's, 42
——— water, 1437
Methane (marsh gas), 2170
'Metlex' wall plugs, 421
Metre gauge railways, 552
Metric carat, 7
——— pressures, 9
——— system, 2
Metrical screw threads, 1146, 1147
——— surfaces, 6
Metropolitan Buildings Act (*see* London Building Act)
——— ——— Referees' gas tests, 2481
——— ——— Vicker's tramway controllers, 690
——— ——— ——— transformers, 2303
——— Water Board standard sockets,
Michell bearings, 2409 [1429
Micrometers, 778
Micrometric theodolite, 63
Mild steel pillars, 375
——— steel, treatment of, 188
Mileage, road, in relation to population and
Mill gearing, 910, 914 [area, 716
Milling cutters, 1024, 1025
——— ——— for copper, 1031
———, flour, 783
———, gold, 2638
——— ———, cost of, 2639
——— machines, 1024
——— ——— ——— power for, 1057
——— speeds and feeds, 1026, 1056, 1057
Mills and millwork, 783
———, ball, gold mining, 2641
——— Chilian, mining, 2640
——— cotton, 784
——— ——— electrical driving, 790
——— crushing, 793, 2640
——— gold, 2640
——— ——— cost of, 2639
——— mule spinning, 787
——— oil, 792
——— paper, 792
——— ring spinning, 787
——— rolls, mining, 2640
——— sugar, 794
——— tube, mining, 2640
——— wind, 1185
Mine, legal definition of a, 2555
——— valuation, 2651
Mineral deposits, classification of, 2557
———, legal definition of, 2555
——— mixtures, calculation of, 2563

MIN

Minerals, metalliferous, 2560
——— ——— ownership of, 2555
Mines, Australian, 2594
——— coal dust in, 2617
——— deep shafts of, 2575
——— electric signalling in, 2619
——— fire extinction in, 2619
——— gases in, 2614
——— legislative enactments relating to, 2556
——— locomotives for, 2602
——— pumping in, 2612, 2645
——— rescue apparatus for, 2617
——— spontaneous combustion in, 2618
——— subsidences from, 2588
——— valuation of, 2651
——— ventilation of, 2614
Mining, 2555
——— air regulator, 2616
——— alluvial or placer, 2565, 2642
——— blasting, 2572
——— boring, 2563
——— branches of, 2558
——— cables, electrical, 2612
——— centrifugal pumps, 2613, 2645
———, coal, 2558, 2589
——— coal cutters, 2583
——— coffering, 2577
——— colliery, electric cables for, 2612
——— compressed air for, 2568, 2644
——— cost of, 2591, 2594
——— detonators, 2573
——— dredging, 2646
——— dressing of ores, 2619
——— drifts, 2573
——— drilling, 2564, 2593
——— electric cables for, 2612
——— elevator, 2645
——— elevators, 2644
——— explosives, 2572, 2797
——— ferro-concrete tubbing, 2579
——— François cementation process, 2581
——— flotation processes and machines, 2627
——— flow sheets, 2628
——— fuses, 2573, 2798
——— gold, 2638
——— haulage, 2593, 2599
——— headgears, 2605
——— hydraulic, 2643
——— inclines, 2599
——— legal, 2555
——— magnetic separation, 2626
——— metallurgy standards, 2641
——— minerals, classification of, 2557, 2559
——— ——— ——— ownership of, 2555
——— monitors or giants, 2643
——— operations, cost of, 2592
——— ore dressing and coal washing, 2619
——— pillars, 2585
——— placer or alluvial, 2565, 2642
——— pulleys, 2605
——— pumping, 2612
——— raising and winzing, 2579
——— rescue apparatus, 2617
——— rock drills for, 2565
——— ropes, 178, 2599, 2606
——— safety lamps, 2616
——— shaft pillars, 2585
——— ——— sinking, 2575
——— shafts, 2575
——— ——— cost of sinking, 2575, 2578

MIN

Mining shafts, deep, 2582
——— ——— lining for, 2576
——— shot holes, 2564
——— sizing, 2622
——— sluices, 2643
——— stamps, 2639
——— standards, 2641
——— subsidences, 2588
——— surveying instruments, 59
——— tubbing, 2577
——— tube mills, 2640
——— tubs, 2598
——— veins and masses, 2590
——— winding, 2608
——— plants, electric, 2604
——— winzing and raising, 2579
——— working costs of coal, 2589
Ministry of Health's requirements *re* sewage, purification plants, 1373
——— Transport, 714 [regulations, 543
——— permanent way
——— regulations, overhead electric lines, 2227
——— railways, 664
——— Roads Dept., 714
——— standard load for highway bridges, 255
Mirrlees' cyclic variation recorder, 1908
——— Diesel engines, 1896
——— fuel pump for tar oil, 2158
——— ejector air-pump, 1846
——— Leblanc air-pump, 1844
——— semi-Diesel engines, 1914
——— surface condenser, 1835
——— Watson Co., air-pumps, 1846
——— evaporators, 795, 796, 797
——— oil mills, 792
——— sugar-crushing plants,
——— water distillers, 797 [794
Mirrors, convex, 119
——— to silver, 2815
Mississippi river bank protection, 1508
———, dredging plant on, 1486, 1511
——— levee on, 1512
Mitis castings, 2680
Mitre wheels, 924
Mixtures, freezing, 105
——— non-freezing, 105
M.L.S. superheaters, 602, 1618
Modulus of elasticity, Young's, 184
Moisture in air, 1865
'Moler' (Fosalsil), 429, 435, 1222
Molesworth's rules for dams, 1411
Molybdenum steels, 2695
Moments, bending, 210, 212
Moments of inertia, 33, 217, 315
Momentum of trains, 268
Mond bituminous suction producers, 2185
——— gas producer plant, 2183, 2185
——— gas, 2183
Monel metal, 1806, 2729
Money, foreign, 18
Monitors or giants, mining, 2643
Mono Recorder, J. Gordon & Co., Ltd., 2084
Monsoon rainfalls, India, 1383
Morris motor car engine, 1936
Morrish's rule for centre of buoyancy of ships,
Morse chain drives, 905 [2355
——— taper measures for twist drills, 1058
Mortar, Portland cement, tinting, 387

NAP

Mortars, 383
——— adhesive strength of, 175 [for, 432
——— and cements, refractory, specification
——— cement, 383
———, choice of, 387
——— coloured, 387
——— lime, 383
——— mixing, 387
——— proportions for, 386
——— raw materials for, 384
——— refractory, 432
——— trass, 383
——— specifications of, 388
Mortise and tenon, 313
Mortising-machines, 1060
Motion, uniform, 27
Motor-cars (*see* 'Automobiles')
——— heavy, regulations on roads, 722
——— power brakes for, 1973
——— steam railway, 626
——— stopping distance of, 724
——— coach trains, British Thompson-Houston Co.'s equipment, 682
——— camshaft equipment
Motor control on tramways, 690 [for, 682
——— converters, A.C.E.C., 2275
——— fuels, 1951
——— spirit, composite, 2156
——— vehicles, legal speed limits, 724
——— new classification of, 762
——— regulations under the Act, 762
Motors, electric, alternating current, 2286
——— electric battery, 705
——— cascade (Hunt), 2293
——— commutator, 2294
——— controllers for, 2298
——— converters, 2274, 2275
——— direct current, 2278, 2282
——— speed regulation
——— detonation, 2153 [of, 2281
——— induction, 2289
——— polyphase, 2290, 2294
——— spirits for, 2152
——— starters for, 2298
——— synchronous, 2286
——— tramway, 690, 2283
Moulded dimensions of ships, 2353
Moulded-in-place concrete piles, 499
Moulding-sand, 2752
Mouldings, 410
Muff couplings, 1176
Muffle furnaces, 2665
Mule spinning mill, 787
Multiple effect evaporator, 795
Multitubular boilers, 1576
Multivane centrifugal fan, 2520
Mumford feedwater regulator, 1677
'Muraline,' water paint, Carson's, 2802
'Muraprime,' Carson's, 2801
Murray's tables of radii of curves, 564
Musgrave mechanical draught system, 2122
——— milling cutters, 1026
Mushet steel, 1063

NAILS, composition, weight of, 163
——— copper, weight of, 162
——— slating, 411
——— wrought-iron, weight of, 162
N.A.P. tyres, 1977

NAP

Naphthalene, 2469
Napier aero engine, 2048
Narrow-gauge, or light, railways, 552 [1871
National Gas Engine Co.'s gas engines,
——— Grid Transmission Lines, 2220
——— oil engines, 1919, 1923
Natural gas, 2171
Nautical measures, 1
Naval architecture, 2353
—— brass, 2724
Navigable waterways, 1505
Navigation of rivers, 1505
Navy, oil fuel for, 2166
'Neckar' de-aeration system, 1640
——— water softening system, 1635
Negligence, engineer's liability for, 2906
Negretti & Zambra's boiler distance thermometer, 1667
——————— low temperature electrical distance thermometer, 1232
Neptune electric tramway rail bond, 698
Neutral bricks, 431
New Boyer long-stroke hammer, 1079
—— Welland canal, 1528
—— York harbour dredging, 1487
—— electrification of railways, 681
—— Zealand type of mining dredger, 2648
Nickel, 2718, 2726
——— chrome steel, 2695
——— chromium alloys, 2727
———, impurities in, 2719
——— silver, 2727
——— steel, 174, 2694
——— steel bars, identification of, 167
———, welding, 2776
Niclausse boiler, 1584
Nimbus aero engines, 2062
Nitro-glycerine, 2797
Noggings, brick, 369
'Nomag' non-magnetic cast iron, 2348
Non-corrodible steel, 2713
Non-ferrous alloys, 2721
Non-freezing mixtures, 105
Non-magnetic alloys, 2743
——————— cast iron, 2348
Non-resonant gears, 919
Northampton iron, 2682
North-Eastern superheaters, 2403
——————— Werkspoor marine Diesel engine, 2419
Norton & Heald grinding machines, 1120
Notch, rectangular, method of calculating quantity of water flowing in a stream, 1286
———, 'V,' automatic water recorder for steam engines, 1849
——————— method of calculating quantity of water flowing in a stream, 1268
Nozzles, 1442, 1770
———, air, 2665
——— flow of steam through, 1538
——————— water through, 1306
——— steam turbine, 1770
Nut and bolt machines, cutting speeds of, 1054
—— locking, 1159
——, to remove a stubborn, 2815 [gonal, 1154
Nuts and bolt-heads, British standard hexa-
——————— weights of, 161
——— and bolts, 538, 1152

OIL

Nuts and bolts, American, 1153 [for, 538
——————— for railway plant, specification
——————— Whitworth, 1152, 1153
——, capped, 1159
—— castle, British standard, 1160
—— cold punched, number per 100 lbs., 161
——, washers, and bolt pins, French, 1158
—— wing, 1161
'Nuvako' pneumatic conveying system, 848

OAK work, preservation of, 2816
Obach dry battery, 2255
Objects at sea, distance of, 76
Observations, timing, 80
Obsolescence of plant, 2897
Octagon rod metal, weight of, 135
——— steel, weight of, 137
Offsets, setting out curves by, 563
Offshore dock, 1477
Oil, action on metals, 980
— burners, safe use of, 2167
— burning system, Prior, 2165
— ——— Thornycroft, 2164
— ——— ——— Wallsend-Howden, 2163
— consumption of locomotives, 624
— elimination from feed-water, 1680
— engine electricity stations, 1884
— engines, Diesel, 1893
——————— Allen, 1899
——————— Beardmore-Tosi, 1898, 2432
——————— Belliss & Morcom, 1897
——————— Burmeister & Wain, 2413
——————— care of, 1904
——————— Carels, 1900
——————— coal-tar fuel for, 2160
——————— cost of, 1904
——————— crank shafts, 1904
——————— Doxford, 2422, 2423 [1902
——————— English Electric Co.'s, 1898,
——————— Fiat British Auxiliaries, Ltd., 2430
——————— ——— Fullagar, 1902
——————— Gleniffer, 1931
——————— Krupp, 2430
——————— M.A.N., 2427
——————— marine, 2411
———————, oil fuel for, 2404
——————— output of, 2434
——————— power of, 2437
——————— types of, 2412
——————— weight of, 2434
——————— Mirrlees, 1896
——————— Morcom & Belliss, 1897
——————— North-Eastern Werkspoor,
——————— packings for, 1903 [2419
——————— power of, 2435
——————— Richardsons-Westgarth, 2428
——————— Schneider, 1899
——————— Scott's, 2424, 2425
——————— shafts of, 1904
——————— submarine type, 2432
——————— Sulzer, 1899, 2420
——————— Swan, Hunter, & Wigham Richardson, 2432
——————— Swiss Locomotive Works, 1899
——————— tar fuel for, 2160
——————— Vickers', 2424, 2426

OIL

Oil engines, Diesel, weight of, 1903
—— —— —— —— Werkspoor, 2419
—— —— —— —— White, 2432
—— ——, failure of parts of, 1889
—— —— —— flywheels of, 1888
—— —— —— generation of electricity by, 1884
—— —— —— high compression, 1916
—— —— —— —— —— —— —— —— Babcock, 1917
—— —— —— —— —— —— —— —— Blackstone, 1920
—— —— —— —— —— —— —— —— Campbell, 1919
—— —— —— —— —— —— —— —— Clayton-Babcock, 1917
—— —— —— —— —— —— —— —— Crossley, 1917
—— —— —— —— —— —— —— —— Maclaren-Benz, 1922
—— —— —— —— —— —— —— —— National, 1919
—— —— —— —— —— —— —— —— Paxman, 1920
—— —— —— —— —— —— —— —— Petter, 1919 [1918
—— —— —— —— —— —— —— —— Ruston-Hornsby,
—— —— —— —— —— —— —— —— Vickers, 1917
—— —— —— low compression, 1922
—— —— —— —— —— —— —— —— Blackstone, 1922
—— —— —— —— —— —— —— —— Dorman, 1924
—— —— —— —— —— —— —— —— Hamworthy, 1923
—— —— —— —— —— —— —— —— National, 1923
—— —— —— —— —— —— —— —— Petter, 1923
—— —— semi-Diesel, 1911
—— —— —— —— —— —— —— Allen, 1911
—— —— —— —— —— —— —— Bolinder, 1913
—— —— —— —— —— —— —— Brotherhood, 1912
—— —— —— —— —— —— —— Clayton-Babcock, 1917
—— —— —— —— —— —— —— Crossley, 1911
—— —— —— —— —— —— —— Fielding & Platt, 1914
—— —— —— —— —— —— —— McLaren-Benz, 1922
—— —— —— —— —— —— —— Marshall, 1911
—— —— —— —— —— —— —— Mirrless' Simplex, 1914
—— —— —— —— —— —— —— Robey, 1912
—— —— —— —— —— —— —— Still, 1916
—— —— —— tar oil for, 2160
—— —— —— valves of, 1888
——, filtration of, 982
—— fired furnaces, 2666
—— fuel atomiser, 2162
—— —— British standard specification for, 2167
—— —— combustion, air required for, 2166
—— —— for Diesel marine engines, 2404
—— —— locomotives, 264
—— —— marine engines, 2404
—— —— Navy, 2166
—— —— furnaces, 2666
—— —— heating, 1068
—— ——, notes on, 2168
—— —— special, 2167
—— —— system, Prior, 2165
—— —— —— —— Thornycroft, 2164
—— —— —— —— Wallsend-Howden, 2163
—— gas heater, Burdon, 1068
——, lubricating for locomotives, 624
—— mills, 792
—— mixtures for external combustion, 2167
——, physical and frictional properties of lubri-
—— pumps, 1275 [cating, 981
—— separators, 1681
—— —— —— —— Baker, 1681
—— —— —— —— Drayton, 1682
——, shale, 2150
—— switch and transformer, 2347
—— *v.* steam engines, 2439
——, watch, 2810
—— well drilling, wire ropes for, 2612

PAC

Oiling leather belts, 862
Oils, 977
—— action on metals, 980
—— filtration of, 982
—— flash temperatures of, 117, 981
—— insulating, 2347
—— lubricating, 980
—— pumping, 1275
—— shale, 2150
—— standard, frictional properties of, 981
—— transformer, 2347
—— vegetable, manufacture of, 792
—— viscous flow of, 994
—— viscosity of, 978, 993
Oilstones, to face, 2814
Olefiant gas, 2171
Omnimeter, 63
Optical apparatus, lighthouse, 1495
—— pyrometers, 2676
Optics, 119
'Optimum' de-aerator, Weir, 1640
Orders, standing, for railway bills, 664
Ordnance scale maps, 56
—— —— Survey scales, 56
Ore dressing, 2619
——, horse-power per ton required to treat
Ores, copper, tin, lead, zinc, 2562 [2641
Orifice of a culvert, discharge from the sub-
Orifices, 1442 [merged, 1359
—— discharge of water through, 1306
—— for test purposes, 1442
—— flow of steam through, 1538
Oscillation, centre of, 31
Osram electric lamps, 2324
Outer rail, superelevation of, 555
Ovens, electric baking, 2333
Overall dimensions of ships, 2353
Overhead electric cranes, 802
—— —— —— tramways, 692, 694
—— —— —— transmission lines, 2220, 2221, 2227
—— —— lines, aluminium, 2220
—— —— —— copper, 2221
—— —— travelling cranes, 802
—— —— structures, anchorage of, 2349
Oxy-acetylene cutting, 2777
—— —— welding, 2759 [gas, 2169, 2475
Oxygen and air required for combustion of
—— —— —— volume necessary for combustion per cubic foot of gas, 2170
—— supply for acetylene welding, 2761

PACKARD motor car engine, 1940
Packing, centrifugal pump, 1268
—— —— condenser tube, 1827
—— —— hydraulic, 1264
—— ——, labyrinth, turbine, 1792
—— —— metallic, for piston rods, 'Beldam,' Asbestos Co., 1755
—— —— —— —— —— Beldam Packing and Rubber Co., 1753, 1754
—— —— —— —— —— 'Britimp,' 1199, 1751, 1793 1888, 1903
—— —— —— —— —— Bell's United Asbestos Co., 1754
—— —— —— —— —— James Walker & Co., 1752 1753

PAC

Packing, metallic, 'Lascar,' 1755
——————— 'Lion,' 1752, 1753
——————— 'Reefer,' 1754
——————— tool for, 1755
——————— United Kingdom, 1750
——————— United States, 1749
——— pump, 1264
——— ring, 1756
——— rings of pistons, metal for, 2740
——————— Rowan, 1756
——————— S.E.A., 1756
———, Woodite, water gauge, 1663
Paint, aluminium, 418
———, brick floor, 2804
——— brushes, to clean, 2811
———, Carson's, 2801, 2802
——— coal tar, 2806
——— felt roofing, 2804
——— fire-resisting, 2805
——— flexible, 2803
——— galvanised roofing, 234
——— glazier's, 416
——— graphite, 2806
——— ground glass, 2803
——— heat-indicating, 962, 2806
——— hydraulic cement, 2803
——— luminous, 2806
——— oil, quick-drying, 2805
——— priming, 2801
——— red, quick-drying, 2804
——— removal of, 2801
——— roof, 2805
——— value of, 417
——— waterproof, 2804
——— waterproofing boards, 2804
——— waterproofing wall, 2804
——— wooden floor, 2804
——————— post preserving, 2805
Painters' work, constants of labour for, 418
——————— valuation and estimate of, 417
Painting, 416, 2801 [
——— floors, 2804
——— fresh cemented surfaces, 416
——— galvanised iron, 234
——— ironwork, 2802
——— Portland cement, 2806
——— ships' bottoms, 2444
———, spray, 417
——— woodwork, 2801
——— zinc, 2803
Palatine Engineering Co.'s water meters, 1436
——————— waste water detection system, 1420
Panama Canal, 1527
——— dredging, 1486
Pantagraph, 953
——— Harling, 90
Panties, 414
Paperhanging, 419 [1222
Paper insulation, cold storage, Willesden, 2346
——————— electrical, 2210
——————— Willesden, 2346
——— mills, 792
———, sound-proofing, Willesden, 342
——— underlining, Willesden, 413
Papers and books, sizes of, 19
Paragon air pump, 2408
Parallax, equatorial, of sun, 76
———, sun's, in altitude, 78, 83
Parallel motion, Scott Russell's, 954

PHO

Parallel slide valves, 1758
Parallelogram of forces, 26
Parian cement, 406
Paris, electrification of suburban lines in, 680
Parkes and Martin, melting points of alloys,
Parsons geared turbines, 2381 [104
——— steam turbine, blades and caulking pieces for, 1790
——— turbine gearing, 2386
Partitions, 341
——— blocks for, 341
——— expanded metal, 335
Parts of gas engines, 1887
——— locomotives, 592
Passages, steam, area of, 1748
Passenger trains, effect of gradients on speed
Pastes, 397 [of, 585
Patent Specification Drawings, 90
Patents, 2833
Pattern composition, 2754
Patterns, weight compared with castings, 161
Pavements, concrete, filler for, 745
——————— tractive effort on, 726
Paving, asphalt, 743, 745
——— cost of, 744, 746
——— deal, 747
——— hardwood, 743
——— Metropolitan Committee's Report on,
——— rubber, 747 [743
——— stone, 745
——— street, 750
——— streets of London, 743, 745
Paxman 'Economic' boiler, 1581, 1606
——— engines, 1724, 1733
——— gas engines, 1874
——— oil engines, 1920
——— portable engines, 1733
——— positive valve gear engine, 1724
Peat, 2105
Peaucellier straight line link motion, 954
Pelton wheel buckets, 1310
——— regulators, 1312
——— wheels, 1309
Pen, drawing, Stanley's patent, 91
Penberthy cellar drainer, 1263
——— injector, 1655
Pendulum, 30
——— ballistic, 34
Percussion, centre of, 31
Perfector reinforced concrete system, 452
Permanent way, railway, 542
——————— lengthening life of track, 543
——————— Ministry of Transport regulations for, 543
——————— tramway and light railway,
Permitted explosives, 2798 [701
Permutit water softener, 1422, 1628
Peters, G. D., & Co.'s plastic arc welding
Petrol, 2150 [system, 2786
——— air gas, 2487
——— electric lorries, 707
——— engines (*see* Automobile engines)
———, efficiency of, 1984
——— pipe joints, 1247
——— vehicles, working costs, 760
Petroleum, 2149
——— fuel, properties of, 2152
Petter's oil engines, 1921, 1923
Phœnix lifting magnets, 823 [2330
——— Office rules for lightning conductors,

PHO

Phosphor bronze, 2733
——————— springs, 1012
Photo-copying, 92
Photo-electric cells and their uses, 2350B
Photographic lens apertures, 119
Photometry, 2527 [tion, 981
Physical properties of standard oils for lubrica-
Picric acid, or lyddite, explosive, 2798
Piers, bridge, 526
——, marine, cost of, 2872
Pig iron, 2680
Pile, Coignet reinforced concrete, 498
—— sinking with water jet, 501
Piled foundations, 497
Piles, 463, 465, 497
—— concrete, 498
——————— moulded-in-place, 499
—— formulæ for safe load on, 502
—— interlocking steel, 504
—— loads carried by, 500
—— reinforced concrete, 463, 465
——————————— loads on, 501
—— safe load on, 502
—— screw, 503
—— sinking by water jet, 501
Piling hammers, 502
——, sheet steel, 504
——————— Ransome type 'D,' 504
——— 'side-groove' steel, 505
——— simplex, 498
Pillars, British standard, 242
—— cast iron, 375
—— mild steel, 375
——— mining, 2585
—— reinforced concrete, 442
——— shaft, 2585
——— wooden, 364
—— wrought iron, 375
'Pilot' jointing, Beldam, 1690
——— packing, Beldam, 1754, 1756
Pine beams, breaking weight of, 185
Pinions, non-resonant, 919
———, raw hide, 919
Pins, crank, diameter of, 1003
Pintsch's illuminated buoys, 1490
Pipe bending, 1092
—— bends, unbalanced pressures on, 143
—— ——, corrugated, 1686
—— —— for steam hammers, 1687
—— ———, steel and wrought iron, 1685
—— couplings for automatic brakes, 633
—— coverings, 1695, 1697
—— flanges, 1166
——————— American standard, 1168
——— ——— British standard, 1166
—— hangers, 1688
—— jointing materials, weight of, 164
—— joints, 1244
——————— cement, 1247
——————— petrol, 1247
——— ——— 'Victaulic,' 1169, 1246
—— lagging, 1695
—— lines, 1345
———, flexible, 'Victaulic,' 1246
——————— (steam) drop of pressure in, 1691
—— sockets, 1429, 1430
—— supports, 1303
—— thickness (steam), 1687
—— thread tap heads, 1173
—— threads, 1150

PIP

Pipe threads, American standard, 1152
——————— British standard, 1151
——————— Whitworth standard, 1152
—— unions, 1244
Pipes, aluminium, 159 [2808
—— Angus Smith's composition for iron
—— brass, weight of, 146 [2497
—— British standard low-pressure heating,
—— cast-iron, 145, 147, 150, 154, 1430, 2477
———————, weight of, 145, 147, 150, 154, 1430, 2477, 2497
—— concrete, 340, 467, 1303, 1363
——————— lined, 1247
—— copper, weight of, 146, 156
—— design of, 1302
—— diameter of, 1361, 1419
—— discharge of water through, 1298, 1299, 1300, 1301, 1364, 1423, 1426, 1429
—— drain, 429, 1379
—— exhaust, 1689
—— expanded steel concrete, 340
—— flanges of, 1689
—— flow of air through, 1082, 1693
——————— steam through, 1691
——————— water through, 1298, 1364, 1423, 1426, 1429
—— for waterworks, 1423
——, gas, 145, 165
—— ——, British standard, 165, 1150
—— —— threads of, 1150
——————— weight of, 145, 2477
—— heating, 2497
—— hydraulic, thickness of, 1242
—— iron, corrosion of, 1419
—— —— soil, weight of, 1381
—— —— water and hydraulic, thickness of,
—— lead, corrosion of, 420, 1419 [1242
——————— soil, 1381
———————, strength of, 192
———————, weight of, 146, 1381, 1431
—— pressure in, 1242
—— rain-water, 420
—— reinforced concrete, 467, 1303
—— salt glazed, for sewers, 430, 1360
—— screw threads of, 1150
—— sewer, British standard, 1360
—— socket, and fittings, 151, 152, 155
—— soil, 1381
—— speed of water through, 1242, 1424
—— steam, 1684
—— ——— bends, 1685
——————— copper, 1687
——————— corrugated, 1686
——————— coverings of, 1695, 1697
——————— exhaust, 1689
——————— flow of steam through, 1691
——————— joints of, 1690
——————— leaks in, 1690
——————— Lloyd's rules for, 1687
——————— loss of heat from, 1693
——————— of boilers, 1684
——————— pressure, drop in, 1691
——————— size of, 1690
——————— steel bends, 1685
——————— thickness of, 1687
——————— valves of, 1668
——————— weight of, 145
——— steel, corrosion of, 1419
——————— design of, 1302
———————, identification of, 167

PIP

Pipes, thicknesses of, 164
—— velocity of water in, 1242, 1298, 1364,
—— water, 145, 164, 1423 [1423, 1428
———— British standard, 165, 166
———— corrosion of, 420, 1419
———— hammer in, 1347
————, incrustation of, 1419
———— joints in, 1244
———— threads of, 1150
———— weight of, 145
———— wooden stave, 1304
—— wrought-iron, 145, 148
———— weight of, 145, 148
Piping for cold-storage rooms, 1220
Piston of steam indicator, 1717
—— nut-lock plate, 1757
—— packing-rings, metal for, 2740
———— Rowan, 1757
—— rod packing rings, Beldam (Beldam Packing & Rubber Co.) 1753, 1754
———— 'Britimp,' 1751
———— 'Lascar,' (Beldam Asbestos Co.), 1755
———— 'Lion' (James Walker & Co., 1752, 1753
———— 'Reefer' (Bell's United Asbestos Co.), 1754
———— United Kingdom metallic, 1749
———— United States metallic, 1748
—— speed of marine engines, 2373
Pistons, 1756
—— motor car, 1956
Pitch of broach teeth, 1026
—— cutter teeth, 1026
—— impulse blades of steam turbines, [1788
Pitot pressure tubes, 1506
Pit type furnaces, 2666
P. I. V. gear, 951
Placer mining, 2642
Plane figures, centre of gravity of, 23
——, inclined, 26
—— surface power required to move against air, 1183
—— table (survey instrument), 68
Planing machine tests, 1023
—— machines, 1021
———— power for, 1055
———— speed table for, 1023, 1024
Planks, timber, 356
Plans, railway, House of Commons rules for,
Plant, depreciation of, 2897 [665
—— ice-making, 1219
Plaster, hardening, 406
—— of Paris, 406
Plastering and lathing, 409
Plasters, 406
——, hard, 406
Plasterwork, expanded metal lathings for, 334
Plate-bending tool, 1093
Plates, bulb, 239, 241 [615
—— copper, for locomotive fire boxes, 614,
—— for locomotive boilers, specification of, 614, 615
——, steel, for locomotive boilers, 617
———— frames, 618

POR

Plates, thickness of, Robey boilers, 1576
—— tin, weight of, 140
—— wall, 329
Platinite, 2727
Platinum, tensile strength of, 170
—— substitute, 2727
Platnam, 2727
Plenum ventilation system, 2523
—— heating system, 2499
Ploughing, power required for, 47
Plugs, fusible boiler, 1609
Plumbing, 419
Plunger pump, 1250
Plymouth breakwater, 1458
Pneumatic air mains, 1081
———— caulking hammers, 1081
———— despatch tubes, 1202
———— drills, 1081
———— foundations, frictional resistance of, 508
———— hammer, Boyer long stroke, 1079
———— Massey's power, 1129
———— hose couplings, 1084
———— house tubes, 1206
———— portable tools, 1079
———— power hammers, 1129
———— Massey, 1129
———— process of sinking cylinders, 503
———— railway signals, 648
———— receivers, 1081
———— riveting, 1086
———— tools, care of, 1085
———— for salvage work, 1081
———— portable, 1079
———— tubing, flexible, for locomotives, 607, 608
———— mining, 2571
———— tools, 1081
———— welding, 2766
———— tyres, 1976
'Poilite,' 323
———— corrugated sheeting, 323
———— flat sheets, 323
———— louvres, 323
———— roofing tiles, 414
———— slates, 412
———— wall tiling, 324
Points and crossings of tramways, 689
——, railway, 657, 658
Polar curves (light), 2528
—— radius of earth, 76
Pole plate of roofs, 320
—— star, to find, 76
———— variation of compass by, 73
Poles for electric tramways, 695, 703
——, strength and stiffness of, 314
—— telegraph and telephone, 2339
—— tramway, 695, 703
Pole's formula for flow of gas in mains, 2478
Polishing, 1123
Pollock, Bolinder oil engine, 1913
Polygraphic copying process, 94
Polyphase motors, 2294
———— efficiencies of, 2294
Ponds, cooling, 1851
Pontoon sectional dock, 1477
Porcelain insulators for overhead lines, 2225, 2350A
Porosity of bricks, 428
Port Natal dredging plant, 1485

POR

Portable boiler, Babcock & Wilcox, 1598
——— electric drills, 1042
——— engines, Paxman, 1733
——— pneumatic tools, 1079
Portal bracing of bridges, 289
Portland cement, 401
——— ——— American, 405
——— ——— blast furnace, 405
——— ——— British standard, 403, 405
——— ——— concrete, American specification for waterproofing material, 405
——— ——— mortar, tints, 387
——— ——— painting, 2806
——— ——— reinforced concrete, 439
——— ——— testing, 401
——— ——— machine, Adie's, 402
——— ——— tinting, 387
Positive valve gear engine, Paxman's, 1724
Posts, timber, weight of, 364
Potter, F. W., & Co.'s wing fan, 2521
Pounds to tons conversion table, 10
Power, absorptive, of coloured glass, 1500
——— animal, 44
——— ——— labour produced by, 44
——— brakes for measurement of, 34
——— ——— ——— Alden's, 38
——— ——— ——— Brotherhood fluid friction, 38
——— ——— ——— fan, 41
——— ——— ——— Froude's, 38
——— ——— ——— Hirn's rope, 34
——— ——— ——— Prony, 35
——— ——— ——— Ringelmann's, 36
——— ——— ——— water-pipe absorption, 37
———, electric driving, in workshops, 773
——— factor improvement, electrical, 2307
——— gas units, 1861
——— hammers, 1125
———, holding, of glue, 176
——— horse, 44
——— in pumps, loss by friction, 1265
——— losses in machine shops, 1061
——— men, 44, 46,
——— of Diesel marine engines, 2435
——— electric locomotives, 553
——— fall of water, to compute, 1283
——— horses, 44
——— locomotives, 587
——— men, 44, 46
——— surfaces, reflecting, 2527
——— windmills, 1185
——— Plant Co.'s helical gearing, 929
——— ——— turbine gears, 933
——— plant, thermal efficiencies of, 1539
——— required by railway brakes, 638
——— ——— for aerofoils, 2006, 2020
——— ——— bolt and nut machines, 1054
——— ——— cotton machinery, 789
——— ——— drilling, boring, and screwing machines, 1054
——— ——— grinding machines, 1056
——— ——— lathes, 1053
——— ——— machine tools, 1053

PRE

Power required for planing machines, 1055
——— ——— ploughing, 47
——— ——— punching, shearing, and cropping machines, 1056
——— ——— shaping and slotting machines, 1055
——— ——— wood-working machines, 1058
——— ——— to compress free air, 1188
——— ——— move plane surface against the air, 1183
——— ——— pump water, 1241
——— ——— work pneumatic despatch tubes, 1202
——— stations, electric, cost of, 2860
———, tide, 1283
——— transmission lines, 2220, 2229
——— ——— of, by chain driving, 899
——— ——— cotton ropes, 874
——— ——— leather belting, 855
——— ——— link belts, 869
——— ——— steel shafting, 1001
——— variation of a blower, 2520
———, water, 1283
——— windmill, 1185
Pre-gasification of coal, 2132
Pre-heater, air, 1648, 2393
Premier electric welding system, 2785
——— gas engine, 1874
——— oil engine, 1918
——— water cooling towers, 1851
Preservation of oak work, 2816
——— ——— ropes, 200
——— ——— timber, 360, 361
Preservative compositions, 2816
——— ——— for iron work, 2812
——— ——— stone work, 2811
——— ——— wood posts, 2816
Press, hydrostatic, 1349
Pressboard, 2346
Pressed bricks, 426
Pressure acceleration produced by, 28 [2015
——— and temperatures of air at heights
——— flanges, cast-iron, 1245
——— for pneumatic tyres, 1976
——— gauge tester, Brotherhood's, 1666
——— gauges, 1666
——— in cylinders, allowable, 192
——— hydraulic mains, 1242
——— ——— pipes, 1302
——— of atmosphere, 1182
——— earth on retaining walls, 514
——— steam, mean, 1706, 2372
——— water, 1239
——— on retaining walls, 514
——— wind and weight of snow on roofs, 322
——— on archstones, 525
——— producing acceleration, 28
——— regulator, steam, 1735
——— required to drive carrier through pneumatic despatch tubes, 1202, 1205
——— tanks, 2572
——— tubes, Pitot, 1506
——— unions, 1245
———, volume of liquids reduced by, 117
——— working, pneumatic despatch tubes, 1202

PRE

Pressures, bearing, 961
——— boiling points of water at various, 109
——— draught, rate of combustion, 1543
——— for riveting and flanging, 1562
———, gas, 2478
——— metric, 9
——— unbalanced, on pipe bends, 1431
Preventives of corrosion in boilers, 1623
——— scale in boilers, 1623, 1637
——— and corrosion in boilers, Cumberland system, 1624
Price's rule for superelevation of outer rail of railway curve, 556
Primary batteries, 2255
Priming, paint, 2801
Prior oil burning system, 2165
Prismatic compass, 68
Producer, bituminous suction gas, 2186
——— gas, 2173, 2174
——— composition of, 2174, 2175
——— heat value of, 2175
——— water, 2189
Producers, gas, 2174
——— Appleby-Bentley, 2179
——— Crossley, 2180
——— Kerpely, 2177
——— Mond, 2183
——— Ruston, 2187
Prony brake, 35
——— formula for water velocity, 1507
Propeller propulsion, 2441
——— shafts, Board of Trade rules, 1005
——— Bureau Véritas rules, 1005
——— Lloyd's rules, 1005
Propellers, aeroplane, 2000
——— screw, metals for, 2008
Properties of aluminium *v.* copper for electrical conductors, 2213
——— gases, 107
——— saturated steam, 1532, 1833
——— steel at high temperatures, 1823
Proportions for concrete, 389, 408
——— mortars, J. A. van der Kloes, 386
Propulsion, reduction gear for, 946
Protection of iron from rust, 2712, 2803
Protective distance of wells from drains, 1409
Puddle, dams, 509
Pugging, 314
Pulley arms and bosses, 1174
——— speeds for cotton mills, 792
Pulleys, 25
——— and drums, rope, sizes of, 880
——— belt, 864
——— how to put belts on, 867
——— mining, 2605
——— rope driving, 870, 872, 880
Pulsometer centrifugal pump, 1269
——— high lift turbine pump, 1270
——— 'Torrent' filter, 1416
——— refrigerating apparatus, 1232
——— steam pump, 1261
Pul-syn-etic electric clock system, 2327
Pulverised coal fuel, 2125
——— advantages of, 2144
——— for locomotives, 622, 2141
——— systems, Clarke Chapman, 2134
——— International, 2125, 2137

PUR

Pulverised coal systems, 'Lopulco,' 2125, 2137
Pump balancing, 1250
——— barrels, 1250
——— buckets, 1250
——— dredgers, 1483
——— governor, Copes, 1680
——— leathers, 1265
——— openings, 1851
——— packings, 1264
——— valves, 1250
——— materials for, 1264
Pumping, boiler pressure required for, 1262
——— by compressed air, 1276
——— engines, 1714
——— from wells, 1251
——— in mines, 2612, 2645
——— machinery, air, 1203
——— oils, 1275
——— water, horse-power required for, 1242
Pumps, 1249, 1849
——— air, 1840
——— lift, 1276
——— artesian well, 1251
——— boiler feed and service, 1254
——— Boulton, 1253
——— capacity of, 1252
——— centrifugal, 1266
——— , packing of, 1268
——— circulating, condenser, 1842
——— Clarke, Chapman steam, 1254
——— compressed air, 1276
——— Cornish, 2612
——— Edwards air, 1842
——— feed, 1254, 1256
——— force, 1250, 1850
——— hot water centrifugal, 1270
——— Leblanc air, 1844
——— lifting, 1249
——— mining, 2612
——— Mirrlees ejector air, 1846
——— oil, 1275
——— plunger, 1250
——— Pulsometer centrifugal, 1269 [1270
——— high-lift turbine pump,
——— steam pump, 1261
——— reciprocating, discharge from, 1252
——— semi-rotary, 1252
——— steam, 1254
——— Stone-Paramor rotary, 1275
——— turbine, 1271
——— suction, 1249
——— turbine, 1270
——— turbo, 1260
——— turbo-feed steam, Weir, 1260
——— 'VSG', 1274
——— vacuum, Bailey's, 1200
——— variable delivery pressure, 1274
——— viscous fluid, 1275
——— Warner Anthony, 1256
——— Weir direct-acting steam, 1256
——— turbo-feed steam, 1260
——— Willcox semi-rotary, 1252
——— ejector steam-jet, 1657
——— wind, 1186
——— Woodeson steam, 1254
Punched nuts, cold, number per 100 lbs., 161
Punching machines, power for, 1055
——— plates, power required for, 1061
Purchase of machinery, 2912
Purification of gas, 2454, 2465

PUR

Purification of water, 1419
Purifier and water softener, Royle's, 1632
Purifiers, gas, cost of, 2469
Purifying water for boilers, 1632, 1634
Purity of sewage effluents, standard of, 1378
——— water, standard of, 1418
Push plate conveyors, 846
Putty, 416, 418, 2801
Pyrometers, fusion, 2674
——— heat radiation, 2676
——— optical, 2676
——— thermoelectric, 2677

QUA

QUARRYING—(*see* Excavation), 483
Quasi-Arc electric welding, 2781
Quay cranes, 1470
——— walls, cost of, 1465
Quayside appliances, 1470
——— burtoning, 1471
——— transporters, 1471
Queen-post roof, 319
——— posts, 316
Quenching baths for tempering cutting tools, 1070
——— liquids, 1070
Quick change universal drill jig, 1047
——— drying oil paint, 2805
——— varnish, 2807
——— return motion, 956

RAC

RACES, head, 1298
Racks, strainer, 1297, 1343
Radial inward flow turbines, 1325
Radiant heat transmission in boiler furnaces, 1704
Radiator shutters for aero engines, 2070
——— tablets, Boilerine, 1953
Radiators, 2503
——— aero engine, weight of, 2070
——— Westinghouse, 2506
Radii of curvature for rails, 556
——— curves, table of, 564
Radio communication, 2343, 2821
——— audio-frequencies, radio and reactance of condensers, 2830
——— capacity chart and resistance, 2829
——— condensers, reactance of, at radio and audio-frequencies, 2830
——— copper wire tables, 2825
——— empirical formulæ, 2826
——— frequency tables and wavelength, 2827
——— glossary of technical terms, 2821
——— radio and audio-frequencies, reactance of condensers at, 2830
——— radio formulæ, 2830A
——— reactance of condensers at radio and audio-frequencies, 2830

RAI

Radio communication, resistance and capacity chart, 3829
——— technical terms, glossary of, 2821
——— useful formulæ, 2823
——— wave length and frequency tables, 2827
——— wire, copper, bare, 2827
——— cotton covered, 2826
——— enamelled 2827
——— silk covered, 2825
——— tables, 2825
Radio-telegraphy, 2342
Radius, equatorial, of earth and sun, 74
——— of bends for steam pipes, 1685
——— gyration, 32
Rafters, 319
Rail bearers of bridges, 281
——— bonds, 697
——— chairs, screws for, 538
——— conductor leakage on electric tramways, [700
——— joints, railway, 537
——— tramway, 688
——— motor cars, 626
——— outer, superelevation of, 555
——— welding, 699
Railless electric traction, 703
Rails, 529
——— American, 533
——— British standard, 529
——— bullhead, 529
——— dimensions of, 529
——— electric tramway, 684
——— flat-bottomed or flanged, 533
——— radii of curvature and versed sines, 556
——— Sandberg, 540
——— steel, Board of Trade report on, 535
——— breaking load of, 536
——— tramway, 684
——— British standard, 684
——— weight of, 535
Railway Bills, Standing Orders of House of Commons for, 664
——— ballast, 542
——— brakes, 638
——— bridges, 248
——— allowance for impact on, 269
——— live loads on, 249
——— rolling loads on, 249
——— standard clearances, 250
——— working stresses on, 262
——— carriages, 627
——— axle boxes, 632
——— couplings, 627, 633
——— electric lighting of, 633
——— heating by steam, 633, 635
——— springs, 632
——— wheels, 632
——— check rails, 538
——— construction, cost of, 543
——— crossings, angles and leads of, 562
——— cross-overs, 538

Railway curves, 562
——— ——— ranging out, 562
——— ——— resistance of, 585
——— ——— superelevation of outer rail for, 555, 556
——— cuttings, volume of earth in, 481
——— ——— walls of, 514
——— electrification, standards for, 673
——— expenditure and receipts in United Kingdom, 661
——— gauges, 552
——— gradients, resistance due to, 585
——— locomotives, American, 576
——— ——— ——— Great Western, 574, 576
——— ——— ——— London Midland and Scottish, 572, 574
——— ——— ——— London and North Eastern, 572, 574
——— ——— Southern, 574, 576
——— motor cars, steam, 626
——— permanent way, 542
——— platforms, 569
——— point movement, British Power Railway Signal Co.'s low-voltage system, 658
——— points, mechanically operated, 659
——— rail joints, wear plate for, 537
——— rails, 529
——— ——— bolts, nuts, screws, and spikes for, 538
——— ——— check, 538
——— ——— maximum allowable axle load on, 536
——— ——— metal keys for, 542
——— ——— superelevation of, 555
——— ——— weight of, 535
——— rolling stock, 627, 661
——— ——— ——— axle boxes, 632
——— ——— couplings, 627, 633
——— ——— springs, 632
——— ——— material, British standard specification, 611-620
——— Signal Co.'s 'Crewe' all-electric signal system, 650
——— signals, 645
——— ———, Railway Signal Co.'s, 650
——— ———, Sykes', 654
——— ———, Westinghouse, 648
——— ———, wire for, 178
——— sleepers, 538
——— stations, 569
——— statistics, 663
——— track renewal, cost of, 664
——— tunnels, 554
——— wagons, 630
——— ——— axle boxes, 632
——— ——— couplings, 627, 633
——— ——— springs, 632
——— ——— wheels, 642
Railways, 529
——— cost of, 543
——— electric, 669
——— gauges of, 552
——— land required for, 552
——— light, 552
——— ——— cost of, 553
——— Ministry of Transport regulations, 543, 664
——— narrow-gauge, 552

Railways, resistance to traction on level, 583
——— tube, London, 678, 679, 2984
——— United Kingdom, expenditure and receipts, 661
Rainfall, 1399
——— available power from, 1407
——— quantities and rates, 1402
——— absorption and evaporation, 1402
——— and forests, 1406
———, flood discharge, 1403
——— heavy, in various countries, 1405
——— in British Isles, 1399
——— ——— foreign countries, 1400
——— monsoon, India, 1383
Rain gauges, 1406
Rain-water pipes, 420
Raising and winzing, mining, 2579
Ralston's stability and trim indicator, ship, [2369
Ram cylinders, 1349
——— leathers, 1264
Rammed furnace linings, 432
Ramoneur boiler tube cleaner, 1683
Rams, hydraulic, 1350
Ranging, 51
——— out curves, 562
Rankine, formula for safe load on piles, 502
——— on masonry, 379
——— rules for dams, 1411
Ransome steel sheet piling, 504
Ransomes, Sims & Jefferies electric trolley buses, 705
Rapid cutting tool steels, 1062
——— hardening ferrocrete, 405, 735, 745
——— magnetic separators, 850, 2626
Ras Serani Lighthouse, Mombasa, 1497,1499
Ratios, cylinder area, ship, 2372
Rattler test for bricks, 428
Rawlplug bolt anchor, 421
Rawlplugs, 420
Reaction turbines, steam, 1788, 1791
——— ——— water, 1315
Receipts, United Kingdom railway, 663
Receivers, air, 2571
Recipes, workshop, 2801
Recipro-turbine machinery, 2374
Reciprocating pumps, discharge from, 1252
Recorders, CO_2, 2083
——— duplex-mono, Gordon's, 2084
——— Lea's automatic water, for steam engines, 1849
——— Mirrlees cyclic variation, 1908
——— weir, 1441
Rectangular beams, 455, 460
——— ——— ——— pine, breaking strain of, 185
——— ——— notch method of measuring quantity of water flowing in a stream, 1286
——— ——— slabs, reinforced concrete, 459
Rectifiers, mercury vapour, electric, 2277
Reddaway, F., & Co.'s belting, 857
Reducing valves, 1671
Reduction gear, 946, 2381
——— of volume of liquids under pressure,
'Reefer' packing, Bell's, 1754 [117
Refacers, valve, 1670
Reflecting level, Abney, 68
——— ——— power of surfaces, 2527
Reflectors, 2535
——— ——— Benjamin, 2544
Refractions, mean, table of, 83

REF

Refractory cements, 398
—— materials, 430
—— melting points of, 104, 430
—— mortars, 432
—— and cement, specification for, 431
Refrigeration, 1209
Refrigerants, 1214
Refrigerating condensers, 1215
—— evaporators, 1215
—— machinery, 1211, 1217, 1223
—— ammonia absorption, 1217
—— compression, 1211
—— carbonic acid, 1213
—— Hall's, 1224
—— Haslam's, 1226, 1230
—— Liverpool Refrigeration Co.'s, 1226
—— Pulsometer, 1232
—— for ships, 1224
—— Lloyd's rules, 1234
—— thermometer for, 1232
—— power, 1219
Refuse, calorific value of, 2148
—— collection of, 757
—— destructors, 2148
—— City of Glasgow, 2148
—— Heenan's test, 2148
—— fuel plants, 2181
Registered dimensions of ships, 2353
Regulations, Danish, for working lifts, 1354
—— reinforced concrete, 443
—— road, for heavy motor-cars, 722
—— traction engines, 722
—— wheels and loads, 722
—— steelwork, 373
Regulator, air, mining, 2616
—— 'Arca' automatic, 1677
—— feed water, 1672
—— Cope's, 1678
—— Crosby, 1672
—— Drayton, 1673
—— Thermo-feed, 1674
—— induction, 2305
—— locomotive, 601
—— Pelton wheel, 1312
—— temperature, 1642
—— turbine, water, 1333
Reinforced concrete, 388, 439
—— arches, 470
—— bars for, 451, 452
—— beams, 442, 453, 455
—— deflection of, 462
—— inverted, 458
—— bins, 467, 468
—— bridges, arches of, 471
—— cost of, 2846
—— calculation methods, 441
—— columns, 463, 465
—— conical coverings, 469
—— dams, 1292
—— designing, 453
—— domes, 469
—— expanded metal for, 334
—— factory, 478
—— failures, causes of, 477
—— fire resistance, 439
—— resisting floors, 452
—— floor slabs, 459

RES

Reinforced concrete, French Government regulations, 467
—— girders, 453
—— L.C.C. regulations, 443
—— lock-woven mesh, 453
—— materials, 439
—— 'Perfector' system, 452
—— piles, 463, 465, 498
—— loads on, 501
—— pillars, 442, 443
—— pipes, 467
—— Report of Royal Institute, of British Architects on, 439
—— reservoirs, 467
—— roads, 735
—— rust prevention in, 393
—— shear reinforcement, 442
—— silos, 467, 468
—— slabs, 442
—— storage bins, 468
—— stresses in beams, 441
—— structures, designs, 453
—— systems, 451
—— table of sectional area of rods, 477
—— walls, 463, 465
—— work, specification for, 474
—— working stresses, 441
—— sheet lead, 174
Reinforcing rods, areas of, 477
Reisert's feed-water softener, 1632
Renewal of waterworks, 1418
Renold process for preventing corrosion, 2712
Repairing worn stone steps, 343
Repeated telephone cable circuits, 2338
Repose, angles of, of soils, 494
Rescue apparatus for mines, 2617
Re-seating and re-facing valves, 1670
Reservoirs, 1415
—— circular reinforced concrete, 467
—— evaporation and absorption from, 1387
—— general arrangement of, 1415
—— impounding, cost of, 2891
—— lining of, 1412
—— pollution by birds, 1418
—— reinforced concrete, 467
—— silting of, 1396
—— storage, 1415, 1510
Re-sharpening files, 1088
Resilient steel gears, Seymour's patent, 921
Resistance, fire, standards of, 2547
—— reinforced concrete, 444
—— frictional, pneumatic foundations, 508
—— of air to train, 588
—— curves to train, 585
—— of gradients on railways, 585
—— locomotives, 586
—— ships, 2363
—— wood to reversal of stress, 179
—— to air of plane surface, 1183
—— compression, strength of materials, 169
—— torsional rupture, 179
—— traction, 583
—— on a level railroad, 583
—— pavements, 726
—— roads, 726
—— tramways, 689
—— welding, 2789
Resistivity, electrical, of metals, 2204

RES

'Resolutor' coal pulveriser, 2133, 2137
Retaining-walls, 514
——— brick, 517 [on, 516
——— effect of weight of building
——— equilibrium of, 515
——— foundations of, 516
——— pressure of earth on, 514
——— pressure of water on, 514
——— rules for, 517
——— table of contents in cubic yards, 520
——— Trautwine's rule for, 518
——— wharf, 520
Retort house governors, 2464
——— of gasworks, 2453
Retorts, gas, 434, 2453, 2462
——— specification for, 434
Reverberatory furnaces, 2665
Reversing drive, automatic, for planing machines, 1021
——— levels, Zeiss, 70
Revolution indicators, 1450
Rib bar for reinforced concrete, 451, 452
'Ribmet,' 335
Ricardo motor-car piston, 1956
Richardson's Westgarth oil engine, Diesel, 2428
Ridge board of roofs, 318
Ridged roof truss multipliers, 233
Rigby patent hammers, 1135
Rims of brakes, cooling, 37
Ring lubrication, 984
——— spinning mills, 787
Ringelmann's brake, 36
——— smoke charts, 2111, 2112
Rings, iron, strength of, 194
——— of pistons, packing metal for, 2740
——— Rowan's, 1756
River banks, erosion of, 1508
——— canalisation, 1511
——— dredging, 1510 [1510
——— regulating works in United States,
——— regulation, cost of, 1510
Rivers, cost of canalisation of, 1513
——— critical velocity in, 1506, 1507
——— flow and discharge of, 1288, 1506, 1507
——— navigation of, 1505
——— to measure base line across, 53
——— width of, 54
——— raise the height of, by a weir, 1307
——— velocity of flow in, 1506
Rivet bars, 1563
Riveted girder-work, 226
——— joints, 226, 228
——— of boilers, 1567
Riveters, hydraulic, 1355
Riveting, 226, 228, 275
——— and flanging, pressures required for, 1562
——— boilers, 1567
——— bridge, 275
——— pneumatic, 1086
Rivets, 186, 226, 275, 617, 1176, 1563
——— and plates, strength of, 226
——— copper, weight of, 163
——— iron, 1563, 1564
——— weight of, 163
——— steel, 186, 617, 1563, 1564
——— strength of, 228, 1567
Road administration, 711
——— bearers of bridges, 281

ROA

Road camber, 728
——— carpet, waterproof, 730
——— construction, 726
——— costs data, 759
——— crusts, 730, 731, 750
——— drainage, 728
——— engineers, legal points for, 714
——— fund, 721
——— highway authorities, 711
——— statistics, 716
——— machinery, 755
——— maintenance costs, 720, 2875
——— materials, tests of, 750
——— , haulage of, 756
——— metal, 736
——— mileage in relation to population and area, 716
——— Ministry of Transport, 711, 714, 762
——— National Physical Laboratory tests of road stones, 749
——— planning and design, 762B
——— regulations, 722
——— resistance to traction, 725
——— rollers, 756
——— specifications, 741
——— statistics, 716
——— surface tarring, 733
——— surface tests, French, 724
——— tar macadam, 732
——— tars, British standard specification for, 762A
——— specifications and analyses, points from, 762A
——— Traffic Act, 1930, 762
——— statistics, 761
——— vehicles, maximum dimensions of, 762A
——— weights, 762A
——— provisions as to tyres, 762
——— waterproof carpet, 730
——— wearing layer, 730
——— wood-paved, 743
Roads, arterial and by-pass, 718
——— bituminous carpeting, 737
——— macadam, 737
——— brick, 739
——— comparative statistics, 716
——— concrete, 735
——— cost of, 717, 2875
——— country, 726
——— district delegation, 762
——— effect of drought on, 729
——— heavy traffic on, 748
——— Improvement Act, 715
——— London, paving, 743
——— main, 716
——— maintenance cost of, 720, 2875
——— radius at bends of, 762B
——— reinforced concrete, 735
——— resistance to traction on, 726
——— road-metal rocks, 736
——— silicated, 734
——— slipperiness of, 734
——— steam rollers on, 756
——— stones, tests of, 749
——— surface tarring of, 733
——— superelevation at bends in, 762B
——— tree planting on, 730, 758
——— visibility at summits, 762C
——— width of, 727
——— important, 717

Robey boilers, 1576
——— drop valve gear, 1726
——— high speed engines, 1728
——— oil engines, 1912
Robinson anemometer, 1184
——— 'Jubilee' worm drive hose clips, 1085
——— —— locomotive superheater, 601
——— —— marine superheater, 2403
Rock blasting, 1487
——— boring, 2563
——— breakers, sub-aqueous, 1488
——— bulk of, 484
——— drilling and blasting, 1487
——— drills, 2565
——— —— Hardy - Simplex air hammer, 2568
——— fill dams, 1291
——— sub-aqueous, removal of, 1488
Rocks used for road metal, 736
Rods, copper, for locomotives, 615
——— round and square for reinforced concrete, 477
'Rok' roofing, 316, 326, 327
Rolled copper, lead, and brass, sheet and bars, weight of, 137
——— girders, 238
——— iron, flat, weight of, 135, 136
——— iron joists, 329
——— steel, 238
——— steel, flat, weight of, 137
Roller bearings, 970
——— ——— lubrication of, 975
——— tube cutter, 1089
Rollers, steam, 724, 756
Rolling loads, on railway bridges, 249
——— mill engine, Galloway, 1731
——— —— gears, 910
——— period of ships, 2357
——— stock of railways, 627, 661
Rolls, gold mining, 2638
Rolls Royce aero engines, 2045
Roman cement, 397
Roofing, corrugated-iron, 231, 234, 365
——— expanded steel concrete, 338, 339
——— galvanised, paint and varnish for, 234
——— 'Italit,' 327
——— materials, 322
——— heat-insulating value of, 322
——— 'Rok,' 316, 326
——— 'Ruberoid,' 316, 325
——— self-supporting, 232
——— sheets, 'Poilite,' 323
——— tiles, 'Poilite,' 414
——— tiles, 'Trafford,' 413
———, zinc, 420
Roofs, 316, 365
——— angles of, 316, 365
——— collar beam, 317
——— components of, 316
——— corrugated-iron for, 234, 365
——— cost of, 2882
——— coverings of, 316
——— dimensions of parts of, 866
——— furnace, 'Liptak,' 1608
——— joints of wooden, 320
——— king-post, 318
——— load on, 365

Roofs, paint for, 2805
——— pole-plate of, 320
——— queen-post, 319
——— ridge-board of, 318
——— ridged, truss multipliers, 233
——— scantlings for, 318
——— self-supporting, 232
——— single-span or couple, 317
——— steel, framing and covering, 239
——— tie beam of, 320
——— wall-plate of, 318
——— weight of components of, 316
——— weight of snow on, 322
——— wind pressure on, 322
——— wooden, parts of, 366
——— workshop, 231, 765
Rooms, battery, 2259
Rope brake, Hirn's, 34
——— drives, adaptability of, 872
——— ——— design of, 873
——— driving, 788
——— ——— drums, 870
——— ——— efficiency of, 876
——— ——— guards for, 876
——— ——— pulleys for, 870
——— ——— speed, 875
——— ——— systems of, 872
——— haulage, 885, 2606
——— sockets, 887
Ropes and cables, lengths, 1
——— cotton, table of, 874
——— crane, 816, 828
——— driving, 788, 872
——— ——— guard for, 876
——— ——— power of, 873
——— ——— sizes of, 873
——— ——— speed of, 872, 875
——— flat, 2611
——— flattened strand, Wright's, 2607
——— flexible steel wire, 828
——— haulage, 178, 836, 885, 2607
——— hemp, strength of, 199
——— ——— preservation of, 200
——— iron, corrosion of, 200
——— ——— weight of, 2611
——— Lang-lay wire, 885
——— locked coil, 836, 2603
——— manila, 200
——— mining, 178, 828, 836, 2608
——— round, 2610
——— rust-resisting steel, 2610
——— splicing, 879, 887
——— steel wire, 178, 828, 2610
——— ——— Wright's, 2607, 2608, 2609
——— strength of, 176, 178, 200, 2608
——— testing, 200
——— transmissible power of, 873
——— weight of, 166, 2608, 2609
——— winding, 2606,
——— wire, 166, 176, 178, 826, 885
——— ——— breaking stresses, 176
——— ——— durability of, 826
——— ——— haulage, 178, 836, 885, 2607
——— ——— preservation of, 200
——— ——— lubrication of, 889
——— working strength of, 2610
——— Wright's, 2607, 2608, 2609
Ropeways, aerial, 834, 2839
Ross, R. G. & Son, Ltd., Rigby patent hammers, 1135

ROS

'Rosseline,' 1887
Rotary air-pump, Leblanc, 1844
——— blowers, 2520
——— converters, 2270
——— pump, semi-, Willcox, 1252
——— pumps, 1275
Rotating body, energy of, 34
Rotoscope. The Ashdown, 957
Round mining ropes, 2610
——— steel, weight of, 137, 2610
——— timber, solidity of, 364
——— wire ropes, 178, 2610
Rowan's pistons, 1756
——— slow action packing, 1757
Row's (Royles) feed-water heater, 1641
Royal Commission on sewage, 1378
——— Edward Dock, Avonmouth, 1462
——— Institute of British Architects Report on reinforced concrete, 439
——— ——— ——— ——— ——— Report on strength of bricks, 175
Royles Ltd. feed-water heater, 1641
——— water softener and purifier, 1631
Rubber bearings, 1741
——— hard, 2814
——— machining, 1095
——— paving, 747
——— specification for vulcanised insulated cable, 2243
——— tensile strength of, 170
——— tyres, 726
Rubble masonry, 379
'Ruberoid' roofing, 316, 325
Ruble elevator, mining, 2645
Rudders, 2366
Rule for leads of railway crossings, 562
Runabout cranes, 809
Runs, barrow and cart, 494
Runways, overhead, 808
Rupture, torsional, of woods by shock, 179
Russian weights and measures, 17
Rust cleaning, 2811
——— prevention in reinforced concrete, 393
——— preventive, 2803
——— proofing, iron, 2811
Rustless iron, 2716
——— steel, 2713
——— turbine blading, 1805, 1806
Ruston crankshaft alignment indicator, 1739
Ruston-Hornsby gas engines, 1879
——— ——— oil engines, 1918
——— ——— suction gas plant, 2187
Ruston (Ruston-Hornsby) excavator, 483, 2862
——— wood refuse gas producer, 2187
Ruth's steam accumulator, 1854
Rylands' table of iron wire, 176
——— ——— steel wire, 177
——— ——— wire ropes, 176

SCL

Safe load on columns, 497
——— ——— floors, 497
——— ——— I-beams, 236
——— ——— materials, 496
——— ——— ordinary foundations, 496
——— ——— piles, 500
——— ——— ——— reinforced concrete, 501
——— ——— stanchions, 224
——— ——— stone walls, 497
——— pressures for lubricants, 962
——— ——— on bearings, 961
——— working load, chains, 193
——— ——— ——— ropes, 200
Safety lamps, 2616
——— oil fuel burning, 2167
——— valves, 1570, 1659
——— ——— areas of, 1659
——— ——— Board of Trade rules, 1659
——— ——— Bureau Veritas rules, 1660
——— ——— Lloyd's rules, 1660
——— ——— springs of, Board of Trade rule, 1660
Sag of conductors, 2220
Sags of aluminium stranded cables, 2220
St. Helens Cable & Rubber Co.'s cab-tyre sheathed cable, 2244
——— ——— ——— ——— wiring system, 2254
Sale and purchase of machinery, 2912
Salinometer, Crockatt, 1637
Salt-glazed pipes, 430, 1360
Salt solutions, boiling-points of, 111
Salts, melting points of, 2676
Salvage work, pneumatic tools for, 1081
Sand, blasting, 1123
——— flow of, 2817
——— mortar, test for impurities in, 384
——— moulding, 2752
——— pump dredgers, 1483
Sandberg rails, 540, 688
Sandblast process of recutting files, 1088
Sanders' formula for safe load on piles, 502
Sand-lime bricks, 426
Sandy Bay breakwater, 1457
Sandycroft Hunt motor, 2293
Sanitary engineering, 1359
——— ware, 429
Sankey, J. H. & Co.'s Fosalsil, 429, 435, 1222
Saturated salt solutions, boiling points of, 111
——— ——— steam, properties of, 1532, 1833
——— ——— vapour in air, 1866
Saving worn parts of machinery, 1095
Sawing, cold steel, 1057
——— ——— machinery, 1058, 1059
——— ——— metal, hand, 1058
Saws, band, 1059
——— fret, 1061
——— hack, 1061
Scaffolding, 511
——— ——— knots, 512
Scale in boilers, 1622, 1637, 1704A
Scales for surveys, 56
——— thermometer, 99
Scandinavian railway electrification, 674
Scantlings for roofs, 318
Scarfing beams, 312
Schmidt superheater, 602
Schmott's table of frictional resistance of pneumatic foundations, 508
Schneider Diesel engines, 1899
Scleroscope, Shore's, 1098

SCO

Scoop wheels, 1276
Scotch pig iron, 2681
Scott and Mountain tables of horse-power for haulage, 2601
Scott Dunbar auxiliary top and side telescope, 63
Scott Russell parallel motion, 954
Scott's marine oil engine, Diesel, 2424, 2425
Scraper conveyors, 845
Scrapers and brushes, boiler tube, 1684
Scraping water mains, cost of, 2892
Screw, force by, 27
——— heads, standard, 1150
——— piles, 503
——— steamers, shafts of, 1005
——— taps and tapping, 1171
——— threads, American, 1139, 1145
——— British Association, 1139, 1143, 1144, 1145
——— British standard Whitworth, 1140
——— Cycle Engineers' Institute standard, 1149
——— fine, British standard, 1141, 1143, 1156
——— forms of, 1139
——— French standard, 1145
——— International or metrical, 1146, 1147
——— on water and gas pipes, 1140
——— Sellers, 1139, 1145
——— standard, 1139
——— Steinlen, 1139
——— Swiss or Thury, 1139, 1144
——— Whitworth, 1139, 1140, 1144
Screwing and tapping machines, power for, 1054
Screws, 1139
——— prevent rusting in, 2813
——— small, heads for, 1140
——— steel, for rail chairs, specification for, 538
——— threads of, 1139
——— weight of, 163
——— wood, 1148
Scribe marks, 357
Scrubbers, gas, 2454
Scullery sink, 1379
S.E.A. packing rings, 1264, 1756
Sea, distance of objects at, 76
——— level, to reduce a measured base to, 58
——— temperature of the, 106
——— water, effect of, on concrete, 392
Seal rings for turbines, 1331
Seam inspection blocks and covers, asbestos cushioned, 1606
Seamless brass tubes, weight of, 158
——— copper tubes, weight of, 156
——— soft copper jointings, Hulburd's, 608
Seasoning timber, 130, 357, 359
——— loss of weight by, 358
——— shrinkage by, 358
——— time required for, 358
——— woods, 130
——— increase of strength by, 184
Seating block, asbestos cushioned, Pearson's, 1606
Secret commissions, 2905
——— joints in flooring boards, 332

SHA

Secrets, disclosure of, 2905
Sectional dock, pontoon, 1477
Sectioning, alidade, Zeiss, 72
——— electric tramways, trolleys, 692
Sections, British standard, 238
——— railway, House of Commons rules for, 665
Seewer Pelton wheel regulator, 1313
Seger cones, 2674
Sellers' bolts and nuts, 1153
——— screw threads, 1139, 1145
Semi-bauxite bricks, 430
Semi-Diesel engines (*see* Oil engines)
Semi-rotary pump, Willcox, 1252
Semi-steel, 2687
Sentinel pyrometers, 2675
Separators, magnetic, 350, 2626
——— oil, 1681
——— steam and water, 1699
'Serpent' packing, Beldam's, 1754
'Serval' asbestos felt, 325
Service reservoirs, 1415
Setting, boiler, 1604
——— out arches, 523
——— stones, 377
Settings, boiler, leaky, 1605
Sewage, bacterial treatment of, 1372
——— disposal, Royal Commission on, standard of purity for sewage effluents, 1378
——— hourly variation of flow of, 1368
——— land treatment of, 1372
——— Ministry of Health's requirements, 1373
——— pipes, British standard, 1361
——— sludge, 1369
——— activated process, 1376
——— treatment of, 1367
——— works, cost of, 2883
Sewerage systems, 1365
Sewers, 1359
——— American, 1360
——— brickwork for, 1408
——— circular, 1360
——— discharging capacity of, 1359, 1364
——— egg-shaped, 1363
——— expanded steel concrete, 335
——— flushing, 1366
——— rate at which rainfall reaches, 1359
——— salt-glazed pipes for, 430, 1360
——— storm water, discharge of, 1404
——— ventilation of, 1367
Sextant, box, 67
Seymour's resilient steel gears, 921
Shaft furnaces, 2660
——— pillars, 2585
——— sinking, mining, 2575, 2578
Shafting, 1001
——— diameter of, 1001, 1004, 1005
——— horse-power of, 1001
——— iron, 1001
——— lubrication of, 985
——— speed of, 1001, 1003
——— steam turbine, 1005, 2374
——— steel, 1001
——— supports of, 1002
——— wrought-iron, 1001
Shafts, chimney, stability of, 521
——— crane, 815
——— crank, 1003

SHA

Shafts, hollow, 1002
——— mine, 2575
——— ——— deep, 2575
——— oil engine, 1904
——— snap rings, fitting, 1007
——— steamship, 1005
——— ——— ——— Board of Trade rules for, 1005
——— ——— Bureau Veritas rules for, 1005
——— ——— Lloyd's rules for, 1005
——— twisting angle of, 1005
——— ——— moment of, 1004
Shale oil, 2150
Shakespear electrical CO_2 recorder, 2087
Shaping and slotting machines, power for, 1055
Sharpening angles of wood machine cutters, 1072
——— files, 1088
——— steel mining bits, 2565
Shaw, Hele-, friction clutch, 890, 895
Shear reinforcement, 442
Shearing machines, power for, 1056
——— stresses, 212, 289, 458
Sheet aluminium, weight of, 140
——— brass, weight of, 137
——— cast iron, 133
——— copper, weight of, 132, 137
——— iron, galvanised corrugated, 231
——— ——— weight of, 133
——— lead, weight of, 137, 138
——— ——— reinforced, 174
——— metals, specification of, 138
——— ——— weights of, 132, 133, 137
——— steel piling, 504
——— zinc, trade gauge, 11
——— ——— weight of, 138
Sheeting for reservoirs, 1412
Sheets and hoops, Birmingham gauge for, 13
——— 'Fibrent,' 324
——— 'Poilite,' 323
Shell or convex surface, centre of gravity of,
Shells, boiler, 1560, 1566 [24
——— ——— strength of, 1561
Sherardising iron, 2711
Shingles, 364
Shipbuilding cranes, 811
——— ——— steel, British standard specification for, 189
Ship canals, 1526
——— construction, Lloyd's rules for the application of electric arc welding to, 2445
——— refrigeration, 1224
——— steel tests, 186, 189, 191
Ships, Admiralty coefficients, 2365
——— boilers, 2391
——— bottoms, painting of, 2444
——— cargo, fuel economy, 2368
——— centre of buoyancy, 2355
——— ——— flotation, 2356
——— coefficients of displacement, 2354
——— concrete, 2444
——— deadweight, 2363
——— dimensions, 2353
——— displacement curves of, 2354, 2355
——— draft, 2353
——— economical speed of, 2366

SIL

Ships, electrical equipment of, Institution of Electrical Engineers regulations, 2240
——— engine and boiler design of, 2372
——— ——— cylinder ratios, 2372
——— estimate sheets of, 2367
——— fuel consumption of, 2366
——— horse-power of, 2363
——— inclining experiment, 2357
——— liners, notable, 2371
——— Lloyd's rules for ferro-concrete vessels, 2445
——— longitudinal metacentre of, 2358
——— machinery, weight of, 2362
——— moulded dimensions of, 2353
——— overall dimensions of, 2353
——— propulsion of, 2441
——— recipro-turbine machinery, 2374
——— refrigerating machinery for, 1224
——— registered dimensions of, 2353
——— resistance of, 2363
——— rolling period of, 2357
——— rudders, 2366
——— skin effective horse-power of, 2364
——— stability of, 2357
——— ——— and trim indicator, Ralston's, [2359
——— standard cargo, 2369
——— steam pressures, 2372
——— steam turbines, 1767, 2375
——— ——— ——— arrangement of, 2376
——— steel for building, 186, 189
——— transverse metacentre of, 2356
——— wave effective horse-power, 2364
——— weights of, 2361
——— weights of machinery of, 2362
——— wetted surface of, 2356
Shore scleroscope, 1098
Shoring buildings, 513
Shovels, excavating, 483, 2861
Shouldered tenon, 313
Shrinkage of castings, 2755
——— ——— materials, 484
——— ——— timber, 358
Siam canal dredging, 1487
Siamese weights and measures, 19
Side-groove steel piling (Annison's), 505
Side slopes, form of, 492
Siemens' dry battery cells, 2255
——— ebonite, 144
——— electric lamps, 2324
——— inert battery cells, 2255
Sight glasses, Hulburd's cartridge, 1663
Signalling, electric, in coal mines, 2619
——— ——— wire, 178
Signals, railway, 645
——— sea, audible, 1500
——— ——— distance audibility of, 1501
——— ——— explosive, 1500
——— submarine sound, 1500
Silencers, Clarkson Thimble Tube boiler, 2198
Silica bricks, 431
——— ——— specification for, 432
Silicate cotton or slag wool, 342, 1222, 1696
Silicated roads, 734
Silico-manganese steels, 2696
Silos, reinforced concrete, 467, 468
Silting of reservoirs, 1396
——— ——— rivers and canals, 1396
Silver and gold tables, 2642

SIL

Silver solder, 2749
——— tensile strength of, 170
——— wire, gauge for, 11
Silvering mirrors, 2815
Silvertown electricity-supply cables, 2211
Simon-Carves boiler, 1592
Simple and compound locomotives, cylinders of equivalent power, 592
——— pendulum, 30
Simplex patent boiler tube brushes and scrapers, 1684
——— piling system, 498
——— semi-Diesel engines, Mirrlees, 1914
——— valve re-seater, 1671
Simson's rule for area computation, 53
Sines, versed, for rails, 556
Singapore floating dock, 1476
Single-acting force pump, 1250
——— altitude of the sun, to find variation of compass by, 73
——— balk dams, 511
——— phase train haulage in America, 670
——— joisted floor, 328
——— joists, Baltic yellow fir, 331
———-span roof, 317
Sink, scullery, 1379
Sinking cylinders, pneumatic process, 503
——— foundations by freezing process, 508
——— shaft, mining, 2575
——— wells, 1408
Sinks and water-closets, 1380
Sirens, 1501
Size of annealed and bright iron wire, 176
——— books and paper, 19
——— bricks, 366, 427, 432
——— clock dials, 2815
——— haulage wire ropes, 178, 836, 885, 2607
——— mining ropes, 2608
——— Paxman 'Economic' boilers, 1581, 1606
——— pulleys and drums for ropes, 880
——— slates, 411, 412
——— steam hammers, 1126, 1128
——— timber, 357
——— type, 19
Sizing, mining, 2622
Skefko ball bearings, 964, 965, 966, 970
——— roller bearings, 972
Skelton's table of broad flange beams, 237
——— safe loads on bridge struts, 243, 244
Skin effective horse-power of ships, 2364
——— friction on bridge foundations, 258
Slabs, floor, reinforced concrete, 459
Slag concrete, 388
——— wool insulation, 342, 1222, 1696
Slates, 'Fibrent' asbestos-cement, 413
——— 'Poilite,' 412
——— sizes of, 411, 412
——— 'Turnail' asbestos-cement, 412
——— underlining, 413
——— weight of, 412
Slating, 411
——— nails, 411
Sleepers, steel, 540
——— wooden, 538
——— ——— length of, 539
——— ——— creosoting, 539
——— ——— plugging, 538
Slide valve gear, 1758
Slides, felt wipers for, 1053

SOU

Slings, strength of, 196
Slipperiness of roads, 734
Slocum's heat unit table, 108
Slopes, angles and lengths of, 493
——— chaining on, 55
——— of excavations and embankments, 492
——— side form of, 492
Sloping ground, chaining over, 55
Slotting and shaping machines, power for, 1055
Sludge, activated, process, 1376
——— sewage, 1369
Sluice gates, 1343
——— measurement, 1290
Sluices, mining, 2643
——— ——— grade and duty of, 2643
Smith, Angus, composition, 2808
Smith, Major & Stevens, electric lifts, 832
Smith's fires, centrifugal fans for, 2663
Smoke and coal combustion, 2110
Smoke and dust precipitation, electrical, 2333
——— boxes, locomotive, 601
——— charts, 2111
——— colour of, 2112
Snercold petrol air gas plants, 2488
Snifting valves, 1201
Snow, weight of, on roofs, 322
Socket, Blackett's wire-rope, 889
——— joints, 'Flexstel' system of, 1084
——— ——— standard Metropolitan Water Board, 1429
——— ——— British Standard for cables, 2243
——— pipes and fittings, 155, 1429, 1430
——— wire rope, 887
Softening water, 1422, 1628
Soft woods, 348
Soil, electric heating of, 2336
——— evaporation, 1388
——— pipes, 1381
Soils, angles of repose of, 494
——— foundations on different, 496
——— weight of cubic yard of, 494
Solder, 416, 2747
——— aluminium, 2751
——— ——— bronze, 2752
——— cold, 1751
——— glazier's, 416
——— sticking to work, to prevent, 494
Soldered joints, strength of, 176
Soldering, 2747
——— cast iron, 2752
——— sockets, electrical, 2243
Solid figures, centre of gravity of, 24
——— fuels, 2090
Solidity of round and square timber, 365
Solids, specific gravity of, 126
——— velocity of sound through, 118
Solutions, boiling-points of, 111
——— glutinous, standard evaporators for, 796
Soot removers, boiler tube, 1683
Sorting out mixed steel bars, 166
Sound, 117
——— distance of transmission of, 118
——— proofing paper, 342
——— signals, submarine, 1500
——— velocity of, through air, 117
——— ——— solids, 118
——— ——— water, 117

SOU

Soundings, surveying, 58
—— to reduce, to low water, 59
Southern Railway carriages, 628
—— locomotives, 574, 576
—— wagons, 630
South, to determine, by watch, 74
Southampton floating dock, 1475
Space, floor, for machine tools, 772
—— stowage of coal, 2095
Span, length of, overhead wires, 2220
—— wire for electric tramways, 697
Spanners, 1164
—— box, 1164
—— British standard, 1164
Speaking-tubes, 118
Specific gravities and weights, 125
—— table of, 126
—— of earths, 127
—— ebonite, 130
—— elastic fluids, 132
—— gases, 131
—— liquids, 126, 132
—— metals, 132
—— solids, 126, 132
—— stones, 127
—— water, 124, 128
—— woods, 128, 129
—— gravity, 125
—— of accumulator acid, 2257
—— coal, 2560
—— heat, 106
—— of gases, 107
—— steam, 1534
Specification, Admiralty, oil fuel, 2166
—— British Standard (see British Standard)
—— bridgework, 265
—— building bricks, 427
—— coal tar fuel for Diesel engines, 2160
—— creosote oil, 363
—— electric generators, 2261, 2263
—— lifts, 831 [2270
—— rotary converter, G.E.C.,
—— fireclay bricks, 432
—— gas retorts, 434
—— locomotive construction, 611
—— nickel steel, 2694
—— plates for locomotive boilers, 617
—— refractory mortars & cement, 432
—— reinforced concrete work, 474
—— roads, 741
—— sheet metals, 138
—— silica bricks, 432
—— steel, 2693
—— bolts, screws for rail chairs, spikes, 538
—— vulcanised rubber electricity supply cables, 2243
Specifications of mortars, 388
Spectrum, decomposition of light in, 119
Speed, critical, turbine (steam), 1802
—— limits of motor vehicles, 724
—— gear, variable, 948
—— reduction gear, turbine, 930, 1818, 2381
—— regulation, direct current motors, 2281
—— of centrifugal pumps, 1267
—— cotton machinery pulleys, 789
—— cutting, carbon steel lathe tools,
—— drills, 1033 [1073

STA

Speed of emery wheels, 1108
—— grinding wheels, 1105, 1108
—— grindstones, 1104
—— lathes, 1019
—— marine engine pistons, 2373
—— milling cutters, 1026
—— pinion, to complete the, 913
—— planing machines, 1023
—— shafting, 1001
—— ships, economical, 2366
—— water through pipes, 1242, 1424
—— wood-working machinery, 1058
'Speedicut' twist drills, 1039
Spelter solder, 2752
Spence's metal, 2743
Spencer, John, Ltd., wrought iron tubes, 148, 149
Spencer-Bonecourt Kirke gas-fired boilers, 2191
—— waste heat boilers,
—— Hopwood boiler, 1600 [2194
Spherical excess, 58
Spigot joints, cast-iron pipes for waterworks,
Spike-gripping power of wood, 184 [1429
Spikes, railway, 538
Spillway dams, 1295
Spillways, automatic syphon, 1398
Spinning mills, 788
——, ring, 788
Spiral or helical springs, 614, 1009
—— points of railway curves, 568
Spirit level glass, to set, 2817
Splicing driving ropes, 879
—— wire ropes, 887
Spontaneous combustion in mines, 2618
—— ignition, temperatures of, 101
Spray painting, 417
—— ponds, 1851
Spraying, metal, 1096
Springs, 1009
—— bearing, 613, 632
—— buffer, 633
—— motor car, 1963, 1970
—— clock type, 1013
—— compression, 1015
—— conical round, 1014
—— heat treatment of, 1015
—— helical, 614, 1009
—— laminated, 620, 1970
—— manufacture of, 1015
—— locomotive, 613
—— railway carriages and waggons, 614, 632
—— safety valves, Board of Trade rules for, 1661
—— phosphor bronze, 1012
—— spiral or helical, 614, 1009
—— steels for, 2698
—— volute flat, 1014
—— volute and helical, for locomotives,
Sprinkler stoker, Bennis, 2113 [613, 614
Spur gearing, 911
Square steel, weight of, 137
—— timber, solidity of, 365
Stability and trim indicator, ship, 2359
—— of chimney-shafts, 521
—— dams, 1294
—— ships, 2357
Staffordshire blue bricks, 426
—— cold-blast iron, 2683
—— semi-cold-blast iron, 2683

STA

Stainless iron, 2716
——— steel, 2713
——— ——— welding of, 2713
Stair-treads Mason's, 343
Stairs, 342
Stamps, drop, 1132, 1135
——— duty of, mining, 2639
——— gold milling, 2639
——— mining, 2639
Stanchions, mild steel, 224, 225
Standard aluminium cables, 2219
——— American screw threads, 1139, 1145, 1146
——— ——— bends for steam pipes, 1685
——— boiler construction, 1560
——— ——— candles, 2525
——— ——— conditions for design of marine boilers, 1543
——— copper conductors, 2221
——— creosote, 363
——— cycle screw threads, 1149
——— driving chains, 899
——— electrical machinery, 2260
——— electricity supply meters, 2315
——— foreign horse-powers, 1708
——— French screw threads, 1148
——— hot-water pipes and fittings, 2497
——— limes, 385
——— load for highway bridges, 252
——— marine boilers, 1543
——— oils for lubrication, 931
——— pipe flanges, American, 1163
——— ——— sockets, 1429, 1430
——— ——— threads, 1150, 1151, 1152
——— ——— Portland cement, 403, 405
——— purity of water, 1418
——— railway electrification, 669
——— ——— ——— rails, 529, 533
——— ——— materials, 611
——— screw gauge, 1139
——— ——— heads, 1150
——— ——— taps, 1170
——— sections, 238
——— signalling wire, 178
——— sizes of conductors for electric supply mains, 2205, 2206
——— steel conduits, 2245
——— trolley wheels, 693
——— tubular poles for electric tramways, 695
——— tungsten electric lamps, 2322
——— ——— weights of cast iron water pipes, 150
——— wire gauge, 11, 12
——— Telephones & Cables, Ltd., loading coils, 2338
——— ——— ——— ——— ——— ——— telephone repeaters, 2339
Standards of deals, 356
——— ——— fire resistance, 2547
——— ——— illumination, 2532
——— ——— light, 2525
——— ——— temperature, 194
———, mining and metallurgy, 2641
Standing Orders of House of Commons for Railway Bills, 664
Stanley's cranked gimbal dial, 66
——— drawing pen, 91
——— gradiometer, 67
——— ——— surveying instruments, 59

STE

Stanton concrete-lined pipes, 1247
Star, Pole, to find, 76
Stars, identification of, 76
Starters for gas-engines, 1886
Starting torque of two-crank engines, 1747
Static balancers, 2304
Statics, 23
Statistics, highway, 716
———, railway, 663
Status, legal, of engineers, 2903
Stay bars of boilers, 1563
Steam, 1531
——— accumulators, 1853
——— and water separators, 1699
——— atomiser, Guyot's, 2162
——— ——— Holden's, 2162
——— boiler plant, efficiency of, 1555
——— boilers, size of chimneys for, 1611
———, buildings warmed by, 2494
——— condensed, measurement of, 1849
——— condensers, 1825
——— consumption, 1708, 1712, 1779, 1808
——— ——— ——— of steam-hammers, 1226
——— ——— ——— winding engines, 1714
——— ——— ———, theoretical, 1708
——— ——— ———, typical, 1712
——— diagram factor, 2373
——— dryers and water separators, 1699
——— engines, back pressure and extraction, 1731
——— ——— Belliss and Morcom high-speed, 1731
——— ——— Davey Paxman portable, 1733
——— ——— ——— ——— positive valve gear, 1724
——— ——— design of, 1737
——— ——— duty of, 1713
——— ——— efficiency ratio of, 1709, 1710
——— ——— Galloway drop piston valve, 1730
——— ——— ——— extraction, 1735
——— ——— ——— uniflow, 1723
——— ——— governor of, 32, 1748
——— ——— horse-power of, 1707
——— ——— indicator of, 1715
——— ——— mechanical efficiency of, 1707
——— ——— portable, 1733
——— ——— Robey drop-valve, 1726
——— ——— high-speed, 1728
——— ——— types of, 1722
——— ——— uniflow, 1722
——— evaporation of, 1582
——— excavating, cost of, 2861
———, exhaust, heating by, 2499
——— ——— pipes, 1689
——— expansion of, 1705
——— Fittings Co. (*see* Drayton)
———, flow of, through orifices, 1538
——— ——— pipes, 1691, 1693
——— ——— in turbines, 1767
——— hammers, 1125
——— ——— formula for, 1128
——— ——— pipe bends, for, 1687
——— heating of railway carriages, 633
——— ——— buildings, 2491
——— high pressure, 1535, 1710
——— jackets, 1711
——— lorries, 759
——— mains for heating, area and diameter of, 2493

Steam meters, 1446
——— nozzles, 1770
——— passages, area of, 1748
——— pipes, 1684
——— ——— bends for, 152, 1685
——— ——— copper, 1687
——— ——— coverings for, 1695, 1697
——— ——— exhaust, 1689
——— ——— flow of steam in, 1691
——— ——— joints of, 1689
——— ——— leaks in, 1690
——— ——— Lloyd's rules for, 1687
——— ——— loss of heat from, 1693
——— ——— pressure drop in, 1691
——— ——— size of, 1690
——— ——— steel bends, 1685
——— ——— ——— corrugated, 1686
——— ——— ——— thickness of, 1687
——— ——— weight of, 145, 150
——— power plant thermal efficiencies, 1539
———, pressure and temperature of, 1532
——— ——— in cylinders of compound and triple-expansion engines, 2373
——— ——— regulator, Galloway, 1735
——— pressures corresponding to temperatures, 1532
——— properties of saturated, 1532, 1833
——— pumps, 1254
——— ——— Clarke, Chapman, 1254
——— ——— Pulsometer, 1261
——— ——— Warner-Anthony, 1256
——— ——— Weir, 1256
——— radiators, 2503
——— railway motor cars, 626
——— regulators, 601
——— rollers, 724, 756
———, saturated, properties of, 1532, 1833
——— saving by superheating, 1710
——— separators, 1699
———, specific heat of, 1534
——— superheated, 1534, 1711
——— ——— effect of, on steel, 171
——— ——— heat of, 1535
——— superheaters, 602, 1612, 2403
——— tables, 1531
———, temperature and pressure of, for vacuum, in inches, 1833
———, temperatures at high vacua, 1534
——— traps, 1699
——— ——— Bradford, 1699
——— ——— Drayton balanced, 1702
——— ——— ——— high pressure, 1700
——— ——— practice, notes on, 1702
——— turbine forgings, 1804
——— ——— shafting, 1005, 2374
——— turbines, 1767
——— vacua, high temperatures of, 1534
——— valves, 1668
——— ———, area of, 1748
——— *v.* oil engines, 2439
——— vehicles, working costs of, 756, 759
——— wagons, working costs of, 756, 759
———, work done by, 1708
Steamers, cargo, 2369
Steamship boilers, various types of, 2391
——— ——— shafts, Board of Trade rules for, 1005
——— ——— Bureau Véritas rules for, 1005
——— ——— Lloyd's rules for, 1005

Steamships, arrangement of turbines in, 2376
Steel, 2691
——— Admiralty, tests for, 186
——— analysis of various, 2699
——— and carbon alloy, 2693
——— angle and tee, weight of, 142
——— at high temperatures, properties of, 1823
——— bars, estimating cost of, 780
——— ——— sorting out mixed, 166
——— ——— tests for, 188
——— beams, Skelton's table of, 235
——— belt conveyors, 813
——— belting, 869
——— bends for steam pipes, 1685
——— bits, mining, sharpening, 2565
——— blueing, 2812
———, boiler, 1561
——— ——— plates, 1568
——— ——— stays, 1568, 1569
——— bridgework, 247
———, British standard for bridges and buildings, 185
——— ——— shipbuilding, 189
——— carbon, 1062, 2691
——— castings for locomotives, 615
———, chrome nickel, 2695
———, chromium plating of, 2704
——— conduits, 2245
——— ——— dimensions of, 2246
——— cored aluminium cables, 2219
———, corrosion of, 174
——— drilling hard, speeds for, 2815
——— ———, speeds for, 1035
——— effect of superheated steam on, 171
——— expanded reinforcements for concrete, 334, 335
———, expansion of, at high temperatures, 2704
———, flat, weight of, 137
——— flexible wire ropes, 828
——— forgings, 1804
——— ——— ——— locomotive, 614
——— girders, rolled, 238
——— ——— safe loads for, 238
———, hardening and tempering, 1062, 1067, 1070, 2697, 2705
———, heat treatment of, 2693, 2697
——— helical springs, 614, 1009
———, hexagon, weight of, 135
——— high speed, scrap, to identify, 1063
——— ——— tool, 1062, 2692
——— ——— temperatures, effect of, on, 2704
——— Invar, 2703
——— manganese, 2696
——— mild, treatment for, 188
——— Molybdenum, 2695
——— mushet, 1063
——— nickel, 2694, 2702
——— ——— chrome, 2695
——— non-corrodible, 2713
——— octagon, weight of, 135
——— piling, 504
——— pillars, 375
——— pipes, design of, 1302
——— ——— ———, identification of, 167
——— ——— water, incrustation of, 1419
——— ——— for locomotive boilers, 616
——— plates for railway rolling stock, 618
——— rails, breaking load, 536
——— railway rails, Board of Trade report on, 535

STE

Steel, rapid-cutting, tool, 1062
—— rivets, 186, 617, 1563, 1564
——, rolled, 238
—— —— flat, weight of, 137
—— —— ——, limiting dimensions of, 238
—— roofs, 229
—— ropes, 178, 828, 2606
——, round, weight of, 137, 2610
—— rust-proof coating for, 2803
—— sections, rolled, British standard, 238
—— shafting, 1001
—— shafts, twisting angle of, 1005
—— sheet piling, 504
—— Silico Manganese, 2696
—— sleepers, 540
—— spring, 2698
——, square, weight of, 137
——, stainless, 2713
—— stanchions, 224, 225
—— steam pipe bends, 1685
—— straight nickel, 2694
——, superheated steam, effect of, on, 171
——, tee and angle, weight of, 142
—— tempered, colours of, 1070
—— tempering and hardening, 1066, 2697, 2705
—— tensile strength of, at high and low temperatures, 171
—— —— ——, at very high temperatures, 173
—— tests for, Admiralty, 186
—— tool, high-speed, 1063
—— —— —— ——, hardening, 1062
—— —— lathe, cutting speeds, 1072
—— tubes, strength of, 191
—— —— for locomotive boilers, 616
—— Vanadium, 2696
——, weight of, 137
—— welding, 2771
—— wire, weight of, 161
—— —— crane ropes, 178, 816, 828
—— —— ropes, 178
—— —— —— flexible, 828
—— —— —— table of, 178
—— —— ——, weight of, 166, 178
Steelwork encasing, expanded metal, 337
—— —— for buildings, 373
Steining, 1407
Steinlen screw thread, 1139
Stephenson's link motion valve gear, 1762
Steps, stone, repairing, 343
Steward hypsometric aneroid, 69, 70
—— —— surveying telemeter, 69
Stewarts & Lloyds steam and gas fittings, 150
—— —— —— —— screwed and socketed joints, 151
Stiffest rectangular beam from log, to cut, 357
Stiffness of beams and poles, 314
—— —— woods, 179
Stiff-plastic bricks, 426
Still oil engine, 1916
Stirling boilers, 1590
Stock bricks, 426
——, railway rolling, 627, 661
Stokers, mechanical, 622, 2112
—— —— —— —— Babcock & Wilcox, 2112
—— —— —— —— Bennis, 2113
—— —— —— —— Erith Roe, 2117

STR

Stokers, mechanical, Illinois (Vickers), 2114
—— —— —— —— locomotive, 622
—— —— —— —— Underfeed, 2119
Stone, artificial, 407
—— bridges and arches, 524
—— stairs, 342
—— steps, repairing, 343
—— walls, safe load on, 497
——, weight of, 382
——, H., specific gravities of woods, 128, 129
Stone-Paramor rotary pump, 1275
—— turbine pump, 1271
Stone's system of electric light for railway carriages, 633
—— 'H-R' gear, 946
—— balance drive gear, 947
Stones for masonry, 381
—— lifting, 376
—— road, tests of, 749
—— setting, 377
—— specific gravity of, 127
Stoneware, 429
Stonework, preservative, 2805
Stop-valves, 1668
Storage battery locomotives, 707
—— bins, reinforced concrete, 468
——, coal, 2452
——, loss of thermal value by, 2094
——, coke, 2452
——, cold, 1219
——, gas, 2456
—— reservoirs, 1415, 1510
Storing logs, 360
—— timber, 360
Storm-water discharge of sewers, 1404
Stoutest rectangular beam from a log, to cut, [357
Stowage space for coal, 2095
Straight axles, locomotive, 596, 611
—— line link motion, 953
Strain, elongation, of woods, 184
Strainer racks, 1297, 1343
Straining beams, joints of, 314
Strains, breaking, copper telegraph wire, 177
—— ——, galvanised iron telegraph wire, 177
Strand ropes, flattened, Wright's, 2607
Stranded cables, sags on aluminium, 2220
—— copper conductors, 2222
Straps, heel, of wooden roofs, 320
Streams, fire, 2548
—— measuring quantity of water flowing in, 1290
Street lighting, 2546
—— ——, cost of, 2871
—— paving, 750
Strength, adhesive, of mortars, 175
—— and stiffness of beams and poles, 314
—— bursting, of tubes, 190
—— compressive, of concrete, 391
—— —— —— —— materials, 169
—— crushing, of bricks, 175, 427, 441
—— —— —— timber, 182
—— lateral, of materials, 179
—— of alloys, 170
—— Australian timber, 356
—— beams and poles, 314
—— boiler flues, 1562
—— —— shells, 1561
—— bricks, 175

STR

Strength of brickwork, 175
—— —— chains, 193
—— columns, 223
—— —— concrete, 391
—— —— copper at various temperatures,
—— —— corrugated iron, 234 [171
—— —— iron at high and low temperatures, 171
—— chains, 193
—— rings, 194
—— wire, 176, 177
—— lead pipes, 192
—— masonry, 175
—— materials, 169
—— materials, absolute and ultimate,
—— lateral, 179 [182
—— transverse, 183
—— metals, 170, 171
—— at various temperatures, 171
—— mortars, adhesive, 175
—— rings, 194
—— rivets, 228, 1567
—— ropes, 176, 178, 200, 2608
—— signalling wire, 178
—— slings, 196
—— soldered joints, 176
—— steel at high temperatures, 173
—— plates, 617
—— rails, 536
—— trolley span wire, 697
—— tubes, 191
—— swivels, 195
—— tubes, bursting, 190
—— welds, electric, 2776
—— woods, 170
—— effect of seasoning, 184
—, tensile, of Bull's metal, 2725
—— copper at different temperatures, 171
—— Delta metal, 170, 171,
—— gun-metal, 174
—— materials, 170
—— metals, alloys, and woods, 170
—— rubber, 170
—— steel at high and low temperatures, 171
—— very high temperature, 173
—— transverse, of materials, 183
Stress, breaking, copper telegraph wire, 177
—— galvanised high purity iron line wire, 177
—, wire and wire ropes, 176
—— due to temperature, 174
—, elongation produced by, 184
Stresses on beams and girders, 207, 211, 458
—, reinforced concrete, 441, 453,
Stresses on dock gates, 1465 [458
—— walls, 1463
—— engines, 1737
—— roof trusses, 232
—— turbines, 1797
—— working, on bridges, 295
—— reinforced concrete, 445
Strip, aluminium, weight of, 139
Stroboscopes, Ashdown Industrial, 957
Structural steelwork for buildings, 185, 373
—— shipbuilding, 189
Structures, reinforced concrete, 453

SUP

Struts, 223
—— between houses, 513
——, British standard channels as, 241, 243
—— safe load on, 243, 244
Strutting, 314
Stubs' iron and steel wire gauges, 14
Sturtevant balanced draught system, 2123
—— blowers and exhausters, 2520
—— dryers (timber), 359, 360
—— forced draught system, 2123
—— induced draught system, 2123
—— plenum system of ventilation, 2523
—— progressive or tunnel dryers, 359
—— table of air for blast furnaces, 2664, 2665
Subaqueous rock removal, 1488
Submarine sound signals, 1500
Submarines, Diesel engines for, 2432 [1359
Submerged orifice of a culvert, discharge from,
Subsidences, mining, 2588
Substances, fire-resisting, 341, 2547
Subtense measurements, tacheometer, 65
Suburban railways, electric traction on, 678
Suction cutter dredgers, 1483
—— dredgers, 1482
—— gas plant, Appleby-Bentley, 2179
—— Crossley, 2180
—— Hornsby-Stockport, 2187
—— Kerpely, 2177
—— Ruston, 2187
—— producers, bituminous, 2186
—— pump, 1249
'Suder' acetylene purifying material, 2765
—— bronze, 2774
—— welding wire, 2771
Suez Canal, 1527
Sugar cane machinery, 794
—— mills, 794
Sulphate of ammonia, 2473
Sulzer Diesel engines, 1899, 2420
—— Maag gears, 925
—— marine Diesel engines, 1899, 2420
—— submarine engines, 2432
—— water-tube boiler, 1601
Sun, the, 76 [altitude of, 73
—— to find variation of compass by single
Sun's parallax in altitude, 78, 83
——radius and distance from earth, 76
Superelevation of outer railway rail, 555
Sun-dial, horizontal, 85
Superheated steam, 1534, 1711
——, degree of, 1711
—— economy of, 1710
—— effect on steel, 171
—— heat of, 1535
Superheater Co.'s superheaters, 602
Superheaters, 601, 1612
—— Babcock & Wilcox, 1617
—— Clarke, Chapman & Co., 1588
—— 'Dee,' 1620
—— Galloway, 1615
—— heating surface of, 1613
—— locomotive, 601
—— marine, 2403 [2403
—— M.L.S. Schmidt type, 602, 1618,
—— McPhail & Simpson's, 1613
—— North Eastern, 2403
—— Robinson, 605
—— Superheater Co.'s, 602
—— John Thompson, 1618

SUP

Superheaters, tubes for water-tube boilers and, 1704
Supermiser, Galloway, 1651
Supply cables, electricity, 2207
Supports of shafting, 1002
——————————— Francis's table, 1003
Surcharged wall, 516
Surface condensers, 1826
——— measures, 5
——— of boiler tubes, 1571
——— tarring of roads, 733
——— —, wetted, of ships, 2356
Surfaces, boiler, 2391
——— Ferodo friction, 819, 850, 897
— ——— reflecting power of, 2527
Surveying, 51
——— chain, 1
——— ——instruments, 59
——————— mining, 86
——— —, rate of, 53
——— soundings, 58
——— telemeter, Steward, 69
Surveys, scales for, 56
Suspended ceilings, expanded metal, 337
Suspension bridges, 305
Swan, Hunter & Wigham Richardson, marine oil engines, 2432
——————————— on floating docks, 1473
——— ——— ——— Singapore floating dock, 1476
Swing bridges, 293
——— ——— stresses on, 263
Swiss Locomotive & Machine Works' Diesel engines, 1899
——— railway electrification, 674
——— screw thread, 1139, 1144
Switch and transformer oil, 2347
———-board for electric traction, Board of Trade, 701
———-boards, 2297
——— gear, British Standard, 2297
Swivels, strength of, 195
Sykes' electro-mechanical railway signals, 654
——— railway signalling devices, 654
Synchronous motors, 2286
Synchroscope, 2314

TACHOGRAPH, 'Jaquet,' 1451
T Bars, 240
T beams, 442
— joint, 312
Tacheometer, subtense measurements, 65
——————— Zeiss, 72
Tail haulage, 2599
Tall chimneys, 522
Tangential angles for chords, table of, 563
Tangye gas engines, 1875
Tank locomotives, 574
Tanks, aero-engine, 2070
——— cost of, 2884
——— pressure, 2572
——— water, 1432
Tap grinding, 1171
——, gun, 1172
——— heads, pipe thread, 1173
Taper measures for twist drills, 1038
Tapping, 1171

TEM

Tapping and screwing machines, power for, 1171
-- ——— drill sizes for, 1171
——— ——— lubricants, 1171
——— ——— screw, 1171
Taps, screw, Whitworth standard, 1170
Tar distillates, 2156
——— distillation, 2459
——— fuels, 2156
——— lignite, 2158
——— macadam, 732
——— oil engines, 2159
——— oil fuel for Diesel engines, 2160
———, properties of, 2158
Tarring roads, 733
'Tecalemit' high-pressure lubrication system, 991
Tee iron, weight of, 141
——— slots in cast-iron tables, 1173
——— steel, weight of, 142
Teeth, bevel wheel, 924
——— cycloidal, 910
——— epicycloidal, 910
——— gearing, helical, 927
——— involute, 908
——— of cutters, pitch of, 1026
——— ——— wheels, construction of, 1024
Telegraph radio plant, 2342
——— ——— wire, breaking strains of, 176
Telegraphs and signals of railways, 645
Telemeter, Steward surveying, 69
Telephones, attenuation and formulæ, 2336
——— ——— cost of, 2885
——— ——— G.E.C. intercommunication, 2341
——— ——— loading coils for, 2338
——— ——— repeaters for, 2339
Telephony, 2336
Telescope, Dunbar Scott's auxiliary top and side, 63
Telford road crust, 731
Temperature, absolute zero of, 107
——————————— steam, for vacuum, in inches, 1833
——— ———, casting, 2688
——— ——— control of furnaces, 2673
——— ——— conversion tables, 99
——— ——— —, effect of, on bridges, 270, 307
——— ———, movement of air caused by difference of, 2519
——— ——— of atmosphere, 106
——— ——— earth, 106
——— ——— lakes and seas, 106
——— ——— regulation, 1642, 2508
——— ——— regulators, 1642, 2508
——— ——— standards, 104
——— ———, stress due to, 174
——— ——— stresses in bridges, 307
——— ——— tables, 99
Temperatures, colour of different, 105
——— ——— contract, 2508
——— ——— -- flash, 117
——— ——— freezing, 105
——— ——— gas works, 2460
——— ——— -- ignition, 101
——— ——— miscellaneous, 101
——— ——— of steam at high vacua, 1534
——— ——— properties of steel at high, 1823
——— ———, tensile strength of copper at different, 173

Temperatures, tensile strength of steel at different, 171, 173
Tempered steel, colours of, 1070
Tempering and hardening steel, 1066, 2697, 2705
——— cutting tools, 1062
——————— quenching baths for, 1070
Tenders, locomotive, 625
Tenon and mortise 313
———, tusk or shouldered, 313
Tenoning machines, 1059, 1061
Tensile strength of alloys, 170
——————— Bull's metal, 2725
——————— copper at different temperatures, 173
——————— Delta metal, 170, 171, 174
——————— gun-metal at different temperatures, 174
——————— materials, 169
——————— metals, alloys, and woods, 170
——————— ——— at different temperatures, 171
——————— rubber, 170
——————— steel at different temperatures, 171, 173
Tension in beams, 458
Terne plating, 2711
Terra-cotta, 428
Testing apparatus, pressure gauge, 1666
——— crankshaft alignment, 1739, 1905
——— electric-light cables, 2319, 2320
——— gas, 2481
——— machines for Portland cement, Adie's, 402
——— Portland cement, 401
——— water, 1418
Tests, Admiralty, for iron, 188
——————— steel, 186
———, bituminous materials, 750
——— engine, typical, 1713
——— for bullhead rails, 532, 533
——— planing machines, 1023
——— steel bars, Admiralty, 188
——— ——— rivets, Admiralty, 186
———, for ships, British standard, 189
——— stones, road, 749
——— tramway rails, 686
——— rattler, for bricks, 428
——— tensile, Bull's metal, 2725
——— Delta metal, 170, 171, 174
——— turbine (steam), 1806
Thames river dredging, 1487
Theodolite, adjustment of, 59
——— Everest, 61
——— micrometric, 63 [562
——— ranging out railway curves with, 59
——— the plain, 59
——— transit, 62
——— tunnelling, 62
——— Zeiss, 71
Thermal efficiencies of fuels, 1539
——————— gas engines, 1884
——————— steam power plant, 1539 [536
——— influences, railway permanent way,
——— units, British, 109, 1542, 2449, 2476, 2494

Thermionic valves, 2343
Thermit welding, 699, 2793
Thermo-electric pyrometers, 2677
Thermofeed feed-water regulator, 1674
Thermo-dynamic efficiency, 1867
Thermometer scale conversion chart, 102
——— scales, 99
Thermometers, expansion type, 2674
——————— Negretti & Zambra's boiler distance, 1667
——————— low temperature electrical distance, 1232
——————— Stewards, swing, 69
Thermostatic metal, 2816
Therms, 109, 1542, 2449, 2476, 2494
Thickness of arches, 525
——————— boiler plates, 1576
——— ——— cylinders of hydrostatic presses, 1349
——————— hydraulic pipes, 150, 1243
——— ——— steam pipes, 1687
——————— walls, 369
Thimble-tube boiler, Clarkson's, 2196
Thomas' gas meter, 1443 [1593
Thompson, John, boilers, 1573, 1577, 1578,
——————— superheater, 1618
——————— water-tube boiler, 1593
'Thompson,' ship insulation system, 1223
Thornycroft boilers, 2399
——————— oil fuel system, 2164, 2397
Thorp gas meter, 1444
Threads of pipes, 1150
——————— screws, 1139, 1145
Three-core cables, 2247
——— phase system for electric railways, 670
Thring-Hopkinson torsionmeter, 42
Thrust bearings, 967
——— block, Michell, 2409
Thury's screw thread, 1139, 1144
Thwaite's blast-furnace gas, 2189
——————— rope brake, 34
Tide power, 1283
Tie-beam of roof, 320
Tilbury floating landing stages, 1502
Tiles, 428
——— concrete, 394
——— roofing, 413
——————— 'Poilite,' 414
——— ——— 'Turner's,' 413
——— wall, 'Poilite,' 324
Tiling, 413
Tilting furnaces, 2667
Timber, 347
——— appliances, 1473
———, Australian, 356
——— beams, weight of, 356
——— , content sizes of, 356
——— creosoting, 361
——— crushing strength of, 182
——— dams, 1292
———, drying apparatus for, 359
——— durability of, 355
——— flooring, weight of, 364
——— joists, weight of, 364
——— measures, 347
——— planks, 356
——— posts, weight of, 364
———, preservation of, 360
———, roofs, 365
——— seasoning, 130, 357, 358, 359

TIM

Timber, selection of, 357
——— shrinkage of, 358
——— solidity of, 364
——— storage and preservation of, 360
———, square and round, solidity of, 365
——— weights and sizes of, 356
——— of green and seasoned, 358
——— work, weight of, 364
Timbering drifts, 2574
——— trenches, 513
Time, equation of, 74
——— required for seasoning timber, 358
——— service, electric, 2327
———, to determine, 79, 84
——— transit, pneumatic dispatch, tube, [1203
Timing observations, 82
Tin, 2562, 2718
——— dredging, 2647
——— ore, 2562
——— plating, 2711
———, tensile strength of, 170
Tinplates, weight of, 160
Tinting Portland cement, 387
Tip clearance loss, steam turbine, 1795
——— wagons, 495
Tirrill automatic voltage regulator, 2266
Tolerances for bolts, 1155, 1156
——— fine screw threads, 1141
——— nuts, 1155, 1156
Tomey, J., & Sons' gauge glass, 1665
Tool shanks, 1072
——— steels, high-speed, 1062
——— for cutting ebonite and fibre,
Tools, 1019 [1095
——— automatic, lubrication of, 985
——— carbon steel for, 1062
——— cutting angles of, 1072
——— grinder, Van Dorn, 1114
———, machine, 1019
——— floor space for, 772
——— power required for, 1053
——— portable pneumatic, 1079
——— quenching baths for tempering cutting,
——— shape of, 1061 [1070
Tooth gearing, 907
——— armature shafts, 918
——— bevel, 924
——— cycloidal, 910
——— epicyclic, 910
——— 'Fabroil,' 919
——— helical, 927, 1818
———, double, 929, 1818
———, triple, 930
——— involute, 908
———, gear cutters for,
——— Lewis formula, 915 [910
——— 'Maag,' 925
——— mill, 914
——— mitre, 924
——— non-resonant, 919
——— resilient steel, 921
——— rolling-mill, 910
——— spur, 911
——— teeth, grinding of, 922
———, hardening, 923
———, lapping, 922
——— turbine, 930
Torque, starting, 1747
Torrent filter, Pulsometer, 1416
Torsion meter, Hopkinson's, 42

TRA

Torsional rupture by shock, of woods, 179
Tosi-Diesel engines, 1898
Totalisers, 1440
Towers, water-cooling, 1851
——— water, cost of, 2890
Tracing, 96
——— leakages, 1420
Traction, canal, 1525
——— electric, 669
———, Board of Trade regulations
for, 701
——— engines, regulation on roads, 722
——— lamps, 2323
——— motors, resistance, on a level railroad,
583
——— British standard, 2282
——— railless electric, 703
Tractive force of locomotives, 587, 592
——— resistance on pavements, 726
——— tramways, 689
Trade-marks, 2834
Traffic, road, 727
——— statistics, 761
Trafford sheets, Turner's, 325
Train acceleration, 587
——— electric lamps, 2323
——— lighting, 633
——— resistance, 582
——— formulæ for, 582, 583
——— wheels, 632
Trains, momentum of, 268
——— resistance of air to, 582
——— table of speed of, 586
Tramway undertakings, Transport Ministry's
return, 762
Tramways, electric, 683
——— Board of Trade memoran-
dum, 700
——— Board of Trade regula-
tions, 701
——— British Standard specifica-
tions for, 703
——— brakes for, 691
——— cables for, 700
——— cars for, 701
——— construction of, 694
——— cost of construction, 689,
2885
——— working, 2885
——— equipment of, 692
——— foundations for, 689
——— guard wire for, 697
——— leakage on, 700
——— motor control, 690
——— motors for, 690
———, English Electric
Co., 2285
——— overhead construction, 694
——— paving, 689
——— permanent way for, 701
——— points and crossings for,
689
——— poles for, 695
——— rail bonding, 697
——— welding, 699
——— rails for, 684
———, joints for, 688
——— sectioning, 692
——— span wire for, 697
——— track leakage on, 700

Tramways, electric, tractive resistance on, 689
——— trolley wheels for, 693
——— wire for, 697
——— fittings for, 694
——— tyres and axles for, 683
Transformers, electric, 2301
——— booster, 2305
——— bushings for, 2304
——— oil for, 2347
——— on the National 'Grid' system, 2350A
Transit theodolite, 62
——— time, pneumatic dispatch tubes, 1203
——— tunnelling theodolite, 62
Transition curves, 567
Transmission, hydraulic, steam turbine, 1821, 2388
——— lines, overhead, 2220, 2221, 2227
——— of power by leather belting, 855
——— ropes, 873
——— steel shafting, 1001
———, telephonic, 2336
Transport, costs and capacities of, 47
——— Ministry of, 712, 762
Transporters, quay side, 1471
Transporting earth, 491
Transverse metacentre of ships, 2356
——— strength of corrugated iron, 234
——— materials, 183
Traps, steam, 1699
——— general notes on, 1702
Trass mortar, 383
Trautwine rule for retaining-walls, 518
Travelling cranes, 802
Treads, Mason's stair, 343
Treatment of mild steel, 188
——— sewage, 1367
Tree planting on roads, 730, 758
Trenches, timbering, 513
Triangulation, 52
Tribrach system, 60
Trim indicator, ship, Ralston's, 2359
Trinitrotoluene explosive, 2798
Trip centrifugal clutch, 893
Triple expansion engines, steam pressure in, 2373
——— valve for Westinghouse brake, 639
Trisection of angles, 92
Trist's (Ronald) 'Thermofeed' feed-water regulator, 1674
——— S.E.A. rings, 1264, 1756
Trolley poles, electric tramway, 695
——— system of electric tramways, 693
——— wheels, electric tramway, 693
——— wire, electric tramway, 697
——— fittings, 694
Trolleys for electric tramways, 693
Tropical rainfall, 1385
Truck weighers, 1440
Trucks, industrial electric, 704, 760
Truss multipliers for ridged roofs, 233
Tubbing of mines, 2577
Tube bending, 1092
——— boilers, water, 1584, 2397
——— cleaners, 1684
——— couplings, 1081
——— cutters, 1089

Tube former, 1093
——— mills, mining, 2640
——— railways, cost of construction, 2874
——— London, 678, 679
Tubes, aluminium, weight of, 159
——— boiler, 600, 616, 1571, 2395
——— brushes and scrapers, 1684
——— charcoal iron lapwelded, 1571
——— locomotive, 600, 616
———, boiler, surface of, 1571
——— weight of, 1571
——— brass, seamless, weight of, 158
——— bursting strength of, 190
——— condenser, 1826
——— copper, weight of, 156
——— diameter of cylindrical marine boiler, 2395
——— for water-tube boilers and superheaters, 1704
——— house, pneumatic, 1206
——— locomotive boiler, 600, 616
——— marine boiler, 2395
——— pneumatic despatch, 1202
——— power required for working, 1203
——— pressure, Pitot, 1506
——— screwed and socketed, thicknesses and weights of, 151
——— small metal, to bend, 2815
——— speaking, 118
——— steel, strength of, 191
——— velocity and discharge of water through, 1298, 1299, 1300, 1301, 1364, 1423, 1426, 1429
——— Venturi, 1442
——— weight of wrought iron, 148, 149
Tubing, aluminium, weight of, 159
——— metallic flexible, 608, 1081, 2571, 2766
Tubs, colliery, 2598
Tubular boilers, multi-, 1576
———poles for electric tramways, 695
——— standard, 695
Tullis, John, on belt-driving, 863
Tunnelling transit theodolite, 62
Tunnels, railway, 554
——— cost of, 2887
Turbine gears, 929, 1818, 2381
——— head races, 1298
——— pumps, 1270
——— Pulsometer, 1270
——— Stone, 1271
——— shafting, steam, 1005, 2374
Turbines,
(Steam), 1767, 2375
——— arrangement of, 2376
——— back pressure, 1810
——— bearings of, 1796
——— blading of, 1787, 1788
——— Britimp metallic packings for, 1793
——— British standard specification for, 1816
——— Brown-Curtis, 2381
——— carry-over energy, 1786
——— consumption of steam in, 1779, 1809
——— corrections for steam conditions, 1806
——— critical speeds of, 1802

TUR

Turbines,
(Steam), design of, 1767
——— disc drum type, 1792
——— friction in, 1794
——— double helical gear for, 1818
——— dummy loss in, 1793
——— efficiency of, 1779
——— electrical transmission, 1823
——— flow of steam in, 1767
——— forgings for, 1804
——— gear, 1818
——— geared, installations of, 1822
———, glands, 1794
——— high-pressure, 1809
——— hydraulic transmission, 1821, 2388
——— impulse, 1783
——— installations of geared, 1822
——— internal combustion fuel, 1926
——— labyrinth packings of, 1792
——— Ljungström, 2385
——— losses in, 1792
——— low pressure, 1809
——— lubrication of, 1801
——— materials for construction of, 1802
——— mixed pressure, 1810
——— nozzles of, 1770
——— packing, labyrinth, 1792
——— Parsons, 2381
——— gearing for, 2386
——— steam blades and caulking pieces for, 1790
——— pitch of impulse blades, 1788
——— power transmission systems, 2388
——— reaction, 1788
——— reducing, 1810
——— reduction gears, 929, 1818, 2381
——— ship, 2375
——— arrangement of, 2376
——— shafting for, 1005, 2374
——— starting large, 1806
——— steam consumption in, 1779
——— stresses in, 1797
——— tests of, 1806
——— tip clearance loss of, 1795
——— types of, 1809
——— variation in efficiency with steam pressure, 1815
——— water glands of, 1794
——— windage loss in, 1794
Turbines,
(Water), 1315
——— American, 1319
——— British, 1320
——— classification of, 1315
——— draft tube of, 1330
——— flumes of, 1298
——— Francis, 1317
——— Gilkes, 1323, 1324, 1337
——— governors of, 1335
——— Howes, 1320
——— Little Giant, 1320
——— radial inward-flow, 1327
——— reaction, on pressure, 1315
——— regulation of, 1331
——— runners of, 1326
——— seal rings for, 1331
——— sluice gates, 1343
——— spiral, 1325
——— strainer racks, 1297, 1343
——— Turgo-Gilkes, 1337

UNI

Turbines,
(Water), types of, 1317
Turbo-alternator, G.E.C., 2263
Turbo-charging system, Büchi, 2435
Turbo pumps, 1260
Turgo impulse wheel, Gilkes, 1337
Turner, E. R. & F., on flour mills, 783
Turning tools, diamond, 1087
Tusk tenon, 313
Tuyeres bosh, 2662
Tweddell's hydraulic riveter, 1355
Twist drills, 1034
——— Firth's, 1035
——— taper measures for, 1038
Twisting angle of shafts, 1005
Two-core cables, 2247
Type metal, 2743
——, sizes and descriptions of, 19
Tyres, carriage and wagon railway, 612
——— for locomotive wheels, 612
——— motor cars, 1976
——— railway-wagon wheels, 612
——— tramway wheels, 683
———, influence of width of, 725
——— N.A.P., 1977
——— pneumatic, 1977
——— Macintosh, 1977
——— rubber and iron, 726
——— widths of, 847, 725

'U' LEATHERS, 1264
Ultimate strength of materials, 182
Underfeed stokers, 2119
Underground haulage, cost of, 2602
——— railways, cost of, 2872
Underlining, 413
Unequal angles, standard sections with, 238
Uniflow steam engine, 1722
———, Galloway, 1723
Uniform motion, 27
Unions, pipe, 1244 [2494
Unit, British Thermal, 109, 1542, 2449, 2476,
United Flexible Metallic Tubing Co., tubing, 608, 1081, 2571, 2766
——— Kingdom metallic piston-rod packing, 1749
——— statistics of railway expenditure, 661
——— waterways in, 1505
——— States Bureau of Standards, melting points, 103
——— of America and railway electrification, 675
——— metallic packing, 1749
——— Metallic Packing Co.'s Bradford steam trap, 1699
——— packing for centrifugal pumps, 1268
——— railway statistics, 661
——— river regulation works, 1510
——— Water Softeners Ltd., Lassen-Hjort system, 1633
——— Permutit process, 1422, 1628
——— Zeolytic exchange process, 1422, 1627
Units, electrical, 2203
——— gas, power, 1861
——— of heat, 109, 1542, 2449, 2476, 2494

UNW

Unwin's formula for flow of gas in mains, 2478
Urinals, 1382
——— painting of, 1382

'V' BELTING, 883
'V' notch method of calculating quantity of water flowing in a stream, 1290
——— water recorder for steam engines, 1849
——— weir recorder, 1441
Vacuum brake, 642
——— correction for steam turbines, 1782
——— cup lifters, 824
——— evaporators for sugar mills, 794
——— gauges, 1666
——— ———, tester, Brotherhood's, 1666
——— pump, Bailey's, 1200
——— ——— Holland, 1201
——— temperature and pressure of steam in inches, 1833
——— evaporators for trade liquids, 795
Valuation of masonry, 381
——— mines, 2651
——— painters' work, 417
Value of degrees of temperature, 102
——— paints, 417
Valve, driver's equalising brake, 639, 641
——— gear, 594, 1758
——— drop, 1726
——— ——— Caprotti, 596, 1857
——— Lentz, 595
——— Walschaert, 594
———, improved triple brake, 639
———, locomotive, 594
——— re-facers, 1670
Valves (steam), 1668
——— 'Arca' compact, 1678
——— area, 1748
——— cylindrical, 593
——— Dexine, 1264
——— Ferranti, 1668
——— Hopkinson's, 1668
——— Klinger, 1669
——— locomotive, 593
——— motor-car, 1962
——— of gas, oil, and petrol engines, 1888
——— pump, 1250
——— reducing, 1671
——— safety, 1570, 1659
———, areas, 1659
——— ———, Board of Trade rules for, 1659
——— ——— Bureau Véritas rules for, 1660
——— ——— Lloyd's rules for, 1660
——— slide, gear, 1758
——— snifting, 1201
——— steam, 1668
——— area of, 1748
——— stop, 1668
——— thermionic, 2343
——— 'Vee-Reg.', 1669
Van Dorn portable electric drill, 1042
——— tool post grinder, 1114
Vans, electric, 704
Variable delivery pressure pumps, 1274
Variable-speed gear, 948
——— Continuously-variable chain driven, 951

VIS

Variable-speed gear, 'VSG,' hydraulic, 948
Variation of compass, annual, 74
——— to find, 73
Varnish for galvanised iron, 234
Varnishes, 2807
——— quick drying, 2807
——— —, waterproofing, 2807
'Vee-Reg.' valve, 1669
Veeder magnetic counter, 1452
Vegetable oils, manufacture of, 792
Vehicles, electric battery, 704
——— on bridges, heavy, 762A
———, petrol, 760
——— -, public service, 757
———, passenger, working costs of, 757
Vein mining, 2590
Velocities of water through bridges, 259
Velocity and discharge of water, 1423
——— coefficient of steam turbines, 1787
———, critical, irrigation canals, 1392
——— mean, of canals, 1390
——— of air, 1183, 1390
——— a blow, steam hammer, 1128
——— light, 119
——— sound through air, 117
——— solid substances, 118
——— water, 117
——— water in channels, 1298
——— pipes, 1298, 1364, 1423, 1426, 1429
——— rivers, 1288, 1506, 1507
——— wind, 1184
——— wood-working machinery, 1058
Vena contracta, 1306
Ventilation, 2513
——— air cooling, 2514
——— distribution of, 2513
——— filtering, 2514
——— necessary for, 2513
——— fans, 2663
———, furnace, mining, 2614
——— of floor joists, 333
——— mines, 2614
———, plenum system, 2523
——— of sewers, 1367
Ventilator, Boyle's air pump, 2523
Venturi tubes, 1442
——— water meter, 1438
Versed sines for rails, table of, 556
Vertical boilers, 1582
——— force of earth's magnetism, 74
——— retorts, 2488
Vice gripping, 1091
Vickers' auto forced lubrication system, 989
——— automatic reversing drive, 1021
——— boilers, 1584, 1586
——— Diesel engine for submarines, 2424, 2426
——— 'Duralumin,' 2735
——— gas engine, 1872
——— Illinois chain grate stoker, 2114, 2117
——— oil engine, 1917
——— stokers, 2114
——— water tube boilers, 1584, 1586
Victaulic flexible pipe lines, 1246
——— pipe joints, 1169, 1246
'Visco' air filter, 2268
Viscosity of air, 1183

VIS

Viscosity of oils, 978, 993
——— water, 1239
Viscous flow of oils, 993
——— liquids, pumps for, 1275
Volga, dredging plant on, 1510
Voltage regulator, Tirrill, 2266
Voltmeters, Weston standard, 2312
Volume, blast, 2664
——— of air to burn gas, 2169, 2475
——— earth in cuttings, 481
——— liquids reduced by pressure, 117
Volute and helical springs for locomotives, 613, 614
——— flat springs, 1014
'VSG' variable delivery pump, 1274
——— speed gear, 948
Vulcan Diesel hydraulic drive, 2389
Vulcanised fibre, British standard, 2234
——— rubber wires and cables, 2209, 2211
Vulcanising wood, 361
Vyrnwy dam, 1413

WAGON axles, railway, 612
——— tyres, railway, 612
Wagons and carts, loads of, 494
——— railway, 630
——— steam, working costs of, 756, 759
——— steel for, 617
——— tip, 495
Walker, J., & Co.'s, Ltd., 'Lion' metallic packing, 1752, 1753
Wallace clutch, 895
Wall-breakwaters, 1458
Wall paint, 2804
Wallpaper, number of pieces required for room, 419
Wall-plates of floors, 320
——— roofs, 318
Wall plugs, 420
——— and sockets, 2242
Walls, 368
——— damp course for, 368
——— dampness of, to remedy, 407, 2804
——— dock, 1463
——— foundations of, 376
——— heat transmission through, 344
——— hollow, 368
——— London Building Act rules, 370
——— quay, cost of, 1465
——— railway cutting, 514
——— reinforced concrete, 463, 465
——— retaining-, 514
——— brick, 517
——— equilibrium of, 515
——— foundations of, 516
——— pressure of earth on, 514
——— water on, 514
——— rules for, 517
——— weight of buildings on, 516
——— rules for building, 368, 370
——— stone and brick, safe loads on, 497
——— surcharged, 516
——— tables of contents in cubic yards, 520
——— thickness of, 369
——— warehouse, London Building Act rules, 370
——— wharf, 520
Wallsend oil burner, 2163

WAT

Wallsend-Howden liquid fuel system, 2163
Walschaert valve gear, 594
Ward Leonard system of electric winding, [832
Ware, electrical, 430
——— sanitary, 429
Warehouse walls, 370
Warner Anthony, steam pump, 1256
——— wind pump, 1256
Warrington wire-gauge, 12
Washable water paint, Carson's, 2802
Washers, air, 2514
——— and scrubbers, gas, 2454
——— nuts, and bolt-pins, French, 1158
——— weight of, 124
Washing coal, 2619
Waste heat boilers, 2194
———, generation of electricity by, [2100
———, recovery of, 2199
——— water detection, 1420
——— meters, 1437
——— weir capacities, 1395
Wastes of sinks, 1380
Watch, determination of south by means of, [74
——— screw threads, 1144
Watches, non-magnetic alloy for, 2743
——— oil for, 2810
——— testing, 2810
Water, 124, 1239
——— air pump, measurement of, 1849
——— and steam separators, 1699
——— Board, Metropolitan, standard pipe sockets, 1429
———, boiler, treatment of, 1623
——— boiling-points of, 111
——— canal, loss of, 1389, 1524
——— chlorinating, cost of, 1419
——— closets, 1380
——— London County Council regulations, 1380
———, consumption of, in locomotives, 624
——— cooler, Heenan, 1232
——— cooling for internal combustion engines, 1895
——— steam condensers, 1832
——— towers, 1851
———, cost of lifting by, 1242
——— data, 1239
——— degassing, 1638
———, discharge of, through orifices, 1306
——— pipes, 1298, 1364, 1423, 1426, 1429, 1430
——— sewers, 1359, 1364
——— distillers, 797
——— dynamometer, Froude, 38
——— elevator, Boulton's, 1253
——— engines, effective power of, 1283
——— evaporation from, 1383
——— feed, 1624
——— elimination of oil from, 1680
——— heating, 1640
——— regulators, 1672
——— softeners, 1628
——— filtration, 1416
———, flow of, through orifices, 1306
——— pipes, 1298, 1364, 1423 1426, 1429
——— flumes, and channels, 1298

WAT

Water flowing in streams, to determine quantity of, 1286, 1290
——————— over weirs, table for, 1288
——————— through pipes, head required for, 1284, 1364
——————— under bridges, 259
——— gas, 2189, 2471
——— gathering grounds, 1416
——— gauges (boiler), 1662
———————, defects in, 1664
——————, 'Hulburds,' 1663
——————, 'Klinger,' 1662
———, general data of, 1239
——— glands, steam turbine, 1794
——— -glass cements, 397
——— hammer in pipes, 1347
———, hardness of, 1421
——— heated by steam, 2494
——— heater, Westinghouse, 2510
———, horse-power of, 1284, 1285
——————————— required to raise, table of, 1240, 1241
——— hot, supply of, 2509, 2511
——— in channels, velocity of, 1298
——————— wells, quantity, 1409
———, irrigation, duty of, 1393
——— leakage detection, 1420
——— losses from irrigation canals, 1389
——————————— canals, 1389, 1524
——— main pipes, cast-iron, weight of, 145, 150
——— mains, cement joints for, 1247
——————— cost of scraping, 2892
——— metering, 1435
——— meters, 1437
——— pipe absorption dynamometer, 37
——————— bends, unbalanced pressure on, 1431
——————— fittings, weight of, 154, 155
——————— joints, 1244, 1429
——————— screw threads, 164
——— pipes, 1419
——————— British standard, 165
——————— incrustation and corrosion of, 1419
——————— iron, weight of, 155
——————— lead, corrosion of, 420
———————————, weight of, 148, 1431
——————— main, cast-iron, weight of, 145
——————— thickness of, 165
——————— threads of, 1150
——————— zinc, corrosion of, 420
——— power, development of, 1283
——————, to compute, 1283
——————— required to pump, 1241
——— pressure, 1239
——————— engines, 1309
——————— in hydraulic mains, 1242
——————— on retaining walls, 514
———————, table of, 1240, 1241
———, purification of, 1419, 1632
——————————— for boilers, 1624
——— purity, standard of, 1418
——— raiser, Willcox's, 1657
——— raising by compressed air, 1276
——— recorder, Lea's automatic, for steam engines, 1849,
———, reduction of volume of, under pressure 117
——— separators and dryers, steam, 1699
———, sluice measurement, 1290

WEI

Water softening, 1422, 1628
——————————— Kennicott system, 1629
——————————— Lassen-Hjort system, 1633
——————————— 'Neckar' system, 1635
——————————— Permutit system, 1422, 1628
——————————— Royles' Reisert's system, 1631
——————————— Zeolytic process, 1422, 1628, 1704a
———, specific gravity of, 124, 128
——— standard of purity of, 1418
——— supply, irrigation, 1383
——— surface, evaporation from, 1383
——— tanks, 1432
——— testing, 1418
——— turbines, 1315
———, reduce sounding to low, 59
——— towers, cost of, 2890
——— troughs, locomotive, 625
——— tube boilers for land service, 1584
——————————— marine service, 2397, 2400
———, velocity of, in pipes and channels, 1242, 1298, 1364, 1423, 1426, 1429
——————————— Prony formula for, 1507
——————————— sound through, 117
——— viscosity of, 1239
——— waste, detection of, 1420
——— weighers, 1439
———, weight of, 124, 1239
Waterproof paints, 2804
Waterproofing boards, 2804
——————————— concrete, 394
Waterway of bridges, 258
Waterways, German, 1515
——————————, United Kingdom, 1505
——————————, navigable, 1505
Waterworks, 1411
——————————— cast-iron sockets for pipes, 1429
——————————— cost of, 2890
——————————— maintenance and renewal of, 1418
——————————— towers, cost of, 2890
Watt parallel motion, 955
Wave action on breakwater, 1457
——— effective horse-power of ships, 2364
Weaving shed, 786
Webb's brine attemperator, 1229
Wedge, 26
Wedges, 313
Weighbridge, Avery's, 1440
Weighing machines, 1439, 1440, 1448
Weight, aero engines, 2070
——————— breaking, of pine, 185
——————— dead, ships, 2363
——————— of aluminium bar and strip, 139
——————————— sheet, 140
——————————— tubes, 159
——————— angle iron, 141
——————————— steel, 142
——————————— annealed and bright iron wire, 176
——————— atmosphere, 1181
——————————— ball cast iron, 133
——————————— balls of various metals, 132, 135
——————— bar cast iron, 134
——————————— boiler tubes, 1571
——————— bolt heads, 161
——————————— brass bars, 137

Weight of brass pipes, 146
——— tubes, seamless, 158
——— wire, 161
——— braziers' copper rivets, 163
——— buildings, effect of, on retaining-walls, 516
——— cast-iron gas pipes, 145, 150, 2477
——— pipe fittings, 154
——— pipes, 147, 150
——— water pipes, 145, 154
——— castings from wood patterns, 161
——— in brass, 161
——— cast iron, 161
——— copper, 161
——— gun-metal, 161
——— zinc, 161
——— coal, 2560
——— composition nails, 163
——— concrete, 391
——— copper bars, 137
——— nails, 162
——— pipes, 146
——— rivets, 163
——— tubes, seamless, 156
——— wire, 161
——— crowds, 44
——— cubic inch of metals, 132
——— yard of soils, 494
——— Diesel engines, 1903, 2434
——— marine engines, 2434
——— ebonite, 144
——— engines, aero, 2070
——— flat rolled iron, 135, 136
——— steel, 137
——— foot square of metals, 132
——— gas, 124
——— engines, 1883
——— gas-pipes, 145
——— hexagon rod metal, 135
——— hoop-iron, 160
——— iron rivets, 163
——— wire, 161
——— wrought, 134
——— lead bars, 137
——— pipes, 146
——— formulæ for, 146
——— locomotives, 589
——— machinery on ships, 2362
——— metal sheets and bars, 135
——— metals, 132
——— nails, 163
——— nut and bolt heads, 161
——— nuts, cold-punched, 161
——— octagon rod metal, 135
——— octagon steel, 137
——— oil engines, 1903
——— pipe-jointing material, 164
——— pipes and tubes, 145
——— pneumatic power hammers, 1130
——— radiators, aero-engine, 2070
——— railway carriages and trains, 627
——— rails, 535
——— rolled flat iron, 135
——— lead, copper, and brass, sheet and bars, 137
——— roofs, 316
——— ropes, steel, 166, 2608, 2609
——— round steel, 166
——— screwed and socketed tubes, 151
——— screws, 163
Weight of seamless brass tubes, 158
——— copper tubes, 156
——— sheet, bar, and ball metals, 133
——— cast iron, 134
——— lead, 138
——— zinc, 138
——— ship machinery, 2362
——— ships, 2361
——— slates, 412
——— snow on roofs, 322
——— socket pipes and fittings, 155
——— soil pipes, 1381
——— soils, 494
——— solids and liquids, 126
——— square steel, 137
——— steam engines, 1733
——— and boilers, 1733
——— hammers, 1226
——— pipes, 145
——— steel, 137
——— wire, 161
——— ropes, 166, 178
——— , wrought, 134
——— Stirling boilers, 1590
——— stone, 382
——— stones, 127
——— tee iron, 141
——— steel, 142
——— timber, 356, 358, 364
——— beams, posts, and joists, [364
——— work, 364
——— tinplates, 160
——— tubes, 145
——— tramway poles, 696
——— washers, 162
——— water, 124, 1239
——— pipe fittings, 154
——— wire, 161
——— woods, 364
——— wrought-copper nails, 162
——— iron and steel, 134
——— nails, 162
——— pipes, 145
——— tubes, 148
Weights, 6
——— and measures, 1
——— foreign, 16
——— Siamese, 19
——— sizes of timber, 356, 358
——— specific gravities, 125
——— , atomic, 123
——— bridgework, 261
——— of and safe loads on compound steel girders, 238, 252
——— girders, 252
Weir, G. & J., air compressors, 2439
——— air pumps, 1849, 2408
——— boiler feed heater, 605, 2406
——— de-aerator, 1640
——— direct-acting steam feed pump, 1256
——— ejector air pump, 1849
——— evaporators, 2406
——— heater, 605
——— regulator, 1676
——— Monel metal, 1806, 2729
——— paragon air pump, 2408
——— surface condenser, 1837
——— turbo-feed pump, 1260

WEI

Weir, G. & J., uniflux condenser, 2406
——— water extractor pump, 1837
—— Bremen, 1511
—— measurement, water, 1287
—— recorders, 1441
——, waste, capacities, 1395
Weirs, to raise a river a certain height by, 1298
——— water flowing over, tables for, 1288
Weisbach's formula for safe load on piles, 503
Weld tests, 2776
Welded boiler seams, 1563
—— bonds, 699
Welders, training, 2776
Welding and cutting, 2759
——— acetylene, 699, 2759, 2793A
———, alloy process, 2784
——— aluminium, 2774, 2793B
———, atomic hydrogen system, 2792
——— blow pipes, 2793A
——— brass, 2775, 2793B
——— carbon steels, 9793A
——— cast iron, 2773, 2793A
——— copper, 2775, 2793A
——, electric, 699, 2779
——— ——— arc, 2780
——— cost of, 2789
——— Lloyd's rules for application to ship construction, 2445
——— filling materials for, 2769, 2793A
——— fluxes, 2770
——— lead, 2775
——— low pressure equipment, 2793A
——— machines, 2776
——— methods of, 2793A
——— miscellaneous metals and alloys, 2793B
——— nickel, 2776
——— oxy-acetylene blowpipe for, 2767
——— progress in 1930..2793
——— rails, 699
——— regulators, 2793A
——— repairs, 2777
——— resistance, 2789
——— special and stainless steels, 2973A
——— speed and cost of, 2973A
——— steel, 2771
——— Thermit, 699, 2793
Weldless steel tubes for locomotive boilers, 616
Welds, testing of, 2776
Well sinking, 1408
Welland Canal, Canada, 1528
Wells, artesian, 1409
——— pumps for, 1251
——— brickwork of, 1408
——— deep, pumping from, 1276
——— gas-engine starter, 1886
———, protective distance of, from drains, 1409
——— pumping from, 1251
——— by compressed air, 1276
——— quantity of water in, 1409
——— steining of, 1407
——— to increase yield of, 1409
Welsh iron, 2682

WIL

Werkspoor-North Eastern Diesel engine, [2419
West Coast iron, 2683
West, true, to determine, 74
Western Electric Co. loading coils, 2338
——— telephone repeaters, 2339
Westinghouse air compressor, 1193
——— automatic brake, 638
——— electro-pneumatic railway signalling system, 645
——— lavatory water heater, 2510
——— mechanical railway signalling system, 645
——— Morse driving chains, 905
——— motor car power brake, 1973
——— radiators, 2506
——— system of heating railway carriages, 635
Weston ammeters and voltmeters, 2312
——— friction coupling, 890
——— synchroscope, 2314
Wet docks, 1459
——— cost of, 1459, 2856
—— grinding, 1107
Wetted surface of ships, 2356
Wharf cranes, 810
——— walls, 520
Wheel base of locomotives, 599
Wheels, abrasive, 435, 1105
——— speed of, 1108
——— emery, guards, 1112
——— speed of, 1108
——— fly, 1742
———, gas engine, 1888
——— grinding, 1105
——— hand, proportions of, 1163
——— locomotive, counterbalancing, 599
——— tyres of, 597, 612
——— Pelton, 1309
——— railway wagon and carriage, 632
——— road regulations for, 722
——— scoop, 1276
——— teeth of, 907
——— trolley tram, 693
Whistling buoys, 1490
White Diesel engines, 2432
White-Foster water-tube boiler, 2397
White metal alloy, 2737
Whitewash, 417, 418
Whiting, 418
Whitworth nuts and bolts, 1152, 1153
——— screw threads, 1139, 1140, 1144
——— standard pipe threads, 1152
——— screws for watch and instrument makers, [1144
——— taps, 1170
——— wire gauge, 12
Width of river, to measure, 54
——— roads, 727
——— tyres, 725
Wigglesworth's clutches, 853
Wilcox & Babcock boilers, 1596, 2397, 2398
Willcox ejector steam-jet pump, 1657
——— -Penberthy cellar drainer, 1262
——— injector, 1655
——— Ramoneur boiler tube cleaner, 1683
——— semi-rotary pump, 1252
——— soot remover, 1683
Willesden paper, 342, 413, 1222, 2346
——— underlining, 413

WIL

Wilson plastic arc welding system, 2786
Wind action, 1185
——— engines, 1187
——— impulse of, 1184
——— mills, 1185
——— mills, power of, 1185
Wind-power control, 1186
——— pressure and weight of snow on roofs, 322
——— ——— on bridges, 265
——— pumps, 1186
——— velocity, 1184
Windage loss in steam turbines, 1794
Winding engines, electric, 832
——— ——— ——— steam consumption of, 1714
———, mining, 2604
——— wire ropes for, 176, 2607, 2608, 2609
Windows, factory, 773
Wing fan, F. W. Potter & Soar, Ltd., 2521
——— nuts, 1161
Winzing and raising in mining, 2579
Wipers for slides, 1053
Wire, annealed and bright iron, Rylands', 176
——— copper, breaking strain of, 177
——— ———, weight of, 161
——— cut bricks, 426
——— drawing, 2816
———, enamelled, copper, 2309
——— gauge indicator, 779
——— ——— standards used in the United States, 15, 16
——— gauges, 11
——— guard for electric tramways, 697
———, hard-drawn copper, 177
———, iron, breaking strains of, 176
———, weight of, 177
——— ropes, 166, 176, 178, 826, 885
——— breaking stresses of, 178, 179
———, durability of, 826
——— flat, 2611
——— flexible steel, 828
——— for winding, 178, 2606
——— lubrication of, 889
——— preservation of, 200
——— socketing, 887
——— splicing, 879, 887
———, Rylands', 176, 177
——— span, for electric tramways, 697
——— springs, safe loads for, 1010
———, steel, breaking strains of, 177
——— strand, for signalling purposes, 178
——— telegraph, 177
——— trolley, for electric tramways, 697
——— weight of, 161
——— Wright's, 2607, 2608, 2609
Wireless or radio telegraphy, 2342
——— signalling in fog, 1501
Wires, insulated, specification for, 2243
——— strength of, for tramways, 697
Wiring, electric-light systems, Institution of Electrical Engineers' rules for, 2231
——— ———Callender's 'Kalibond' system, 2252
——— for electric light, Henley system, 2247
——— ———Johnson & Phillips' system, 2251
——— ———Macintosh system, 2254

WOR

Wiring for electric light, St. Helen's Cable & Rubber Co.'s C.T.S. system, 2244
——— ——— ——— ——— specification of, 2245
Witness, engineer as, 2905
Witton interpole tramway motor, 2283
Wollaston producer boiler, 1584, 1585
Wood beams, breaking weight of, 185
——— charcoal fuel, 2106
———, creosoting of, 361, 539
——— fuel, 2106
——— girders for floors, 329
———, machine cutters, sharpening angles of, 1072
——— pillars, 364
——— post paint, 2805
——— preservative processes, 360, 361, 2816
——— pulleys, 864
——— refuse suction gas producer, 2181, 2187
——— resistance to reversal of stress, 179
——— roadways, 743
——— roofs, 366
——— roofs, joints of, 320
——— seasoning, increase of strength by, 184
——— screws, 1148
———, spike gripping power of, 184
——— stairs, 342
——— stave pipes, 1304
——— vulcanising, 361
——— work painting, 2801
——— working machinery, power required for, 1058
——— ——— ——— ——— speeds of, 1058
——— ——— tools used for metal, 1090
Woodeson boilers, 1588, 2400
——— — feed pumps, 1254
Woodite water gauge packing, 1663
Woods, 347
——— drying and seasoning of, 130, 357, 359
——— hard, 348
——— resistance of, to torsional rupture by shock, 179
——— seasoning, 358
——— ——— ——— ——— increase of strength by, 184
——— soft, 348
——— specific gravities of, 128, 129
——— stiffness of, 179
——— tensile strength of, 170
——— weight per cubic foot, 128, 129, 130, 364
Wool, slag, 342, 1222, 1696
Work, 29
———, accumulated, 29
——— done by men and animals, 44
——— ——— steam, 1708
——— on inclined plane, 28
Work-tables of machine tools, 1172
Working aluminium, 2815
——— costs of getting coal, 2583, 2589
——— load on crane chains, 193
——— strength of boiler flues, 1562
——— ——— ——— ropes, 199, 2610
——— stresses, reinforced concrete, 441
——— ——— on bridges, 262, 264
Works contract, powers of an engineer, 2606
———, gas, cost of, 2451, 2867
———, use of high-pressure gas in, 773

WOR

Workshop lighting, 2530
——— practice, 774
——— recipes, 2801
——— roofs, 231, 765
Workshops, engineering, construction of, 765
——— use of high-pressure gas in, 773
Worm and its cutter, 943
——— wheel inspection, 944
——— worm wheel manufacture, 942
——— gear applications, 945
——— gearing, 937
——— horse-power formulæ, 945
——— wheel generating, 943
Worn parts, machinery, saving of, 1095
W. R. Patents, CO_2 combustion indicator (now International Gas Detectors, Ltd.), 2086
Wright Whirlwind aero engines, 2058
Wrights' Ropes Ltd., flattened strand wire ropes, 2607
———, locked coil wire ropes, 2608
———, rust resisting steel ropes, 2610
Wrought-copper nails, weight of, 162
Wrought-iron, 134, 2680
——— bends for steam pipes, 1685
——— castings (Mitis), 2680
———, corrosion of, 174, 2710
——— joists, 331
——— nails, weight of, 162
——— pillars, 375
——— pipes, 145, 148, 2477
——— shafting, 1001
——— tubes, weight of, 148, 149, 2477
———, weight of, 134

ZIN

X-RAYS in industry, 2349

YARD, cubic, different soils, weight of a, 494
Yarn cotton measurement, 788
———, Egyptian, 786
Yarrow boiler casings, 2403
——— boilers, land type, 1602
———, marine type, 2399, 2401
Yellow fir girders, 330
——— pine, breaking weight of, 185
Yorkshire cold-blast iron, 2682
——— pig iron, 2682
Young's hydraulic bolt forcer, 1354
——— modulus of elasticity, 184, 1825

ZEISS reversing levels, 70
——— sectioning alidade, 72
——— self-reducing tacheometer, 72
——— theodolite, 71
——— tacheometer, 72
Zeolites, water filtration over, 1422
Zeolytic exchange process, water softening, 1422, 1627, 1704A
Zero of temperature, absolute, 107
Zeuner valve gear, 1758
Zinc, 2718
——— castings, weight of, 161
———, impurities in, 2718
——— ore, 2562
———, painting on, 2803
——— roofing, 420
———, sheet, trade gauge, 11
———, weight of, 138
——— tensile strength of, 170
——— water pipes, corrosion of, 420

THE ENGINEER'S YEAR-BOOK ALPHABETICAL DIRECTORY OF ADVERTISERS' NAMES, WITH ADDRESSES, 1931.

INDEX ALPHABÉTIQUE DES ANNONCEURS, AVEC LEURS ADRESSES, 1931.

(Pour le Répertoire analytique des spécialités annoncées voir pages 3001 à 3039.)

INDICE ALFABETICO DELLE DITTE ANNUNZIATRICI COI LORO INDIRIZZI, 1931.

(Per la Guida classificata delle specialità annunziate vedansi pagine 3001 a 3039.)

GUÍA ALFABÉTICA DE NOMBRES DE ANUNCIADORES CON SUS SEÑAS, 1931.

(Véase la Guía clasificada de artículos especiales de los anunciadores en las páginas 3001 á 3039.)

АЛФАВИТНЫЙ УКАЗАТЕЛЬ ФИРМЪ ЗА 1931 ГОДЪ.

(Подробный Указатель Спеціальностей анонсодателей—см. стр. 3001–3039.)

PAGE

Allen-Liversidge, Ltd. (with which are amalgamated Imperial Light, Ltd., and Dissolved Acetylene Co., Ltd.), Victoria Station House, Westminster, S.W. 1 *facing pages* 2762, 2763

Anderson, D., & Son, Ltd., Lagan Felt Works, Belfast; Roach Road Works, Old Ford, London, E. 3; and Park Road Works, Stretford, Manchester *facing pages* 316, 317

Anderson, Thomas, Ltd., 53-62 Derby Road, Canada Dock, Liverpool; and at London, Belfast, Glasgow, Southampton, Manchester, and Barrow ... *facing pages* 1224, 1225

FOR CLASSIFIED DIRECTORY OF ADVERTISERS' SPECIALITIES, SEE PP. 2999, &c.

PAGE

Asbestos Cement Building Products, Ltd., Trafford Park, Manchester *facing page* 324

Atlas Diesel Co., Ltd., New Oxford House, Hart Street, W.C. 1 ... *facing page* 1930

Auld, David, & Sons, Ltd., Whitevale Foundry, Rowchester Street, Glasgow, S.E. ... ... *facing page* 1671

Austin & Co., Dunn Street, Newcastle-on-Tyne ... ... *facing page* 801

Babcock & Wilcox, Ltd., Babcock House, Farringdon Street, London, E.C. 4 ; Renfrew and Dumbarton, Scotland ; and Oldbury, England ... ... *facing pages* 2398, 2399

Beldam Packing and Rubber Co., Ltd., 16 Gracechurch Street, London, E.C. 3. Works : Brentford, Middlesex ... ... *facing page* 1754

Bennis, Ed., & Co., Ltd., Little Hulton, Bolton, and 28 Victoria Street, Westminster, S.W. 1 ... ... *last grey page at end of book* 3041

Berk, F. W., & Co., Ltd., 1 Fenchurch Avenue, London, E.C.3. Works : Abbey Mills Chemical Works, Stratford, London, E.15 ; and Pentrepoth, near Swansea *facing page* 2257

Bingham, Charles, & Co., 34 Victoria Street, London, S.W. 1 *grey page in front of book* xxv

Blackstone & Co., Ltd., Rutland Works, Stamford ; and Aldwych House, Aldwych, London, W.C. 2 ... ... *facing page* 1920

Boby, Robert, Ltd., Bury St. Edmunds, England ... *grey page in front of book* xxxi

Booth, James, & Co. (1915), Ltd., Argyle Street, Nechells, Birmingham. London Office : Vickers House, Broadway, S.W. 1 ... ... *grey pages in front of book* xxi, xxii

Braby, Fredk., & Co., Ltd., 352-364 Euston Road, London, N.W. 1 *Buyers' Guide Section*

Brett's Patent Lifter Co., Ltd., Foleshill Works, Coventry *grey page in front of book* vii

British Aluminium Co., Ltd., Adelaide House, King William Street, London, E.C. 4. Stores : 22 Aldersgate Street, London, E.C. 1 ; and 21 Barwick Street, Birmingham ... ... *grey page in front of book* vi

British Industrial Gases, Ltd. (Chas. Bingham & Co.), 34 Victoria Street, London, S.W. 1 ... ... *grey page in front of book* xxv

British Metallic Packings Co. (1929), Ltd., 7 Princes Street, Westminster, London, S.W. 1 ... ... *facing page* 1751

British Power Railway Signal Co., Ltd., Caxton House, Westminster, London, S.W. 1. Works : Slough, Bucks. ... ... *grey page in front of book* xxviii

Brotherhood, Peter, Ltd., Peterborough ... ... *facing pages* 1921, 2378

Callender's Cable & Construction Co., Ltd., Hamilton House, Victoria Embankment, London, E.C. 4 ... ... *facing page* 2252

Carbic, Ltd., Walworth House, 51 Holborn Viaduct, London, E.C. 1 *facing page* 2775

FOR CLASSIFIED DIRECTORY OF ADVERTISERS' SPECIALITIES, SEE PP. 2999, &c.

PAGE

Carter, J., Sons & Co., Ltd., New Bailey Street, Salford, Manchester *grey page in front of book* xi

'Chemist & Druggist,' The, 28 Essex Street, London, W.C. 2 ... *facing page* 2256

Clarke, Chapman & Co., Ltd., Victoria Works, Gateshead-on-Tyne; 50 Fenchurch Street, London, E.C. 3; and at Glasgow, Manchester, Cardiff ... *facing pages* 2134, 2135

Clarkson Thimble Tube Boiler Co., Ltd., 613 Australia House, Strand, London, W.C. 2 ... ... *facing page* 2197

Coan, Robert W., Ltd., Coan House, Duncan Street, London, N. 1; Works: 219 Goswell Road, London, E.C. 1 ... ... *facing page* 2734

Cochran & Co., Annan, Ltd., Annan, Scotland; and 137 Victoria Street, Westminster, London, S.W. 1 ... ... *facing page* 1582

Consolidated Brake & Engineering Co., Ltd., Caxton House, Westminster, London, S.W. 1; Works: Slough, Bucks ... ... *grey page in front of book* xxvii

Consolidated Pneumatic Tool Co., Ltd., Egyptian House, 170 Piccadilly, London, W. 1; Works: Fraserburgh, Scotland ... ... *facing pages* 1043, 1079

Copes Regulators, Ltd., 231 Strand, London, W.C. 2 ... ... *facing page* 1678

Cork Manufacturing Co., Ltd., Folly Lane, South Chingford, London, E. 4 *facing page* 1541

Coventry Chain Co., Ltd., Spon End Works, Coventry, Eng.; 47 Victoria Street, Westminster, London, S.W. 1; and 308 Deansgate, Manchester *grey page in front of book* x

Craigpark Electric Cable Co., Ltd., Flemington Street, Springburn, Glasgow; 17 Victoria Street, Westminster, S.W. 1; and 49 Deansgate, Manchester *facing page* 2211

Crockatt, W., & Sons, Ltd., 64 Darnley Street, Glasgow; 12 Bath Street, Liverpool; and 50 Whitechapel Road, London, E. 1 ... *grey page in front of book* xxix

Crosby, Lockwood & Son, 7 Stationers' Hall Court, London, E.C. 4 *facing pages* 1042, 1679, 2269

Dampney, J. & Co., Ltd., Britannic Works, Newcastle-on-Tyne ... *facing page* 1624

Delta Metal Company, Ltd., Delta Works, East Greenwich, London, S.E. 10; and Dartmouth Street, Birmingham ... ... *facing page* 171

Demag Aktiengesellschaft, Duisburg ... ... *inside back cover* 3042, 3043

Dick, R. & J., Ltd., Greenhead Works, Glasgow; and 46 Watling Street, London, E.C. 4 ... ... *facing pages* 855, 884, 885

Dickinson, F. E., St. Andrew's House, Holborn Circus, London, E.C. 1 *grey page in front of book* xxiii

District Chemical Co., Ltd., 106 Fenchurch Street, London, E.C. 3 *facing page* 2769

Dorman, Long & Co., Ltd., Middlesbrough, and 4 Central Buildings, Westminster, London, S.W. 1 ... ... *facing page* 247

FOR CLASSIFIED DIRECTORY OF ADVERTISERS' SPECIALITIES, SEE PP. 2999, &c.

PAGE

Drayton Regulator & Instrument Co., Ltd. (formerly **Steam Fittings Co., Ltd.**), West Drayton, Middlesex ... ... *facing page* 1670

'Electrical Review,' The, 4 Ludgate Hill, London, E.C. 4 *grey page in front of book* xxx

Engineer,' The, 28 Essex Street, Strand, London, W.C. 2 ... *facing pages* 795, 1593, 2686

English Steel Corporation, Ltd., Vickers Works, Sheffield ... *Buyers' Guide Section*

Ferro-Arc Welding Co., Ltd., 12, Grosvenor Gardens, Westminster, London, S.W. 1 *facing page* 2786

Ferrox Company, Ltd., 12 Grosvenor Gardens, London, S.W. 1. Works: Wolverhampton ... ... *facing page* 2768

François Cementation Co., Ltd., Bentley Works, Doncaster. Registered Office: 701 Salisbury House, London Wall, London, E.C. 2 ... *grey page in front of book* v

Fry's Metal Foundries, Ltd., 25/42 Holland Street, Blackfriars, London, S.E. 1; and at Manchester, Bristol, Glasgow, and Dublin ... ... *facing page* 2757

Glenfield & Kennedy Ltd., Kilmarnock ... ... *facing pages* 1270, 1271

Gleniffer Engines, Ltd., Temple Works, Annieland, Glasgow ... *facing page* 1931

Graphite Products, Ltd., 218-220 Queen's Road, Battersea, London, S.W. 8 *grey page in front of book* xxxii

Guest & Chrimes, Ltd., Rotherham. London Address: Cecil Chambers, 76 Strand, W.C. 2 ... ... *facing page* 1435

Habershon, J. J., & Sons, Ld., Holmes Mills, Rotherham ... ... *facing page* 1983

Hart, Thomas, Ltd., Lambeth Rope Works, Blackburn ... ... *facing page* 879

Heenan & Froud Ltd., Worcester ... ... *facing pages* 2267, 2268

Henderson & Glass, Vulcan Street, Liverpool ... ... *grey page in front of book* xxxiii

Hendry, James, Ltd., 252 Main Street, Bridgeton, Glasgow; 16 Union Court, Old Broad Street, E.C.2; Empire House, Gt. Charles Street, Birmingham; and Consolidated Buildings, Johannesburg ... ... *between pages* 860, 861

Henley's, W. T., Telegraph Works Co., Ltd., 11 Holborn Viaduct, London, E.C. 1; North Woolwich, E. 16; and Gravesend, Kent *grey page facing title-page* xli

Heriot-Watt College, Edinburgh ... ...*grey page in front of book* xxxviii

Hoffmann Manufacturing Co., Ltd., New Street, Chelmsford, Essex. Branches: London, Birmingham, Manchester, Leeds, and Glasgow ... ... *facing page* 961

Hopkinsons, Ltd., Britannia Works, Huddersfield; 34 Norfolk Street, Strand, London, W.C. 2 ... ... *facing pages* 1668, 1669

FOR CLASSIFIED DIRECTORY OF ADVERTISERS' SPECIALITIES, SEE PP. 2999, &c.

PAGE

Howden, James, & Co., Ltd., 195 Scotland Street, Glasgow, C. 5 ; Fenton House, 112 Fenchurch Street, London, E.C. 3 ; Caxton House, Westminster, S.W. 1 ; Cunard Building, Liverpool ; 86 Cross Street, Manchester ; Milburn House, Newcastle-on-Tyne ... ... *facing page* 2392

Howden-Ljungström Preheaters (Land), Ltd., Caxton House, Westminster, London, S.W. 1 ; 195 Scotland Street, Glasgow ; 86 Cross Street, Manchester ; and Milburn House, Newcastle-on-Tyne ... ... *facing page* 2393

Humphreys, J. H., & Sons, Horsedge Electrical Works, Oldham, England *facing page* 1050

Imperial Chemical Industries, Ltd., Millbank, London, S.W. 1 *grey page in front of book* iv

Imperial College of Science and Technology, South Kensington, London, S.W. 7 ... ... *grey page in front of book* xl

Incandescent Heat Co., Ltd., British Mills, Cornwall Road, Smethwick, Birmingham ... ... *facing page* 2659

International Combustion, Ltd., Aldwych House, Aldwych, London, W.C. 2. Works : Barrow-in-Furness, and Derby ... ... *facing pages* 2136, 2137

International Gas Detectors, Ltd., (Late W. R. Patents, Ltd.) Great Western Street, Leeds ... ... *grey page in front of book* xv

'Ironmonger,' The, 28, Essex Street, London, W.C. 2 ... ... *facing pages* 985, 1625, 2735

Johnson & Phillips, Ltd., Victoria Works, Charlton, S.E. 7 ; and 12 Union Court, Old Broad Street, London, E.C. 2 ... ... *grey page in front of book* xxxiv

Kent, George, Ltd., Biscot Road Works, Luton, Beds. ... ... *facing page* 1437

Kenyon, Wm., & Sons, Ltd., Chapel Field Works, Dukinfield, near Manchester ; and 167 Moorgate, London, E.C. 2 ... ... *facing page* 878

Klinger, Richard, Ltd., 120 Southwark Street, London, S.E. 1 ... *facing pages* 1662, 1663

Latch & Batchelor, Ltd., Hay Mills, Birmingham ... ... *facing page* 2609

Le Grand, Sutcliffe & Gell, Ltd., Southall, Middlesex ... ... *facing page* 1239

Leyland & Birmingham Rubber Co., Ltd., Leyland, near Preston, Lancs. ; 24-30 Duke Street, Aldgate, London, E. 1 ; 124 New Street, Birmingham ; 29 Brown Street, Manchester ; Cardiff, Glasgow, Edinburgh, Dublin ; Johannesburg, Calcutta, and Buenos Aires ... ... *grey page in front of book* ii

Ley's Malleable Castings, Co., Ltd., Derby ... ... *facing page* 2687

FOR CLASSIFIED DIRECTORY OF ADVERTISERS' SPECIALITIES, SEE PP. 2999, &c.

PAGE

Liptak Furnace Arches, Ltd., 38 Victoria Street, London, S.W. 1 *grey page in front of book* xvi

Liverpool Electric Cable Co., Ltd., Linacre Lane, Bootle, Liverpool *facing page* 2210

Liverpool Refrigeration Co., Ltd. (Associated with H. J. West & Co., Ltd.), Colonial House, Water Street, Liverpool; Swinton House, Gray's Inn Road, London, W.C. 1. Works: Coalbrookdale, Warrington, and Saxilby *facing page* 1209

Loughborough Technical College, Loughborough ... *grey page in front of book* xxxvii

'Machinery Market,' The, Wardrobe Chambers, 146A Queen Victoria Street, London, E.C. 4 ... ... *white page at end of book* 2641

Manchester College of Technology, Deansgate, Manchester *grey page in front of book* xxxix

Massey, B. & S., Ltd., Openshaw, Manchester; 28 Victoria Street, Westminster, S.W. 1; 110 Bothwell Street, Glasgow; Dock Street, Newport; and 93 Ann Street, Belfast ... ... *grey page in front of book* iii

'Metallurgist,' The, 28 Essex Street, London, W.C. 2 ... ... *facing pages* 170, 1051

Mirrlees Watson Co., Ltd., Scotland Street, Glasgow; and 7 Grosvenor Gardens, London, S.W. 1 ... ... *facing pages* 794, 1846

Morgan Bros. (Publishers), Ltd., 28 Essex Street, London, W.C. 2 *facing pages* 1436, 2774

Musgrave & Co., Ltd., St. Ann's Ironworks, Belfast; Sardinia House, Kingsway, London, W.C. 2; 42 Deansgate, Manchester; 24 Queen Street, Cardiff; and at Paris ... ... *facing page* 2122

Newalls Insulation Co., Ltd., Washington Station, Co. Durham ... *facing pages* 1696, 1697

Palatine Engineering Co., Ltd., Hawthorne Road, Bootle, near Liverpool *grey page in front of book* xxxv

Parsons Marine Steam Turbine Co., Ltd., Turbinia Works, Wallsend-on-Tyne *facing page* 2379

Patent Lightning Crusher Co., Ltd., 14A Rosebery Avenue, London, E.C. 1 *facing page* 2640

Peters, G. D., & Co., Ltd., Caxton House, Westminster, London, S.W. 1 *grey page in front of book* xiv

Potter, F. W., & Soar, Ltd., Phipp Street, Great Eastern Street, London, E.C. 2 *grey page in front of book* xxvi

Premier Cooler and Engineering Co., Ltd., Shalford, near Guildford, Surrey; and 3 & 5 Queen Street, Cheapside, London, E.C. 4 ... ... *facing page* 1852

Pulsometer Engineering Co., Ltd., Nine Elms Ironworks, Reading, Berks. London Office: 11 Tothill Street, Westminster, S.W. 1; and at Birmingham, Cardiff, Glasgow, Leeds, Liverpool, and Manchester ... ... *facing page* 1261

FOR CLASSIFIED DIRECTORY OF ADVERTISERS' SPECIALITIES, SEE PP. 2999, &c.

PAGE

Quasi-Arc Co., Ltd., 15 Grosvenor Gardens, London, S.W. 1
grey page in front of book xviii

Reddaway, F., & Co., Ltd., Pendleton, Manchester; and Bush House, Aldwych, London, W.C. 2 ... ... *facing pages* 858, 859

Renold, Hans, Ltd., Burnage Works, Didsbury, Manchester; Bush House, Aldwych, London, W.C. 2; Chamber of Commerce Building, Birmingham; 27 Blythswood Square, Glasgow; Leeds, Bristol, and Liverpool ... *facing page* 904

Robinson, L., & Co., London Chambers, Gillingham, Kent; and 300 Langside Road, Glasgow ... ... *facing page* 1085

Rotherham & Sons, Ltd., Coventry; London Office, 1 Holborn Circus, E.C. 1; also at La Chaux-de-Fonds, Switzerland ... ... *grey page in front of book* xxiv

Ruberoid Company, Ltd., Lincoln House, High Holborn, London, W.C. 1; and Brimsdown, Middlesex ... ... *facing page* 325

Saville, J. J., & Co., Ltd., Triumph Steel Works, Sheffield
grey page in front of book ix

Shardlow, Ambrose & Co., Ltd., Ealing Works, Sheffield
grey page in front of book xiii

Simon-Carves, Ltd., 20 Mount Street, Manchester ... ... *facing page* 1592

Simplex Conduits, Ltd., Garrison Lane, Birmingham; 113–117 Charing Cross Road, London, W.C. 2; Manchester, Glasgow, Newcastle, Liverpool, and Bristol *facing page* 2297

Sparklets, Ltd., Angel Road, Upper Edmonton, London, N. 18... ... *facing page* 2756

Spencer-Bonecourt, Ltd., Broadway Buildings, 50–64 Broadway, Westminster, London, S.W. 1. Works: Hitchin, Herts ... ... *facing page* 2196

Spencer-Hopwood, Ltd., Scotswood Works, Newcastle-upon-Tyne *facing page* 1600

Stein, John G., & Co., Ltd., Bonnybridge, Scotland ... ... *facing page* 425

Stirling Boiler Co., Ltd., 32–33 Farringdon Street, London, E.C. 4; and 45 Hope Street, Glasgow ... ... *facing page* 1583

Stone, J., & Co., Ltd., Deptford, London, S.E. 14; Oceanic House, 1A Cockspur Street, London, S.W. 1; and Branch Works, Charlton, London, S.E. 7 *facing 'Contents'*

Sulzer Bros., 31 Bedford Square, London, W.C. 1; and Manchester, Glasgow, and Swansea. Works: Winterthur, Switzerland ... *grey page in front of book* viii

Summerson, Thomas, & Sons, Ltd., Albert Hill Foundry, Darlington; 32 Victoria Street, Westminster, London, S.W. 1 ... ... *facing pages* 530, 531

FOR CLASSIFIED DIRECTORY OF ADVERTISERS' SPECIALITIES, SEE PP. 2999, &c.

PAGE

Superheater Company, Ltd., 195 Strand, London, W.C. 2 ... ... *facing page* 1618

Swan, Hunter & Wigham Richardson, Ltd., Wallsend-on-Tyne
grey page in front of book xvii

Tecalemit, Ltd., Mitre House, Scrubbs Lane, Willesden, London, N.W. 10
facing page 984

Thornycroft, John I., & Co., Ltd., Thornycroft House, Smith Square, Westminster, London, S.W. 1; Southampton and Basingstoke
grey page in front of book xxxvi

Tullis, John, & Son, Ltd., St. Ann's Leather Works, Glasgow; 5 Lloyd's Avenue, London, E.C. 3; 116 Queen Victoria Street, London, E.C. 4; and Manchester, Birmingham, and Leeds ... ... *facing pages* 864, 865

Underfeed Stoker Co., Ltd., Aldwych House, Aldwych, London, W.C. 2
facing pages 2118, 2119

United States Metallic Packing Co., Ltd., Soho Works, Allerton Road, Bradford, England; 116 Fenchurch Street, London, E.C. 3, and branches
facing page 1750

Variable Speed Gear, Ltd., 6 Broadway, Westminster, S.W. 1 ... *facing pages* 950, 1274

Walker, James, & Co., Ltd., Lion Works, Maybury Hill, Woking, Surrey (late Garford Street, West India Dock Road, London, E. 14); Liverpool, Glasgow, Newcastle, and Cardiff ... ... *facing pages* 2912, 2913

Wallsend Slipway & Engineering Co., Ltd., Wallsend-on-Tyne
grey page in front of book xix

Weir, G. & J., Ltd., Cathcart, Glasgow; 77 Billiter Buildings, Billiter Street, London, E.C. 3; and Bank Chambers, 26 Mosley Street, Newcastle-on-Tyne
facing page 1260

Welbecson Press, Ltd., 137 Blenheim Crescent, London, W. 11 ... *facing pages* 18, 19

Wells, A. C., & Co., Ltd., Carnarvon St., Cheetham, Manchester; and 4 Bradley's Bldgs., White Lion Street, London, N. 1 ... ...*grey page in front of book* xx

Weston Electrical Instrument Co., Ltd., 15 Gt. Saffron Hill, London, E.C. 1; and Newark, N.J., U.S.A. ... ... *facing page* 2312

West's Gas Improvement Co., Ltd., Albion Ironworks, Miles Platting, Manchester; London Office: Regent House, Kingsway, W.C. 2 ... *facing page* 2449

FOR CLASSIFIED DIRECTORY OF ADVERTISERS' SPECIALITIES, SEE PP. 2999, &c.

PAGE

Willcox, W. H., & Co., Ltd., 32-38 Southwark Street, London, S.E. 1 *first grey page inside front cover* i

Willesden Paper & Canvas Works, Ltd., Willesden Junction, London, N.W.10 *grey page in front of book* xii

Wrights' Ropes, Ltd., Universe Works, Birmingham; Salisbury House, London Wall, London, E.C. 2; Park Place, Cardiff; 15-17 Maclellan St., Glasgow *facing page* 2608

Yarrow & Co., Ltd., Scotstoun, Glasgow (formerly of Poplar, London) *facing page* 1601

FOR CLASSIFIED DIRECTORY OF ADVERTISERS' SPECIALITIES, SEE PP. 2999, &c.

'THE ENGINEER'S YEAR-BOOK' CLASSIFIED DIRECTORY OF LEADING ENGINEERING AND ALLIED FIRMS ADVERTISING IN THE 1931 EDITION.

A RELIABLE GUIDE FOR THE USE OF BUYERS OF ENGINEERING MATERIALS, MACHINERY, TOOLS, PLANT, APPLIANCES, ETC.

For Advertisers' Addresses, and Positions of their Announcements in the 'Year-Book,' consult foregoing Alphabetical List, pages 2989 to 2997.

The aim of the Proprietors of 'The Engineer's Year-Book' is to include in this publication only the most reputable firms' announcements, so that readers desiring to order, or specify, the various specialities advertised, may do so with confidence.

CLASSIFIED DIRECTORY OF ADVERTISERS' MANUFACTURES AND SPECIALITIES.

A RELIABLE GUIDE FOR BUYERS OF ENGINEERING MATERIALS, MACHINERY, TOOLS, PLANT, APPLIANCES, ETC.

Accumulator Acid.

Berk, F. W., & Co., Ltd.

Accumulators.

See BATTERIES.

Acetylene.

Allen-Liversidge, Ltd.
Bingham, Charles, & Co.
District Chemical Co., Ltd.

Acetylene Flares.

Braby, Fredk., & Co., Ltd. (Portable).
Wells, A. C., & Co., Ltd.

Acetylene Generating Plants.

Carbic, Ltd.
Ferrox Company, Ltd.

Admiralty Gunmetal Ingots.

Delta Metal Co., Ltd.

Aerial Grinders (Electric).

Humphreys, J. H., & Sons.

Aeroplane Accessories.

Cork Manufacturing Co., Ltd.
Ferrox Company, Ltd.

Agricultural Tools and Machinery.

Blackstone & Co., Ltd.

Air Compressors.

Brotherhood, Peter, Ltd.
Consolidated Pneumatic Tool Co., Ltd.
Demag Aktiengesellschaft.
Holland, Bernard, & Co.
Liverpool Refrigeration Co., Ltd. (High Pressure).
United States Metallic Packing Co., Ltd.

Air-Cooling and Humidifying Apparatus.

Heenan & Froud, Ltd.
Potter, F. W., & Soar, Ltd.
Premier Cooler & Engineering Co., Ltd.

Air Filters and Purifiers.

Heenan & Froud, Ltd.
Premier Cooler and Engineering Co., Ltd.

FOR ADVERTISERS' ADDRESSES AND POSITIONS OF THEIR ANNOUNCEMENTS, CONSULT FOREGOING ALPHABETICAL LIST, PAGES 2989 TO 2997.

Air Heaters and Dryers.

Howden, James, & Co., Ltd.
Howden-LjungstromPreheaters(Land) Ltd.
Incandescent Heat Co., Ltd.
Underfeed Stoker Co., Ltd.

See also HEATING & VENTILATING.

Air Hoists.

Consolidated Pneumatic Tool Co., Ltd.
United States Metallic Packing Co., Ltd.

Air Meters.

Demag Aktiengesellschaft.

Air Preheaters.

Howden-LjungstromPreheaters(Land) Ltd.

Air Pumps.

Clarke, Chapman & Co., Ltd.
Mirrlees Watson Co., Ltd. (Mirrlees Leblanc Rotary Air Pumps and Steam Ejector Air Pumps).
Rotherham & Sons, Ltd.
Weir, G. & J., Ltd.

Alloys.

Bingham, Charles, & Co.
Booth, James, & Co. (1915), Ltd., ('Duralumin,' Regd. Trade Mark, Vickers' Special Alloy).
British Aluminium Co., Ltd.
Coan, Robert W., Ltd. (Non-corrosive).
Delta Metal Co., Ltd.

Alternators.

Johnson & Phillips, Ltd.

Aluminium (Ingots, Sheets, Tube, Wire, Cable, Sections, etc.).

British Aluminium Co., Ltd.

Aluminium Castings and Fittings for All Trades.

Coan, Robert W., Ltd.
Fry's Metal Foundries, Ltd.
Sparklets, Ltd.

Aluminium Die Castings.

Coan, Robert W., Ltd.
Fry's Metal Foundries, Ltd.
Sparklets, Ltd.

Aluminium Powder.

Ferrox Company, Ltd.

Aluminium Vessels.

Braby, Fredk., & Co., Ltd.

Ammeters.

See INSTRUMENTS (ELECTRICAL).

Ammunition.

Imperial Chemical Industries, Ltd.

Analysis of Gases.

International Gas Detectors, Ltd. (Late 'W. R.' Patents, Ltd.)

Anti-friction Metals.

See ALLOYS.

Anti-vibration.

Cork Manufacturing Co., Ltd.

Arc Lamps.

See ELECTRIC LIGHT FITTINGS.

Arches (Furnace).
Liptak Furnace Arches, Ltd.

Artesian Well Engineers.
Le Grand, Sutcliffe & Gell, Ltd.

Asbestos Cement Corrugated Roofing Sheets.
Asbestos-Cement Building Products, Ltd.

Asbestos Cement Flat Sheets for Walls, Partitions, Ceilings, etc.
Asbestos Cement Building Products, Ltd.

Asbestos Cement Roofing Slates & Tiles.
Asbestos Cement Building Products, Ltd.

Asbestos Coverings & Mattresses.
Anderson, Thomas, Ltd.
Kenyon, Wm., & Sons, Ltd.

Asbestos Goods, Fittings, Packing, etc.
Anderson, Thomas, Ltd.
Beldam Packing & Rubber Co., Ltd.
Kenyon, Wm., & Sons, Ltd.
Klinger, Richard, Ltd.
Trist, Ronald, & Co., Ltd.
Walker, James, & Co., Ltd. ('Lion').
Willcox, W. H., & Co., Ltd.

Asbestos Packing ('Lion').
Walker, James, & Co., Ltd.

Automatic Feed Water Regulators.
Copes Regulators, Ltd.
Drayton Regulator & Instrument Co., Ltd.

Automatic Signals.
British Power Railway Signal Co., Ltd.

Balata Belting.
Dick, R. & J., Ltd. (Original Balata Brand).
Tullis, John, & Son, Ltd.

Balls and Ball Bearings.
Hoffmann Manufacturing Co., Ltd.

Barometers.
Heath & Co.

Barrows.
Braby, Fredk., & Co., Ltd. (Steel).

Bars, Non-Ferrous.
Delta Metal Co., Ltd.

Batteries (Electrical Storage).
Berk, F. W., & Co., Ltd. (Acids for).

Bearing Metals, etc.
Delta Metal Co., Ltd.
Fry's Metal Foundries, Ltd.
Shardlow, Ambrose, & Co., Ltd.

Bearings (Ball and Roller).
Hoffmann Manufacturing Co., Ltd.

Belt and Rope Pulleys.
Hick, Hargreaves & Co., Ltd.

Belting, etc.—Leather, Cotton, Hair, India Rubber, etc.
Dick, R. & J., Ltd. (Original Balata).
Hendry, James, Ltd. (Leather).

FOR ADVERTISERS' ADDRESSES AND POSITIONS OF THEIR ANNOUNCEMENTS, CONSULT FOREGOING ALPHABETICAL LIST, PAGES 2989 TO 2997.

Belting, etc.—Leather, Cotton, Hair, India Rubber, etc.— *continued.*

Henley's, W. T., Telegraph Works Co., Ltd.
Leyland and Birmingham Rubber Co., Ltd.
Reddaway, F., & Co., Ltd.
Tullis, John, & Son, Ltd.
Walker, James, & Co., Ltd.
Willcox, W. H., & Co., Ltd.

Biflex Chains for Automobile Camshaft Drives.

Coventry Chain Co., Ltd.

Bitumen Damp-proof Sheeting.

Ruberoid Co., Ltd.

Blast-Furnace Plant.

Demag Aktiengesellschaft.
Incandescent Heat Co., Ltd.

Blast-Furnace Tuyeres.

Palatine Engineering Co., Ltd.

Blasting Powder.

Imperial Chemical Industries, Ltd.

Blowers.

See FANS.

Blowpipes (Acetylene Welding & Cutting).

Ferrox Company, Ltd.

Boiler Discrustants.

Dampney, J. & Co., Ltd.

Boiler and Pipe Coverings.

Anderson, Thomas, Ltd.
Bell's United Asbestos Co., Ltd.
Kenyon, Wm., & Sons, Ltd.
Newalls Insulation Co., Ltd.

Boiler Cleaners (Mechanical and Electrical).

Crockatt, W., & Sons, Ltd.
Demag Aktiengesellschaft.
Hulburd Patents, Ltd.
Willcox, W. H., & Co., Ltd.

Boiler Composition (for inside use).

Dampney, J., & Co., Ltd.

Boiler Feed Filters.

See FEED WATER.

Boiler Feed Pumps.

Clarke, Chapman & Co., Ltd.
Pulsometer Engineering Co., Ltd.
Royles, Ltd.
Sulzer Bros.
Weir, G. & J., Ltd.

Boiler Fittings.

See ENGINE AND BOILER FITTINGS.

Boiler Furnaces.

Bennis, Ed., & Co., Ltd.
International Combustion, Ltd.
Superheater Co., Ltd. (Tubes, etc.).
Underfeed Stoker Co., Ltd.

Boiler and Furnace Insulators.

Newalls Insulation, Ltd.

Boiler Manufacturers.

Babcock & Wilcox, Ltd.
Clarke, Chapman & Co., Ltd.
Clarkson Thimble Tube Boiler Co., Ltd.
Cochran & Co. (Annan), Ltd.
Davey, Paxman & Co., Ltd.

FOR ADVERTISERS' ADDRESSES AND POSITIONS OF THEIR ANNOUNCEMENTS, CONSULT FOREGOING ALPHABETICAL LIST, PAGES 2989 TO 2997.

Boiler Manufacturers— *continued.*

Howden, James, & Co., Ltd.
Simon-Carves, Ltd.
Spencer-Bonecourt, Ltd.
Spencer-Hopwood, Ltd.
Stirling Boiler Co., Ltd.
Sulzer Bros.
Swan, Hunter & Wigham Richardson, Ltd. (Marine).
Thornycroft, John I., & Co., Ltd.
Wallsend Slipway & Engineering Co., Ltd. (Multitubular & Water-tube).
Ward, Thos. W., Ltd.
Yarrow & Co., Ltd.

Boiler Mountings.

Auld, David, & Sons, Ltd.
Cochran & Co. (Annan), Ltd.
Hopkinsons, Ltd.

See also ENGINE AND BOILER FITTINGS.

Boiler Preservatives.

Dampney, J., & Co., Ltd.

Boiler Tubes.

Booth, James, & Co. (1915), Ltd.

Boilers (Gas Fired).

Clarkson Thimble Tube Boiler Co., Ltd.

Boilers (Waste Heat & Silencer).

Clarkson Thimble Tube Boiler Co., Ltd.

Bolts and Nuts (Iron, Steel, Copper, Naval Brass, etc.).

Bull's Metal and Melloid Co., Ltd.
Davis & Timmins, Ltd.

Books and Journals, Scientific.

'Electrical Review' (The).
'Engineer' (The).
Griffin, Charles, & Co., Ltd.
'Ironmonger' (The).
Lockwood (Crosby) & Son.
'Machinery Market' (The).

Borehole (Deep Well) Pumps.

Le Grand, Sutcliffe & Gell, Ltd.
Sulzer Bros.

Boring Machines for Rock and Coal.

Consolidated Pneumatic Tool Co., Ltd.
Demag Aktiengesellschaft.

Boring Plant (Well).

Le Grand, Sutcliffe & Gell, Ltd.

Brake Blocks.

Summerson, Thomas, & Sons, Ltd.

Brakes (Electro-Mechanical).

Austin & Co.
Elliston, Evans & Jackson, Ltd.

Brakes (Railway).

Consolidated Brake & Engineering Co., Ltd.

Brass and Bronze.

Booth, James, & Co. (1915), Ltd.
Delta Metal Co., Ltd.
Ferrox Company, Ltd.
Imperial Chemical Industries, Ltd.
Kynoch, Ltd.

Brass and Copper Bolts and Nuts.

See BOLTS AND NUTS.

Brass Founders and Finishers.

Guest & Chrimes, Ltd.

Bridge Builders.

Demag Aktiengesellschaft.
Henderson & Glass.

'BRITIMP' Metallic Packings.

British Metallic Packings Co. (1929), Ltd.

Bronze.

See Brass and Bronze.

Bush Valves for Air Pumps.

Beldam Packing and Rubber Co., Ltd.

Cable Laying Machinery.

Clarke, Chapman & Co., Ltd.
Johnson & Phillips, Ltd.

Cables: Telegraph, Telephone, Electric Light, and Submarine.

British Aluminium Co., Ltd.
Callender's Cable and Construction Co., Ltd.
Craigpark Electric Cable Co., Ltd.
Henley's, W. T., Telegraph Works Co., Ltd.
Johnson & Phillips, Ltd.
Liverpool Electric Cable Co., Ltd.

'Cab Tyre Sheathed' Cables.

Callender's Cable and Construction Co., Ltd.
Henley's, W. T., Telegraph Works Co., Ltd.
Johnson & Phillips, Ltd.

Calorimeters for Fuel Testing.

Drayton Regulator & Instrument Co., Ltd.

'Camel' Hair, 'Camel' Brand Belting (sole makers of).

Reddaway, F., & Co., Ltd.

Carbide of Calcium.

Allen-Liversidge, Ltd.
Bingham, Charles, & Co.
Braby, Fredk., & Co., Ltd.
District Chemical Co., Ltd.

Carbonizing Plant.

West's Gas Improvement Co., Ltd.

Case Hardening.

District Chemical Co., Ltd.

Castings and Forgings.

Boby, Robert, Ltd.
Coan, Robert W., Ltd. (Aluminium).
Delta Metal Co., Ltd. (Brass & Bronze).
English Steel Corpn., Ltd.
Ley's Malleable Castings Co., Ltd.
Sparklets, Ltd.
Stone, J., & Co., Ltd.
Summerson, Thomas, & Sons, Ltd. (Iron & Steel Castings).

For Advertisers' Addresses and positions of their announcements, consult foregoing Alphabetical List, pages 2989 to 2997.

Cement Machinery.
Howes, S.

Cement (Refractory).
Stein, John G. & Co., Ltd.

Cementation Specialities.
François Cementation Co., Ltd.

Centre Grinders (Electric).
Humphreys, J. H., & Sons.

Centrifugal Pumps.
See PUMPS.

Chain Wheels.
Coventry Chain Co., Ltd.
Renold (Hans), Ltd.

Chains and Chain Driving.
Coventry Chain Co., Ltd.
Renold (Hans), Ltd.

Chasers—Machine.
Carter, J., Sons & Co., Ltd.

Chemical Plant and Machinery.
British Aluminium Co., Ltd.

Chemicals.
Berk, F. W., & Co., Ltd.
Bingham, Charles, & Co.
District Chemical Co., Ltd.
Imperial Chemical Industries, Ltd.

Chimney Stacks.
Babcock & Wilcox, Ltd.
Bennis, Ed., & Co., Ltd.
Braby, Fredk., & Co., Ltd.
Incandescent Heat Co., Ltd.

Chucks (Magnetic).
Humphreys, J. H., & Sons.

Clips ('Jubilee' Worm-Drive (Patent)).
Robinson, L. & Co.

Coal, Coke, and Ash Handling Plants.
Babcock & Willcox, Ltd.
Boby, Robert, Ltd.
Demag Aktiengesellschaft.
West's Gas Improvement Co., Ltd.

Coal Drills and Cutters.
Consolidated Pneumatic Tool Co., Ltd.
Demag Aktiengesellschaft.

Coal Economy Specialists.
International Combustion, Ltd.

Coal Furnaces.
See FURNACES.

Coal Grinding, Elevating, and Conveying Machinery.
Simon-Carves, Ltd.

Cocks (Brass) for Water, Steam, and Gas.
Guest & Chrimes, Ltd.
Rotherham & Sons, Ltd.

Coke Oven Plant.
Simon-Carves, Ltd.

Coke Screening & Grading Plant.
West's Gas Improvement Co., Ltd.

FOR ADVERTISERS' ADDRESSES AND POSITIONS OF THEIR ANNOUNCEMENTS, CONSULT FOREGOING ALPHABETICAL LIST, PAGES 2989 TO 2997.

CO_2 Recorders.

Underfeed Stoker Co., Ltd.
International Gas Detectors, Ltd. (late 'W. R.' Patents, Ltd.).

Cold Storage.

Anderson, Thomas, Ltd.
Brotherhood, Peter, Ltd.
Liverpool Refrigeration Co., Ltd.
Newalls Insulation, Ltd.
Ruberoid Co., Ltd.
Willesden Paper & Canvas Works, Ltd.

Collapsible Gates.

Potter, F. W., & Soar, Ltd.

Colleges (Engineering).

Heriot-Watt College, Edinburgh.
Imperial College of Science and Technology, London.
Loughborough Technical College.
Manchester College of Technology.

Collieries (Steel Wire Ropes for).

Latch & Batchelor, Ltd.
Wrights' Ropes, Ltd.

Colliery Plant.

Bennis, Ed., & Co., Ltd.
Clarke, Chapman & Co., Ltd.
Demag Aktiengesellschaft.
Hart, Thomas, Ltd.
Henley's, W. T., Telegraph Works Co., Ltd.
Kenyon, Wm., & Sons, Ltd.
Liverpool Electric Cable Co., Ltd.
Patent Lightning Crusher Co., Ltd.
Simon-Carves, Ltd.
Spencer-Bonecourt, Ltd.

'COLOUR-TIPT' Welding Metals.

Peters, G. D. & Co., Ltd.

Colours, Paints, and Varnishes.

Imperial Chemical Industries, Ltd.

Composition Flooring.

Asbestos Cement Building Products, Ltd.

Condenser Tubes (Copper).

See COPPER TUBES.

Condensers, Evaporators, Water Softening and Distilling Apparatus, etc.

Babcock & Wilcox, Ltd.
Clarke, Chapman & Co., Ltd.
Crockatt, W., & Sons, Ltd.
Davey, Paxman & Co., Ltd.
Mirrlees Watson Co., Ltd.
Weir, G. & J., Ltd.

Conduits.

See 'STEEL CONDUITS' and 'WIRING SYSTEMS.'

Constructional Engineers.

Babcock & Wilcox, Ltd.
Bennis, Ed., & Co., Ltd.
Braby, Fredk., & Co., Ltd.
Dorman, Long, & Co., Ltd.
Henderson & Glass.
Incandescent Heat Co., Ltd.

Consulting Engineers.

François Cementation Co., Ltd.

Contractors.

François Cementation Co., Ltd.

Contractors' Tools and Plant.

Consolidated Pneumatic Tool Co., Ltd.
Holland, Bernard, & Co.

Conveyors.

Babcock & Wilcox, Ltd.
Bennis, Ed., & Co., Ltd.
Boby, Robert, Ltd.
Hart, Thomas, Ltd.
Reddaway, F., & Co., Ltd.
Tullis, John, & Son, Ltd.
Underfeed Stoker Co., Ltd.
West's Gas Improvement Co., Ltd.

See also BELTING.

Cooling Towers.

Brotherhood, Peter, Ltd.
Demag Aktiengesellschaft.
Premier Cooler and Engineering Co., Ltd.

Copper.

Booth, James, & Co. (1915), Ltd.

Copper, Brass, & Gunmetal Tubes.

Booth, James, & Co. (1915), Ltd.

Copper Forgings.

Delta Metal Co., Ltd.

Copper Rods and Sections.

Delta Metal Co., Ltd.

Copper Smelters and Wire Manufacturers.

Delta Metal Co., Ltd.
Shropshire Iron Co., Ltd.

Coppersmiths.

Stone, J., & Co., Ltd.

'CORK' Packings, Gaskets, and Washers.

Cork Manufacturing Co., Ltd.

Corrugated Iron.

Braby, Fredk., & Co., Ltd.

Corrugated Tiles and Sheets (Asbestos Cement).

Asbestos Cement Building Products, Ltd.

Cotton Belting.

Reddaway, F., & Co., Ltd.
Tullis, John, & Son, Ltd.
Willcox, W. H., & Co., Ltd.

Cotton Driving Ropes.

Hart, Thomas, Ltd. ('Lambeth').
Kenyon, Wm., & Sons, Ltd. (Patent Inter-Stranded).

Counting Instruments.

Dickinson, F. E.

Couplings (Flexible).

See FLEXIBLE COUPLINGS.

Coventry Chain Couplings.

Coventry Chain Co., Ltd.

Coverings for Boilers, etc.

Kenyon, Wm., & Sons, Ltd.

Cranes.

See LIFTING AND HOISTING MACHINERY.

Crude Oil Engines.

See OIL ENGINES.

Crushing and Pulverising Machinery.

International Combustion, Ltd.
Patent Lightning Crusher Co., Ltd.

Cylinders.
Allen-Liversidge, Ltd.

Damp Course & Bitumen Sheeting.
Ruberoid Co., Ltd.

'Delta' Brand Metals and Alloys.
Delta Metal Co., Ltd.

Demagnetisers.
Humphreys, J. H., & Sons.

Diamond Drillers.
François Cementation Co., Ltd.

Die Castings.
Coan, Robert W., Ltd. (Aluminium).
Fry's Metal Foundries, Ltd.
Dorman, W. H., & Co., Ltd.
Sparklets, Ltd.

Diesel and Semi-Diesel Engines.
Atlas Diesel Co., Ltd.
Brotherhood, Peter, Ltd.
Doxford, William, & Sons, Ltd.
Gleniffer Engines, Ltd.
Holland, Bernard, & Co.
Sulzer Bros.

Distilling Apparatus (Fresh Water).
Mirrlees Watson Co., Ltd.

Distribution Boards.
Simplex Conduits, Ltd.

'Dixtampo' Yellow Metal Rods for Hot Stamping and Forging.
Delta Metal Co., Ltd.

'Dixtrudo' High-speed Brass Screwing and Turning Rods.
Delta Metal Co., Ltd.

Docks (Floating).
Swan Hunter & Wigham Richardson, Ltd.

Draught Gauges.
Hopkinsons, Ltd.

Dredge Plant.
Clarke, Chapman & Co., Ltd.

Drilling Machines.
Clarke, Chapman & Co., Ltd.
Consolidated Pneumatic Tool Co., Ltd.
Humphreys, J. H., & Sons (Portable Electric).
United States Metallic Packing Co., Ltd. (Portable Electric).

Drilling Pillars.
Carter, J., Sons & Co., Ltd.

Drills (Twist).
Saville, J. J., & Co., Ltd. (High Speed Steel).
English Steel Corpn., Ltd.

Driving Bands, Ropes, etc.
Hart, Thomas, Ltd.
Henley's, W. T., Telegraph Works Co., Ltd.
Kenyon, Wm. & Sons, Ltd.
Reddaway, F., & Co., Ltd.
Tullis, John, & Son, Ltd.
Willcox, W. H., & Co., Ltd.

Driving Chains.
Coventry Chain Co., Ltd.
Renold (Hans), Ltd.

Drop Stamps and Presses.
Bretts Patent Lifter Co., Ltd.
Massey, B. & S., Ltd.

For Advertisers' Addresses and positions of their announcements, consult foregoing Alphabetical List, pages 2989 to 2997.

Drying Apparatus.
Musgrave & Co., Ltd.
Potter, F. W., & Soar, Ltd.

'DU-EX' Double Expansion Nipples.
Stone, J., & Co., Ltd.

Duff Jacks.
Consolidated Pneumatic Tool Co., Ltd.

'Duralumin' (Regd. Trade Mark, Vickers' Special Alloy).
Booth, James, & Co. (1915), Ltd.

Dyestuffs.
Imperial Chemical Industries, Ltd.

Dynamite.
Imperial Chemical Industries, Ltd.

Dynamo Brushes
Potter, F. W., & Soar, Ltd.

Dynamos and Motors.
Clarke, Chapman & Co., Ltd.
Johnson & Phillips, Ltd.

Dynamometers.
Heenan & Froud, Ltd.

Economisers (Fuel).
Babcock & Wilcox, Ltd.
Bennis, Ed., & Co., Ltd.
Howden-Ljungström Preheaters (Land) Ltd.
International Gas Dectectors (late 'W. R.' Patents, Ltd.).
Spencer-Bonecourt, Ltd.

Economisers (Fuel)—*continued.*
Vickers Boiler Co., Ltd.

Educational Institutions.
See COLLEGES and SCHOOLS.

Electric Arc Cutting, and Welding Plant and Accessories.
Imperial Chemical Industries, Ltd.

Electric Cables.
See CABLES: TELEGRAPH, TELEPHONE, ELECTRIC LIGHT AND SUBMARINE.

Electric Cables ('C.T.S.' Type).
See 'CAB TYRE SHEATHED' CABLES.

Electric Fans.
See FANS.

Electric Furnaces.
Demag Aktiengesellschaft.
Imperial Chemical Industries, Ltd.
Incandescent Heat Co., Ltd.

Electric Generators.
See DYNAMOS AND MOTORS.

Electric Heaters.
Consolidated Pneumatic Tool Co., Ltd. (for Rivets and Tyres).

Electric Lifts and Cranes.
Austin & Co.
Babcock & Wilcox, Ltd.
Clarke, Chapman & Co., Ltd.
Demag Aktiengesellschaft.

FOR ADVERTISERS' ADDRESSES AND POSITIONS OF THEIR ANNOUNCEMENTS. CONSULT FOREGOING ALPHABETICAL LIST, PAGES 2989 TO 2997.

Electric Light Fittings and Electrical Accessories.

Callender's Cable and Construction Co., Ltd.
Clarke, Chapman & Co., Ltd.
Craigpark Electric Cable Co., Ltd.
Guest & Chrimes, Ltd. (Incandescent Lamps).
Henley's, W. T., Telegraph Works Co., Ltd.
Johnson & Phillips, Ltd.
Liverpool Electric Cable Co., Ltd.
Potter, F. W., & Soar, Ltd.
Robinson, L., & Co.
Simplex Conduits, Ltd.

Electric Lighting Contractors.

Clarke, Chapman & Co., Ltd.
Craigpark Electric Cable Co., Ltd.
Henley's, W. T., Telegraph Works Co., Ltd.
Johnson & Phillips, Ltd.
Liverpool Electric Cable Co., Ltd.
Weston Electrical Instrument Co., Ltd.

Electric Motors.

See Dynamos and Motors.

Electric Portable Drills and Hammers.

Consolidated Pneumatic Tool Co., Ltd.
Humphreys, J. H., & Sons.
United States Metallic Packing Co., Ltd.

Electric Railways and Tramways.

Callender's Cable and Construction Co., Ltd.
Clarke, Chapman & Co., Ltd.

Electric Railways and Tramways—*continued.*

Craigpark Electric Cable Co., Ltd.
Henley's, W. T., Telegraph Works Co., Ltd. (Electric Cables and Gear).
Liverpool Electric Cable Co., Ltd.

Electric Welding.

Braby, Fredk., & Co., Ltd.
Consolidated Pneumatic Tool Co., Ltd.
Ferro-Arc Welding Co., Ltd.
Ferrox Company, Ltd.
Imperial Chemical Industries, Ltd.
Quasi-Arc Co., Ltd.

Electric Wiring Systems.

See Wiring Systems (Flexible).

Electrical Counters.

Dickinson, F. E.

Electrical Engineers.

British Power Railway Signal Co., Ltd. (Signals).
Callender's Cable and Construction Co., Ltd.
Clarke, Chapman & Co., Ltd.
Craigpark Electric Cable Co., Ltd.
Henley's, W. T., Telegraph Works Co., Ltd.
Johnson & Phillips, Ltd.
Liverpool Electric Cable Co., Ltd.
Siemens Bros. & Co., Ltd.
Simplex Conduits, Ltd.
Weston Electrical Instrument Co., Ltd.

For Advertisers' Addresses and positions of their announcements, consult foregoing Alphabetical List, pages 2989 to 2997.

Electrical Instruments & Apparatus.

See INSTRUMENTS AND APPARATUS (ELECTRICAL).

Electrical Storage Cells and Batteries.

See STORAGE CELLS.

Electrodes (for Electric Arc Welding).

Consolidated Pneumatic Tool Co., Ltd. (Weldrick's Patent).
Ferro-Arc Welding Co., Ltd.
Imperial Chemical Industries, Ltd.
Peters, G. D., & Co., Ltd.
Quasi-Arc Co., Ltd.

Elevators.

See LIFTING AND HOISTING.

Enamels, Colours, and Varnishes.

Imperial Chemical Industries, Ltd.

'Endural' Electric Cables.

Henley's, W. T., Telegraph Works Co., Ltd.

Engine and Boiler Fittings, Boiler Furnaces, Tubes, etc.

Babcock & Wilcox, Ltd.
Beldam Packing and Rubber Co., Ltd.
Bennis, Ed., & Co., Ltd. (Stokers).
Clarke, Chapman & Co., Ltd.
Crockatt, W., & Sons, Ltd.
Hopkinsons, Ltd.
Howden, James, & Co., Ltd.
Howden-LjungströmPreheaters(Land) Ltd.
Massey, B. & S., Ltd.

Engine and Boiler Fittings, Boiler Furnaces, Tubes, etc.—*continued.*

Mirrlees Watson Co., Ltd.
Spencer-Bonecourt, Ltd.
Spencer-Hopwood, Ltd.
Stirling Boiler Co., Ltd.
Superheater Company, Ltd.
Underfeed Stoker Co., Ltd.
United Kingdom Metallic Packing Syndicate, Ltd.
United States Metallic Packing Co., Ltd.
Walker, James, & Co., Ltd.
Weir, G. & J., Ltd.
Willcox, W. H., & Co., Ltd.

Engine House Telegraphs.

Hopkinsons, Ltd.

Engine Packing.

Anderson, Thomas, Ltd.
Beldam Packing & Rubber Co., Ltd.
British Metallic Packings Co. (1929), Ltd. ('Britimp').
Klinger, Richard, Ltd.
Leyland & Birmingham Rubber Co., Ltd.
Reddaway, F., & Co., Ltd.
United States Metallic Packing Co., Ltd.
Walker, James, & Co., Ltd. ('Lion').
Willcox, W. H., & Co., Ltd.
Willesden Paper & Canvas Works, Ltd.

Engine Packing ('Lion').

Walker, James, & Co., Ltd.

Engineering and Railway Supplies.

See RAILWAY AND TRAMWAY PLANT.

Engineering Training.

See COLLEGES.

Engineers (Electrical).

See ELECTRICAL ENGINEERS.

Engineers (Mechanical).

Imperial Chemical Industries, Ltd.
Shardlow, Ambrose & Co., Ltd.
Spencer-Bonecourt, Ltd.
Spencer-Hopwood, Ltd.
Walker, James, & Co., Ltd.

Engineers' Tools and Fittings.

Clarke, Chapman & Co., Ltd.
Consolidated Pneumatic Tool Co., Ltd.
Crockatt, W., & Sons, Ltd.
English Steel Corpn., Ltd.
Massey, B. & S., Ltd.
Shardlow, Ambrose & Co., Ltd.
United States Metallic Packing Co., Ltd.
Willcox, W. H., & Co., Ltd.

Engines (Steam).

See STEAM ENGINES.

Engravers and Printers.

Spottiswoode, Ballantyne & Co., Ltd.
Welbecson Press, Ltd.

Evaporators, Condensers, Water Distilling Apparatus, etc.

Babcock & Wilcox, Ltd.
Clarke, Chapman & Co., Ltd.
Mirrlees Watson Co., Ltd.
Spencer-Bonecourt, Ltd.
Spencer-Hopwood, Ltd.
Weir, G. & J., Ltd.

Evaporators for Sugar and Trade Liquors.

Mirrlees Watson Co., Ltd.

Excavators and Transporters.

Demag Aktiengesellschaft.

Exhaust Heat Boilers.

Clarkson Thimble Tube Boiler Co., Ltd.

Expanding Clips.

Robinson, L., & Co.

Explosives.

Imperial Chemical Industries, Ltd.

Extruded Metals.

Booth, James, & Co. (1915), Ltd.
British Aluminium Co., Ltd.
Delta Metal Co., Ltd.
Imperial Chemical Industries, Ltd.

Factories (Lighting of).

Allen-Liversidge, Ltd.
Simplex Conduits, Ltd.

Fans: Blowing, Exhaust, Electric, etc.

Howden, James, & Co., Ltd.
Musgrave & Co., Ltd.
Potter, F. W., & Soar, Ltd.

Feed-Water Controllers.

Copes Regulators, Ltd.
Hopkinsons, Ltd.

FOR ADVERTISERS' ADDRESSES AND POSITIONS OF THEIR ANNOUNCEMENTS, CONSULT FOREGOING ALPHABETICAL LIST, PAGES 2989 TO 2997.

Feed-Water Heaters, Filters and Purifiers.
Babcock & Wilcox, Ltd.
Clarke, Chapman & Co., Ltd.
Crockatt, W., & Sons, Ltd.
Weir, G. & J., Ltd.

Feed-Water (Measurement of).
Kent, George, Ltd.
Lea Recorder Co., Ltd.

Felt and Felt Roofs.
Anderson, D., & Son, Ltd.
Reddaway, F., & Co., Ltd.
Ruberoid Co., Ltd.
Willesden Paper & Canvas Works, Ltd.

Felts for Printing Machinery, Paper Making, and Calico Printing Machinery.
Reddaway, F., & Co., Ltd.

Fibre (Vulcanised).
Reddaway, F., & Co., Ltd.

File Manufacturers.
English Steel Corpn., Ltd.
Saville, J. J., & Co., Ltd.

Filtering Apparatus.
Mirrlees Watson Co., Ltd.
Pulsometer Engineering Co., Ltd.

Fire Bricks and Fire Clay Goods.
Stein, John G., & Co., Ltd.

Fire Extinguishing Appliances.
Guest & Chrimes, Ltd.
Reddaway, F., & Co., Ltd.

Flexible Couplings.
Dick, R. & J., Ltd.

Flue Covers and Linings.
Stein, John G. & Co., Ltd.

Fluxes for Oxy-acetylene Welding.
Thorn & Hoddle, Ltd.

'Foliac' Boiler Graphite.
Graphite Products, Ltd.

'Foliac' Graphite Pipe Joint Compound.
Graphite Products, Ltd.

'Foliac' Graphite Specialities.
Graphite Products, Ltd.

'Foliac' Silica-Graphite Paint.
Graphite Products, Ltd.

Forced and Induced Draught Apparatus.
Bennis, Ed., & Co., Ltd.
Howden, James, & Co., Ltd.
Musgrave & Co., Ltd.
Potter, F. W., & Co.

Forged Boiler Drums and Vessels.
English Steel Corpn., Ltd.

Forging Furnaces.
See FURNACES.

Forging Hammers.
Brett's Patent Lifter Co., Ltd.
Demag Aktiengesellschaft.
Massey, B. & S., Ltd.

Forging Presses.
Brett's Patent Lifter Co., Ltd.

Forgings and Castings.
See Castings and Forgings.

Founders.
Coan, Robert W., Ltd. (Aluminium).
Imperial Chemical Industries, Ltd.

Foundry Plant.
Willcox, W. H., & Co., Ltd.

Friction Clutches.
Reddaway, F., & Co., Ltd.

Fuel Economisers.
See Economisers.

Fuel Economy Systems.
Howden, James, & Co., Ltd.
Howden - Ljungström Preheaters (Land), Ltd.
International Combustion, Ltd.

Furnace Arches.
Liptak Furnace Arches, Ltd.

Furnaces (Boiler).
See Engine and Boiler Fittings.

Furnaces (Electric).
Demag Aktiengesellschaft.
Imperial Chemical Industries, Ltd.
Incandescent Heat Co., Ltd.

Furnaces (Gas and Oil Heated and Industrial).
Bennis, Ed., & Co., Ltd.
Howden, James, & Co., Ltd.
Incandescent Heat Co., Ltd.
Massey, B. & S., Ltd.

Galvanised Iron and Steel Manufacturers.
Braby, Fredk., & Co., Ltd.

Galvanisers.
Braby, Fredk., & Co., Ltd.
Wesson, W., & Co., Ltd.

Gas Engine Packing.
United States Metallic Packing Co., Ltd.

Gas Flue Pipes & Fittings.
Asbestos Cement Building Products, Ltd.

Gas Furnaces.
See Furnaces.

Gas Meters.
Guest & Chrimes, Ltd. (Wet and Dry).
Kent, George, Ltd.

Gas, Oil, and Petrol Engines.
Atlas Diesel Co., Ltd.
Bennis, Ed., & Co., Ltd.
Blackstone & Co., Ltd.
Brotherhood, Peter, Ltd.
Clarke, Chapman & Co., Ltd.
Clayton & Shuttleworth, Ltd.
Consolidated Pneumatic Tool Co., Ltd.
Demag Aktiengesellschaft.
Gleniffer Engines, Ltd.
Thornycroft, John I., & Co., Ltd.

For Advertisers' Addresses and positions of their announcements, consult foregoing Alphabetical List, pages 2989 to 2997.

Gas-Producers.

Bennis, Ed., & Co., Ltd.
Blackstone & Co., Ltd.
Demag Aktiengesellschaft.
Incandescent Heat Co., Ltd.

Gas Retorts.

Stein, John G., & Co., Ltd.

Gas Works Plant.

West's Gas Improvement Co., Ltd.

Gauge Glasses.

Crockatt, W., & Sons, Ltd.
Hopkinsons, Ltd.
Klinger, Richard, Ltd.

Gauge Glass Protectors.

Hopkinsons, Ltd.

Gauge Glass Rings.

Crockatt, W., & Sons, Ltd.
Hopkinsons, Ltd.
Walker, James, & Co., Ltd. ('Wallico' Cone).

Gauges.

See STEAM GAUGES.

Gear Boxes (Increasing & Reducing).

Coventry Chain Co., Ltd.
Demag Aktiengesellschaft.

Geared Turbine Auxiliaries.

Brotherhood, Peter, Ltd.
Parsons' Marine Steam Turbine Co., Ltd.

Geared Turbine Auxiliaries—*continued.*

Weir, G. & J., Ltd.

Gearing.

Demag Aktiengesellschaft.
Parsons' Marine Steam Turbine Co., Ltd.

Gears (Hydraulic, Oil).

Variable Speed Gear, Ltd.

General Foundry Castings.

Fry's Metal Foundries, Ltd.

Generators (Electric).

See DYNAMOS AND MOTORS.

Girders, Bridges, etc.

See STRUCTURAL IRON AND STEEL WORK.

Granulating Machinery.

Patent Lightning Crusher Co., Ltd.

Graphite.

Graphite Products, Ltd. ('Foliac').

'Graphol' Lubricant.

Peters, G. D., & Co., Ltd.

Grease (Lubricating).

Willcox, W. H., & Co., Ltd.

Grinders.

See GRINDING MACHINERY.

Grinding Machinery.

Consolidated Pneumatic Tool Co., Ltd.
Humphreys, J. H., & Sons (Portable Electric).
Patent Lightning Crusher Co., Ltd.
Renold (Hans), Ltd. (Mortise Chain Grinding Machinery).

Gunmetal Ingots.

Delta Metal Co., Ltd.

Hammers (Steam, Compressed Air, Pneumatic Power, etc.).

Brett's Patent Lifter Co., Ltd.
Consolidated Pneumatic Tool Co., Ltd.
Demag Aktiengesellschaft.
Massey, B. & S., Ltd.
United States Metallic Packing Co., Ltd.

Hand Cranes.

See LIFTING AND HOISTING MACHINERY, ETC.

Hauling Plant.

See WINDING AND HAULING.

Heat Treatment of Metals.

Incandescent Heat Co., Ltd.

Heating for Railway Carriages.

Westinghouse Brake and Saxby Signal Co., Ltd.

Heating, Ventilating, etc.

Musgrave & Co., Ltd.
Potter, F. W., & Soar, Ltd.
Spencer-Bonecourt, Ltd.
Sulzer Bros.

High-Speed Tool Steel.

English Steel Corpn., Ltd.
Sanderson Bros. & Newbould, Ltd.
Saville, J. J., & Co., Ltd.

Hoists.

See LIFTING AND HOISTING.

Hollow Rods and Sections (Brass, Copper, etc.).

Delta Metal Co., Ltd.

Hose Pipe Clips.

Robinson, L., & Co.
Rotherham & Sons, Ltd.

Hose for Fire Brigades, Mills, etc.

Leyland and Birmingham Rubber Co., Ltd.
Reddaway, F., & Co., Ltd.
Tullis, John, & Son, Ltd.

Hose Pipes, Leather and Canvas Hose, etc.

Reddaway, F., & Co., Ltd.
Tullis, John, & Son, Ltd.
Willcox, W. H., & Co., Ltd.
Willesden Paper & Canvas Works, Ltd.

Hot Brass Pressings.

Delta Metal Co., Ltd.

Hot Water Supply Boilers.

Royles, Ltd.

Hydraulic Gauges.

Hopkinsons, Ltd.

FOR ADVERTISERS' ADDRESSES AND POSITIONS OF THEIR ANNOUNCEMENTS, CONSULT FOREGOING ALPHABETICAL LIST, PAGES 2989 TO 2997.

Hydraulic Lifts.

See LIFTING AND HOISTING.

Hydraulic Machines, Tools, and Appliances, etc.

Massey, B. & S., Ltd.
Tangyes, Ltd.
Variable Speed Gear, Ltd.

Hydraulic Pumps.

Variable Speed Gear, Ltd.

Hydrochloric Acid.

Berk, F. W., & Co., Ltd.
Imperial Chemical Industries, Ltd.

Hydro-Electric Control Valves.

Glenfield & Kennedy, Ltd.

Ice-making Machinery.

See REFRIGERATING MACHINERY.

Illuminated Signs.

Simplex Conduits, Ltd.

India Rubber, Gutta Percha, and Bitumen Manufacturers.

Beldam Packing & Rubber Co., Ltd.
Callender's Cable & Construction Co., Ltd.
Craigpark Electric Cable Co., Ltd.
Henley's, W. T., Telegraph Works Co., Ltd.
Johnson & Phillips, Ltd.
Leyland & Birmingham Rubber Co., Ltd.
Liverpool Electric Cable Co., Ltd.
Macintosh Cable Co., Ltd.
Reddaway, F., & Co., Ltd.
Walker, James, & Co., Ltd. ('Lion').

Ingot Metals.

British Aluminium Co., Ltd.
Delta Metal Co., Ltd.

Injectors, Ejectors, and General Jet Appliances.

See STEAM INJECTORS.

Instruments and Apparatus (Electrical).

Coan, Robert W., Ltd.
Dickinson, F. E.
Johnson & Phillips, Ltd.
Weston Electrical Instrument Co., Ltd.

Instruments: Surveying, Mathematical, Drawing, etc.

Harling, W. H.

Insulating Materials.

Anderson, Thomas, Ltd.
Beldam Packing and Rubber Co., Ltd.
Callender's Cable & Construction Co., Ltd.
Craigpark Electric Cable Co., Ltd.
Henley's, W. T., Telegraph Works Co., Ltd.
Johnson & Phillips, Ltd.
Kenyon, Wm., & Sons, Ltd.
Newalls Insulation, Ltd.
Ruberoid Co., Ltd.
Willesden Paper & Canvas Works, Ltd.

Insulation Specialities.

Newalls Insulation, Ltd.

FOR ADVERTISERS' ADDRESSES AND POSITIONS OF THEIR ANNOUNCEMENTS, CONSULT FOREGOING ALPHABETICAL LIST, PAGES 2989 TO 2997.

Insulators.
Anderson, Thomas, Ltd.

Internal Combustion Engines.
See GAS, OIL, AND PETROL ENGINES.

Inverted Tooth Chains.
Coventry Chain Co., Ltd.

Iron and Steel Manufacturers and Merchants.
Braby, Fredk., & Co., Ltd. (Sheets).
Habershon, J. J., & Sons, Ltd.
Henderson & Glass.

Iron (Rustless).
Habershon, J. J., & Sons, Ltd.
English Steel Corpn., Ltd.
Saville, J. J., & Co., Ltd.

Ironfounders—Malleable.
Ley's Malleable Casting Co., Ltd.

Irrigation Appliances.
Glenfield & Kennedy, Ltd.

Jointing ('Beldamite').
Beldam Packing & Rubber Co., Ltd.

Jointing Cement.
Hopkinsons, Ltd. ('Metallic').
Kenyon, Wm., & Sons, Ltd. ('Kisol').

Jointing ('Klingerit').
Klinger, Richard, Ltd.

Jointing Materials.
Beldam Packing and Rubber Co., Ltd.
Cork Manufacturing Co., Ltd.
Graphite Products, Ltd.
Hopkinsons, Ltd. ('Koru').

Jointing Materials—*continued.*
Kenyon, Wm., & Sons, Ltd.
Klinger, Richard, Ltd.
Walker, James, & Co., Ltd. ('Lion' and 'Walkerite').
Willcox, W. H., & Co., Ltd.

Joints ('Lion' and 'Walkerite').
Walker, James, & Co., Ltd.

Joists (Steel).
Dorman, Long & Co., Ltd.
Henderson & Glass.

Journals (Technical).
See BOOKS AND JOURNALS.

'JUBILEE' Worm-Drive Clips (Patent).
Robinson, L., & Co.

'Kisol' Cement for Boilers, Steam Pipes, etc.
Kenyon, Wm., & Sons, Ltd.

Lamps (Electric).
See ELECTRIC LIGHT FITTINGS, ETC.

Lamps (Industrial).
Allen-Liversidge, Ltd.
Carbic, Ltd.
Wells, A. C., & Co., Ltd.

Lamps (Rear).
Coan, Robert W., Ltd.

Lathes.
See TOOL MAKERS.

Leather Belting.
See BELTING.

FOR ADVERTISERS' ADDRESSES AND POSITIONS OF THEIR ANNOUNCEMENTS, CONSULT FOREGOING ALPHABETICAL LIST, PAGES 2989 TO 2997.

Lift Enclosures.

Potter, F. W., & Soar, Ltd.

Lifting and Hoisting Machinery and Lifting Tackle.

Austin & Co.
Babcock & Wilcox, Ltd.
Boby, Robert, Ltd.
Carter, J., Sons & Co., Ltd. (Friction Hoists).
Clarke, Chapman & Co., Ltd.
Consolidated Pneumatic Tool Co., Ltd.
United States Metallic Packing Co., Ltd.

Lifting Magnets.

Demag Aktiengesellschaft.

Lighting and Heating (Acetylene Gas Plants for).

Allen-Liversidge, Ltd.
Bingham, Charles, & Co.
Carbic, Ltd.

Lighting (Train).

Carbic, Ltd.
Imperial Chemical Industries, Ltd.
Stone, J., & Co., Ltd.

Line Shaft Bearings (Ball and Roller).

Hoffmann Manufacturing Co., Ltd.

'Lion' Packing.

Walker, James, & Co., Ltd.

Loading and Transport Plants.

Demag Aktiengesellschaft.

Locomotive Superheated Packing.

United States Metallic Packing Co., Ltd.

Locomotive Superheaters.

Superheater Company, Ltd.

Locomotives.

Blackstone & Co., Ltd.

Lubricating Devices.

Rotherham & Sons, Ltd.
Tecalemit, Ltd. (High Pressure).

Lubricating Systems (High Pressure).

Tecalemit, Ltd.

Lubrication Experts.

Tecalemit, Ltd.

Lubricator Glasses.

Tomey, Joseph, & Sons, Ltd.

Lubricators and Lubricants.

Graphite Products, Ltd. ('Foliac').
Peters, G. D., & Co., Ltd. ('Graphol').
Rotherham & Sons, Ltd.
Tecalemit, Ltd.
United Kingdom Metallic Packing Syndicate, Ltd.
Willcox, W. H., & Co., Ltd.

Machine and Hand Tools.

See ENGINEERS' TOOLS AND FITTINGS, *also* TOOL MAKERS.

FOR ADVERTISERS' ADDRESSES AND POSITIONS OF THEIR ANNOUNCEMENTS, CONSULT FOREGOING ALPHABETICAL LIST, PAGES 2989 TO 2997.

Machine Output Recorders.
Dickinson, F. E.

Machinery Guards.
Braby, Fredk., & Co., Ltd.
Potter, F. W., & Soar, Ltd.

Magnetic Chucks.
Humphreys, J. H., & Sons.

Magnets.
English Steel Corpn., Ltd.

Malleable Castings.
Ley's Malleable Castings Co., Ltd.

Manganese Bronze.
Delta Metal Co., Ltd.

Marine Engines, Boilers, and Auxiliary Machinery.
Babcock & Wilcox, Ltd.
Clarke, Chapman & Co., Ltd.
Cochran & Co. (Annan), Ltd.
Howden, James, & Co., Ltd.
Mirrlees Watson Co., Ltd.
Spencer-Bonecourt, Ltd.
Superheater Company, Ltd.
Swan, Hunter & Wigham Richardson, Ltd.
Thornycroft, John I., & Co., Ltd.
Weir, G. & J., Ltd.
Yarrow & Co., Ltd.

Marine Glues.
Crockatt, W., & Sons, Ltd.

Marine Oil Engines.
Gleniffer Engines, Ltd.
Swan, Hunter & Wigham Richardson, Ltd.
Thornycroft, John I., & Co., Ltd.

Marine Superheaters.
Superheater Company, Ltd.

Measuring Instruments (Electrical).
See Instruments.

Mechanical Stokers.
Babcock & Wilcox, Ltd.
Bennis, Ed., & Co., Ltd.
Spencer-Bonecourt, Ltd.
Stirling Boiler Co., Ltd.
Underfeed Stoker Co., Ltd.
Vickers Boiler Co., Ltd.

Metal Cutting and Welding.
Allen-Liversidge, Ltd.
District Chemical Co., Ltd.
Ferro-Arc Welding Co., Ltd.
Ferrox Company, Ltd.
Imperial Chemical Industries, Ltd.
Peters, G. D., & Co., Ltd.

Metal Windows (Sashes & Casements).
Braby, Fredk., & Co., Ltd.

Metallic Packing.
See Engine Packing.

Metallic Packing ('Britimp').
British Metallic Packings Co. (1929), Ltd.

Metallic Packing ('Osmanite').
Klinger, Richard, Ltd.

Metallic Piston Rod Packing ('Lion').

Walker, James, & Co., Ltd.

Metallic Piston Rod Packing ('United States').

United States Metallic Packing Co., Ltd.

Metallurgical Engineers.

Ferro-Arc Welding Co., Ltd.

Metals: Bearing, Anti-friction, etc.

Booth, James & Co. (1915), Ltd. ('Duralumin,' Regd. Trade Mark, Vickers Special Alloy).
British Aluminium Co., Ltd.
Delta Metal Co., Ltd.
Fry's Metal Foundries, Ltd.
Imperial Chemical Industries, Ltd.
Stone, J., & Co., Ltd.
Thermit, Ltd.

Meters (Air, Gas, Steam, and Water).

Glenfield & Kennedy, Ltd.
Guest & Chrimes, Ltd.
Kent, George, Ltd.
Palatine Engineering Co., Ltd.

Micrometers.

Shardlow, Ambrose & Co., Ltd.

Mill Bandings and Twines.

Hart, Thomas, Ltd.
Kenyon, Wm., & Sons, Ltd.

Mill Gearing, Shafting, etc.

See SHAFTING AND GEARING.

Mills (Pulverizing).

Patent Lightning Crusher Co., Ltd.
Simon-Carves, Ltd.

Mine Signal Indicators.

Hopkinsons, Ltd.

Mineral Wool.

Kenyon, Wm., & Sons, Ltd.

Mining and Hauling Plant.

Babcock & Wilcox, Ltd.
Bennis, Ed., & Co., Ltd.
Clarke, Chapman & Co., Ltd.
Consolidated Pneumatic Tool Co., Ltd.
Demag Aktiengesellschaft.
Hart, Thomas, Ltd.
Henley's, W. T., Telegraph Works Co., Ltd. (Electrical Plant).
Kenyon, Wm., & Sons, Ltd.
Patent Lightning Crusher Co., Ltd.
Spencer-Bonecourt, Ltd.
United Flexible Metallic Tubing Co., Ltd.

Mining Cables and Accessories.

Callender's Cable and Construction Co., Ltd.
Henley's, W. T., Telegraph Works Co., Ltd.
Johnson & Phillips, Ltd.
Macintosh Cable Co., Ltd.
Robinson, L., & Co.

Mining Engineers.

François Cementation Co., Ltd.

Mining Explosives.

Imperial Chemical Industries, Ltd.

Mining Machinery.

See MINING AND HAULING PLANT.

Mining Tools.

Consolidated Pneumatic Tool Co., Ltd.
Demag Aktiengesellschaft.

Motor Car Accessories.

Booth, James & Co. (1915), Ltd. ('Duralumin' for, Registered Trade Mark, Vickers' Special Alloy).
Coan, Robert W., Ltd.
Imperial Chemical Industries, Ltd.
Robinson, L., & Co.
Rotherham & Sons, Ltd.

Motor Lubricants.

Graphite Products, Ltd.
Willcox, W. H., & Co., Ltd.

Motor Step Mats.

Coan, Robert W., Ltd.

Motor Vehicles (Commercial).

Thornycroft, John I., & Co., Ltd.

Multiple Effect Evaporators.

Mirrlees Watson Co., Ltd.

Name Plates for all Trades.

Coan, Robert W., Ltd.

Nitric Acid.

Berk, F. W., & Co., Ltd.

Non-Ferrous Metals.

Delta Metal Co., Ltd.

Number Plates.

Coan, Robert W., Ltd.

Oil Burning Installations.

Howden, James, & Co., Ltd.
Thornycroft, John I., & Co., Ltd.
Wallsend Slipway & Engineering Co., Ltd.

Oil Coolers.

Heenan & Froud, Ltd.
Incandescent Heat Co., Ltd.

Oil Eliminators.

Premier Cooler & Engineering Co., Ltd.

Oil Furnaces.

See FURNACES.

Oil, Gas, and Petrol Engines.

Bennis, Ed., & Co., Ltd.
Blackstone & Co., Ltd.
Brotherhood, Peter, Ltd.
Clarke, Chapman & Co., Ltd.
Clayton & Shuttleworth, Ltd.
Consolidated Pneumatic Tool Co., Ltd.
Gleniffer Engines, Ltd.
Sulzer Bros.
Thornycroft, John I., & Co., Ltd.
Wallsend Slipway & Engineering Co., Ltd.

Oil Heating & Brazing Plants.

Wells, A. C., & Co., Ltd.

Oil Locomotives (Internal Combustion).

Blackstone & Co., Ltd.

Oils, Lubricating Compounds, etc.

Graphite Products, Ltd.
Willcox, W. H., & Co., Ltd.

Overhead Cranes and Runways.

Demag Aktiengesellschaft.

Oxy-Acetylene Welding and Cutting.

Allen-Liversidge, Ltd.
Bingham, Charles, & Co.
Braby, Fredk., & Co., Ltd.
Carbic, Ltd.
Coan, Robert W., Ltd.
District Chemical Co., Ltd.
Ferro-Arc Welding Co., Ltd.
Ferrox Company, Ltd.

Oxygen.

Bingham, Charles, & Co.
Woolley, James, Sons & Co., Ltd.

Oxygen and Hydrogen Generating Plants.

Bingham, Chas., & Co.

Packing ('Britimp').

British Metallic Packings Co. (1929), Ltd.

Packing ('Langite' Oil Resisting).

Cork Manufacturing Co., Ltd.

Packing ('Lion').

Walker, James, & Co., Ltd.

Packing ('Pilot').

Beldam Packing & Rubber Co., Ltd.

Packing (Steam).

See ENGINE PACKING.

Packing ('United States').

United States Metallic Packing Co., Ltd.

Painting Machines.

Consolidated Pneumatic Tool Co., Ltd.
Wells, A. C., & Co., Ltd. (and Lime-washing Machines)

Paints, Colours, Enamels, and Varnishes.

Dampney, J., & Co., Ltd.
Imperial Chemical Industries, Ltd.

Paints for Preserving Iron and Steel Work.

Dampney, J., & Co., Ltd.
Graphite Products, Ltd.
Imperial Chemical Industries, Ltd.
Ruberoid Co., Ltd. ('Flexduro').

Paper Makers' Felts.

Reddaway, F., & Co., Ltd.

Paper (Waterproof).

Willesden Paper & Canvas Works, Ltd.

Perforated Metals.

Braby, Fredk., & Co., Ltd.

Petrol Engines.

See OIL AND GAS ENGINES.

Phosphor Bronze.

Delta Metal Co., Ltd.
Ferrox Company, Ltd.
Fry's Metal Foundries, Ltd.

Pipe Coils.

Liverpool Refrigeration Co., Ltd.

Pipe Coverings.
Kenyon, Wm., & Sons, Ltd.
Willcox, W. H., & Co., Ltd.

Pipe Scraping Appliances.
Glenfield & Kennedy, Ltd.

Pipes and Fittings.
See TUBES AND FITTINGS.

Piston Packing.
Walker, James, & Co., Ltd. ('Lion').

Planished Steel Sheets.
Braby, Fredk., & Co., Ltd.

Plaster.
Sankey, J. H., & Son, Ltd.

Plumbers' Brasswork.
Guest & Chrimes, Ltd.

Pneumatic Conveying Plants.
Boby, Robert, Ltd.

Pneumatic Tools and Appliances.
Consolidated Pneumatic Tool Co., Ltd.
Demag Aktiengesellschaft
Massey, B. & S., Ltd.
United States Metallic Packing Co., Ltd.

'Poilite' Asbestos Sheets and Roofing Tiles.
Asbestos Cement Building Products, Ltd.

Portable Electric Drills and Hammers.
See ELECTRIC PORTABLE DRILLS AND HAMMERS.

Portland Cement.
Sankey, J. H., & Son, Ltd.

Powdered Coal Plants.
Howden, James, & Co., Ltd.
International Combustion, Ltd.

Power Hammers.
See HAMMERS.

Power Presses.
Brett's Patent Lifter Co., Ltd.
Imperial Chemical Industries, Ltd.

Power Transmission Appliances.
Hart, Thomas, Ltd.
Hoffmann Manufacturing Co., Ltd.
Kenyon, Wm., & Sons, Ltd.
Reddaway, F., & Co., Ltd.

Preservatives (Boiler).
Dampney, J., & Co., Ltd.
Graphite Products, Ltd.

Preservatives (Wood).
Anderson, D., & Son, Ltd. ('Sidol').

Pressure Pipes for Water Mains.
Asbestos Cement Building Products, Ltd.

Printers, Lithographers, and Engravers.
Spottiswoode, Ballantyne & Co., Ltd.
Welbecson Press, Ltd.

FOR ADVERTISERS' ADDRESSES AND POSITIONS OF THEIR ANNOUNCEMENTS, CONSULT FOREGOING ALPHABETICAL LIST, PAGES 2989 TO 2997.

Producer-Gas Plants.
See GAS-PRODUCERS.

Propellers.
Premier Cooler and Engineering Co., Ltd. (Aeroplane).
Stone, J., & Co., Ltd.
Thornycroft, John I., & Co., Ltd.

Publications (Technical).
See BOOKS AND JOURNALS.

Pulleys.
See BELT AND ROPE PULLEYS.

Pulverised Fuel Equipment.
Howden, James, & Co., Ltd.
Underfeed Stoker Co., Ltd.
International Combustion, Ltd.
Simon-Carves, Ltd.

Pulverizer Mills.
Patent Lightning Crusher Co., Ltd.
Simon-Carves, Ltd.

Pump Buckets.
Crockatt, W., & Sons, Ltd.

Pumps and Pumping Machinery.
Beldam Packing and Rubber Co., Ltd. (Valves for).
Blackstone & Co., Ltd.
Carter, J., Sons, & Co., Ltd. (Test Pumps).
Clarke, Chapman & Co., Ltd.
Clayton & Shuttleworth, Ltd.
Consolidated Pneumatic Tool Co., Ltd. (Vacuum).
Davey, Paxman & Co., Ltd.
Holland, Bernard, & Co. (Vacuum).
Le Grand Sutcliffe & Gell, Ltd.
Mirrlees Watson Co., Ltd. (Vacuum).

Pumps and Pumping Machinery— *continued.*
Pulsometer Engineering Co., Ltd. (Vacuum and Centrifugal).
Stone, J., & Co., Ltd.
Sulzer Bros.
Weir, G. & J., Ltd.
Willcox, W. H., & Co., Ltd.

Pumps (Deep Well).
Le Grand Sutcliffe & Gell, Ltd.
Sulzer Bros.

Pumps (Variable Delivery, Oil).
Variable Speed Gear, Ltd.

Punching, Shearing, and Bending Machines.
Demag Aktiengesellschaft.

Pyrometers.
Incandescent Heat Co., Ltd.

Quarrying Tools.
Demag Aktiengesellschaft.

Quasi-Arc Welding Accessories and Electrodes.
Ferrox Company, Ltd.
Quasi-Arc Co., Ltd.

Radiator Clips.
Robinson, L., & Co.

Rail Manufacturers.
Dorman, Long & Co., Ltd.
Summerson, Thomas, & Sons, Ltd.

FOR ADVERTISERS' ADDRESSES AND POSITIONS OF THEIR ANNOUNCEMENTS, CONSULT FOREGOING ALPHABETICAL LIST, PAGES 2989 TO 2997.

Rails and Fastenings.

Summerson, Thomas, & Sons, Ltd.

Railway Brakes.

Consolidated Brake & Engineering Co., Ltd.

Railway Points and Crossings.

Summerson, Thomas, & Sons, Ltd.

Railway Ramps.

Summerson, Thomas, & Sons, Ltd.

Railway and Tramway Plant.

British Aluminium Co., Ltd.
British Power Railway Signal Co., Ltd. (Signals).
Clarke, Chapman & Co., Ltd.
Liverpool Electric Cable Co., Ltd.
Reddaway, F., & Co., Ltd.
Summerson, Thomas, & Sons, Ltd.

Railway Signals and Safety Appliances.

British Power Railway Signal Co., Ltd.

Railway Stop Blocks.

Summerson, Thomas, & Sons, Ltd.

Rainwater Goods.

Asbestos Cement Building Products, Ltd.

Recorders (Speed).

Consolidated Pneumatic Tool Co., Ltd. (for Locomotives and Trains).

Recorders (Water).

Palatine Engineering Co., Ltd.

Recording Instruments.

Dickinson, F. E.
Drayton Regulator & Instrument Co., Ltd.
International Gas Detectors, Ltd. (late 'W. R.' Patents, Ltd.).
Palatine Engineering Co., Ltd.
Rotherham & Sons, Ltd.
Weston Electrical Instrument Co.,Ltd.

Reducing Valves.

See VALVES AND COCKS.

Reduction Gears.

Demag Aktiengesellschaft.
Parsons Marine Steam Turbine Co., Ltd.
Stone, J., & Co., Ltd.

Reflectors and Lanterns.

Simplex Conduits, Ltd.

Refractories.

Stein, John G., & Co., Ltd.
West's Gas Improvement Co., Ltd.

Refrigerating Machinery.

Boby, Robert, Ltd.
Brotherhood, Peter, Ltd.
Liverpool Refrigeration Co., Ltd.
Pulsometer Engineering Co., Ltd.
Sulzer Bros.

Refrigerating Tubes and Coils.

Booth, James, & Co. (1915), Ltd.
Liverpool Refrigeration Co., Ltd.

FOR ADVERTISERS' ADDRESSES AND POSITIONS OF THEIR ANNOUNCEMENTS, CONSULT FOREGOING ALPHABETICAL LIST, PAGES 2989 TO 2997.

Refuse Utilization Plants.

Heenan & Froud, Ltd.
Patent Lightning Crusher Co., Ltd.

Regenerative Air Preheaters.

Howden - Ljungstrom Preheaters (Land), Ltd.

Regulators (Automatic).

Drayton Regulator & Instrument Co., Ltd.
Incandescent Heat Co., Ltd.

Riveters (Electric).

United States Metallic Packing Co., Ltd.

Riveters (Hydraulic & Pneumatic).

Consolidated Pneumatic Tool Co., Ltd.

Rivets (Iron, Steel, Copper, Naval Brass, etc.).

British Aluminium Co., Ltd.
Stone, J., & Co., Ltd.

Rock Drills.

Consolidated Pneumatic Tool Co., Ltd.
Demag Aktiengesellschaft.
English Steel Corpn., Ltd.

'Rok' Roofing.

Anderson, D., & Son, Ltd.

Roller Bearings.

Hoffmann Manufacturing Co., Ltd.

Roller Chains (Heavy).

Coventry Chain Co., Ltd.

Rolling Mill Engines.

Demag Aktiengesellschaft.

Rolling Mills.

Demag Aktiengesellschaft.

Rolling Stock.

See RAILWAY AND TRAMWAY PLANT.

Roofing Contractors.

Anderson, D., & Son, Ltd.
Ruberoid Co., Ltd.
Willesden Paper & Canvas Works, Ltd.

Roofing Felts.

Anderson, D., & Son, Ltd.
Asbestos Cement Building Products, Ltd.
Ruberoid Co., Ltd.
Turner Brothers Asbestos Co., Ltd.

Roofs, Iron Buildings, etc.

Anderson, D., & Son, Ltd.
Braby, Fredk., & Co., Ltd.
British Aluminium Co., Ltd.
Henderson & Glass.
Ruberoid Co., Ltd.
Willesden Paper & Canvas Works, Ltd.

Rope Pulleys.

See BELT AND ROPE PULLEYS.

Ropes: Steel Wire, Locked Coil and Flattened Strand.

Latch & Batchelor, Ltd.

Ropes: Wire, Hemp, Manilla, Leather, Cotton, and Asbestos.

Hart, Thomas, Ltd. ('Lambeth' Cotton).
Kenyon, Wm., & Sons, Ltd. (Patent Inter-Stranded Cotton).
Latch & Batchelor, Ltd. (Wire).
Whitecross Co., Ltd.
Willcox, W. H., & Co., Ltd.
Wrights' Ropes, Ltd.

Ropeways (Aerial).

See AERIAL ROPEWAYS.

Rubber Flooring and Tiling.

Leyland & Birmingham Rubber Co., Ltd.

Rubber Goods (General).

Leyland & Birmingham Rubber Co., Ltd.

'Ruberoid' Roofing.

Ruberoid Co., Ltd.

Rust Preventives.

Crockatt, W., & Sons, Ltd. (Anti-Rust Varnish).
Graphite Products, Ltd.

Sandwashers.

Glenfied & Kennedy, Ltd. ('Peeble's' Patent).

Scale and Corrosion in Boilers (Preventives of).

Dampney, J., & Co., Ltd.

Scaling Tools.

Crockatt, W., & Sons, Ltd.
Demag Aktiengesellschaft.

Schools and Colleges (Engineering).

Heriot-Watt College, Edinburgh.
Imperial College of Science and Technology, London.
Loughborough Technical College.
Manchester College of Technology.

Scientific Instruments.

Booth, James, & Co. (1915), Ltd. ('Duralumin' for, Regd. Trade Mark, Vickers' Special Alloy).

Scrap Presses.

Demag Aktiengesellschaft.

Screw Propellers.

Stone, J., & Co., Ltd.
Thornycroft, John I., & Co., Ltd.

Sections and Rods (Extruded), Brass, Bronze, etc.

Delta Metal Co., Ltd.

Separation of Oil from Condensed Steam.

Premier Cooler and Engineering Co., Ltd.

Sewage Disposal Appliances.

Stone, J., & Co., Ltd.

Shaft Sinkers.

François Cementation Co., Ltd.

Shafting and Gearing.

English Steel Corpn., Ltd.
Foster, Joseph & Sons, Ltd.

Sheet Metal Work, Cylinder Cases, etc.

Braby, Fredk., & Co., Ltd.
Kenyon, Wm., & Sons, Ltd.

Sheets (Iron and Steel).

Braby, Fredk., & Co., Ltd.
English Steel Corpn., Ltd.

Ship Builders.

Cochran & Co. (Annan), Ltd.
Spencer-Bonecourt, Ltd.
Swan, Hunter & Wigham Richardson, Ltd.
Thornycroft, John I., & Co., Ltd.
Yarrow & Co., Ltd.

FOR ADVERTISERS' ADDRESSES AND POSITIONS OF THEIR ANNOUNCEMENTS, CONSULT FOREGOING ALPHABETICAL LIST, PAGES 2989 TO 2997.

Ship and Engine Repairers.

Swan, Hunter & Wigham Richardson, Ltd.
Thornycroft, John I., & Co., Ltd.
Yarrow & Co., Ltd.

Ships' Insulation.

Anderson, Thomas, Ltd.

Sieves and Screens.

Potter, F. W., & Soar, Ltd.

Signals.

See RAILWAY SIGNALS.

Silica Bricks.

Stein, John G., & Co., Ltd.

Silicate Cotton and Slag Wool Coverings.

Kenyon, Wm., & Sons, Ltd.

Skylights.

Braby, Fredk., & Co., Ltd.

Slate Punches.

Carter, J., Sons & Co., Ltd.

Slide Rules.

See INSTRUMENTS, MATHEMATICAL, ETC.

Slide Valves (Parallel).

Hopkinsons, Ltd.

Sluice Gates and Valves.

Guest & Chrimes, Ltd. (Valves for Water, Steam, Sewage, and Gas).
Premier Cooler and Engineering Co., Ltd.

Smiths' Hearths and Plant.

Massey, B. & S., Ltd.

Smoke Preventers and Steam Generators.

Howden, James, & Co., Ltd.

Solder.

Fry's Metal Foundries, Ltd.

Solid Rubber Cables.

See CABLES.

Solid Rubber Tyres for Commercial Vehicles.

Leyland and Birmingham Rubber Co., Ltd.

Soot Removers.

Willcox, W. H., & Co., Ltd.

Spur and Bevel Gearing.

Demag Aktiengesellschaft.

Staircases (Fire-resisting).

Braby, Fredk., & Co., Ltd.

Stair Treads.

British Aluminium Co., Ltd.
Coan, Robert W., Ltd. (Aluminium).
Potter, F. W., & Soar, Ltd.

Stampings.

English Steel Corpn., Ltd.
Delta Metal Co., Ltd.
Shardlow, Ambrose & Co., Ltd.

Stamps (Friction and Steam Drop).

Brett's Patent Lifter Co., Ltd.
Massey, B. & S., Ltd.

Stanchions (Steel).

Henderson & Glass.

Steam Boilers.

See Boiler Manufacturers.

Steam, Compressed Air, and Pneumatic Power Hammers.

Brett's Patent Lifter Co., Ltd.
Demag Aktiengesellschaft.
Massey, B. & S., Ltd.

Steam De-Superheaters.

Drayton Regulator & Instrument Co., Ltd.
Superheater Company, Ltd.

Steam Dryers and Purifiers.

Hopkinsons, Ltd.
United States Metallic Packing Co., Ltd.

Steam Engines.

Babcock & Wilcox, Ltd.
Brotherhood, Peter, Ltd.
Clarke, Chapman & Co., Ltd.
Cochran & Co. (Annan), Ltd.
Demag Aktiengesellschaft.
Howden, James, & Co., Ltd.
Sulzer Bros.
Thornycroft, John I., & Co., Ltd.
Wallsend Slipway & Engineering Co., Ltd.
Yarrow & Co., Ltd.

Steam Gauges, Cocks, Water Gauges, Safety Valves, Slide Valves, Indicators, Steam Traps, etc.

Auld, David, & Sons, Ltd.
Crockatt, W., & Sons, Ltd.
Hopkinsons, Ltd.
Klinger, Richard, Ltd.
Premier Cooler and Engineering Co., Ltd. (Steam Traps).
United States Metallic Packing Co., Ltd. ('Bradford' Steam Trap).
Willcox, W. H., & Co., Ltd.

Steam Hammers.

See Hammers.

Steam Injectors.

Hopkinsons, Ltd.
Willcox, W. H., & Co., Ltd.

Steam Jointing.

See Jointing Materials.

Steam Packing.

Beldam Packing and Rubber Co., Ltd.
British Metallic Packings Co. (1929), Ltd. ('Britimp').
Henley's, W. T., Telegraph Works Co., Ltd.
Klinger, Richard, Ltd. ('Osmanite').
Reddaway, F., & Co., Ltd.
United Kingdom Metallic Packing Syndicate, Ltd.
Walker, James, & Co., Ltd. ('Lion').
Willcox, W. H., & Co., Ltd.
Willesden Paper & Canvas Works, Ltd.

Steam Packing ('Lion').

Walker, James, & Co., Ltd.

For Advertisers' Addresses and positions of their announcements, consult foregoing Alphabetical List, pages 2989 to 2997.

Steam Pipes.
See TUBES AND FITTINGS.

Steam Pumps.
See PUMPS.

Steam Separators.
Hopkinsons, Ltd.

Steam Ships, Launches, and Yachts.
Cochran & Co. (Annan), Ltd.
Thornycroft, John I., & Co., Ltd.
Yarrow & Co., Ltd.

Steam Traps.
See STEAM GAUGES, ETC.

Steam Turbines and Fittings.
Brotherhood, Peter, Ltd.
Howden, James, & Co., Ltd.
Parsons Marine Steam Turbine Co., Ltd.
Thornycroft, John I., & Co., Ltd.
Wallsend Slipway & Engineering Co., Ltd.
Yarrow & Co., Ltd.

Steam Valves (Automatic).
Auld, David, & Sons, Ltd.
Drayton Regulator & Instrument Co., Ltd.

Steel Balls.
English Steel Corpn., Ltd.
Hoffmann Manufacturing Co., Ltd.
Ransome & Marles Bearing Co., Ltd.

Steel Bridges.
Keay, E. C. & J. (1926), Ltd.

Steel Conduits for Electric Wiring.
Simplex Conduits, Ltd.

Steel Flooring.
Braby, Fredk., & Co., Ltd.

Steel Forgings and Castings.
See CASTINGS AND FORGINGS.

Steel Manufacturers.
Haberskon, J. J., & Sons, Ltd.
Dorman, Long & Co., Ltd.
Henderson & Glass.
Saville, J. J., & Co., Ltd. (also Acid & Heat Resisting Steels).
English Steel Corpn., Ltd.

Steel Oil Drums.
Braby, Fredk., & Co., Ltd.

Steel Rails and Accessories.
Dorman, Long & Co., Ltd.
Summerson, Thomas, & Sons, Ltd.

Steel Rollers.
English Steel Corpn., Ltd.
Habershon, J. J., & Sons Ltd.
Hoffmann Manufacturing Co., Ltd.

Steel Sections.
Bennis, Ed., & Co., Ltd.
Dorman, Long & Co., Ltd.
Henderson & Glass.

Steel (Stainless).
Saville, J. J., & Co., Ltd.

Steel Tubes.
See TUBES AND FITTINGS.

Stokers (Mechanical).
See MECHANICAL STOKERS.

Stone Pulverizing Mills.

Patent Lightning Crusher Co., Ltd.

Storage Bins.

Braby, Fredk., & Co., Ltd. (Metal).

Structural Iron and Steel Work, etc.

Babcock & Wilcox, Ltd.
Bennis, Ed., & Co., Ltd.
Demag Aktiengesellschaft.
Henderson & Glass.

Submarine Cables.

See CABLES.

Sugar Mills and Machinery.

Mirrlees Watson Co., Ltd.

Sulphuric Acid.

Berk, F. W., & Co., Ltd.
District Chemical Co., Ltd.
Imperial Chemical Industries, Ltd.

Superheaters and Steam Pipes.

Babcock & Wilcox, Ltd.
Clarke, Chapman & Co., Ltd.
Galloways, Ltd.
Spencer-Bonecourt, Ltd.
Stirling Boiler Co., Ltd.
Superheater Company, Ltd.
Vickers Boiler Co., Ltd.
Yarrow & Co., Ltd.

Surface Combustion.

Spencer - Bonecourt, Ltd. (Kirke's Patents).

Switch Gear.

Callender's Cable and Construction Co., Ltd.
Henley's, W. T., Telegraph Works Co., Ltd.
Johnson & Phillips, Ltd.
Potter, F. W., & Co.
Simplex Conduits, Ltd.

Tachometers.

Dickinson, F. E.

Tanks and Cisterns.

Braby, Fredk., & Co., Ltd.

Taps, Square Thread.

Carter, J., Sons & Co., Ltd.

Technical Training.

See SCHOOLS AND COLLEGES.

Telegraph Engineers.

Callender's Cable and Construction Co., Ltd.
Craigpark Electric Cable Co., Ltd.
Henley's, W. T., Telegraph Works Co., Ltd.
Johnson & Phillips, Ltd.
Liverpool Electric Cable Co., Ltd.

Telegraph Insulator Bolts and all kinds of Ironwork for Telegraph and Telephone.

Henley's, W. T., Telegraph Works Co., Ltd.
Johnson & Phillips, Ltd.

FOR ADVERTISERS' ADDRESSES AND POSITIONS OF THEIR ANNOUNCEMENTS, CONSULT FOREGOING ALPHABETICAL LIST, PAGES 2989 TO 2997.

Telegraph Wire.

See WIRE.

Telephone Equipment.

Henley's, W. T., Telegraph Works Co., Ltd.

Temperature Controls and Regulators.

Drayton Regulator & Instrument Co., Ltd.
Incandescent Heat Co., Ltd.
Kent, George, Ltd.

Tempering Furnaces.

See FURNACES.

Testing Instruments (Electrical).

See INSTRUMENTS.

'Thermit' Welding.

Nobel Industries, Ltd.

Thermometers.

Drayton Regulator & Instrument Co., Ltd.

Thermostatic Valves.

Drayton Regulator & Instrument Co., Ltd.
Hopkinsons, Ltd.

Thrust Bearings (Ball).

Hoffmann Manufacturing Co., Ltd.

Tip Waggons.

Braby, Fredk., & Co., Ltd.
Demag Aktiengesellschaft.

Tool Makers, Machinists, etc.

Clarke, Chapman & Co., Ltd.
Consolidated Pneumatic Tool Co., Ltd.
Crockatt, W., & Sons, Ltd.
English Steel Corpn., Ltd.
Massey, B. & S., Ltd.
Shardlow, Ambrose & Co., Ltd.
Simplex Conduits, Ltd.
United States Metallic Packing Co., Ltd.
Willcox, W. H., & Co., Ltd.

Tool Steel.

English Steel Corpn., Ltd.
Saville, J. J., & Co., Ltd.

Town's Gas.

See GAS WORKS PLANT.

Trade Mark Plates.

Coan, Robert W., Ltd.

Train Lighting Plant.

Stone, J., & Co., Ltd.

Tramways & Railways (Electrical).

See ELECTRIC RAILWAYS AND TRAMWAYS.

Tramways (Steel Wire Ropes for).

Wrights' Ropes, Ltd.

Transformer Gaskets.

Cork Manufacturing Co., Ltd.

Transformers (Electric Power).
Johnson & Phillips, Ltd.

Transmission (Hydraulic, Oil).
Variable Speed Gear, Ltd.

Trench Braces.
Consolidated Pneumatic Tool Co., Ltd.

Trucks—Petrol.
Brett's Patent Lifter Co., Ltd.

Tube Cleaners.
Crockatt, W., & Sons, Ltd.
Willcox, W. H., & Co., Ltd.

Tubes and Fittings.
Babcock & Wilcox, Ltd.
British Aluminium Co., Ltd.
English Steel Corpn., Ltd. (Weldless Steel).
See also STEEL CONDUITS.

Tubes (Pneumatic).
Booth, James, & Co. (1915), Ltd.

Tunnelling Plants.
Demag Aktiengesellschaft.

Turbine Gears.
See GEARED TURBINES.

Turbine Pumps.
Pulsometer Engineering Co., Ltd.

Turbines (Steam).
See STEAM TURBINES.

Turbo Generators.
See DYNAMOS AND MOTORS.

Turbo Pumps.
Weir, G. & J., Ltd.

Turntables.
Demag Aktiengesellschaft.
Summerson, Thomas, & Sons, Ltd.

'Unbreakable' Lamps and Oil Filters.
Wells, A. C., & Co., Ltd.

Underslating.
Willesden Paper & Canvas Works, Ltd.

Vacuum and Steam Traps.
Auld, David, & Sons, Ltd.
Drayton Regulator and Instrument Co., Ltd.
Hopkinsons, Ltd.
Premier Cooler and Engineering Co., Ltd.
United States Metallic Packing Co., Ltd.

Vacuum Evaporators for Chemical Liquids.
Mirrlees Watson Co., Ltd.

Vacuum Gauges.
Hopkinsons, Ltd.

Vacuum Pans (Concentrating).
Mirrlees Watson Co., Ltd.

Vacuum Pumps.
See PUMPS.

Valve Reseaters.

Crockatt, W., & Sons, Ltd.

Valves and Cocks.

Auld, David, & Sons, Ltd.
Beldam Packing and Rubber Co., Ltd.
Crockatt, W., & Sons, Ltd.
Dick, R. & J., Ltd.
Drayton Regulator & Instrument Co., Ltd.
Hopkinsons, Ltd.
Klinger, Richard, Ltd.
Palatine Engineering Co., Ltd.
Premier Cooler and Engineering Co., Ltd.
Reddaway, F., & Co., Ltd.
United States Metallic Packing Co., Ltd.
Walker, James, & Co., Ltd.
Weir, G. & J., Ltd.
Willcox, W. H., & Co., Ltd.

Valves for Water, Oil, etc.

Glenfield & Kennedy, Ltd.

Variable Speed Oil Transmission Gears.

Variable Speed Gear, Ltd.

Varnishes.

Crockatt, W., & Sons, Ltd. (Anti-Rust Varnish).
Imperial Chemical Industries, Ltd.

Ventilating Engineers.

Boyle, Robert, & Son.
Musgrave & Co., Ltd.
Potter, F. W., & Soar, Ltd.

Ventilating, Heating, etc.

Boyle, Robert, & Son.
Musgrave & Co., Ltd.
Potter, F. W., & Soar, Ltd.

Ventilating Shafts.

Braby, Fredk., & Co., Ltd.

Voltmeters.

See INSTRUMENTS (ELECTRICAL).

Vulcanite and Vulcanised Fibre.

Reddaway, F., & Co., Ltd.

Wall and Ceiling Sheets.

Willesden Paper and Canvas Works, Ltd.

Waste Heat Boilers.

Clarkson Thimble Tube Boiler Co., Ltd.
Spencer-Bonecourt, Ltd.

Waste Oil Filters, Cabinets and Cisterns

Wells, A. C., & Co., Ltd.

Waste Water Meters.

Palatine Engineering Co., Ltd.

Water Cooling Apparatus.

Heenan & Froud, Ltd.
Premier Cooler and Engineering Co., Ltd.

Water Distilling Apparatus, etc.

Mirrlees Watson Co., Ltd.
Stone, J., & Co., Ltd.

See also CONDENSERS.

FOR ADVERTISERS' ADDRESSES AND POSITIONS OF THEIR ANNOUNCEMENTS, CONSULT FOREGOING ALPHABETICAL LIST, PAGES 2989 TO 2997.

Water Gauges, etc.

See STEAM GAUGES.

Water Level Indicators.

Kent, George, Ltd.
Klinger, Richard, Ltd.

Water Meters and Water Fittings.

Glenfield & Kennedy, Ltd.
Guest & Chrimes, Ltd.
Kent, George, Ltd.
Palatine Engineering Co., Ltd.

Water Recorders.

Kent, George, Ltd.
Palatine Engineering Co., Ltd.

Water Regulators.

Copes Regulators Ltd. (Feed Water).

Water Softening and Purifying Apparatus.

Babcock & Wilcox, Ltd.
Underfeed Stoker Co., Ltd. ('Usco' System).

See also FEED WATER HEATERS, ETC.

Water Towers.

Premier Cooler and Engineering Co., Ltd.

Water-Tube Boilers.

See BOILER MANUFACTURERS.

Wedge Gate Valves.

Hopkinsons, Ltd.

Weighing Machines.

Bennis, Ed., & Co., Ltd.
Howes, S.

Welding and Cutting of Metals.

Allen-Liversidge, Ltd.
Carbic, Ltd.
Coan, Robert W., Ltd.
District Chemical Co., Ltd.
Ferro-Arc Welding Co., Ltd.
Ferrox Company, Ltd.
Harvey, G. A., & Co. (London), Ltd.
Imperial Chemical Industries, Ltd.
Peters, G. D., & Co., Ltd.
Quasi-Arc, Co., Ltd.

'Wells,' Lights.

Wells, A. C., & Co. Ltd.

Wheel Gearing.

Demag Aktiengesellschaft.

Wheels (Steel).

Renold (Hans), Ltd.

White Lines (Permanent).

Coan, Robert W., Ltd.

'Willesden' Paper and Canvas.

Willesden Paper & Canvas Works, Ltd.

Winding and Hauling Engines and Plant.

Clarke, Chapman & Co., Ltd.
Demag Aktiengesellschaft.

Winding and Hauling Ropes.

Hart, Thomas, Ltd. ('Lambeth' Cotton Ropes).
Kenyon, Wm., & Sons, Ltd. (Cotton Ropes).
Wrights' Ropes, Ltd.

See also ROPES.

Wire Gauze.

Potter, F. W., & Soar, Ltd.

Wire Manufacturers.

British Aluminium Co., Ltd.
Delta Metal Co., Ltd.
Dorman, Long & Co., Ltd. (Steel).
Latch & Batchelor, Ltd.
Wrights' Ropes, Ltd.

Wire (Insulated) Manufacturers.

Callender's Cable & Construction Co., Ltd.
Craigpark Electric Cable Co., Ltd.
Henley's, W. T., Telegraph Works Co., Ltd.
Johnson & Phillips, Ltd.
Liverpool Electric Cable Co., Ltd.
Macintosh Cable Co., Ltd.

Wire Rope Tramways.

Wrights' Ropes, Ltd.

Wire Ropes.

Latch & Batchelor, Ltd.
Wrights' Ropes, Ltd. (Flattened Strand & Locked Coil).

Wire Ropeways.

Wrights' Ropes, Ltd.

Wire Work and Woven Wire.

Braby, Fredk., & Co., Ltd.
Potter, F. W., & Soar, Ltd.

Wiring Systems (Flexible).

Callender's Cable & Construction Co., Ltd.
Henley's, W. T., Telegraph Works Co., Ltd.
Macintosh Cable Co., Ltd.
Simplex Conduits, Ltd.

Wood Preservatives.

Anderson, D., & Son, Ltd. ('Sidol').
Carson, Walter, & Sons.

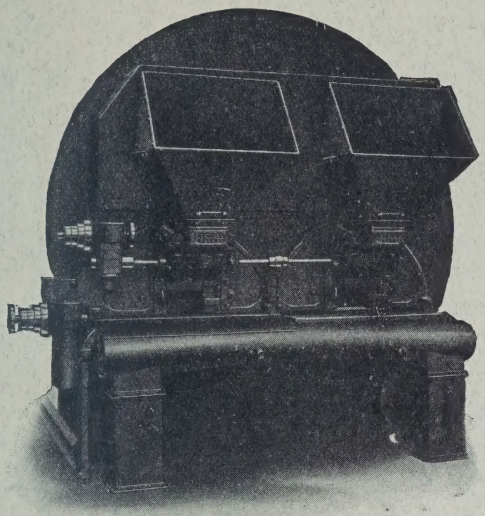
Workshops (Lighting of).

Allen-Liversidge, Ltd.

Wrought and Sheet Ironwork.

Braby, Fredk., & Co., Ltd.
Kenyon, Wm., & Sons, Ltd.
Potter, F. W., & Soar, Ltd.

Printed in England at The Ballantyne Press
Spottiswoode, Ballantyne & Co. Ltd.
Colchester, London & Eton



Lancashire Boiler gives

82·5% efficiency

with the new

Bennis Air Draught Stoker

No Steam Jets

Absolutely Smokeless.

ED. BENNIS & CO. LTD.,

LITTLE HULTON, BOLTON,

And 28 Victoria Street, Westminster, London, S.W. 1

With
Demag
Products
you work quicker
and cheaper—
You save time
and wages.

We are suppliers of:

ALL KINDS OF MACHINES FOR ROAD MAKING,
RAILWAY CONSTRUCTION AND TUNNEL BUILDING

Demag Universal Shovel Excavator, Type U, for the rapid removal of large quantities of earth, also for working in quarries.

Demag Scraper Plants, the most practical and simple conveying device for removing dumps and making cuttings.

Demag Motor Compressors, single- or double-axle, the light portable compressed-air producer with direct motor drive.

Demag Pneumatic Tools, for all purposes and for every industry, Drill Hammers, Riveting- and Chisel-Hammers, Concrete Breakers. Ask for our illustrated catalogues. Also regarding Large Volume Compressors, Toothed-wheel gears, Machinery and Hauling Engines for mines.

DEMAG Aktiengesellschaft DUISBURG

British Agents for Excavators, Steam Cranes and Gears

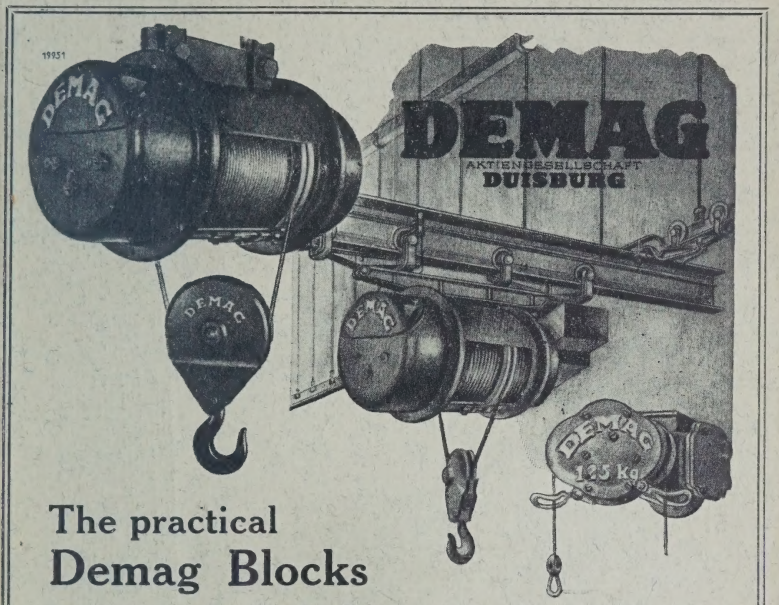
MAURICE WARLOW & Co.

12 Victoria Street, London, S.W.1, England

British Agents for Air-Compressors

THE ROTARY AIR COMPRESSORS Co., Ltd.

12 Victoria Street, London, S.W.1, England



The practical

Demag Blocks

are now operating in thousands in all the industrial countries of the world.

We are suppliers of:

MODERN ELECTRIC HIGH-SPEED HOISTING DEVICES AND TRANSPORTING PLANTS OF ALL KINDS

Demag Blocks, Model K, new construction with considerable improvements, they work quicker, easier and cheaper.

Demag Suspension Cranes (Patented), the new, practical crane construction design, with which the runways can be assembled by the customer's staff, as they consist of standardized parts.

Demag Double-acting Hoists (Patented), with 2 load hooks, 250 kg. carrying capacity, the small, practical lifting device for sacks and cases.

Demag Steam Cranes, running on narrow gauge track, for quick transport, working in factories and on stockyards. Write for our illustrated catalogues, including those for modern harbour cranes, level luffing cranes, loading bridges, shipyard cranes, floating cranes, factory overhead travelling cranes and cranes for other special purposes.

DEMAG Aktiengesellschaft DUISBURG

British Agents for DEMAG Blocks and other Electric Hoists

AABACAS ENGINEERING Co., Ltd., 10 Canning Place, Liverpool

British Agents for Mining Machinery

J. E. HAYDEN, Ltd., 58 Mitchell Street, Sheffield

3 0112 115706803